Fundamentals of
INTERNAL COMBUSTION ENGINES

as applied to

Reciprocating, Gas Turbine, and Jet Propulsion Power Plants

Fourth Edition, Revised 1959

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PREFACE TO THE 1954 EDITION

This edition does not differ from the previous one as far as the subject matter is concerned. It does differ, however, in the arrangement of the material presented. There are no longer two distinct parts to the book. In this edition all of the material is arranged in twenty consecutive chapters.

Changes and additions where necessary were made throughout the textbook in order to clear up points that were either not apparent or needed additional information to give a better understanding of the subject under discussion. Since the greatest interest was centered, during the past two years, in the fields of gas turbines, jet and rocket engines, the chapters dealing with these types of motors were revised and additional material was added. The chapter on Nuclear Power for Ship Propulsion was completely rewritten.

The authors wish to thank Commander W. H. Cullin, USN, for revision of Chapter XVI, Commander John W. Crawford, USN, for the writing of Chapter XX, Lt. W. W. K. Miller, Jr., USN, for revising Chapter XVII, Lt. P. M. Leavy, Jr., USN, for supplying much of the new material used in Chapters XVI, XVII, and XVIII, and to Mr. S. D. Heron, consulting engineer, for his many suggestions. The authors also wish to acknowledge their indebtedness to the instructional staff of the Department of Marine Engineering, United States Naval Academy for their valuable criticisms and help during the course of the revision of this textbook. Acknowledgment is also made to various concerns and authors for the use of their illustrations in the text.

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PREFACE

This textbook was prepared primarily for use in the course "Internal Combustion Engines" given to the midshipmen during their first class year at the U. S. Naval Academy. The purpose of the text is to present the basic theory, fundamental principles, and performance characteristics of the three major categories of internal combustion engines, i.e., *the spark ignition engine, the compression ignition engine, and the gas turbine.* A secondary purpose is to acquaint the student with the nomenclature of the various component parts of these engines, the capabilities and limitations of the various types of power plants, current development trends, and future applications.

It is not the purpose of the course, or the text, to produce design engineers or specialists in any one field of propulsion engineering. Such specialized training in a particular field is considered to be graduate work.

The text is arranged in two distinct parts. Part I covers the reciprocating types of internal combustion engines, i.e., the compression ignition and spark ignition power plants. The second part of the text is devoted to the gas turbine, particularly as applied to shipboard installations and the jet propulsion field. The latest types of propulsion plants, namely, hydrogen peroxide and nuclear power, are covered briefly.

The authors are indebted to the Head of the Department of Marine Engineering and the instructional staff, United States Naval Academy, for their interest, assistance, and helpful suggestions. Appreciation is expressed for the cooperation of the many industrial firms and engineers in industry for contributing material and illustrations; in particular, to Mr. S. D. Heron, consulting engineer, and to The Buda Company, the Hercules Motors Corporation, and The Texas Company, for their constructive comments. Personnel of the Bureau of Aeronautics, Bureau of Ships, and U. S. Naval Engineering Experiment Station of the Department of the Navy and the National Advisory Committee for Aeronautics contributed valuable technical assistance and advice. Chapter IV of Part II was prepared by the Nuclear Power Section of the Bureau of Ships.

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SYMBOLS AND BASIC UNITS

A	Area (sq ft or sq in)
AR	Air rate (lb air per hp-hr)
B	Combustion chamber or combustor
C	Compressor Coefficient or mathematical constant Number of cylinders
c	Velocity of sound (fps or fpm) Specific heat (Btu per lb-F abs)
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
D	Total piston displacement of engine (cu in)
d	Diameter (in)
F	Force (lb) Degrees Fahrenheit, temperature
f	Friction Weight fraction of exhaust residual to total mixture Frequency of pulse jet (cycles per sec)
g	Acceleration of gravity ($=32.2$, ft per sec^2)
H	Total enthalpy (Btu)
h	Specific enthalpy (Btu per lb)
I	Impulse
J	Mechanical equivalent of heat or Joule's constant ($=778$, ft-lb per Btu)
k	Ratio of specific heats (c_p/c_v)
L	Stroke (ft or in)
l	Length (ft or in)
M	Molecular weight (lb)
m	Mass (slugs or lb-sec^2 per ft) Moisture (per cent)
N	Revolutions per minute (rpm)
N_m	Mach Number
n	Polytropic exponent ($pv^n = C$) Number of revolutions per power stroke

SYMBOLS AND BASIC UNITS

P	Propulsive power (ft-lb per sec)
p	Pressure (psi or psf)
Q	Total quantity of heat (Btu) Rate of heat flow (Btu per hr)
q	Quantity of heat per unit weight of working fluid (Btu per lb)
R	Specific gas constant (=53.3, ft-lb per lb-F abs) Degrees Rankine, temperature
r	Radius (ft) Compression ratio Ratio
S	Total entropy (Btu per F)
s	Specific entropy (Btu per lb-F)
T	Absolute temperature (F abs or R) Torque (lb-ft) Thrust (lb) Turbine, component of gas turbine
t	Temperature (degrees Fahrenheit, F) Time (sec, min, hr) Thickness (ft or in)
U	Total internal energy (Btu)
u	Specific internal energy (Btu per lb)
V	Velocity (fps or fpm) Total volume (cu ft)
v	Specific volume (cu ft per lb)
W	Total weight (lb)
w	Weight rate (lb per unit time)
Wk	Total work (ft-lb or Btu)
wk	Specific work (ft-lb per lb or Btu per lb)

Greek Symbols

α	Alpha	Nozzle angle (deg) Work ratio
β	Beta	Angle of divergence of exit nozzle (deg) Blade angle (deg)

SYMBOLS AND BASIC UNITS

γ	Gamma	Air density (lb per cu ft) Specific weight ($1/v$, lb per cu ft)
Δ	Delta	Change or increment, usually of a property during a process
η	Eta	Efficiency
	η	Over-all efficiency
	η_a	Adiabatic efficiency
	η_B	Combustion chamber efficiency
	η_C	Compression or compressor efficiency
	η_g	Reduction gear efficiency
	η_m	Mechanical efficiency
	η_{int}	Intercooler effectiveness
	η_p	Propulsive efficiency
	η_{prop}	Propeller efficiency
	η_t	Indicated thermal efficiency
	η_{tb}	Brake thermal efficiency
	η_{Tu}	Turbine efficiency
	η_v	Volumetric efficiency
	η_X	Regenerator efficiency
λ	Lambda	Factor for divergence of exit nozzle (deg)
π	pi	pi (3.14)
ρ	rho	Mass density (slug per cu ft or lb-sec^2 per ft^4)
Σ	Sigma	Summation
ϕ	phi	Velocity coefficient

Subscript

a	Air; entrance to intercooler; adiabatic
act	Actual
B	Combustion chamber or combustor
b	Brake; brake mean effective pressure; exit to the intercooler; peripheral
C	Compressor
c	Compression; entrance to reheater
d	Exit from reheater
E	Expansion ratio
e	Exit or exhaust nozzle

ABBREVIATIONS

GM	General Motors
gpm	Gallons per minute
HHV	Higher heating value of fuel (Btu per lb)
Hg	Mercury
HV	Heating value of fuel (Btu per lb)
hp	Horsepower
hr	Hour
i	Indicated; ideal work
ihp	Indicated horsepower
imep	Indicated mean effective pressure (psi)
isfc	Indicated specific fuel consumption (lb fuel per ihp-hr)
iwk	Ideal work
KE	Kinetic energy
kw	Kilowatt(s)
LHV	Lower heating value of fuel (Btu per lb)
lb	Pound
ln	Logarithm to base e
mep	Mean effective pressure (psi)
mg	Milligrams
min	Minute
ml	Milliliters
mph	Miles per hour
PD	*Piston displacement (cu in)*
PN	Performance Number
psf	Pounds per square foot
psfa	Pounds per square foot absolute
psi	Pounds per square inch
psia	Pounds per square inch absolute
rpm	Revolutions per minute
rps	Revolutions per second
SAE	Society of Automotive Engineers

ABBREVIATIONS

sfc	Specific fuel consumption (lb fuel per hp-hr)
shp	Shaft horsepower
SI	Spark ignition
sp gr	Specific gravity
sq	Square
S/V	Surface to volume ratio
TDC	Top dead center
TEL	Tetraethyl lead
temp	Temperature
Thp	Thrust horsepower
VI	Viscosity index
X	Regenerator

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CONTENTS

CHAPTER I

INTRODUCTION TO RECIPROCATING ENGINES

The rise in civilization may be closely tied to improvements in transportation. Developments pertaining to transportation have enabled peoples of the world to exchange their goods more freely, to travel widely, and to understand better the customs and traditions of other nations. With the advent of more rapid forms of transportation, greater exchange has resulted and more people have profited. Cultural and scientific gains have been particularly gratifying.

The development of transportation may be divided, generally, into four stages: the discovery of the principle of the wheel, which made possible the construction of a practical vehicle; the utilization of wind to propel ships; the development of the combustion engine as a power plant to drive both ships and vehicles, and, finally, the practical application of aeronautical sciences and refinements in the combustion engine, which led to the creation of the airplane.

Our present transportation system, consisting of railroads, ships, automobiles, and airplanes, depends on mechanical power. Practically all of this power is derived from combustion engines, which convert the chemical energy of fuel into the mechanical energy utilized.

The combustion engine is a complex apparatus. Plate 1-1 indicates the multitude of subjects which enter into the design of a combustion engine. It can be seen that a *detailed* understanding of this type of engine requires a knowledge of many engineering subjects. An understanding of the *basic principles* involved, however, may be acquired in a relatively short time. This book will deal only with the material necessary for such an understanding.

1-1. Familiarization. Combustion engines may be divided into two general classes, external combustion engines and internal combustion engines.

In the *external combustion engine*, a working fluid is utilized to transfer some of the heat of combustion to that portion of the engine wherein this heat is transformed into mechanical energy. A common example of this type is the steam power plant employing a boiler and a turbine or reciprocating engine, and in which the working fluid is water (steam). Such an arrangement is not generally desirable for mobile power plants, since it entails the use of heavy and bulky heat exchangers, as well as the transportation of the supply of working fluid.

The *internal combustion engine* inducts air from the atmosphere,

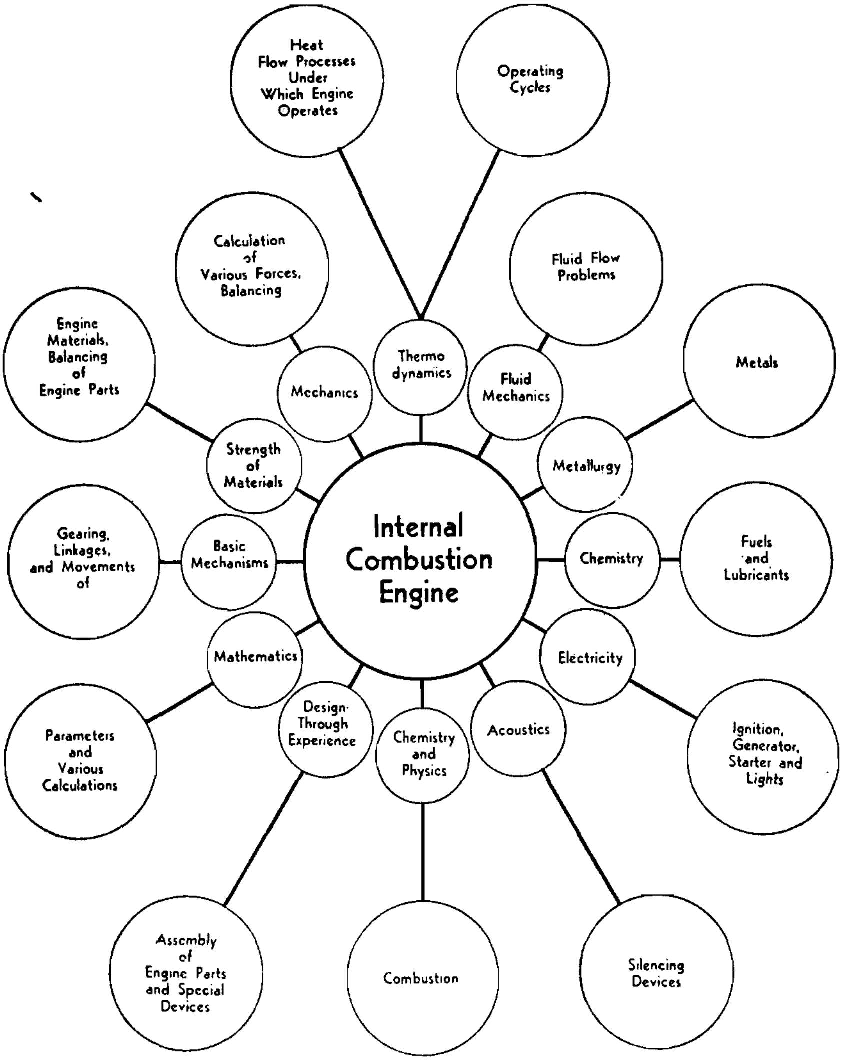


PLATE 1-1. Subjects involved in design of an internal combustion engine.

and the combustion of the fuel and air occurs in or near that portion of the engine which converts heat to mechanical energy. The expanding gases drive the engine directly, and the products of combustion are rejected back to the atmosphere. There is no necessity for an intermediate heat transferring apparatus, thus eliminating the need for heavy and bulky heat exchangers and the necessity of transporting the

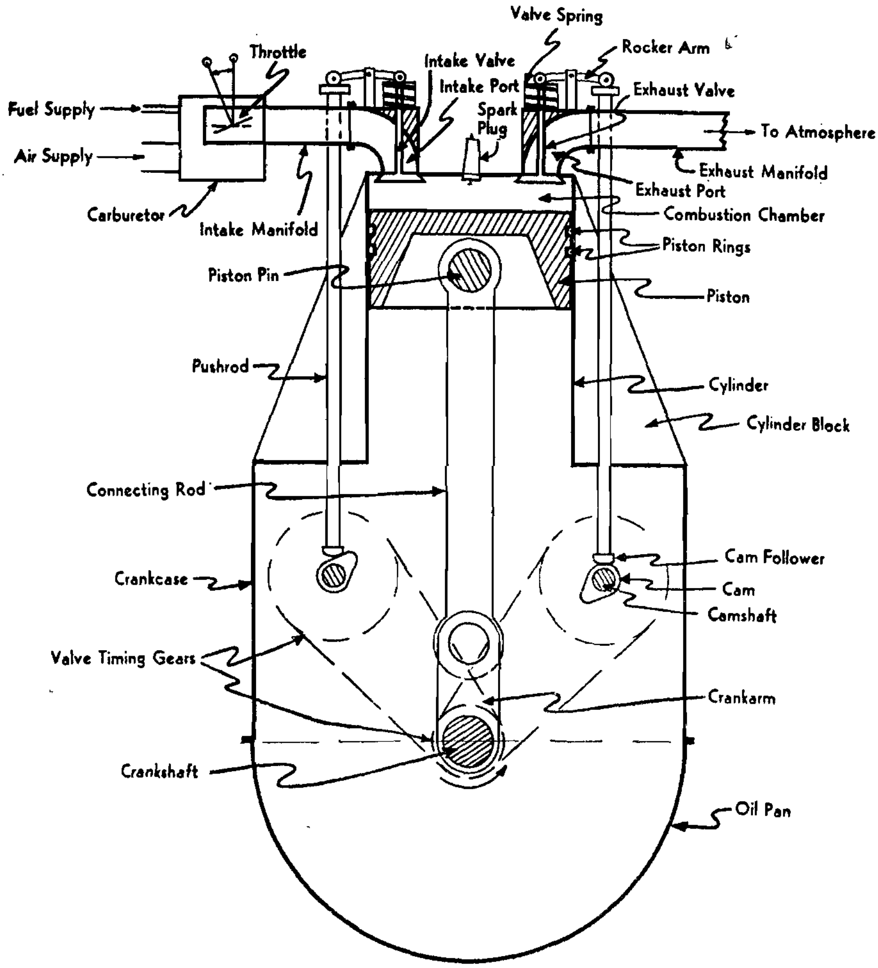


FIG. 1-1. Diagram indicating nomenclature of basic spark ignition engine parts (not to scale).

working fluid. These factors, plus the reasonably high thermal efficiencies obtained, give rise to the wide use of internal combustion engines for mobile power plants. In fact, they provide a decidedly major portion of America's power.

The first part of this book will cover internal combustion *reciprocating* engines, while the latter part will deal mainly with internal combustion engines of the *non-reciprocating* type.

Internal combustion reciprocating engines may be further divided into two basic types, namely, *spark ignition* and *compression ignition.* Normally, the abbreviations "SI" for "spark ignition" and "CI" for "compression ignition" types of reciprocating engines will be used throughout the text. Also, the words "reciprocating engine" will mean "internal combustion reciprocating engine," as distinguished from the external combustion reciprocating type.

1-2. Basic Engine Nomenclature. The names of the various parts of a reciprocating engine necessary for an understanding of the subject matter are discussed in this article and are shown in Fig. 1-1. Most of these basic parts apply equally well to both SI and CI engines, although Fig. 1-1 represents a 4-stroke cycle SI engine.

The **cylinder,** as the name implies, is the cylindrically shaped container within which the piston travels in reciprocating linear motion. The cylinder is supported in position in the **cylinder block,** attached to, or an integral part of, the **crankcase.** The volume enclosed by the upper part of the cylinder and the top of the piston during the combustion process is called the **combustion chamber.**

In the SI engine, a mixture of fuel and air from the **carburetor** enters the cylinder through the **intake manifold** and **intake port.** A **throttle** in the carburetor controls the mass of mixture entering the combustion chamber. An **intake valve** is located at the junction of the intake port and the cylinder.

A **spark plug,** located near the top of the cylinder, initiates combustion. The **piston** and **piston rings** prevent the escape of the expanding gases from the combustion chamber. Energy of the expanding gases is transmitted by the piston, through the **piston pin,** to the **connecting rod.** The connecting rod and **crank arm** of the **crankshaft** translate the linear motion of the piston into rotational motion of the crankshaft. The crankshaft rides in bearings attached to the crankcase, and that end of the crankshaft at which the power is "taken off" is usually termed the **driveshaft.**

The passage through which the products of combustion leave the combustion chamber consists of the **exhaust port** and the **exhaust manifold.** An **exhaust valve** is located at the junction of the exhaust port and the cylinder.

Both intake and exhaust valves are operated by valve mechanisms. A **camshaft** is operated through timing gears driven by the rotation of the crankshaft. Lobed **cams,** integrally connected to the camshaft, actuate the **pushrods** and **rocker arms** against the force of the **valve**

springs. The valve spring holds the valve closed except when the timed rotation of the cam mechanism forces the valve open.

In a CI engine, the fuel is injected directly into the combustion chamber through a **fuel injection nozzle.** The quantity of fuel entering is controlled by a **fuel control lever.** The air supply enters the cylinder

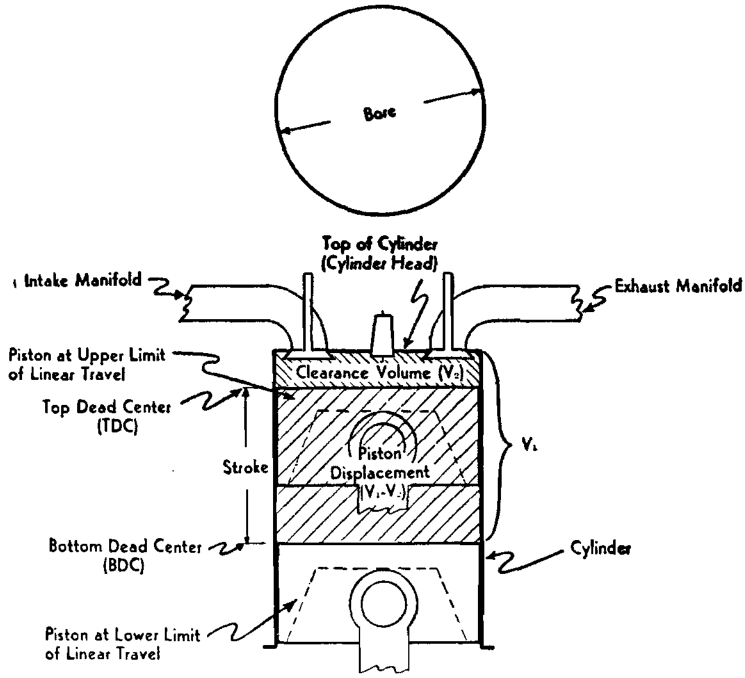


FIG. 1-2. Diagram indicating nomenclature of volumes associated with the cylinder (not to scale).

from a manifold through an intake port located in the side of the cylinder. No carburetor or throttle is therefore necessary. Also, since combustion is initiated by the high temperature of the compressed air trapped in the cylinder, no spark plug is needed.

All reciprocating engines do not have an arrangement identical to that described above. The major parts are similar, however, and the operating principles of the various assemblies are comparable.

In addition to the above nomenclature, certain standard terminology concerning volumes and measurements in the cylinder region are presented herewith, and shown in Fig. 1-2.

Bore (b)—The inside diameter of the cylinder is called the bore, and is measured in inches.

Stroke (L)—During the travel of the piston in reciprocating linear motion, there is both an upper and lower limiting position at which the direction of motion of the piston is reversed. The linear distance, measured parallel to the axis of the cylinder, between the extreme upper and lower positions of the piston is termed the stroke, and is measured in inches.

Top Dead Center (TDC)—The position of the piston when, during its linear travel, it is closest to the top of the cylinder, is called top dead center.

Bottom Dead Center (BDC)—The position of the piston when, during its linear travel, it is farthest from the top of the cylinder, is called bottom dead center.

Clearance Volume (V_2)—The volume contained in the cylinder above the top of the piston when the piston is at TDC is called the clearance volume, and is measured in cubic inches.

Piston Displacement ($V_1 - V_2$)—The volume swept through by the piston in moving between TDC and BDC; or the difference between the volume contained in the cylinder above the top of the piston when the piston is at BDC (V_1), and the clearance volume (V_2), is called the piston displacement. It is measured in cubic inches.

Compression Ratio (r)—The ratio of the volume when the piston is at BDC (V_1) to the volume when the piston is at TDC (V_2) is called the compression ratio.

$$\text{Compression ratio} = r = \frac{V_1}{V_2} \cdot$$

1-3. Engine Classification by Cylinder Arrangement. A commonly used standard for classifying multicylinder reciprocating engines is by cylinder arrangement. Some of the basic types are indicated in Fig. 1-3.

The **"in-line"** type has all cylinders arranged linearly, transmitting power to a single crankshaft. The **"V"** type is essentially two "in-line" engines set at an angle, and utilizing a common crankshaft. The **"X"** type is a more complicated version of the "V" type, essentially four "in-line" arrangements utilizing a single crankshaft. The **"opposed cylinder"** engine consists of two or more cylinders on opposite sides of a common crankshaft. It can be visualized as two "in-line" arrangements 180 degrees apart. When a single cylinder houses two pistons, each of which drives a separate crankshaft, it is called an **"opposed**

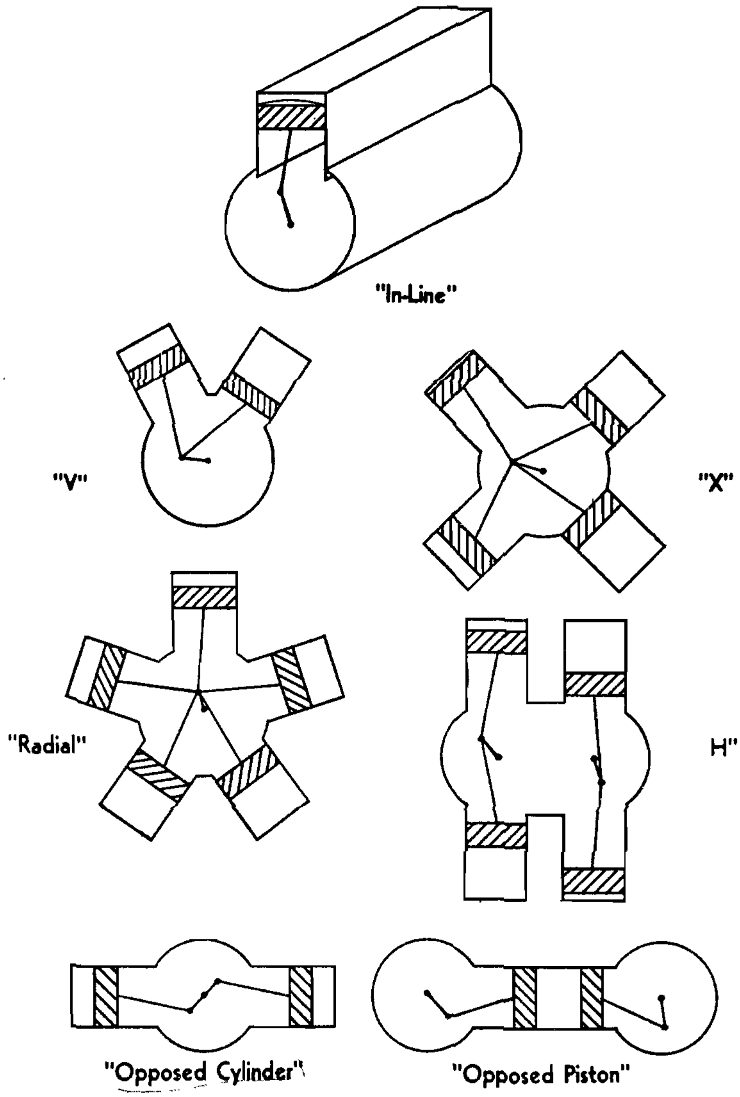


FIG. 1-3. Engine classification by cylinder arrangement (not to scale).

piston" type of engine. The **"H"** type is essentially two "opposed cylinder" types, utilizing two separate, but interconnected, crankshafts. The **"radial"** type consists of cylinders placed radially, and equally spaced, around a common crankshaft.

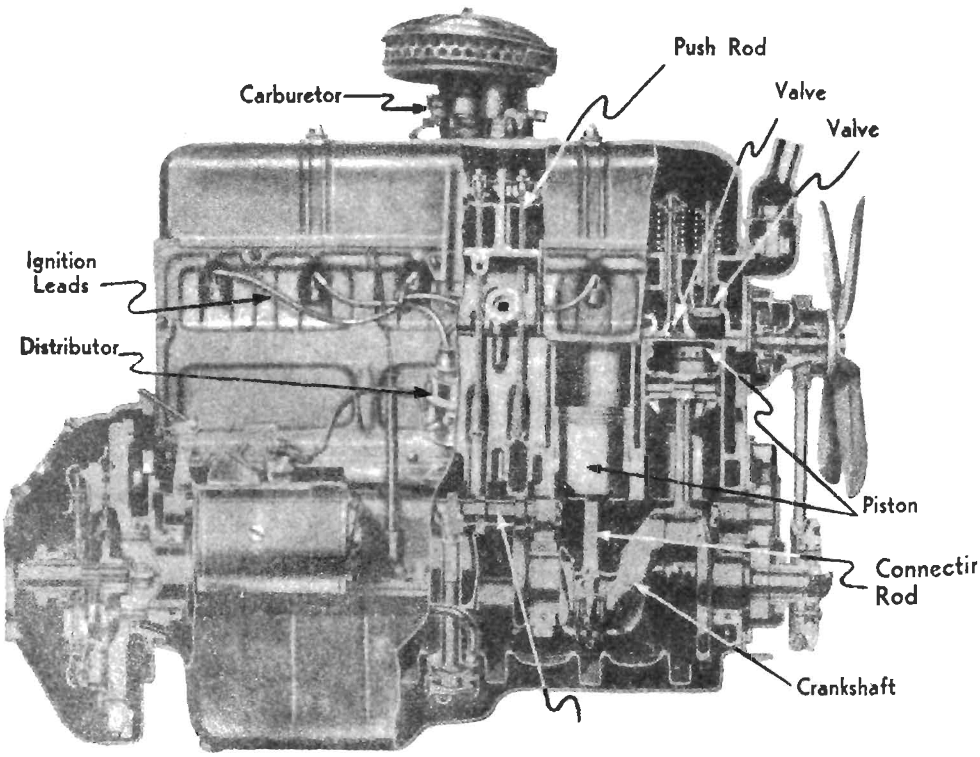


FIG. 1-4(a). "In-line" type spark ignition automobile engine (courtesy of Chevrolet Motor Division, General Motors).

In general, automobiles utilize the "in-line" and "V" type engines. The "radial" engine is used widely in medium and large aircraft, although the "in-line" type is used to some extent. Smaller aircraft generally use the "opposed cylinder" type, or small "in-line" or "V" types. The "opposed piston" type is widely used in large diesel installations. The "H" and "X" types do not presently find wide use, except in some diesel installations. A variation of the "X" type is referred to as the "pancake" engine.

1-4. Spark Ignition Engine. The SI engine is based on the principles of operation stated by Beau de Rochas, a Frenchman, and utilized in an actual engine by Otto, a German, commencing in 1862. Gradually, this cycle of events has come to be known as the Otto Cycle. Figures 1-4(a) and 1-4(b) are pictures of typical SI automobile engines, showing some of the accessory equipment attached. The SI

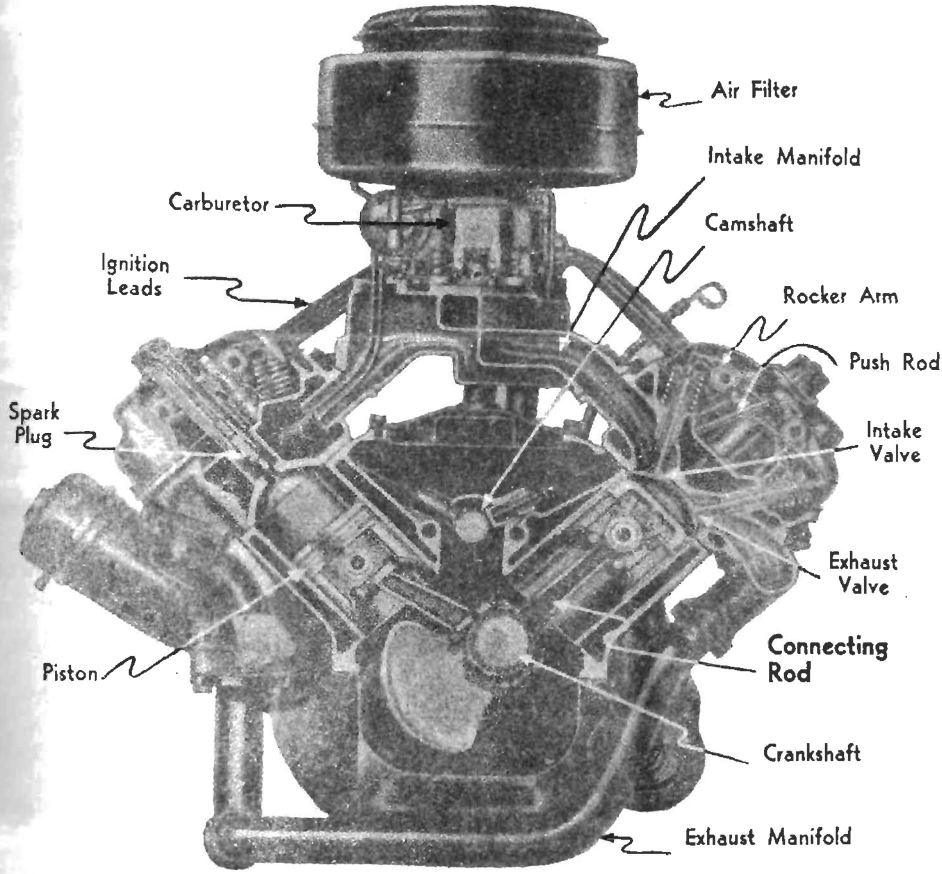


FIG. 1-4(b). "V" type spark ignition automobile engine (courtesy of Chrysler Corporation).

engine is utilized primarily in passenger cars, trucks, buses, boats, aircraft, and agricultural equipment.

SI engines may operate on the **2-stroke cycle** (one *power* stroke for each revolution of the crankshaft), or the **4-stroke cycle** (one *power* stroke for every two revolutions of the crankshaft). Due to the excessive loss of combustible mixture through escape with the exhaust gases, and the resultant high fuel consumption, 2-stroke cycle SI engines are not widely used, except in isolated cases such as outboard motors or in some motorcycles. Most SI engines, therefore, operate on the 4-stroke cycle. Figure 1-5 illustrates the four strokes of this cycle.

The order of events is as follows:

(1) **Intake stroke**—Intake valve open—exhaust valve closed—piston moving down brings in fresh combustible mixture of fuel and air from carburetor.

(2) **Compression stroke**—Both valves closed—combustible charge is compressed by the upward moving piston.

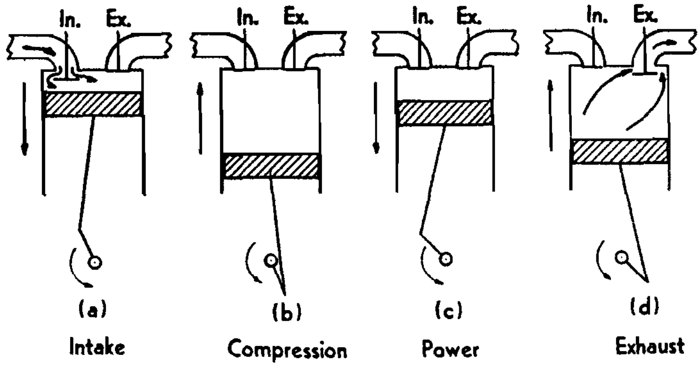


FIG. 1-5. Events in four-stroke cycle engine.

(3) **Power stroke**—Both valves closed—compressed combustible charge is ignited by the spark plug and expanding gases force piston down.

(4) **Exhaust stroke**—Exhaust valve open—intake valve closed—products of combustion forced out through the exhaust valve by the upward moving piston.

One revolution of the crankshaft occurs during the intake and compression strokes, and one revolution during the power and exhaust strokes; or, *for one complete cycle, there is only one* **power** *stroke, but the crankshaft passes through two revolutions.* These two revolutions constitute four strokes of the piston, hence the name "4-stroke cycle."

1-5. Compression Ignition Engine (4-stroke cycle). The CI engine is based on the work of Rudolph Diesel, a German, commencing about 1892, and operates on a cycle known as the Diesel cycle. Its field of use, primarily, is heavy trucks and buses, agricultural equipment, heavy construction and earth moving machinery, some stationary power plants, and certain types of marine propulsion. Since it operates at higher pressures than the SI engine, it is at present a basically heavier engine, and does not find wide use where weight is a primary consideration. Figures 1-6(a) and 1-6(b) are pictures of typical CI engines, showing some of the accessory equipment attached.

CI engines operate on either the 4-stroke or the 2-stroke cycle. The

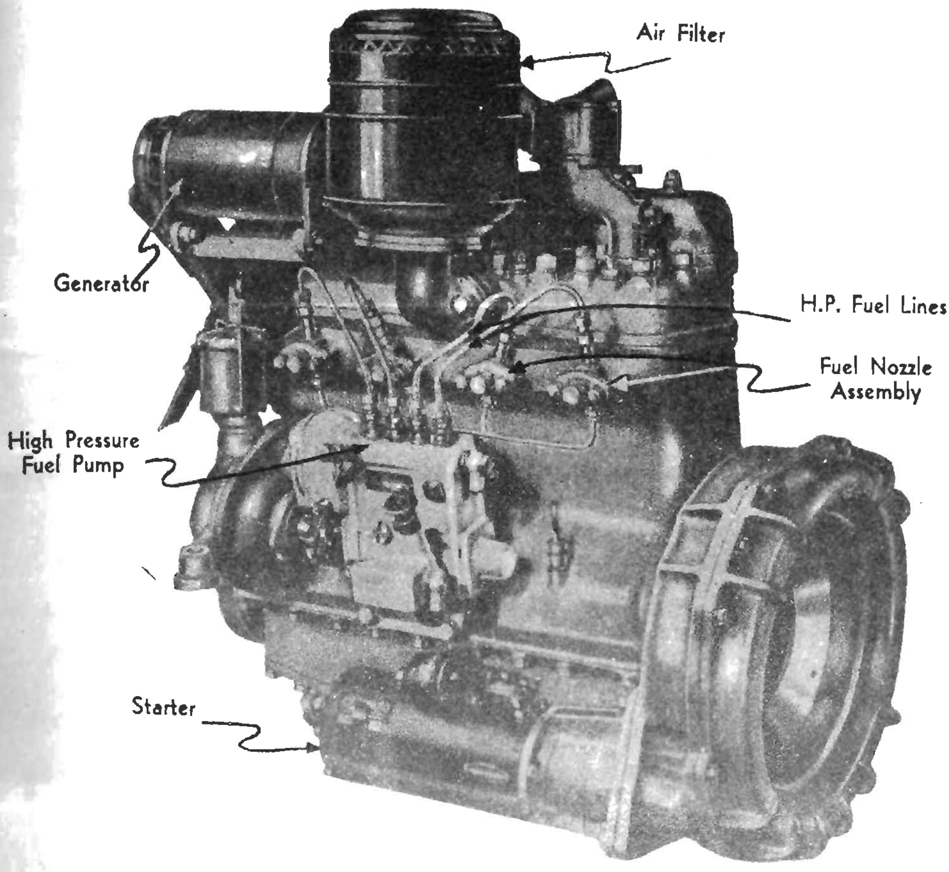


FIG. 1-6(a). "In-line" type compression ignition engine (courtesy of Hercules Motors Corporation).

4-stroke cycle consists of four distinct events, similar to those described in Article 1-4 for the SI engine, namely:

(1) Intake
(2) Compression
(3) Power
(4) Exhaust

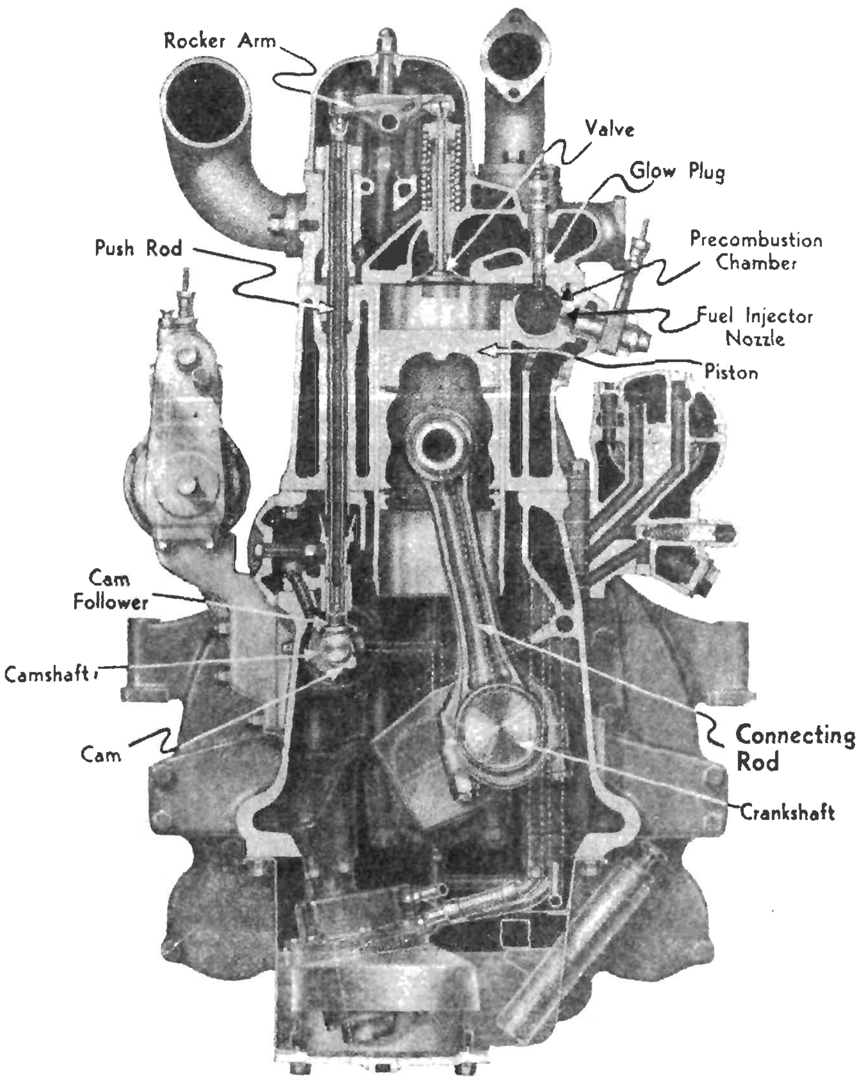
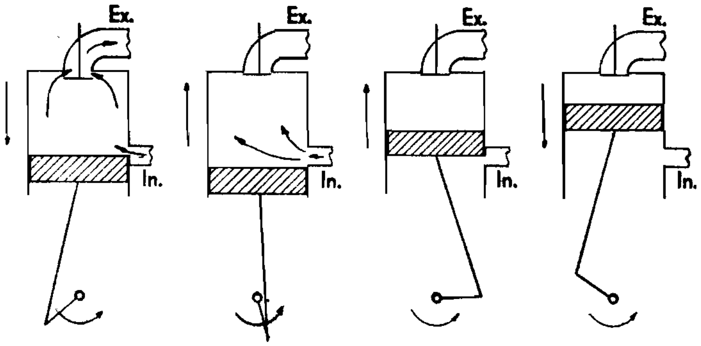


FIG. 1-6(b). Cross section of an "in-line" type compression ignition engine (courtesy of Hercules Motors Corporation).

Again, there is but *one* **power** *stroke for every two revolutions of the crankshaft.*

While the basic strokes are the same for both the CI and SI 4-stroke engines, there are certain differences in the operation of the cycles. In a CI engine, air only is inducted during the intake stroke, and only air is compressed during the compression stroke. The compression ratio of CI engines runs higher than that of SI engines, resulting in higher pressures and temperatures in the combustion chamber. Fuel is injected directly into the cylinder near the end of the compression stroke, and combustion is initiated due to the high temperature and pressure



(a) (b) (c) (d)

FIG. 1-7. Events in two-stroke cycle compression ignition engine.

of the compressed air. The CI engine, then, does not require a carburetor to mix the fuel and air, nor an ignition system to ignite the charge. A fuel injection system is required to force the liquid fuel into the cylinder against the relatively high pressures existing therein.

1-6. Compression Ignition Engine (2-stroke cycle). In Article 1-4, it was mentioned that the 2-stroke cycle is not widely used with SI engines due to the loss through the exhaust valve of some of the freshly inducted air and fuel mixture from the carburetor, with resultant high fuel consumption. The 2-stroke cycle CI engine inducts *air only*, with the fuel injected after the exhaust valve is closed. The major disadvantage of the 2-stroke SI cycle is thus not encountered, and the 2-stroke cycle CI engine is consequently widely used.

The 2-stroke cycle actually includes the same four basic events as the 4-stroke cycle, namely, intake, compression, power, and exhaust. However, *these events are performed in only two strokes of the piston, or one revolution of the crankshaft.* Figure 1-7 is a schematic diagram which indicates the sequence of events in a 2-stroke cycle engine.

As the piston moves toward BDC, with the exhaust valve open, the intake port is uncovered, and air is introduced into the cylinder under pressure, forcing most of the remaining exhaust gases out through the exhaust valve. The exhaust valve then closes. The piston, travelling on the succeeding stroke toward TDC, closes off the intake port and compresses the air, trapped in the cylinder, to a high pressure and consequent high temperature. The fuel is injected as the piston nears TDC. The fuel ignites, and the expansion of the burning gases forces the piston down on the power stroke. Just before the piston uncovers *the intake ports, the exhaust valve opens, and part of the exhaust gases* escape through the exhaust valve, due to the pressure differential existing between the cylinder and the exhaust manifold. The intake port is then uncovered, the remaining exhaust gases cleared out by the pressure of the incoming, or **scavenging** air, and the cycle repeated. One complete cycle, and thus *one* **power** *stroke, occurs on each revolution of the crankshaft.*

The type of 2-stroke engine described above is generally termed the "uniflow" type, since the scavenging air tends to flow in one direction, from the intake near the bottom of the cylinder to the exhaust at the top. There are other basic types which have no valves, both intake and exhaust being controlled by ports uncovered by the piston. In some, the intake and exhaust ports are located on opposite sides of the cylinder, while in others, they are located on the same side; but in each type, the scavenging air enters through the intake ports, circulates up to the top of the cylinder, then down and out through the exhaust ports.

The 2-stroke cycle reciprocating engine was conceived in order to realize both valve simplification and a greater power output from the same size engine. Since one power stroke occurs on every revolution of the crankshaft, the rate of occurrence of power strokes and consequently the power output, theoretically appears double that of a comparable 4-stroke cycle engine. But increasing the rate of power strokes tends to cause excessive heating of the surrounding engine parts, and can produce breakdown of the lubrication and damage to the engine. The speed of the 2-stroke cycle engine must, therefore, be reduced somewhat below that necessary to realize a doubling of the power output. Furthermore, the 4-stroke cycle offers greater fuel economy as its primary asset, with the result that both 2-stroke and 4-stroke cycle CI engines are widely used.

1-7. Fundamental Differences Between SI and CI Engines. While SI and CI engines have much in common, there are certain fundamental differences that cause their operation to vary considerably.

(1) Basic cycle—The SI engine, in general, is based on the Otto cycle, while the CI engine, in general, is based on the Diesel cycle. A study of these variations will be covered in Chapter III in connection with the analysis of cycles.

(2) Introduction of fuel—In most SI engines, the fuel and air are introduced into the combustion chamber as a gaseous mixture. A carburetor is necessary to provide the mixture, and the throttle controls the quantity of mixture introduced.

In the CI engine, the fuel is introduced directly into the combustion chamber through a fuel injection nozzle, and the air is inducted through the air manifold. Mixing takes place in the combustion chamber. The need for a carburetor is eliminated, and the fuel control valve regulates the quantity of fuel introduced.

(3) Ignition—The SI engine requires an ignition system, culminating in a spark plug within the combustion chamber, to initiate combustion. The CI engine utilizes conditions of high temperature and pressure, produced by compression of the air in the cylinder, to initiate combustion when the fuel is injected.

(4) Compression ratio range—The present day range of compression ratios is from about 5 to 10.5 for the SI engine and 12 to 20 for the CI engine. (There are exceptions to this generality which will be covered later in the text.) The upper limit of compression ratio for the SI engine is set mainly by the anti-detonant quality of present day economically feasible fuels. For the CI engine, it is limited mainly by the rapidly increasing weight of the engine structure as the compression ratio is further increased.

(5) Weight—The CI engine, since it operates at considerably higher pressures, must be built with greater strength, and is, generally speaking, a heavier engine.

1-8. Energy Flow Through a Reciprocating Engine. Prior to a study of the events and relationships in the component sections of a reciprocating engine system, it appears advisable to take a "bird's-eye" view of the energy flow through the over-all system. Figure 1-8 is a schematic diagram of this flow.

The fuel is provided to the combustion chamber (A), where it is burned, converting chemical energy into heat.[1] It would be desirable

[1] The amount of potential heat in the fuel may be designated as Q_s expressed in BTU's per unit time or as hp_s (horsepower supplied). The hp_s may be obtained through a conversion factor that relates the amount of heat necessary per hour to produce one horsepower (2545 Btu/hp-hr). The term hp_s is, in a way, a misnomer, since the fuel by itself does not supply any horsepower. The ratio of Q_s and the conversion factor, indicated above, being an equivalent to hp_s, often appears in computations of thermal efficiencies when the indicated horsepower and Q_s are known.

to have all this liberated energy drive the piston, but there are heat losses, primarily to the exhaust, to the coolant, and to radiation. The remaining energy, which may be converted to **indicated horsepower (ihp),** is utilized to drive the piston. The ratio of this energy to the energy originally supplied in the fuel, in comparable units, is termed **indicated thermal efficiency** (η_t).

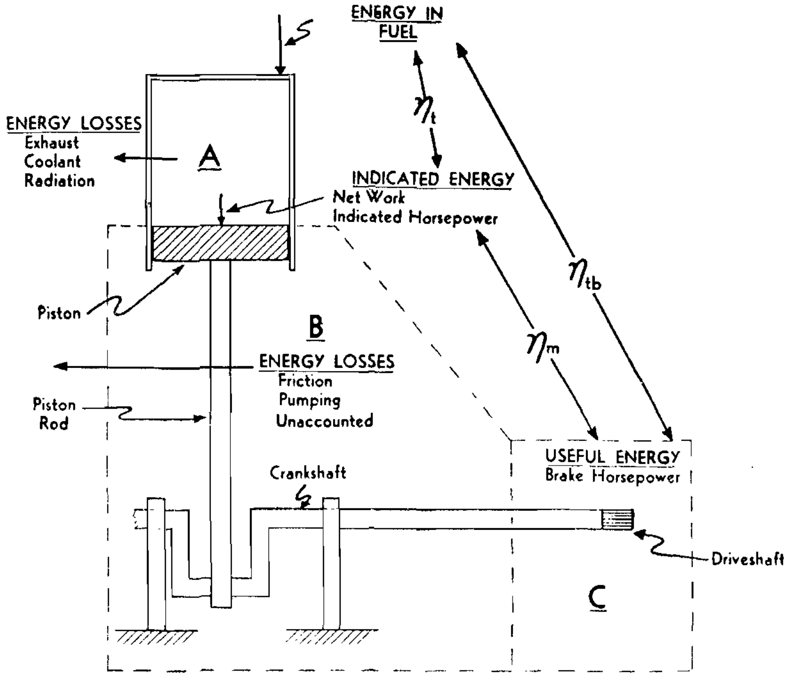


FIG. 1-8. Diagram of energy flow through reciprocating engine.

The energy applied to the piston passes through the connecting rod and crankshaft (area B), to the driveshaft. Again, there are energy losses due to friction, pumping, and other causes—the sum of all these losses, converted to power, is termed **friction horsepower (fhp).** The remaining energy (delivered to area C) is that which can be utilized mechanically. This energy may be converted to power, and is termed **brake horsepower (bhp).** The ratio of bhp (delivered power) to ihp (power provided to the piston) is called **mechanical efficiency** (η_m).

The ratio of the delivered energy, in comparable units, to the energy originally provided by the burning fuel is known as the **brake thermal efficiency** (η_{tb}) of the engine. The brake thermal efficiency (η_{tb}) equals the product of the indicated thermal efficiency (η_t) and the mechanical efficiency (η_m). Also the ihp developed at the piston, minus the fhp lost in the engine, equals the bhp delivered at the driveshaft of the engine.

Throughout the following study of reciprocating engines, these relationships must be kept foremost, since the entire subject of reciprocating engines is concerned, in some way, with factors pertaining to these energies, energy losses, or efficiencies; and the engineer is trying continuously to improve the efficiencies, increase the useful energies, and reduce the energy losses. A general visualization of this over-all picture of the energy flow should materially assist in an understanding of the subject matter which follows.

1-9. Energy Supply to Engine. In Article 1-8, it was mentioned that fuel is provided to the combustion chamber and burned to produce heat. *While the subject of the fuel and air supply, or engine "feed,"* will be covered in more detail later in the book, a brief mention of some associated parameters will be made here.

The energy supplied to the engine is in the form of chemical energy contained in the fuel. If the fuel is burned in the presence of the oxygen in the atmosphere, chemical energy is liberated in the form of heat. The amount of heat which one pound of a fuel is capable of liberating when it burns in the presence of oxygen is termed the **heating value (HV)** of the fuel, and is usually expressed in British Thermal Units (Btu) per pound of fuel.

In order to realize this liberation of energy in the form of heat, the fuel and air must present a combustible mixture in the combustion chamber. The composition of the mixture is called the **fuel-air ratio (F/A)** or the **air-fuel ratio (A/F)**, either of which expresses the proportion of fuel and air, by weight, in the mixture. The fuel-air ratio is obtained by dividing the weight of fuel consumed by the engine per given time by the weight of air used during the same period of time.

$$\text{Fuel-air ratio} = F/A = \frac{w_f(\text{lb fuel/time})}{w_a(\text{lb air/time})} = \frac{w_f}{w_a}.$$

The air-fuel ratio is the reciprocal of the fuel-air ratio:

$$\text{Air-fuel ratio} = A/F = \frac{w_a(\text{lb air/time})}{w_f(\text{lb fuel/time})} = \frac{w_a}{w_f}.$$

Either of these terms may be used, but the procedure in this text will be to refer, in general, to the air-fuel ratio (A/F).

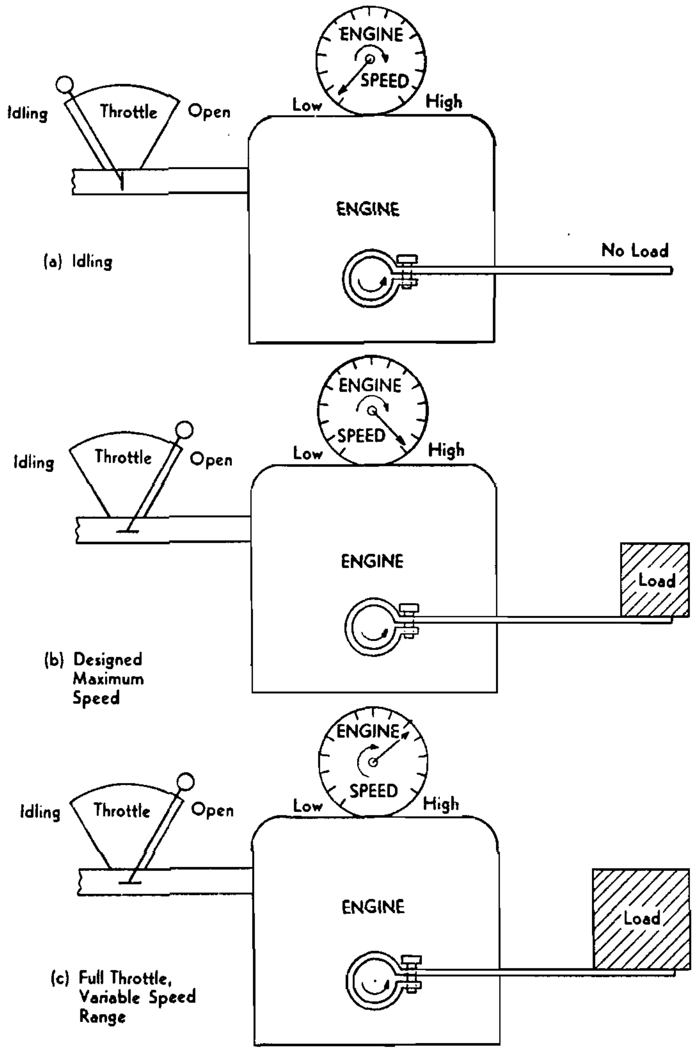


FIG. 1-9. Schematic diagram of engine speed and load control.

There is a definite limited range of air-fuel mixtures which are combustible. The proper proportions of fuel and air depend upon the chemical composition of the fuel and the conditions under which the engine is operating. *A mixture that contains just enough air to support complete combustion of all of the fuel is called a* **chemically correct,** *or* **stoichiometric,** *air-fuel ratio.* One that contains an excess of fuel is termed a "**rich**" mixture and one that contains an excess of air is termed a "**lean**" mixture. An average gasoline requires about 15 lbs of air per lb of fuel for the chemically correct mixture.

1-10. Reciprocating Engine Speed and Load Control. The control of an engine is a question of balance between the forces causing the crankshaft to rotate, and the opposing forces tending to prevent the crankshaft from rotating.

The forces causing the crankshaft to rotate are controlled essentially by regulating the mass of fuel burned in each cycle. In the SI engine, this control is accomplished by a throttle in the carburetor, which regulates the mass of combustible mixture of fuel and air entering the combustion chamber. In the CI engine, the fuel control lever regulates the mass of fuel injected into the cylinder.

The force opposing the rotation of the crankshaft is composed of the internal resistance of the engine and the external resistance applied. This external resistance is called the **load.**

The combination of throttle setting (or fuel control lever setting) and the load, determine the engine speed. If either load or throttle setting is changed an unbalance in the opposing forces is set up, and the other must be changed to set up a balance. Otherwise, the speed of the engine will change to compensate for the unbalance, and will attempt to reach a speed where a balance is reestablished.

If the throttle is almost closed, and there is no load, the relatively small amount of force produced is used to balance the internal resistance (or friction) of the engine, and the engine is said to be idling (no load). This condition is illustrated in Fig. 1-9(a).

If the throttle is opened wide and no load is applied, the engine speed will increase to a point where the engine might be destroyed. If a constant load is now applied of such magnitude that the engine operates at its maximum designed speed, then the forces produced and the load applied will be balanced, and the engine speed will remain constant. This condition is illustrated in Fig. 1-9(b).

Between these two conditions, there is an infinite number of combinations of throttle setting and applied load which will give the desired speed. This area is called the **part throttle operating range.**

After the throttle has been fully opened and the load for designed maximum speed applied, as in Fig. 1-9(b), a further increase in load will obviously reduce the speed, as illustrated in Fig. 1-9(c). This is called the **full throttle, variable load (or speed) operating range.**

When the engine is operating at a wide open throttle, it will produce maximum brake horsepower at one given engine speed. At this speed, it will be carrying a certain load, which if increased, will decrease the speed and thus the brake horsepower. On the other hand, if the load is decreased the engine speed will increase and the brake horsepower will decrease again. This speed at which maximum horsepower occurs is known as the **speed of maximum horsepower.** The statement that the engine is operating at half the load means that the load carried by the engine at the speed producing maximum horsepower was divided by two. Since we are not anxious to operate at speeds higher than the speed for maximum horsepower, the throttle will have to be reduced in order to decrease the mass of fuel used in order to establish the equilibrium, as stated previously, between the external resistance and the forces causing the crankshaft to rotate.

These principles hold equally well for SI and CI engines, since the load is not particularly concerned with the type of engine against which it is operating. The difference between the two lies essentially in the method of controlling the amount of fuel entering the cylinder and thus controlling the forces produced. *In the SI engine, the throttle controls the mass of fuel and air mixture which enters the cylinder. In the CI engine, the "throttle" (fuel control lever) regulates the mass of fuel entering the cylinder.*

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

1-1. Lester C. Lichty, "*Internal Combustion Engines,*" McGraw-Hill Book Company, Inc.

1-2. E. C. Magdeburger, "Diesel Engine in United States Navy," Journal of American Society of Naval Engineers, Inc., Volume 61, No. 1, February 1949.

EXERCISES

1-1. What are the two general classes of combustion engines, and how do they basically differ in principle?

1-2. What are the two basic types of internal combustion engines?

1-3. What is meant by the following terms:

(a) Bore	(d) Stroke	(g) "In-line" engine
(b) TDC	(e) Compression ratio	(h) "Radial" engine
(c) BDC	(f) Piston displacement	(i) "V" engine

1-4. What is the essential difference between a 4-stroke cycle engine and a 2-stroke cycle engine?

1-5. Name the four basic strokes of a 4-stroke cycle engine and describe the process entailed in each.

1-6. Why do 2-stroke cycle CI engines find wide use, whereas 2-stroke cycle SI engines are used only to a very limited extent?

1-7. What are the fundamental differences between SI and CI engines?

1-8. What do the following terms indicate:

(a) fhp	(d) η_t
(b) ihp	(e) η_m
(c) bhp	(f) η_{tb}

1-9. What is meant by the heating value of a fuel?

1-10. Explain A/F ratio.

1-11. What is meant by a "rich" mixture? A "lean" mixture?

1-12. How is speed and load control accomplished in a SI engine? In a CI engine?

1-13. What is meant by the "part throttle operating range"? The "full throttle, variable load operating range"?

1-14. A certain engine produces 125 ihp and delivers 100 bhp.

Find: (a) fhp (b) mechanical efficiency

Ans: (a) 25 hp
(b) 80%

1-15. A 4-stroke cycle SI engine delivers 700 bhp with a mechanical efficiency of 87.5 per cent. The measured fuel consumption is 360 lbs of fuel during one hour, and the air consumption is 910 pounds during a ten minute period. The heating value of the fuel is 19,000 Btu/lb fuel.

Find: (a) ihp (b) fhp (c) A/F ratio (d) n_t (e) η_{tb}

Ans: (a) 800 hp
(b) 100 hp
(c) 15.166
(d) 29.8%
(e) 26.1%

1-16. A 2-stroke cycle CI engine delivers 500 bhp while using 100 hp to overcome friction losses. It consumes 400 pounds of fuel per hour at an air-fuel ratio of 20 to 1. The heating value of the diesel fuel is 18,200 Btu/lb fuel.

Find: (a) ihp (b) η_m (c) Air consumption (lb/hr) (d) η_t (e) η_{tb}

Ans: (a) 600 hp
(b) 83.3%
(c) 8000 lb. air/hr
(d) 21%
(e) 17.5%

CHAPTER II

ENGINEERING THERMODYNAMICS

A prerequisite for a basic and a complete understanding of internal combustion engines is a working knowledge of the principles of thermodynamics. This chapter will review certain aspects of basic engineering thermodynamics that will be of importance in this course. Engineering thermodynamics deals mainly with the concepts of heat and work and the transformation of one into the other.

2-1. Energy, Work and Heat. Since an internal combustion engine is merely a device for transforming energy from one form to another, energy in its various forms is of paramount importance. The term **energy** may be defined *as the capacity of matter to produce a change from the existing conditions.* It is measured by its action or capacity to cause changes in matter or to perform work.

Energy in all its forms falls into two general categories, **stored energy** and **transitory energy**. Transitory energy is a momentary energy form intermediate between two or more stored forms and eventually becomes stored energy. Stored energy is energy within the matter of a system. A **system** is defined as a *collection of matter enclosed within prescribed boundaries.* Energies are also classified as **mechanical energy** and **thermal energy.** Thermal energy is usually expressed in British thermal units[1] (Btu), while mechanical energy is usually expressed in foot-pounds (ft-lbs).

The relationship between thermal units and mechanical units was established by Joule in his historic experiment.[2] This relationship is known as **Joule's equivalent or the mechanical equivalent of heat.**

$$J \approx 778 \text{ ft-lb per Btu}^{3} \qquad \underline{(2\text{-}1)}$$

Work. Work is defined as mechanical energy in transition. It is transitional in nature and cannot be stored in a matter or in a system. After work is completed there is no work present in the system, only the re-

[1] The mean Btu is defined as 1/180 of the energy required to raise one lb of water from 32° F to 212° F.

[2] In 1845, J. P. Joule, an English physicist, found in his experiment that churning water by paddles, and dissipating the mechanical work supplied in the form of friction and turbulence, would warm the water and give the same results that could be accomplished by the transfer of heat from an external source.

[3] J is equal to 778.26 ft-lb per Btu. This has been rounded off to 778 for simplification of calculations.

sult of work which may manifest itself in another form of energy, such as potential energy or heat. Work is usually expressed as the product of force times distance, in terms of ft-lbs (Wk) or ft-lbs per lb (wk); however, it is frequently useful to express work in terms of Btu or Btu per lb.

Heat. Heat may be defined as thermal energy in transition across the boundary of a system. In order to have a heat transfer, a temperature difference must exist between the system and its surroundings. Heat is usually expressed in terms of Btu (Q) or in terms of Btu per lb (q).

Power. Power may be defined as the time rate of energy expenditure or the time rate of doing work. If work is accomplished at the rate of 33,000 ft-lbs per minute, or 550 ft-lbs per second, the power developed is arbitrarily defined as one horsepower.

2-2. Properties, States, and Processes. It is by means of working fluids, or media, that energy is transported and transformed in an internal combustion engine. *Air is the principal constituent of the working media supplied to the engine.* It is in the gaseous state and at atmospheric conditions. The fuel flowing to the engine is in the liquid state until it leaves the jet of the carburetor or the fuel injection nozzle. After leaving the jet, the fuel may be partly or wholly vaporized or be superheated before being ignited in the combustion chamber. All states of media from liquid to gaseous are encountered in an engine.

The thermodynamic **state** or condition of the working media or the system may be defined by its properties. The six properties of a system are pressure, temperature, volume or specific volume,[4] enthalpy, internal energy, and entropy.[5] The values of several properties in combination establish a state. In general, however, two properly chosen properties are sufficient to define the state of the substance, i.e., to fix the other properties. If a property changes, a change in state occurs. A **property,** then, may be defined as *a function of the state and depends only on the state regardless of the process between states.*

Whenever a system undergoes a change of state, a thermodynamic **process** occurs. The process involved in a change of state is described in terms of the path, i.e., a series of thermodynamic states through which the system passes from the initial to the final state. Processes are named to show the consistency of a property. The processes of interest in internal combustion engines are *constant pressure, constant volume, isentropic or reversible adiabatic, and irreversible adiabatic.*

[4] **Specific volume** (v) is the volume occupied by a unit weight of a substance, usually in cubic feet per lb. The reciprocal of specific volume is **density.**

[5] Enthalpy, internal energy, and entropy are discussed later in this chapter.

A **reversible process** is one which, after completion, may be made to reverse itself exactly, with the system returning to the initial state over the identical state path. Thus, a reversible process may also be termed a frictionless process. This is an ideal or perfect process and is an approximation to the processes that occur within an actual engine. If a process is **irreversible** a greater amount of energy than originally supplied will be required to return the system to its initial state due to the presence of either mechanical friction or fluid friction.

2-3. General Energy Equation. The physical law of conservation of energy and the work of Joule led to the gradual acceptance of the premise now termed the **First Law of Thermodynamics:**

Thermal energy and mechanical energy can neither be created nor destroyed but only converted[6] from one form to the other.

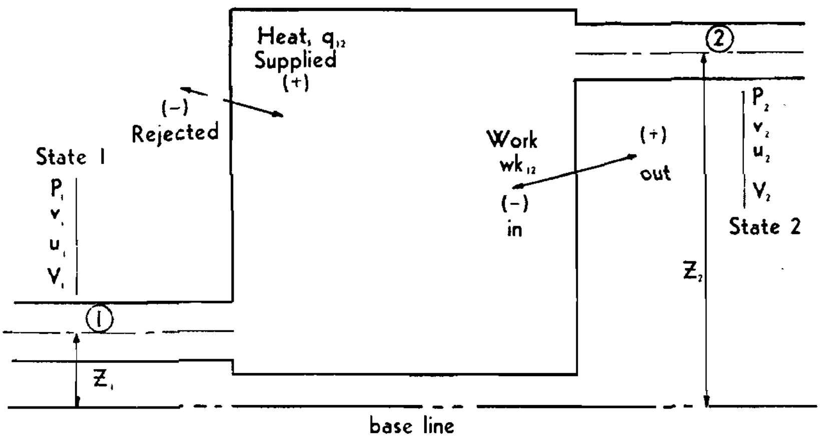


FIG. 2-1. Steady flow system.

The General Energy Equation is an algebraic expression of the first law and includes all the applicable forms of energy that may be encountered within a system. A **steady flow** system is shown in the schematic sketch of Fig. 2-1. The **general energy equation** from state 1 to state 2 in terms of Btu per unit weight of the working media becomes

$$\frac{Z_1}{J} + \frac{V_1^2}{2gJ} + \frac{p_1v_1}{J} + u_1 + q_{12} = \frac{Z_2}{J} + \frac{V_2^2}{2gJ} + \frac{p_2v_2}{J} + u_2 + \frac{wk_{12}}{J}. \quad (2\text{-}2)$$

[6] Conversion factor is 778 ft-lb per Btu.

Each term represents energy in some form as follows:

Z—**Potential energy** (ft-lb per lb) or stored mechanical energy possessed by the substance by virtue of its position in relation to a reference plane.

$\frac{V^2}{2g}$—**Kinetic energy** (ft-lb per lb) or stored mechanical energy possessed by the substance by virtue of its velocity, V (fps), or its motion.

pv—**Flow energy** (ft-lb per lb) or stored mechanical energy possessed by virtue of the substance passing into and out of the system in steady flow. An element of media that enters the system must be pushed by the element behind it with a force equal to pressure times cross sectional area, pA, through distance dL, which results in work being done. Flow energy may also be termed flow work. *Flow must be present to have flow energy.*

u—**Internal energy**[7] (Btu per lb) or stored thermal energy possessed by virtue of the motion and position of the molecules within the substance itself.

q_{12}—**Heat** (Btu per lb) or thermal energy in transition. Heat supplied to the system is considered to have a positive sign and heat rejected from the system is considered to have a negative sign.

wk—**Work** (ft-lb per lb) or mechanical energy in transition. Work done by the system is considered to have a positive sign and work done on the system is considered to have a negative sign.

The General Energy Equation may be applied to any steady flow process. It is important to note that whenever flow exists the pv term is present in the equation. Since the u and pv terms appear often, it is convenient to introduce another thermodynamic property, **enthalpy,** which is defined as

$$h = u + \frac{pv}{J} \qquad \textbf{(Btu/lb).} \qquad \textbf{(2-3)}$$

Substituting equation (2-3) into equation (2-2), the General Energy Equation becomes

$$\frac{Z_1}{J} + \frac{V_1^2}{2gJ} + h_1 + q_{12} = \frac{Z_2}{J} + \frac{V_2^2}{2gJ} + h_2 + \frac{wk_{12}}{J}\,.$$

In internal combustion engines, the potential energy change is very small or zero and is, therefore, negligible. The **flow energy equation**

[7] See reference 2-2.

for steady flow internal combustion engines such as the gas turbine and jet propulsion engines becomes

$$\frac{V_1^2}{2gJ} + h_1 + q_{12} = \frac{V_2^2}{2gJ} + h_2 + \frac{wk_{12}}{J} \qquad \text{(Btu/lb)}. \qquad (2\text{-}4)$$

This equation is applicable to the gas turbine engine and to its component parts (compressor, combustion chamber, turbine).

2-4. Non-flow Energy Equation. When certain energy terms in the General Energy Equation do not apply or are negligible, they may be deleted resulting in a simplified energy equation. This is exemplified by

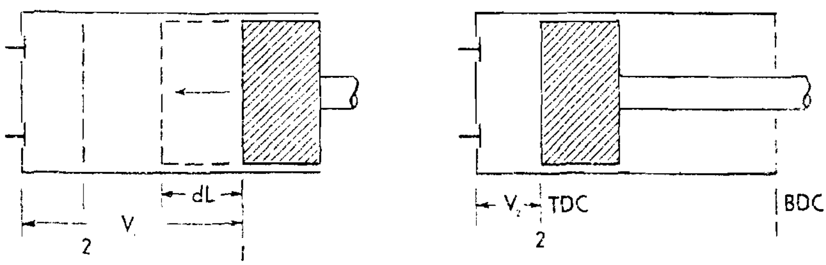


FIG. 2-2. Cylinder and piston arrangement of a reciprocating internal combustion engine.

the processes that occur within the cylinder of a reciprocating engine which is shown in a schematic sketch in Fig. 2-2. The change in the potential energy and the kinetic energy are negligible compared to the changes in q, u, h, and wk and are, therefore, dropped from the general energy equation. Since no flow exists, the flow energy term is deleted from the equation (2-2). These deletions result in a simplified equation that is important in the study of reciprocating engines and is termed the non-flow energy equation:

$$u_1 + q_{12} = u_2 + \frac{wk_{12}}{J} \qquad \text{(Btu/lb)}$$

Rearranging terms

$$q_{12} = u_2 - u_1 + \frac{wk_{12}}{J} \qquad \text{(Btu/lb)}. \qquad (2\text{-}5)$$

Note that if there is no heat transferred in the process, the work is equal to the change in the internal energy; or if no work transpires, the heat is equal to the change in internal energy.

2-5. Work and the p-v Diagram. The determination of the mechanical work performed during a non-flow reversible process, such as the compression and expansion processes that take place in a frictionless piston-cylinder arrangement (Fig. 2-2), are of primary interest in the study of reciprocating engines. The work done while the piston moves through the distance dL may be written

$$d(Wk) = pAdL$$

where A is the cross-sectional area of the piston face and p is the pressure in the cylinder. Substituting $dV = AdL$, the equation for work becomes

$$d(Wk) = pdV.$$

Integrating and establishing the limits, the entire process, then, is

$$Wk_{12} = \int_1^2 pdV \qquad \text{(ft-lb).}$$

In terms of work per lb of working medium, the equation becomes

$$wk_{12} = \int_1^2 pdv \qquad \text{(ft-lb per lb)} \qquad (2\text{-}6)$$

where

wk_{12} = the mechanical work (ft-lb per lb) done during a non-flow reversible process between states 1 and 2.
p = absolute pressure (psfa)
v = specific volume (cu ft per lb)

The area underneath a process line on a *p-v* diagram, shaded area in Fig. 2-3, is also equal to the $\int_1^2 pdv$. Therefore, the area underneath the process line on a *p-v* diagram is equal to the work done in ft-lb per lb. When the process proceeds from right to left, as in the compression stroke, the work is being *done on* the working media and is considered to be negative. When the process proceeds from left to right, as in the expansion or power stroke, the work is being *done by* the working media, i.e., gases of combustion, and is considered to be positive.

It is important to note that the magnitude of work done is a function of the path or process between the initial and final steps; therefore, the process must be known in order to determine the work (See Table 2-1).

2-6. Heat, Entropy, and the T-s Diagram. It is desirable to represent heat transferred during a process as an area. Since heat is thermal energy in transition as a result of a temperature difference, the absolute temperature is chosen as one coordinate. The other coordinate is an arbitrarily chosen thermodynamic property, entropy, so defined that for a reversible process

$$ds = \frac{dq}{T}.$$

Integrating between the limits of the initial and final states of any reversible process:

$$\int_1^2 ds = \int_1^2 \frac{dq}{T}.$$

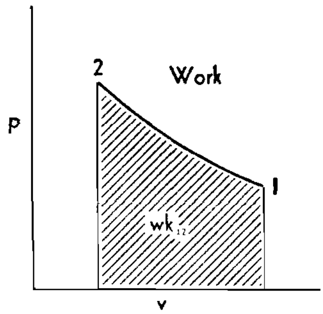


FIG. 2-3. Work and the p-v diagram.

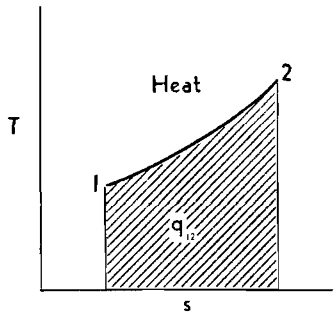


FIG. 2-4. Heat and the T-s diagram.

Therefore, by rearranging,

$$q_{12} = \int_1^2 Tds \qquad \text{(Btu/lb).} \qquad (2\text{-}7)$$

Since entropy was so chosen that heat could be represented as an area, the area underneath the path of a reversible process on the T-s diagram represents heat transferred, supplied or rejected, in Btu per lb. This is shown as a shaded area in Fig. 2-4. If the process line moves from left to right, heat is being *supplied* to the system and it is considered positive. If the process line proceeds from right to left, heat is being *rejected* by the system and is considered negative. Heat is a path function depending for its magnitude on the process used in going from the initial to the final state. (See Table 2-1.)

2-7. Specific Heats. For any reversible process, specific heat may be defined as the quantity of heat required to raise the temperature of

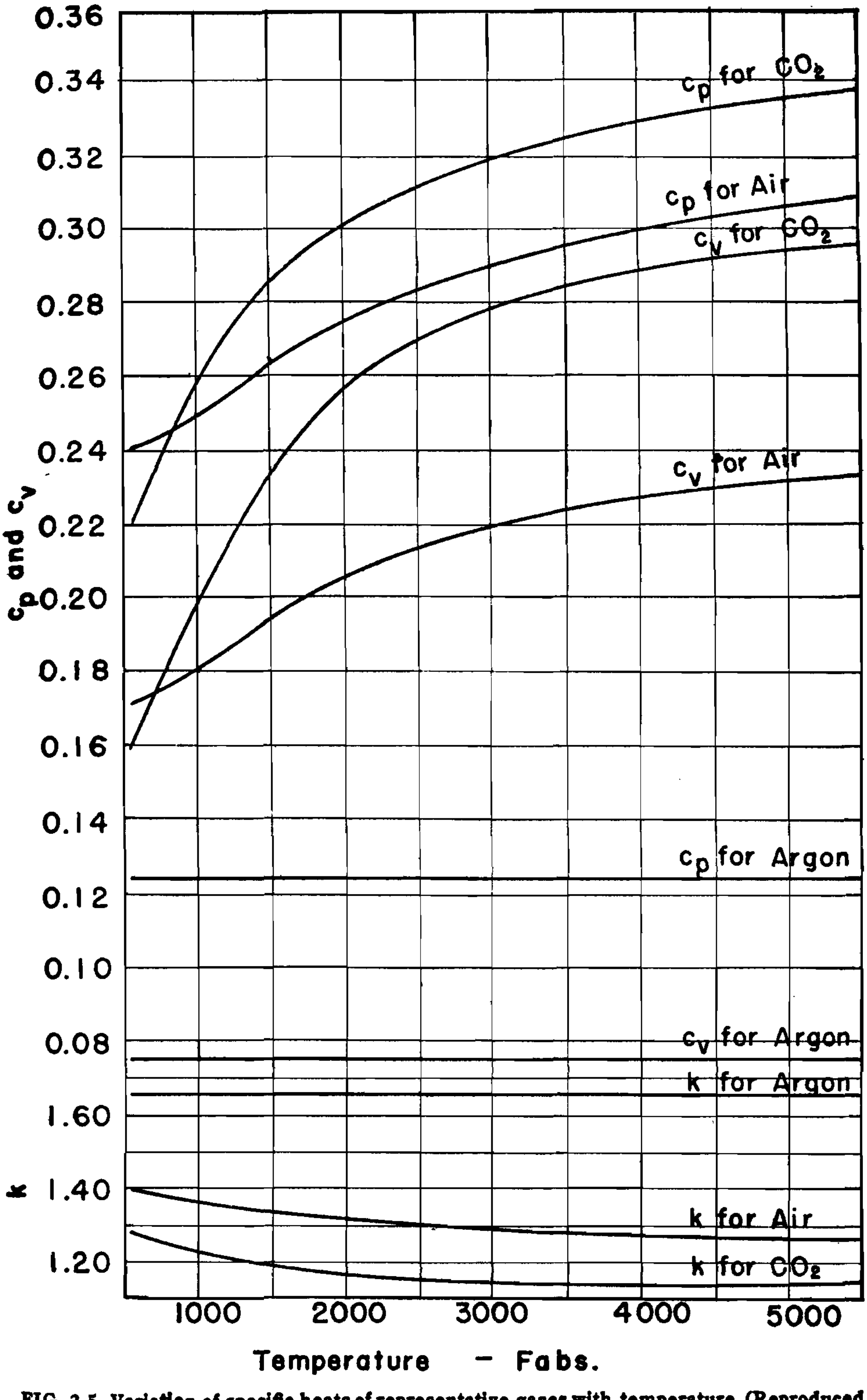


FIG. 2-5. Variation of specific heats of representative gases with temperature. (Reproduced from *Elements of Applied Thermodynamics*, **Johnston, Brockett, Bock, by special permission of U. S. Naval Institute, publisher.)**

one pound of a substance one degree. In engineering units, specific heat or heat capacity is written as

$$q_{12} = c_x(T_2 - T_1) \qquad \text{(Btu/lb)} \tag{2-8}$$

where c_x = specific heat for process x(Btu per lb-° F abs). For a **constant pressure process,** equation (2-8) becomes

$$q_{12} = c_p(T_2 - T_1) \qquad \text{(Btu/lb).} \tag{2-9}$$

For a **constant volume process,** equation (2-7) is

$$q_{12} = c_v(T_2 - T_1) \qquad \text{(Btu/lb).} \tag{2-10}$$

The specific heats at constant pressure (c_p) and at constant volume (c_v), along with the ratio of specific heats (k) defined as

$$k = \frac{c_p}{c_v} \tag{2-11}$$

are of great importance in the study of internal combustion engines. For given temperature limits, *c_p is greater than c_v*; for the heat added at constant pressure is greater than the heat added at constant volume because of the work done in the expansion or the compression of the working media. Thus, *the process lines for constant volume are steeper in slope than the process lines for constant pressure on the T-s diagram.*

The specific heats of gases, except the monatomic gases, vary widely with a change in temperature. This is shown graphically in Fig. 2-5. It should be emphasized that the specific heats for air increase with an increase in the temperature, while the ratio of specific heats decreases with an increase in temperature. Therefore, **equations (2-9), (2-10), and (2-11) can only be used with ideal gases where c_p and c_v are assumed to be constant over the temperature range.** These equations may be used in other cases *if* some method, such as tables or empirical equations, is provided to determine the mean value of the specific heats over the temperature range encountered. In this text, equations (2-9), (2-10), and (2-11) will be used only for ideal cycles (Article 3-2) in which it is *assumed* that the c_p, c_v, and k are held constant over the temperature range at the following values: $c_p = 0.24$ Btu per lb-°F abs, $c_v = 0.171$ Btu per lb-°F abs, and $k = 1.4$.

2-8. The Gas Laws. The gas laws of Boyle and Charles for a perfect gas may be combined and summarized as follows:

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} = \frac{pv}{T}\,. \tag{2-12}$$

This equation gives the relationship between the properties at two state points regardless of the process connecting the two state points. Hence, *equation (2-12) may be applied to any process.*

From Boyle and Charles' laws, the **Equation of State** for a state point of a perfect gas was developed and defined as

$$pv = RT \qquad \textbf{(2-13)}$$

or

$$pV = WRT. \qquad (2\text{-}14)$$

Here, R, specific gas constant, is a constant that is dependent on the type of gas involved. The gas constant is defined as

$$R = \frac{1544}{m} \qquad (\text{ft-lb/lb-°F abs}) \qquad (2\text{-}15)$$

where m is the molecular weight of the gas. For air, *R is equal to 53.3 ft-lb per lb° F abs and may be taken as a constant over the range of temperatures encountered in an internal combustion engine.*

2-9. Perfect Gas Relationships. For a perfect gas the internal energy is directly proportional to its temperature. Thus on p-v coordinates, lines of constant temperature are also lines of constant internal energy. Further, the change in internal energy between any two state points is the same regardless of the process involved, i.e., internal energy is a point function depending for its values only on the state of the gas.

A simple method of evaluating the change in internal energy is available in the non-flow energy equation, equation (2-5),

$$q_{12} = u_2 - u_1 + \frac{wk_{12}}{J} \qquad (\text{Btu/lb}).$$

Any process between the two state points could be chosen, but for further simplification the constant volume process will be utilized. In the constant volume process the area under the process line on the p-v diagram is zero. Therefore, the $wk_{12} = 0$ for this process and the non-flow energy equation becomes

$$q_{12} = u_2 - u_1 \qquad (\text{Btu/lb}). \qquad (2\text{-}16)$$

Equating equations (2-16) and specific heat equation for a constant volume process, equation (2-10),

$$u_2 - u_1 = c_v(T_2 - T_1) \qquad \textbf{(Btu/lb).} \qquad \textbf{(2-17)}$$

This equation provides a method of determining a change in the internal energy of a perfect gas in terms of a temperature change *regardless of the process or path between the two states* if the mean value c_v is a known quantity.

Similar to internal energy, the enthalpy of a perfect gas is directly proportional to its temperature. Thus on the T-s coordinates, the lines of constant temperature are also lines of constant enthalpy. A simple method of evaluating the change in enthalpy is available with a constant pressure non-flow process. In a constant pressure process the work is

$$\frac{wk_{12}}{J} = \frac{p_2v_2}{J} - \frac{p_1v_1}{J} \qquad \text{(Btu/lb)}.$$

Thus the non-flow energy equation becomes

$$q_{12} = (u_2 - u_1) + \left(\frac{p_2v_2}{J} - \frac{p_1v_1}{J}\right).$$

Since by definition

$$h_1 = u_1 + \frac{p_1v_1}{J} \quad \text{and} \quad h_2 = u_2 + \frac{p_2v_2}{J},$$

the above equation results in,

$$q_{12} = h_2 - h_1 \qquad \text{(Btu/lb)}. \qquad (2\text{-}18)$$

Equating equation (2-18) with the specific heat equation for a constant pressure process, equation (2-9), gives

$$h_2 - h_1 = c_p(T_2 - T_1) \qquad \text{(Btu/lb)}. \qquad (2\text{-}19)$$

This equation provides a method of determining the change in the enthalpy of a perfect gas in terms of temperature change *regardless of the process between the two states* if the mean value of c_p is known.

A useful relationship of specific heats and the gas constant R may be derived. Enthalpy is defined as

$$h = u + \frac{pv}{J}$$

wherein

$$u = c_vT$$
$$pv = RT$$
$$h = c_pT$$

Substituting

$$c_pT = c_vT + RT/J$$

Eliminating T,

$$c_p = c_v + \frac{R}{J} \qquad \textbf{(Btu/lb-°F abs).} \qquad \textbf{(2-20)}$$

2-10. Non-Flow Processes for a Perfect Gas. The most important processes in the study of *reciprocating* internal combustion engines are constant volume, constant pressure, and isentropic. A complete understanding of these processes and the method of analyzing them is a requisite for further study.

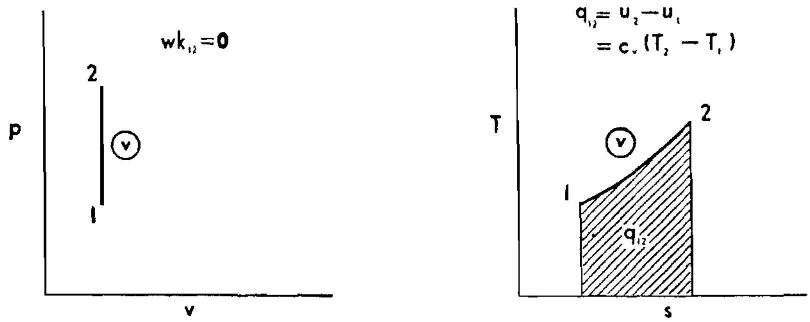


FIG. 2-6. Constant volume process.

Constant Volume Process (v = constant). The non-flow energy equation (2-5) states that

$$q_{12} = u_2 - u_1 + \frac{wk_{12}}{J} \cdot$$

Since there is no area under the constant volume process line on the p-v diagram (Fig. 2-6), the work is equal to zero and the equation may be written as

$$q_{12} = u_2 - u_1 = c_v(T_2 - T_1) \qquad \text{(Btu/lb).}$$

Thus, **in a constant volume heating process,** the **internal energy of the working medium is changed.** It is represented by the shaded area on the T-s diagram. Also, since

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} \quad \text{and} \quad v_1 = v_2,$$

$$\frac{p_1}{T_1} = \frac{p_2}{T_2} \cdot$$

Constant Pressure Process (p = constant). Starting with the non-flow energy equation for a constant pressure process

$$q_{12} = u_2 - u_1 + \frac{wk_{12}}{J}$$

and the fact that work under the process line on the p-v diagram is

$$\frac{wk_{12}}{J} = \frac{p_2v_2}{J} - \frac{p_1v_1}{J},$$

the equation becomes

$$q_{12} = \left(u_2 + \frac{p_2v_2}{J}\right) - \left(u_1 + \frac{p_1v_1}{J}\right)$$

$$q_{12} = h_2 - h_1 = c_p(T_2 - T_1) \qquad \text{(Btu/lb)}.$$

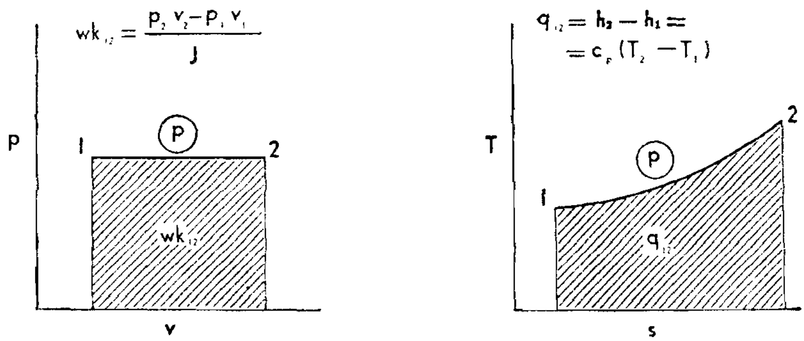


FIG. 2-7. Constant pressure process.

Thus, **in a constant pressure heating process, the enthalpy of the working medium is changed.** The change in the enthalpy is represented by the shaded area on the T-s diagram of Fig. 2-7. Also since

$$p_2 = p_1$$

and

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2},$$

$$\frac{v_1}{T_1} = \frac{v_2}{T_2}.$$

Isentropic or Reversible Adiabatic Process (pv^k = constant). An isentropic process by definition has no heat transferred across the

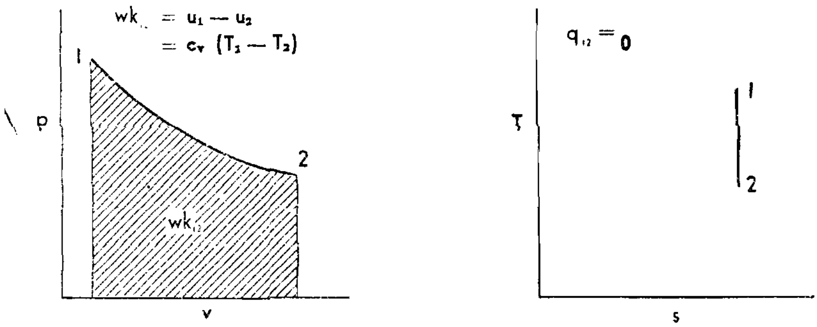


FIG. 2-8. Isentropic (reversible adiabatic) process.

boundaries of the system, hence $ds=0$; as a result, $q=0$ for an isentropic process. The non-flow energy equation becomes

$$\frac{wk_{12}}{J} = u_1 - u_2 = c_v(T_1 - T_2) \qquad \text{(Btu/lb)}$$

as indicated by the shaded area in the p-v diagram[3] of Fig. 2-8.

Other useful relationships may be derived using the process equation $pv^k = \text{constant}$ and the above equation.

$$p_1 v_1^k = p_2 v_2^k$$

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^k \quad \text{or} \quad \left(\frac{p_2}{p_1}\right)^{1/k} = \frac{v_1}{v_2}$$

$$\frac{T_2}{T_1} = \frac{p_2 v_2}{p_1 v_1} = \frac{p_2}{p_1}\left(\frac{v_1}{v_2}\right)^{-1}$$

Substituting for p_2/p_1

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^k \left(\frac{v_1}{v_2}\right)^{-1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

Substituting for v_1/v_2

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)\left[\left(\frac{p_2}{p_1}\right)^{1/k}\right]^{-1} = \left(\frac{p_2}{p_1}\right)^{1-(1/k)} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k}$$

Thus

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k} = \left(\frac{v_1}{v_2}\right)^{k-1} \tag{2-21}$$

[3] In the above expressions for work we have adhered to the convention stated in Article 2-5.

2-11. Summary. For the analysis of the non-flow processes involved in the cylinder of a reciprocating engine, the following basic thermodynamic equations apply for perfect gases:

(1) non-flow energy equation

$$q_{12} = u_2 - u_1 + \frac{wk_{12}}{J}$$

(2) $u = c_v T$

(3) $h = c_p T = u + \dfrac{pv}{J}$

(4) $pv = RT$

(5) $\dfrac{p_1 v_1}{T_1} = \dfrac{p_2 v_2}{T_2} = \dfrac{pv}{T}$.

These equations are valid regardless of the process or path between any two state points.

Table 2-1 is a summary of the equations involved with *perfect gas non-flow processes.*

TABLE 2-1

SUMMARY FOR NON-FLOW PERFECT GAS PROCESSES

Process	Work (Btu/lb)	Heat (Btu/lb)	**Remarks** (1) For a perfect gas $pv = RT$ (2) Non-Flow General Energy Equation $q = u_2 - u_1 + \frac{wk_{12}}{J}$
Constant volume	0	$u_2 - u_1 = c_v(T_2 - T_1)$	$v_1 = v_2$ $dv = 0$ $\frac{p_1}{T_1} = \frac{p_2}{T_2}$
Constant Pressure	$\frac{p_2 v_2 - p_1 v_1}{J} = \frac{R}{J}(T_2 - T_1)$	$h_2 - h_1 = c_p(T_2 - T_1)$	$p_1 = p_2$ $\frac{v_1}{T_1} = \frac{v_2}{T_2}$
Isentropic	$u_1 - u_2 = c_v(T_1 - T_2)$	0	$p_1 v_1^k = p_2 v_2^k$ $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k} = \left(\frac{v_1}{v_2}\right)^{k-1}$

The p-v and T-s diagrams are important aids in the analysis of such processes and should always be used.

The following factors will be useful in solving problems involving power cycles and performance characteristics:

R = 53.3 ft-lb per lb-° F abs for air

1 hp = 33,000 ft-lb per min. = 550 ft-lb per sec.

1 Btu = 778 ft-lb

1 hp = 2545 Btu per hour = 42.42 Btu per min.

1 kw = 1.341 hp

1 kw = 3413 Btu per hour

g = 32.2 ft per sec^2

1 gal = 231 in^3

1 ft^3 = 7.48 gal

1 atm. = 14.7 lb/in^2 = 29.92 in. Hg.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

2-1. E. F. Obert, *Thermodynamics*, McGraw-Hill Book Co., Inc., New York, 1948.

2-2. R. M. Johnston, W. A. Brockett, A. E. Bock, *Elements of Applied Thermodynamics*, U. S. Naval Institute, Annapolis, Md., 1951.

2-3. P. J. Kiefer, M. C. Stuart, *Principles of Engineering Thermodynamics* John Wiley and Sons, Inc., New York, 1930.

2-4. J. H. Keenan, *Thermodynamics*, John Wiley and Sons, Inc., New York, 1941.

2-5. G. A. Hawkins, *Thermodynamics*, John Wiley and Sons, Inc., New York, 1946.

2-6. E. F. Obert, *Internal Combustion Engines*, International Textbook Co., Scranton, 1950.

2-7. L. C. Lichty, *Internal Combustion Engines*, McGraw-Hill Book Co., Inc., New York, 1951.

EXERCISES

2-1. A horsepower is defined as work done at the rate of 33000 ft-lb per minute. Express the horsepower in

(a) Btu per hr; Btu per min.
(b) Mile pounds per hr.
(c) Ft-lbs per second.

Ans:

(a) 2545 Btu/hr, 42.4 Btu/min, (b) 375 mile-lb/hr, (c) 550 ft-lb/sec

2-2. One half pound of air is heated during a non-flow constant volume process until the pressure is doubled. The initial pressure and temperature were $p = 25$ psia and $t = 120$ °F.

Find: (a) Initial volume, ft^3
(b) Final temperature, °F abs.
(c) Change in internal energy, Btu/lb.

Ans:
(a) 4.29 ft^3, (b) 1160° F abs., (c) 99.3 Btu/lb

2-3. One pound of air is initially at 14.7 psia and 60° F. Thirty Btu are supplied in a non-flow constant volume process.
(a) Sketch the process on *p-v* and *T-s* coordinates.

Find: (b) Work done, ft-lb per lb
(c) Change in internal energy Btu per lb
(d) Final temperature, °F abs.

Ans:
(b) 0, (c) 30 Btu/lb, (d) 695° F abs.

2-4. Heat is supplied in a non-flow constant pressure process to air initially at 20 psia and 70° F until the volume is doubled.
(a) Sketch the process on *p-v* and *T-s* coordinates.

Find: (b) Work done, ft-lb per lb
(c) Change in enthalpy, Btu per lb
(d) Change in internal energy, Btu per lb.
(e) Heat supplied, Btu per lb.

Ans:
(b) 28200 ft-lb/lb, (c) 127 Btu/lb, (d) 90.7 Btu/lb, (e) 127 Btu/lb.

2-5. Two pounds of air initially at 14.7 psia and 60° F are compressed isentropically in a non-flow process to 60 psia.
(a) Sketch the process on *p-v* and *T-s* coordinates.

Find: (b) The final volume, ft^3.
(c) The final temperature, °F abs.
(d) Change in internal energy, Btu per lb
(e) Heat added, Btu per lb
(f) Work done on the air, Btu per lb
(g) Change in enthalpy, Btu per lb

Ans:
(b) 9.58 ft^3, (c) 778° F abs., (d) 44,2 Btu/lb, (e) 0, (f) 44.2 Btu/lb, (g) 62 Btu/lb

2-6. An air compressor operating with steady flow delivers 30 pounds of air per second at 60 psia and 720° F abs. The air is initially at 14.7 psia and 60° F and 41200 ft-lb per lb work is done on air. The change in velocity across the compressor is negligible.

Find: (a) Change of enthalpy, Btu per lb
(b) Heat supplied or rejected, Btu per min
(c) Power delivered to the air, hp.

Ans:
(a) 48 Btu/lb, (b) 9000 Btu/min rejected, (c) 2250 hp

CHAPTER III

POWER CYCLES

Since the complexity and the cost of engine test work is high, engine cycle analysis has become an important tool for every engineer. Even though the internal combustion engine does not operate on a thermodynamic cycle,[1] still, the concept of theoretical cycles may be useful to show the effect of changing operating conditions on the performance characteristics, to indicate ultimate performance, and to evaluate one engine relative to another. The theoretical cycle treatment taken up in this chapter, then, is not strictly of academic importance, but has prominent practical applications.

3-1. Methods of Cycle Analysis. In an internal combustion engine, the heat or energy is obtained from the combustion of a hydrocarbon *fuel in air. As the working media passes through the engine and com*bustion takes place, complicated chemical, thermal, and physical changes occur. Friction occurs within the working medium itself, between the working medium and the engine, and between the working parts of the engine. To examine all these changes quantitatively and to account for all the variables creates a very complex problem; therefore, the usual method of approach is through the use of certain theoretical approximations, each of which is based on different simplifying assumptions but with increasing accuracy. The three commonly employed approximations of an actual engine in order of their increasing accuracy to the conditions of an actual engine are termed the (1) **ideal cycle,** (2) **air cycle,** and (3) **fuel-air cycle.** A cycle based on data as obtained on the actual operating engine would constitute an **actual cycle.** The differences among the four cycles are described below.

Ideal Cycle. In the ideal cycle analysis, the working medium is assumed to be a *perfect gas.* The values of the **specific heats are assumed to be constant** and to be those of air at standard conditions, i.e., $c_v = 0.171$ Btu per lb-° F abs; $c_p = 0.24$ Btu per lb-° F abs; $k = 1.4$. It is also assumed that *the working medium, a perfect gas, does not change throughout the cycle and that heat is supplied or rejected ideally and may, if necessary, be supplied or rejected instantaneously.* The ideal cycle represents the upper limit of the performance which an engine may theoretically attain. This cycle allows a simple mathematical analysis based on the perfect gas laws. Peak temperatures and pressures and thermal

[1] *A thermodynamic cycle may be defined as a series of processes through which* the working medium progresses which will eventually return the medium to its original state.

efficiencies based on an ideal cycle analysis are higher than those found in an actual engine and in the other theoretical cycles. No heat loss is assumed for this cycle.

Air Cycle. The air cycle analysis is one step closer to the conditions existing in an actual engine. In the air cycle analysis, the working medium is assumed to be air and **the values of the specific heats are variable** over the temperature range encountered (Fig. 2-5). Heat is supplied and rejected ideally and may, if necessary, be supplied or rejected instantaneously. There is no heat loss in this cycle.

Calculation of a mean specific heat for each temperature range and process is tedious and complicated. The *Thermodynamic Properties of Air*, by Keenan and Kaye (ref. 3–4), which account for variation in the specific heats of air, are used to obtain values of the properties of air at state points and during isentropic processes. Heat and work are expressed directly in terms of internal energy and enthalpy.[2]

The allowance for variable specific heats reduces the peak temperatures and pressures below those calculated in the ideal cycle. The net work and thermal efficiency are also reduced. However, the peak temperatures and pressures and the thermal efficiencies calculated by an air cycle analysis are higher than those found in an actual engine.

Fuel-Air Cycle. The fuel-air cycle represents the closest approximation to the actual engine commonly performed by mathematical analysis. During the compression process the working medium in the spark ignition engine is assumed to be a mixture of fuel, air, and residual gas, and a mixture of air and residual gas in the compression ignition engine. After combustion occurs, the working medium contains the products of combustion such as CO_2, CO, H_2O, N_2, which results in a greater variation in specific heats (Fig. 2-5). A further variation in the specific heats is caused by the dissociation of some of the lighter molecules that occur at high temperatures. Dissociation reactions are endothermic and absorb part of the heat released by combustion. The increase in specific heats and the endothermic dissociation reactions reduce the peak temperatures and pressures and the thermal efficiency below those calculated by the air cycle analysis. In order to make a fuel-air analysis it is necessary to use experimentally determined thermodynamic data for the combustion products. These are usually presented in the form of charts. One form of such charts, commonly called "Hottel Charts" after one of the authors, may be found in references 3-1 and 3-2. These charts take into consideration the effects of the products of combustion and dissociation.

[2] See article 3-2.

It is assumed in the fuel-air cycle that heat may be supplied or rejected instantaneously and that no heat is lost across the boundaries of the system.

Actual Cycle. The actual cycle is obtained by experimental means using any one of several types of engine indicators to obtain a diagram of the pressures and volumes existing within the cylinder of an operating engine. These indicators produce the so-called indicator card diagrams. A detailed discussion of these indicators may be found in any standard work on engine testing. They are discussed in Article 10-4.

Indicator card diagrams may be taken which account for all previous variations and assumptions and, in addition, bring in the effect of heat losses, finite combustion rates, internal fluid frictional losses, valve timing, spark or injection timing, and exhaust blowdown losses. The pumping losses may be determined from a low pressure indicator card. The indicated net work of the actual engine may be computed from these cards. (See pp. 13–25.)

In this text, the analysis of cycles will be limited to a consideration of the ideal, air, and actual cycles. The losses and variations occurring in the actual cycle will be considered individually in Chapter X.

3-2. Tables of Thermodynamic Properties of Air. The air cycle, which has a working medium of air with variable specific heats, gives a closer approximation to the actual engine than the ideal cycle, which has a perfect gas with constant specific heats as its working medium. In some types of engines, such as the gas turbine and jet propulsion engines, where the air-fuel ratio is relatively high,[3] the air cycle closely approximates the conditions within the actual engine. This is also applicable for diesel engines at light load. To calculate the variable specific heats required for an air cycle analysis by empirical equations is complex and unwieldly. However, the time and effort involved in the calculations of an air cycle are greatly reduced by the use of the *Air Tables* prepared by Keenan and Kaye (ref. 3-4), which take into account the variable specific heats of air. An abstraction from the Air Tables is given in Table 3-1.

The Equation of State for gases over the range in temperatures encountered in normal internal combustion engines may be accurately represented by

$$pv = RT.$$

[3] Gas turbine engines normally operate with air-fuel ratios over 100 to 1, which means for every 100 lbs of air only one lb of fuel is burned. Thus, the working medium differs little from air.

TABLE 3-1

THERMODYNAMIC PROPERTIES OF AIR*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
500	23.97	2.182	−10.30	5727
510	26.37	2.339	− 8.58	5451
520	28.77	2.504	− 6.87	5192
530	31.17	2.676	− 5.15	4951
540	33.57	2.857	− 3.44	4725
550	35.98	3.047	− 1.72	4512
560	38.38	3.246	− 0.01	4313
570	40.78	3.454	1.71	4126
580	43.19	3.671	3.43	3950
590	45.59	3.898	5.15	3784
600	48.00	4.135	6.87	3628
610	50.41	4.382	8.59	3481
620	52.81	4.639	10.32	3341
630	55.22	4.907	12.04	3209
640	57.63	5.187	13.76	3085
650	60.04	5.478	15.49	2967
660	62.46	5.780	17.22	2855
670	64.87	6.094	18.95	2749
680	67.28	6.421	20.67	2648
690	69.70	6.760	22.40	2552
700	72.11	7.112	24.13	2461
710	74.53	7.477	25.87	2374
720	76.95	7.855	27.60	2291
730	79.38	8.248	29.34	2213
740	81.80	8.654	31.08	2138
750	84.23	9.075	32.82	2066
760	86.65	9.511	34.56	1997.7
770	89.08	9.962	36.30	1932.3
780	91.51	10.428	38.05	1869.9
790	93.95	10.911	39.80	1810.1
800	96.38	11.410	41.55	1752.9
810	98.82	11.925	43.30	1698.2
820	101.26	12.457	45.05	1645.7
830	103.70	13.006	46.81	1595.4
840	106.14	13.574	48.57	1547.1
850	108.59	14.160	50.33	1500.7
860	111.04	14.764	52.09	1456.3
870	113.49	15.387	53.86	1413.6
880	115.94	16.029	55.62	1372.5
890	118.40	16.692	57.40	1333.0

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
900	120.86	17.374	59.17	1295.0
910	123.32	18.077	60.94	1258.5
920	125.78	18.802	62.72	1223.3
930	128.25	19.548	64.50	1189.4
940	130.72	20.316	66.29	1156.7
950	133.19	21.106	68.07	1125.3
960	135.67	21.919	69.87	1094.9
970	138.14	22.756	71.66	1065.7
980	140.63	23.617	73.45	1037.4
990	143.11	24.502	75.25	1010.1
1000	145.59	25.41	77.05	938.8
1010	148.09	26.35	78.86	958.3
1020	150.58	27.31	80.66	933.7
1030	153.07	28.30	82.47	910.0
1040	155.57	29.31	84.29	887.0
1050	158.07	30.35	86.10	864.8
1060	160.58	31.42	87.92	843.4
1070	163.09	32.52	89.74	822.6
1080	165.60	33.64	91.57	802.5
1090	168.11	34.80	93.40	783.0
1100	170.62	35.99	95.23	764.1
1110	173.15	37.21	97.06	745.8
1120	175.67	38.45	98.90	728.1
1130	178.20	39.74	100.74	710.9
1140	180.73	41.05	102.59	694.3
1150	183.26	42.40	104.43	678.1
1160	185.79	43.78	106.28	662.5
1170	188.33	45.19	108.14	647.3
1180	190.87	46.64	109.99	632.5
1190	193.42	48.12	111.85	618.2
1200	195.96	49.64	113.71	604.3
1210	198.52	51.20	115.58	590.8
1220	201.07	52.79	117.45	577.7
1230	203.63	54.43	119.32	565.0
1240	206.19	56.10	121.20	552.6
1250	208.75	57.81	123.07	540.6
1260	211.32	59.56	124.96	528.9
1270	213.89	61.35	126.34	517.5
1280	216.47	63.19	128.73	506.4
1290	219.04	65.07	130.62	495.7

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
1700	327.36	188.29	210.84	225.7
1710	330.06	192.69	212.85	221.9
1720	332.77	197.18	214.87	218.1
1730	335.47	201.74	216.89	214.4
1740	338.19	206.40	218.92	210.8
1750	340.90	211.13	220.95	207.2
1760	343.61	215.95	222.97	203.7
1770	346.33	220.86	225.01	200.4
1780	349.05	225.86	227.04	197.0
1790	351.78	230.94	229.09	193.8
1800	354.50	236.1	231.12	190.59
1810	357.23	241.4	233.17	187.45
1820	359.96	246.7	235.21	184.40
1830	362.70	252.2	237.26	181.41
1840	365.43	257.7	239.31	178.47
1850	368.17	263.4	241.36	175.58
1860	370.92	269.1	243.43	172.78
1870	373.66	275.0	245.48	170.01
1880	376.41	280.9	247.55	167.32
1890	379.16	286.9	249.61	164.67
1900	381.90	293.1	251.68	162.07
1910	384.66	299.3	253.74	159.51
1920	387.42	305.7	255.81	157.03
1930	390.18	312.1	257.89	154.58
1940	392.94	318.7	259.96	152.18
1950	395.70	325.4	262.04	149.82
1960	398.47	332.2	264.12	147.52
1970	401.23	339.1	266.20	145.24
1980	404.00	346.1	268.28	143.03
1990	406.78	353.2	270.38	140.85
2000	409.55	360.4	272.46	138.72
2020	415.11	375.3	276.65	134.57
2040	420.67	390.6	280.84	130.58
2060	426.25	406.4	285.05	126.72
2080	431.83	422.7	289.26	123.01
2100	437.42	439.5	293.48	119.45
2120	443.02	456.9	297.71	116.01
2140	448.62	474.7	301.94	112.69
2160	454.23	493.2	306.18	109.50
2180	459.86	512.2	310.43	106.41

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
1300	221.62	66.98	132.52	485.2
1310	224.21	68.95	134.42	475.0
1320	226.79	70.95	136.32	465.1
1330	229.38	73.01	138.22	455.4
1340	231.98	75.10	140.13	446.0
1350	234.57	77.25	142.04	436.9
1360	237.17	79.44	143.95	428.0
1370	239.78	81.68	145.87	419.3
1380	242.38	83.97	147.79	410.8
1390	244.99	86.32	149.71	402.6
1400	247.60	88.71	151.64	394.6
1410	250.22	91.15	153.57	386.7
1420	252.84	93.64	155.51	379.1
1430	255.46	96.19	157.44	371.7
1440	258.08	98.79	159.38	364.4
1450	260.71	101.45	161.32	357.3
1460	263.34	104.16	163.27	350.4
1470	265.98	106.93	165.22	343.7
1480	268.61	109.75	167.17	337.1
1490	271.25	112.64	169.12	330.7
1500	273.89	115.58	171.08	324.5
1510	276.54	118.58	173.04	318.3
1520	279.19	121.64	175.00	312.4
1530	281.84	124.77	176.97	306.6
1540	284.50	127.96	178.94	300.9
1550	287.15	131.21	180.91	295.3
1560	289.81	134.52	182.88	289.9
1570	292.47	137.91	184.86	284.6
1580	295.14	141.36	186.84	279.4
1590	297.81	144.87	188.83	274.4
1600	300.48	148 46	190.81	269.4
1610	303.16	152.11	192.80	264.6
1620	305.84	155.83	194.80	259.9
1630	308.52	159.63	196.79	255.3
1640	311.20	163.50	198.79	250.8
1650	313.88	167.44	200.79	246.4
1660	316.58	171.45	202.79	242.0
1670	319.27	175.55	204.80	237.8
1680	321.96	179.72	206.81	233.7
1690	324.66	183.96	208.82	229.7

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
3000	695.77	1953	490.14	38.39
3020	701.64	2010	494.64	37.56
3040	707.51	2068	499.14	36.75
3060	713.38	2126	503.64	35.98
3080	719.26	2186	508.14	35.22
3100	725.14	2248	512.66	34.47
3120	731.03	2311	517.18	33.75
3140	736.92	2376	521.70	33.05
3160	742.82	2441	526.22	32.36
3180	748.72	2509	530.75	31.69
3200	754.62	2577	535.27	31.04
3220	760.52	2647	539.81	30.41
3240	766.43	2719	544.35	29.79
3260	772.35	2792	548.90	29.19
3280	778.27	2867	553.45	28.60
3300	784.19	2943	557.99	28.03
3320	790.12	3021	562.55	27.48
3340	796.04	3100	567.11	26.94
3360	801.98	3182	571.67	26.40
3380	807.92	3264	576.24	25.89
3400	813.85	3349	580.80	25.38
3420	819.80	3435	585.38	24.89
3440	825.75	3523	589.96	24.41
3460	831.70	3613	594.54	23.94
3480	837.66	3705	599.12	23.48
3500	843.61	3798	603.71	23.04
3520	849.58	3893	608.31	22.60
3540	855.54	3991	612.89	22.18
3560	861.51	4090	617.49	21.76
3580	867.48	4191	622.09	21.36
3600	873.45	4293	626.70	20.96
3620	879.43	4399	631.30	20.57
3640	885.41	4506	635.91	20.20
3660	891.40	4615	640.53	19.827
3680	897.39	4726	645.15	19.467
3700	903.38	4839	649.76	19.117
3720	909.37	4954	654.39	18.772
3740	915.37	5072	659.02	18.434
3760	921.37	5192	663.64	18.104
3780	927.37	5314	668.28	17.782

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
2200	465.48	531.7	314.68	103.43
2220	471.12	551.9	318.95	100.56
2240	476.76	572.6	323.22	97.79
2260	482.41	594.0	327.50	95.11
2280	488.07	616.0	331.79	92.53
2300	493.73	638.7	336.08	90.03
2320	499.41	662.0	340.39	87.62
2340	505.08	685.9	344.69	85.29
2360	510.77	710.5	349.01	83.03
2380	516.46	735.9	353.32	80.85
2400	522.16	761.9	357.65	78.75
2420	527.86	788.7	361.98	76.71
2440	533.57	816.2	366.32	74.74
2460	539.29	844.5	370.67	72.83
2480	545.01	873.5	375.02	70.98
2500	550.74	903.3	379.38	69.19
2520	556.48	934.0	383.75	67.46
2540	562.22	965.4	388.12	65.78
2560	567.97	997.7	392.50	64.15
2580	573.72	1030.8	396.88	62.57
2600	579.48	1064.8	401.26	61.05
2620	585.24	1099.6	405.66	59.56
2640	591.01	1135.4	410.06	58.13
2660	596.79	1172.1	414.46	56.74
2680	602.57	1209.7	418.87	55.39
2700	608.36	1248.3	423.29	54.07
2720	614.15	1287.8	427.71	52.80
2740	619.95	1328.3	432.14	51.57
2760	625.75	1369.9	436.57	50.37
2780	631.56	1412.4	441.01	49.21
2800	637.37	1456.0	445.45	48.08
2820	643.19	1500.7	449.90	46.98
2840	649.01	1546.4	454.35	45.91
2860	654.84	1593.2	458.80	44.88
2880	660.67	1641.2	463.26	43.87
2900	666.51	1690.3	467.73	42.89
2920	672.36	1740.5	472.21	41.94
2940	678.20	1791.9	476.68	41.02
2960	684.05	1844.5	481.16	40.12
2980	689.91	1898.4	485.65	39.24

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
4600	1175.86	12655	860.56	9.087
4620	1181.97	12902	865.30	8.952
4640	1188.09	13153	870.04	8.819
4660	1194.20	13408	874.78	8.689
4680	1200.32	13667	879.53	8.561
4700	1206.44	13929	884.28	8.436
4720	1212.56	14196	889.03	8.312
4740	1218.69	14467	893.79	8.191
4760	1224.81	14741	898.54	8.073
4780	1230.94	15020	903.30	7.956
4800	1237.07	15303	908.06	7.841
4820	1243.20	15591	912.82	7.729
4840	1249.33	15882	917.58	7.618
4860	1255.47	16178	922.35	7.510
4880	1261.61	16478	927.12	7.404
4900	1267.75	16783	931.88	7.299
4920	1273.89	17092	936.65	7.196
4940	1280.03	17405	941.42	7.096
4960	1286.18	17724	946.20	6.997
4980	1292.33	18046	950.98	6.899
5000	1298.48	18373	955.75	6.803
5100	1329.25	20080	979.68	6.349
5200	1360.07	21920	1003.65	5.932
5300	1390.94	23880	1027.66	5.549
5400	1421.85	25980	1051.72	5.197
5500	1452.80	28220	1075.81	4.872
5600	1483.80	30620	1099.95	4.572
5700	1514.83	33170	1124.13	4.296
5800	1545.91	35890	1148.35	4.040
5900	1577.02	38790	1172.61	3.803
6000	1608.16	41870	1196.90	3.583
6100	1639.35	45140	1221.23	3.379
6200	1670.57	48610	1245.60	3.189
6300	1701.83	52290	1270.00	3.012
6400	1733.12	56180	1294.44	2.848
6500	1764.45	60310	1318.91	2.694

* Reproduced by permission from *Thermodynamic Properties of Air*, by J. H Keenan and J. Kaye, published by John Wiley and Sons, Inc., 1945.

TABLE 3-1—*Continued*

T °F abs	h Btu/lb	p_r	u Btu/lb	v_r
3800	933.38	5438	672.91	17.468
3820	939.39	5565	677.55	17.161
3840	945.40	5694	682.19	16.861
3860	951.42	5825	686.84	16.566
3880	957.44	5959	691.49	16.278
3900	963.46	6095	694.14	15.998
3920	969.48	6234	700.79	15.723
3940	975.51	6375	705.45	15.453
3960	981.54	6518	710.11	15.189
3980	987.57	6664	714.77	14.931
4000	993.61	6813	719.43	14.678
4020	999.65	6964	724.10	14.431
4040	1005.69	7118	728.77	14.189
4060	1011.73	7275	733.44	13.952
4080	1017.78	7434	738.12	13.721
4100	1023.82	7596	742.80	13.494
4120	1029.88	7761	747.48	13.271
4140	1035.93	7929	752.16	13.053
4160	1041.99	8100	756.85	12.840
4180	1048.05	8273	761.54	12.631
4200	1054.11	8450	766.23	12.427
4220	1060.18	8629	770.92	12.226
4240	1066.24	8812	775.61	12.029
4260	1072.31	8997	780.31	11.837
4280	1078.39	9186	785.02	11.648
4300	1084.46	9377	789.72	11.464
4320	1090.54	9572	794.43	11.283
4340	1096.62	9771	799.14	11.104
4360	1102.70	9972	803.85	10.931
4380	1108.79	10177	808.57	10.760
4400	1114.87	10384	813.28	10.593
4420	1120.96	10596	818.00	10.428
4440	1127.05	10810	822.72	10.268
4460	1133.15	11029	827.44	10.110
4480	1139.24	11250	832.16	9.955
4500	1145.34	11475	836.89	9.804
4520	1151.44	11704	841.62	9.655
4540	1157.54	11936	846.35	9.509
4560	1163.65	12171	851.09	9.366
4580	1169.75	12412	855.82	9.225

This equation leads to a great simplification of the presentation of the properties of the working medium, i.e., air, involved in engine cycle analysis as compared to the tables and charts required for an adequate statement of the properties of a vapor in the cycle of a steam plant. It has been proven[4] from the Equation of State that *the internal energy* (u) *and the enthalpy* (h) *are functions of temperature only.* This statement was the basis used in the preparation of the Air Tables in which internal energy and enthalpy are listed as a function of temperature at state points along an **isentropic process.**

Keenan and Kaye[5] also proved that the ratio of pressures corresponding to given pair of temperatures along a given isentropic process line is equal to the ratio of the relative pressures for the same pair of temperatures. This applies equally as well to ratio of volumes and their relative volumes. This relationship for an **isentropic process** may be shown by

$$\frac{p_1}{p_2} = \frac{p_{r1}}{p_{r2}} \tag{3-1}$$

and

$$\frac{v_1}{v_2} = \frac{v_{r1}}{v_{r2}} \tag{3-2}$$

where p and v are absolute pressures and specific volumes respectively, p_r *and* v_r *are relative pressures and volumes,* and subscripts 1 and 2 refer to any two state points along a given isentropic. The relative pressures (p_r) and volumes (v_r) are functions of temperature only.

The relative pressures and relative volumes are merely the ratios of the value of the property at that particular state point to the value of the property at the arbitrarily chosen base for which the tables were made. Thus

$$p_{r1} = \frac{p_1}{p_0}$$

and

$$v_{r1} = \frac{v_1}{v_0}$$

where the subscript 0 denotes the arbitrarily chosen base condition. The base conditions for the Air Tables are

$$T_0 = 400° \text{ F abs}, \qquad h_0 = 0 \text{ Btu/lb}$$

[4] Ref. 3-5.
[5] Ref. 3-6.

for which

$$p_{r0} = 1.0, \quad v_{r0} = 10^4, \quad \text{and} \quad u_0 = -27.42 \text{ Btu/lb}.$$

The Air Tables list values of enthalpy (h), relative pressure (p_r), internal energy (u), and relative volume (v_r) as functions of the temperature (T) for any state points along an isentropic. *Since the process has been specified as one of constant entropy, i.e., an isentropic process, only one other property need be specified in order to fix the remaining thermodynamic properties. Therefore, if any one of the listed properties is known for any state, the values of the other properties at that state may be found by entering the Air Tables with the one known property.* This is shown in the example problem on the use of the Air Tables that is given in Article 3-8.

3-3. Thermal Efficiency. Since it is assumed that the internal combustion engine operates on a theoretical cycle, the working medium must return to its initial state after passing through a series of thermodynamic processes. When the working medium returns to its initial state, the change in the potential, kinetic, flow, and internal energies for the cycle are zero. If there is no change in these energies, the General Energy Equation, equation (2-2), becomes

$$q \text{ (during cycle)} = \frac{wk}{J} \text{ (during cycle)}.$$

This is the basis for the Second Law of Thermodynamics which may be defined for a heat engine as *no engine, actual or ideal, operating in a cycle can convert all the heat supplied into mechanical work.* Therefore, the heat supplied (q_S) to the cycle minus the heat rejected (q_R) from the cycle, then, must equal the net work of the cycle

$$q_S - q_R = \text{net work.} \tag{3-3}$$

The thermal efficiency of any thermodynamic cycle is then defined as net work of the cycle divided by the heat supplied, or

$$\eta_t = \frac{q_S - q_R}{q_S} = \frac{\text{net work}}{q_S}. \tag{3-4}$$

The heat supplied, the heat rejected, and the net work may be represented graphically on a T-s diagram. The net work may also be represented graphically on a p-v diagram. An ideal theoretical spark ignition engine cycle (Otto cycle) will be used to illustrate the graphical representation of these factors. During the compression stroke, the piston,

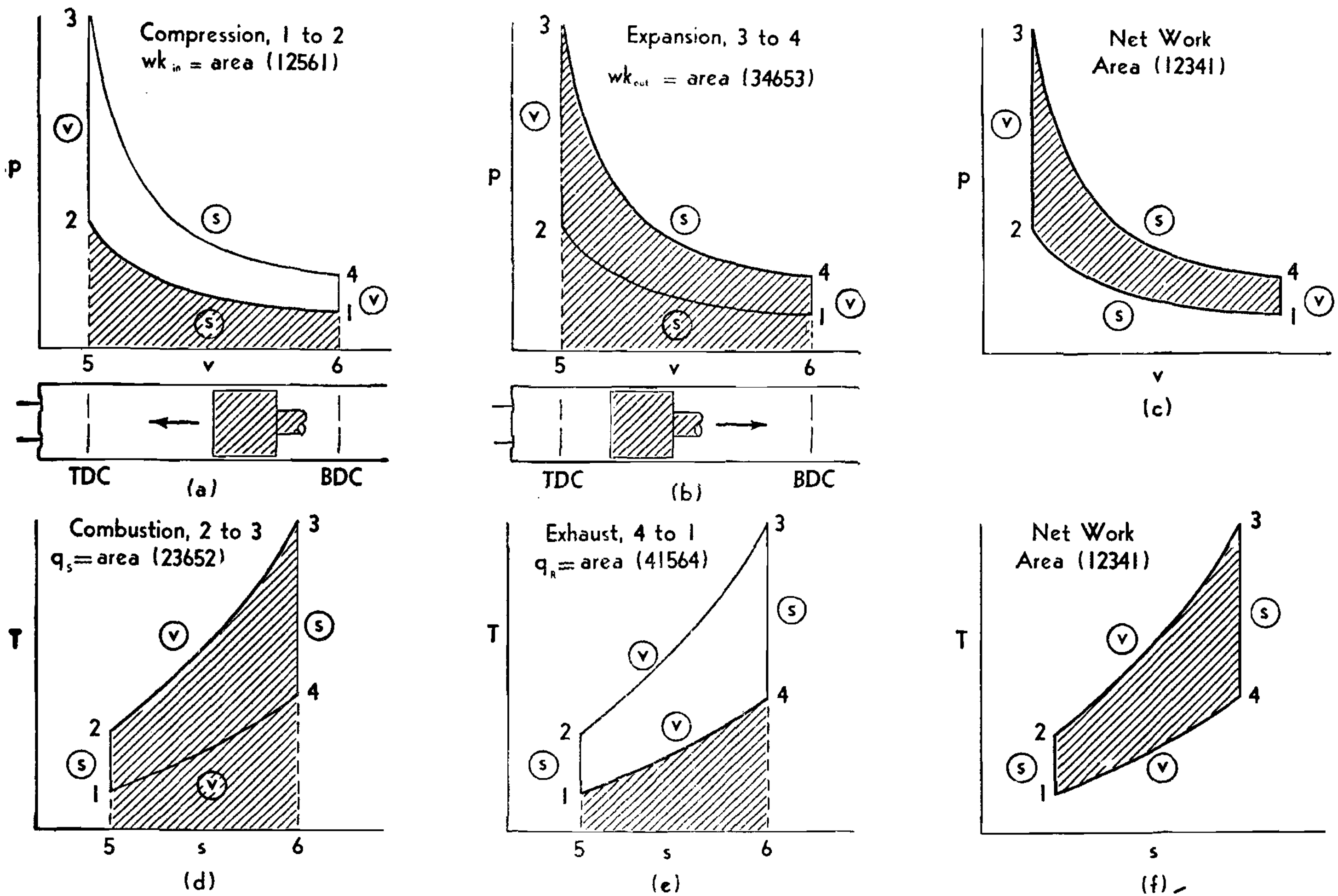


Fig. 3-1. Area representation of work, heat, and net work.

as shown in Fig. 3-1(a), is moving from right to left and is doing work on the air by compressing it. This is termed "work in" (wk_{in}) and it is represented by the shaded area under the isentropic compression process line, states 1 to 2, in Fig. 3-1(a). Heat is now supplied instantaneously to the cycle at constant volume. The heat supplied is represented by the shaded area under the constant volume process line, states 2-3, in Fig. 3-1(d). The heated gases of combustion then expand driving the piston from left to right (power stroke), as shown in Fig. 3-1(b), and producing the "work out" (wk_{out}) of the cycle. Work is being done by the working medium on the piston and the quantity of work is represented by the shaded area under the isentropic expansion process line, states 3 to 4, in Fig. 3-1(b). The heat is rejected instantaneously from the cycle at constant volume, states 4 to 1. The quantity of heat rejected is represented by the shaded area in Fig. 3-1(e).

The net work of the cycle is the shaded area in the T-s diagram, Fig. 3-1(f). The net work is the difference between the heat supplied and the heat rejected, shaded areas in Figs. 3-1(d) and (e). The net work may also be represented on the p-v diagram as shown by the shaded area in Fig. 3-1(c). This area is the difference between the "work out" and the "work in" of the cycle, shaded areas Figs. 3-1(a) and (b). Therefore, net work may be stated as

$$\text{net work} = wk_{out} - wk_{in} = q_S - q_R. \tag{3-5}$$

The calculations for the net work and thermal efficiency of various cycles are simplified by the use of q_S and q_R as compared to the use of wk_{out} and wk_{in}. Therefore, in the cycles to follow only the q_S and q_R method of cycle analysis will be used.

3-4. Otto Cycle. The Otto cycle is the theoretical cycle for the spark-ignition engine. The cycle is graphically shown in Fig. 3-2. The following thermodynamic processes take place during the cycle:

Process 1-2. **Isentropic.** Compression or work done on the working medium by the piston (wk_{in}).

Process 2-3. **Constant Volume.** Heat supplied instantaneously (q_s).

Process 3-4. **Isentropic.** Expansion or work done by the working medium on the piston (wk_{out}).

Process 4-1. **Constant Volume.** Heat rejected instantaneously (q_R).

It should be noted that the exhaust stroke, states 1 to a, and the intake stroke, states a to 1, of a four-stroke cycle SI engine, follow the same path and appear as a horizontal line on the p-v diagram. The effects of the two processes cancel each other so that there is no loss or gain of either heat or work. These two strokes are not usually shown on the *p-v* diagrams of the ideal and air cycles. Normally, the Otto cycle will be represented as a theoretical closed cycle (12341) with the working medium returning to its initial state at the end of the heat rejection process, states 4 to 1.

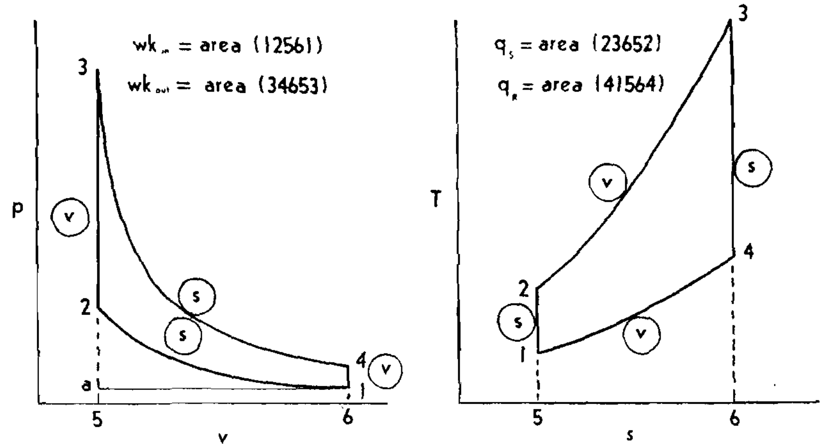


FIG. 3-2. Otto cycle.

Since the heat supplied (q_S) is at **constant volume,** the work during this process is zero and the non-flow energy equation

$$q_{23} = u_3 - u_2 + \frac{wk_{23}}{J}$$

becomes

$$q_S = q_{23} = u_3 - u_2 \qquad \text{(Btu/lb).} \qquad \text{(3-6)}$$

For an ideal cycle, which uses a perfect gas (constant specific heats), the heat supplied may be found from

$$q_S = q_{23} = c_v(T_3 - T_2) \qquad \text{(Btu/lb).} \qquad \text{(3-7)}$$

Since the heat rejected (q_R) is also a **constant volume** process with $wk_{41} = 0$, the heat rejected is

$$q_R = q_{41} = u_4 - u_1 \qquad \text{(Btu/lb).} \qquad \text{(3-8)}$$

For the ideal cycle with a perfect gas,

$$q_R = q_{41} = c_v(T_4 - T_1) \qquad \text{(Btu/lb).} \qquad \text{(3-9)}$$

Therefore, the thermal efficiency of the Otto cycle becomes

$$\eta_t = \frac{\text{net work}}{q_S} = \frac{q_S - q_R}{q_S}$$

$$\eta_t = \frac{(u_3 - u_2) - (u_4 - u_1)}{(u_3 - u_2)}. \tag{3-10}$$

For the ideal cycle with a perfect gas, the thermal efficiency may be written as

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

$$\eta_t = 1 - \left(\frac{T_4 - T_1}{T_3 - T_2}\right). \tag{3-11}$$

This equation may be further simplified[6] to give

$$\eta_t = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{\left(\frac{T_2}{T_1}\right)}.$$

[6] Since the compression process, state 1 to 2, and the expansion process, state 3-4, are isentropic, it may be shown that

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} \text{and} \quad \frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{k-1}$$

Since $v_1 = v_4$ and $v_2 = v_3$, then

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}.$$

Rearranging the equation and subtracting unity from each side results in

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

which becomes

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

Thus,

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2}.$$

Substituting into equation (3-11) gives

$$\eta_t = 1 - \frac{T_1}{T_2}.$$

Since the process from states 1 to 2 is isentropic, the thermal efficiency equation for a *perfect gas* becomes

$$\eta_t = 1 - \frac{1}{\left(\frac{T_2}{T_1}\right)} = 1 - \frac{1}{\left(\frac{v_1}{v_2}\right)^{k-1}}$$

$$\eta_t = 1 - \frac{1}{(r)^{k-1}} \tag{3-12}$$

The **compression ratio (r)** is defined as the volume at the beginning of the compression stroke divided by the volume at the end of the compression stroke, i.e.,

$$r = \frac{V_1}{V_2} \tag{3-13}$$

Compression ratio can be also expressed in terms of specific volumes, thus,

$$r = \frac{v_1}{v_2} \tag{3-13a}$$

Equation (3-12) indicates that the thermal efficiency of the Otto cycle is a function of the compression ratio and k, the ratio of the specific heats of the working medium. An increase in the compression ratio will increase the thermal efficiency of the Otto cycle (Fig. 3-5). An increase in the specific heats will decrease the ratio of specific heats (k) and, therefore, will decrease the thermal efficiency. Thus, the ideal cycle, where k is assumed constant at 1.4, has a higher thermal efficiency than the air cycle, which has variable specific heats that give a lower mean value of k (Fig. 2-5) throughout the operating temperature range of the cycle.

The **expansion ratio (r_E)** is defined as the volume at the end of expansion (power) stroke divided by the volume at the end of the heat supplied process. The expansion ratio for the Otto cycle becomes

$$r_E = \frac{V_4}{V_3} \text{ or} \tag{3-14}$$

$$r_E = \frac{v_4}{v_3} \tag{3-14a}$$

Example Problem 3-1.

An Otto cycle is operating with a perfect gas as the working medium, i.e., an ideal cycle with constant specific heats ($c_p = 0.24$, $c_v = 0.171$, and $k = 1.4$). The compression ratio is 7 to 1 and the heat supplied is 320 Btu/lb. At the beginning of compression process, state 1, the pressure and temperature are 14.7 psia and 60° F.

(a) Sketch the p-v and T-s diagrams.

Calculate the following for an ideal cycle:

(b) Values of p, v, and T at state points 1, 2, 3, and 4.

(c) Heat rejected, Btu/lb.

(d) Net work, Btu/lb.

(e) Thermal efficiency, %.

Solution:

(a) See Fig. 3-2.

(b)

State		1	2	3	4
p	psia	14.7	224	595	39
v	ft³/lb	13.1	1.87	1.87	13.1
T	°F abs	520	1132	3002	.1380

Given values are in bold face in the table.

(1) $$v_1 = \frac{RT_1}{p_1} = \frac{53.3(520)}{14.7(144)} = 13.1 \text{ ft}^3/\text{lb}.$$

(2) $$v_2 = \frac{v_1}{r} = \frac{13.1}{7} = 1.87 \text{ ft}^3/\text{lb}.$$

(3) $$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = 520(7)^{0.4} = 520(2.18) = 1132° \text{ F abs}.$$

(4) $$p_2 = p_1\left(\frac{v_1}{v_2}\right)^{k} = 14.7(7)^{1.4} = 14.7(15.3) = 224 \text{ psia}.$$

$$\text{or } p_2 = \frac{RT_2}{v_2} = \frac{53.3(1132)}{1.87(144)} = 224 \text{ psia}.$$

(5) $$v_3 = v_2 = 1.87 \text{ ft}^3/\text{lb}.$$

(6) $$q_a = c_v(T_3 - T_2) = 320 \text{ Btu/lb (given quantity)}.$$

$$T_3 = T_2 + \frac{q_a}{c_v} = 1132 + \frac{320}{0.171} = 1132 + 1870 = 3002° \text{ F abs}.$$

(7) $$p_3 = \frac{RT_3}{v_3} = \frac{53.3(3002)}{1.87(144)} = 595 \text{ psia}.$$

(8) $$v_4 = v_1 = 13.1 \text{ ft}^3/\text{lb}.$$

(9) $$\frac{v_3}{v_4} = \frac{v_2}{v_1} = \frac{1}{r} = \frac{1}{7}.$$

$$T_4 = T\left(\frac{v_3}{v_4}\right)^{k-} = 3002\left(\frac{1}{7}\right)^{0.4} = \frac{3002}{2.18} = 1380° \text{ F abs}.$$

(10) $$p_4 = p_3\left(\frac{v_3}{v_4}\right)^{k} = 595\left(\frac{1}{7}\right)^{1.4} = \frac{595}{15.3} = 38.9 \text{ psia}.$$

$$\text{or } p_4 = \frac{RT_4}{v_4} = \frac{53.3(1380)}{13.1(144)} = 39 \text{ psia.}$$

(c) $q_R = q_{41} = c_v(T_4 - T_1) = 0.171(1380 - 520) = 147$ Btu/lb.

(d) Net Wk $= q_S - q_R = 320 - 147 = 173$ Btu/lb.

(e) $$\eta_t = \frac{q_S - q_R}{q_S} = \frac{\text{net wk}}{q_S} = \frac{173}{320} = 54\%.$$

also $$\eta_t = 1 - \frac{1}{(r)^{k-1}} = 1 - \frac{1}{(7)^{0.4}} = 1 - 0.46 = 54\%.$$

Example Problem 3-2:

An Otto cycle has air as its working medium, i.e., air with variable specific heats. The compression ratio is 7 to 1; the initial temperature is 60° F; initial pressure is 14.7 psia; and the peak temperature (T_3) is 2673° F abs. By using the method of calculation as shown in the example problem in Article 3-8, the following tabulated values were determined from the Air Tables (Table 3-1).

State		1	2	3	4
p	psia	14.7	220	529	39.7
v	ft³/lb	13.1	1.87	1.87	13.1
T	°F abs	520	1112	2673	1404
u	Btu/lb	−7	97	417	152
h	Btu/lb	29	174	600	248

(a) Sketch the cycle on p-v and T-s coordinates.
Calculate the following for an air cycle:
(b) Compression ratio.
(c) Heat supplied, Btu/lb.
(d) Heat rejected, Btu/lb.
(e) Net work, Btu/lb.
(f) Thermal efficiency, %.

Solution:

(a) See Fig. 3-2.

(b) $$r = \frac{v_1}{v_2} = \frac{13.1}{1.87} = 7.$$

(c) $q_S = u_3 - u_2 = 417 - 97 = 320$ Btu/lb.

(d) $q_R = u_4 - u_1 = 152 - (-7) = 159$ Btu/lb.

(e) net wk $= q_S - q_R = 320 - 159 = 161$ Btu/lb.

(f) $$\eta_t = \frac{q_S - q_R}{q_S} = \frac{\text{net wk}}{q_S} = \frac{161}{320} = 50.4\%.$$

3-5. Diesel Cycle. The Diesel cycle is the theoretical cycle for the diesel or compression ignition engine. A graphical sketch of the cycle on the p-v and T-s coordinates is shown in Fig. 3-3. The basic difference between the Otto and Diesel cycles is in the heat supplied process. In the Otto cycle heat is supplied at constant volume, while in the Diesel cycle the heat is supplied at **constant pressure,** states 2 to 3 in Fig. 3-3. The actual spark ignition and compression ignition engines have other basic differences (Article 1-7). One of these differences that affects cycle analyses is the high compression ratio range of the diesel engines (12 to 20) as compared to the compression ratio range of the spark ignition engines (5 to 10.5).

The Diesel cycle, as shown in Fig. 3-3, has the following thermodynamic processes:

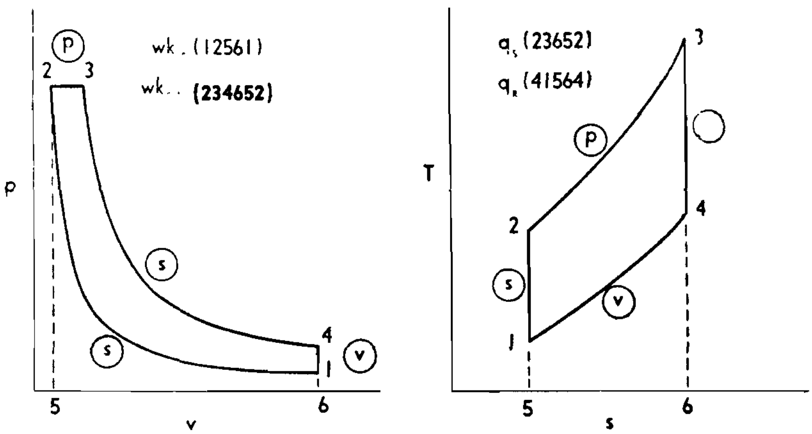


FIG. 3-3. Diesel cycle.

Process 1-2. **Isentropic.** Compression or work done on the working medium by the piston (wk_{in}).

Process 2-3. **Constant Pressure.** Heat supplied (qs), and part of work done ($wk_{out_{23}}$)

Process 3-4. **Isentropic.** Expansion or remaining work done by the working medium on the piston ($wk_{out_{34}}$).

Process 4-1. **Constant Volume.** Heat rejected instantaneously (q_R).

During the constant pressure process, states 2 to 3, the piston moves from position 2 to position 3 so that work is done by the working medium on the piston, i.e.,

$$\frac{wk_{23}}{J} = \int_2^3 pdv = \frac{p_2v_3 - p_2v_2}{J} \qquad \text{(Btu/lb).}$$

Therefore the heat supplied at constant pressure from the non-flow energy equation

$$q_{23} = u_3 - u_2 + \frac{wk_{23}}{J} \qquad \text{(Btu/lb)}$$

becomes

$$q_S = q_{23} = (u_3 - u_2) + \left(\frac{p_3v_3 - p_2v_2}{J}\right) \text{ (Btu/lb).}$$

Since $h = u + (pv/J)$, the equation for the heat supplied at **constant pressure** is written as

$$q_S = q_{23} = h_3 - h_2 \qquad \textbf{(Btu/lb).} \qquad \textbf{(3-15)}$$

For an ideal cycle, which has a perfect gas with constant specific heats as its working medium, the heat supplied may be determined by

$$q_S = q_{23} = c_p(T_3 - T_2) \qquad \text{(Btu/lb).} \qquad \text{(3-16)}$$

It should be emphasized that in a **constant pressure heating process the enthalpy of the working medium changes,** while in a **constant volume heating process the internal energy of the working medium changes.** Thus, the heat rejected at constant volume, states 4 to 1, is

$$q_R = q_{41} = u_4 - u_1 \qquad \textbf{(Btu/lb).} \qquad \textbf{(3-17)}$$

For an ideal cycle with a perfect gas, the heat rejected may be found from

$$q_R = q_{41} = c_v(T_4 - T_1) \qquad \text{(Btu/lb).} \qquad \text{(3-18)}$$

The **thermal efficiency** of the Diesel cycle is determined from

$$\eta_t = \frac{q_S - q_R}{q_S} = \frac{(h_3 - h_2) - (u_4 - u_1)}{(h_3 - h_2)}. \qquad \textbf{(3-19)}$$

For an ideal cycle having a perfect gas as the working medium, the thermal efficiency may be written as

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

$$\eta_t = 1 - \frac{1}{k}\left(\frac{T_4 - T_1}{T_3 - T_2}\right).$$

The compression ratio (r) for the Diesel cycle is defined as

$$r = \frac{v_1}{v_2}.$$

The expansion ratio (r_E) is

$$r_E = \frac{v_4}{v_3}.$$

The **cut-off ratio (r_L),** which is sometimes termed the load ratio, is defined as the volume at the end of the heat supplied (constant pressure combustion) process divided by the volume at the beginning of the same process. For the Diesel cycle

$$r_L = \frac{V_3}{V_2} \text{ or} \tag{3-20}$$

$$r_L = \frac{v_3}{v_2} \tag{3-20a}$$

The thermal efficiency equation may be written in terms of the cutoff ratio, the compression ratio, and the ratio of specific heats.[7] Thus,

[7](a) $$\eta_t = 1 - \frac{1}{k}\left(\frac{T_4 - T_1}{T_3 - T_2}\right)$$

(b) $$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2}\left[\frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}\right]$$

(c) $$\eta_t = 1 - \frac{1}{k}\left(\frac{T_1}{T_2}\right)\left[\frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}\right]$$

(d) For a constant pressure process

$$\frac{v_2}{T_2} = \frac{v_3}{T_3} \quad \text{or} \quad \frac{T_3}{T_2} = \frac{v_3}{v_2} = r_L$$

(e) For an isentropic process

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1}$$

and

$$T_1 = T_2\left(\frac{v_2}{v_1}\right)^{k-1}$$

(f) $$\frac{T_4}{T_1} = \frac{T_3\left(\frac{v_3}{v_4}\right)^{k-1}}{T_2\left(\frac{v_2}{v_1}\right)^{k-1}}$$

(g) Since $v_4 = v_1$

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}\left(\frac{v_3}{v_2}\right)^{k-1} = \frac{v_3}{v_2}\left(\frac{v_3}{v_2}\right)^{k-1} = (r_L)^k$$

(h) $$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \frac{1}{(r)^{k-1}}$$

(i) Substituting (d), (g), and (h) into (c) gives

$$\eta_t = 1 - \frac{1}{(r)^{k-1}}\left[\frac{(r_L)^k - 1}{k(r_L - 1)}\right].$$

$$\eta_t = 1 - \frac{1}{(r)^{k-1}}\left[\frac{(r_L)^k - 1}{k(r_L - 1)}\right]. \qquad (3\text{-}21)$$

Equation 3-21 shows that the thermal efficiency of the Diesel cycle is a function of the compression ratio, the cut-off ratio, and the ratio of the specific heats. The term in brackets, i.e., the function of the cut-off ratio, is always greater than unity. Since the thermal efficiency equation of the Otto cycle is same as that of the Diesel cycle *except for the bracketed term,* the Otto cycle will have a higher thermal efficiency than the Diesel cycle *at the same compression ratio* (Fig. 3-5). As the cut-off ratio is decreased (i.e., less heat is added at constant pressure), the thermal efficiency of the Diesel cycle will approach the thermal efficiency of the Otto cycle. An increase in the compression ratio will increase the thermal efficiency of the Diesel cycle (Fig. 3-5).

Example Problem 3-3.

A theoretical engine is operating on the Diesel cycle and has a perfect gas as the working medium. The compression ratio is 16 to 1 and the maximum temperature (T_3) in the cycle is 2912° F abs. At the beginning of the compression process, the pressure and temperature are 14.7 psia and 60° F. This is an ideal cycle analysis in which the specific heats are constant: $c_p = 0.24$, $c_v = 0.171$, $k = 1.4$.

(a) Sketch the p-v and T-s diagrams.

Calculate the following for an ideal cycle:

(b) Values of p, v, and T at state points 1, 2, 3, and 4.
(c) Heat supplied, Btu/lb.
(d) Heat rejected, Btu/lb.
(e) Net work, Btu/lb.
(f) Thermal efficiency, %.

Solution:

(a) See Fig. 3-3.
(b)

State		1	2	3	4
p	psia	**14.7**	714	714	34.7
v	ft³/lb	13.1	0.82	1.51	13.1
T	°F abs	**520**	1580	**2912**	1230

Given values are in bold face in the table.

(1) $v_1 = \dfrac{RT_1}{p_1} = \dfrac{53.3(520)}{14.7(144)} = 13.1$ ft³/lb

(2) $v_2 = \dfrac{v_1}{r} = \dfrac{13.1}{16} = 0.82$ ft³/lb.

(3) $T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = 520(16)^{0.4} = 520(3.04) = 1580°$ F abs.

(4) $p_2 = p_1\left(\frac{v_1}{v_2}\right)^{k} = 14.7(16)^{1.4} = 14.7(48.5) = 714$ psia.

or $p_2 = \frac{RT_2}{v_2} = \frac{53.3(1580)}{(0.82)(144)} = 714$ psia.

(5) $p_3 = p_2 = 714$ psia.

(6) $v_3 = \frac{RT_3}{p_3} = \frac{53.3(2912)}{714(144)} = 1.51$ ft³/lb.

(7) $v_4 = v_1 = 13.1$ ft³/lb.

(8) $T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = 2912\left(\frac{1.51}{13.1}\right)^{0.4} = 2912(0.422) = 1230°$ F abs.

(9) $p_4 = p_3\left(\frac{v_3}{v_4}\right)^{k} = 714\left(\frac{1.51}{13.1}\right)^{1.4} = 714(0.0485) = 34.7$ psia.

or $p_4 = \frac{RT_4}{v_4} = \frac{53.3(1230)}{13.1(144)} = 34.7$ psia.

(c) $q_S = q_{23} = c_p(T_3 - T_2) = 0.24(2912 - 1580) = 320$ Btu/lb.

(d) $q_R = q_{41} = c_v(T_4 - T_1) = 0.171(1230 - 520) = 121$ Btu/lb.

(e) net wk $= q_S - q_R = 320 - 121 = 199$ Btu/lb.

(f) $\eta_t = \frac{q_S - q_R}{q_S} = \frac{\text{net wk}}{q_S} = \frac{199}{320} = 62.2\%.$

Example Problem 3-4.

A Diesel cycle, operating with air as its working medium, has a compression ratio of 16 to 1. The heat supplied is 320 Btu/lb. Following is a tabulation of properties at change of process points as determined from the Air Tables. (Table 3-1)

State		1	2	3	4
p	psia	14.7			35.8
v	ft³/lb	13.1	0.82	1.45	13.1
T	°F abs	520	1500		1268
u	Btu/lb	−7	171	412	126
h	Btu/lb	29	274		213

(a) Sketch the cycle on p-v and T-s coordinates.

Calculate the following for an air cycle:

(b) Values of properties not given in the above table.

(c) Heat rejected, Btu/lb.

(d) Net work, Btu/lb.

(e) Thermal efficiency, %.

Solution:

(a) See Fig. 3-3.

(b) (1) $p_2 = \dfrac{RT_2}{v_2} = \dfrac{53.3(1500)}{(0.82)(144)} = 676$ psia.

(2) $p_3 = p_2 = 676$ psia.

(3) $T_3 = \dfrac{p_3 v_3}{R} = \dfrac{676(144)(1.45)}{53.3} = 2650°$ F abs.

(4) $q_S = h_3 - h_2$ or $h_3 = h_2 + q_S$

$h_3 = 274 + 320 = 594$ Btu/lb.

(c) $q_R = u_4 - u_1 = 126 - (-7) = 133$ Btu/lb.

(d) net wk $= q_S - q_R = 320 - 133 = 187$ Btu/lb.

(e) $\eta_t = \dfrac{\text{net wk}}{q_S} = \dfrac{187}{320} = 58.5\%$.

3-6. Dual Combustion (Sabathé) Cycle. In a diesel engine, the combustion process (heat supplied process) does not approach a constant pressure process, such as states 2 to 3 in Fig. 3-3, except in an exceptionally large, slow speed engine. Indicator card diagrams (Articles 11-3 and 13-2) of diesel engines show that the combustion process approaches more closely a combination of a constant volume and a constant pressure process. This is also true for the spark ignition engine, for the combustion process cannot take place instantaneously, i.e., at constant volume, as in an Otto cycle. Therefore, the dual combustion cycle, also known as the Sabathé cycle and as the limited pressure cycle, provides a closer approximation to a theoretical cycle for both

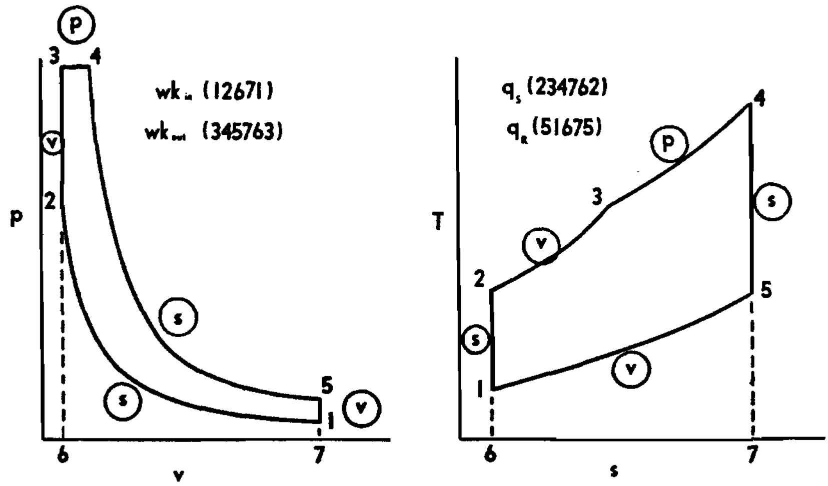


FIG. 3-4. Dual combustion cycle.

the compression ignition (diesel) engine and the spark ignition engine than do the Otto and Diesel cycles.

The dual combustion cycle, as shown in Fig. 3-4, has the following thermodynamic processes:

Process 1-2. **Isentropic.** Compression or work done on the working medium by the piston (wk_{in}).

Process 2-3. **Constant Volume.** Part of the heat supplied (q_{S23}).

Process 3-4. **Constant Pressure.** Remainder of the heat supplied (q_{S34}), and part of work done ($wk_{out_{34}}$).

Process 4-5. **Isentropic.** Expansion or remaining work done by the working medium on the piston ($wk_{out_{45}}$).

Process 5-1. **Constant Volume.** Heat rejected instantaneously (q_R).

The difference between this cycle and the other two theoretical cycles (Otto and Diesel) is that the dual combustion cycle has part of the heat supplied at constant volume and the remainder at constant pressure.

The heat supplied to the cycle is the sum of the heat supplied at constant volume, states 2 to 3, and the heat supplied at constant pressure, states 3 to 4. *Since in a constant volume heating process the internal energy changes and in a constant pressure heating process the enthalpy changes*, the heat supplied to the cycle is

$$q_S = q_{S23} + q_{S34} = (u_3 - u_2) + (h_4 - h_3) \qquad \text{(Btu/lb)}. \qquad (3\text{-}23)$$

The heat is rejected instantaneously at constant volume, states 5 to 1,

$$q_R = q_{51} = (u_5 - u_1) \qquad \text{(Btu/lb)}. \qquad (3\text{-}24)$$

The heat supplied and heat rejected for an ideal cycle having a perfect gas with constant specific heats may be calculated from

$$q_S = q_{S23} + q_{S34} = c_v(T_3 - T_2) + c_p(T_4 - T_3) \qquad \text{(Btu/lb)} \qquad (3\text{-}25)$$

and

$$q_R = q_{51} = c_v(T_5 - T_1) \qquad \text{(Btu/lb)}. \qquad (3\text{-}26)$$

The thermal efficiency of the dual combustion cycle is written as

$$\eta_t = \frac{q_S - q_R}{q_S}$$

$$\eta_t = \frac{[(u_3 - u_2) + (h_4 - h_3)] - (u_5 - u_1)}{(u_3 - u_2) + (h_4 - h_3)} \cdot \qquad (3\text{-}27)$$

For an ideal cycle (perfect gas), the thermal efficiency may be determined from

$$\eta_t = \frac{q_S - q_R}{q_S}$$

$$\eta_t = \frac{[c_v(T_3 - T_2) + c_p(T_4 - T_3)] - c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$\eta_t = 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + k(T_4 - T_3)} \cdot \tag{3-28}$$

Considering the three theoretical cycles (Otto, Diesel, dual combustion) *at the same compression ratio*, the thermal efficiency of the dual combustion cycle will fall between the thermal efficiencies of the Otto and Diesel cycles (Fig. 3-5). As the quantity of heat supplied at constant volume, states 2 to 3 in Fig. 3-4, is increased while the heat supplied at constant pressure, states 3 to 4, is decreased, the thermal efficiency of the dual combustion cycle will approach that of the Otto cycle. If the cut-off ratio:

$$r_L = \frac{v_4}{v_3}, \tag{3-29}$$

is increased, i.e., the quantity of heat supplied at constant pressure, is increased while the quantity of heat supplied at constant volume is decreased, the thermal efficiency of the dual combustion cycle will be decreased and will approach that of the Diesel cycle.

The compression ratio of the dual combustion cycle is

$$r = \frac{v_1}{v_2}$$

while the expansion ratio is

$$r_E = \frac{v_5}{v_4} \cdot$$

Example Problem 3-5:

A theoretical engine is operating on the dual combustion cycle. The specific volume at the end of isentropic compression is 1.09 ft^3/lb and the highest temperature reached in the cycle (T_4) is 2444° F. The heat supplied at constant volume is 100 Btu/lb. At the beginning of the compression process, the pressure and temperature are 14.7 psia and 60° F. The working medium is a perfect gas with the following constants: $c_v = 0.171$, $c_p = 0.24$, $k = 1.4$.

(a) Sketch the cycle on p-v and T-s coordinates.

Calculate the following for an **ideal cycle**:

(b) Values of p, v, and T at change of process points throughout the cycle.

(c) Compression ratio.

(d) Heat supplied to the cycle, Btu/lb.

(e) **Heat rejected, Btu lb.**
(f) **Net work, Btu/lb.**
(g) **Thermal efficiency, %.**
(h) Cut-off ratio.

Solution:

(a) See Fig. 3-4
(b)

State		1	2	3	4	5
p	psia	**14.7**	477	675	675	35.2
v	ft³/lb	13.1	1.09	1.09	1.59	13.1
T	°F abs	**520**	1403	1988	**2904**	1250

Given quantities are in bold face in the table.

(1) $v_1 = \dfrac{RT_1}{p_1} = \dfrac{53.3(520)}{14.7(144)} = 13.1$ ft³/lb.

(2) $T_2 = T_1\left(\dfrac{v_1}{v_2}\right)^{k-1} = 520\left(\dfrac{13.1}{1.09}\right)^{0.4} = 520(2.7) = 1403°$ F abs.

(3) $p_2 = \dfrac{RT_2}{v_2} = \dfrac{53.3(1403)}{1.09(144)} = 477$ psia.

(4) $q_{S23} = c_v(T_3 - T_2) = 100$ Btu/lb. (given quantity)

$$T_3 = T_2 + \left(\frac{q_{S23}}{c_v}\right) = 1403 + \left(\frac{100}{0.171}\right)$$
$$= 1403 + 585 = 1988° \text{ F abs.}$$

(5) $v_3 = v_2 = 1.09$ ft³/lb.

(6) $p_3 = \dfrac{RT_3}{v_3} = \dfrac{53.3(1988)}{1.09(144)} = 675$ psia.

(7) $p_4 = p_3 = 675$ psia.

(8) $v_4 = \dfrac{RT_4}{p_4} = \dfrac{53.3(2904)}{675(144)} = 1.59$ ft³/lb.

(9) $v_5 = v_1 = 13.1$ ft³/lb.

$$p_5 = p_4\left(\frac{v_4}{v_5}\right)^k = 675\left(\frac{1.59}{13.1}\right)^{1.4} = 675(0.052) = 35.2 \text{ psia.}$$

$$T_5 = T_4\left(\frac{v_4}{v_5}\right)^{k-1} = 2904\left(\frac{1.59}{13.1}\right)^{0.4} = 2904(0.43) = 1250° \text{ F abs.}$$

or $T_5 = \dfrac{p_5v_5}{R} = \dfrac{35.2(144)(13.1)}{53.3} = 1250°$ F abs.

(c) $r = \dfrac{v_1}{v_2} = \dfrac{13.1}{1.09} = 12.$

(d) $q_S = q_{S23} + q_{S34} = c_v(T_3 - T_2) + c_p(T_4 - T_3).$

$q_S = 100 + 0.24(2904 - 1988) = 100 + 220 = 320$ Btu/lb.

(e) $q_R = q_{51} = c_v(T_5 - T_1) = 0.171(1250 - 520) = 124.5$ Btu/lb.

(f) net wk $= q_S - q_R = 320 - 124.5 = 195.5$ Btu/lb.

(g) $$\eta_t = \frac{q_S - q_R}{q_S} = \frac{\text{net wk}}{q_S} = \frac{195.5}{320} = 61\%.$$

(h) $$r_L = \frac{v_4}{v_3} = \frac{1.59}{1.09} = 1.46.$$

Example Problem 3-6.

A dual combustion cycle, operating with air as its working medium, has the following properties:

	State	1	2	3	4	5
p	psia	14.4	215	750	750	81.4
v	ft^3/lb	13.65	1.95	1.95	2.46	13.65
T	°F abs	530	1132	3950	4980	3000
u	Btu/lb	−5	101	708	951	490
h	Btu/lb	31	179	978	1292	696

(a) Sketch the cycle on the p-v and T-s coordinates.

Calculate the following for an air cycle:

(b) Heat supplied, Btu/lb.
(c) Heat rejected, Btu/lb.
(d) Net work, Btu/lb.
(e) Thermal efficiency, %.
(f) Compression ratio.
(g) Expansion ratio.
(h) Cut-off ratio.

Solution:

(a) See Fig. 3-4.

(b) $$\begin{aligned} q_S &= q_{S23} + q_{S34} = (u_3 - u_2) + (h_4 - h_3) \\ &= (708 - 101) + (1292 - 978) \\ &= 607 + 314 = 921 \text{ Btu/lb.} \end{aligned}$$

(c) $q_R = q_{51} = u_5 - u_1 = 490 - (-5) = 495$ Btu/lb.

(d) net wk $= q_S - q_R = 921 - 495 = 426$ Btu/lb

(e) $$\eta_t = \frac{q_S - q_R}{q_S} = \frac{426}{921} = 46.2\%.$$

(f) $$r = \frac{v_1}{v_2} = \frac{13.65}{1.95} = 7.$$

(g) $$r_E = \frac{v_5}{v_4} = \frac{13.65}{2.46} = 5.5.$$

(h) $$r_L = \frac{v_4}{v_3} = \frac{2.46}{1.95} = 1.3.$$

3-7. Comparison of Cycles. The important variable factors which are used as a basis for comparison of the cycles are compression ratio, peak pressure, heat supplied, heat rejected, and net work. In order to compare the performance of the Otto, Diesel, and dual combustion cycles, some of the variable factors must be fixed.

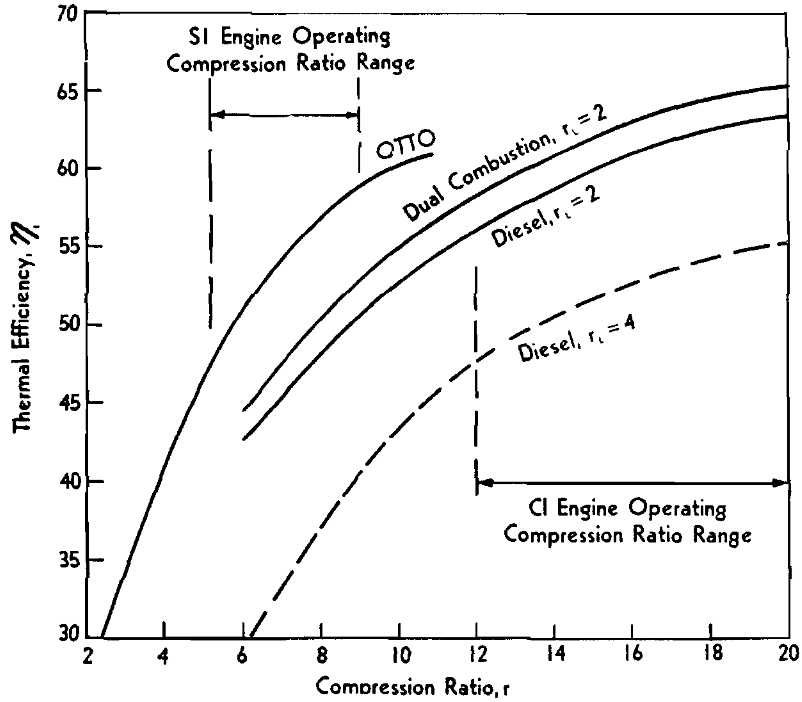


FIG. 3-5. **Comparison of the thermal efficiencies of the theoretical cycles at various compression ratios.**

A comparison of the thermal efficiencies of the three cycles at various compression ratios and with given cut-off ratios for the Diesel and dual combustion cycles is shown in Fig. 3-5. The thermal efficiencies of the theoretical cycles increase with an increase in the compression ratio. At a given compression ratio, the Otto cycle is the most efficient while the Diesel cycle is the least efficient. However, diesel engines operate in a compression ratio range from 12 to 20 while the spark ignition engines operate in a compression ratio range from about 5 to 10.5. The maximum compression ratio for the spark ignition engine is limited by detonation (Chapter VIII). In their respective compression ratio ranges, the Diesel cycle is more efficient than the Otto cycle.

Fig. 3-6 gives a comparison of the theoretical cycles on the p-v and T-s coordinates. In this case, each cycle has *the same compression ratio and the same quantity of heat supplied.* All the cycles start with the same initial conditions, state 1, and have the same isentropic compression to state 2. Heat is then supplied through a different type or types of processes for each cycle; however, the T-s areas 23″ 5″ 62, 22′ 3′ 5′ 62, and 2 3 5 6 2 must be equal in order that the quantity of heat supplied to each cycle be the same. Since all the cycles reject their heat at the same specific volume, process line from states 4 to 1, the quantity

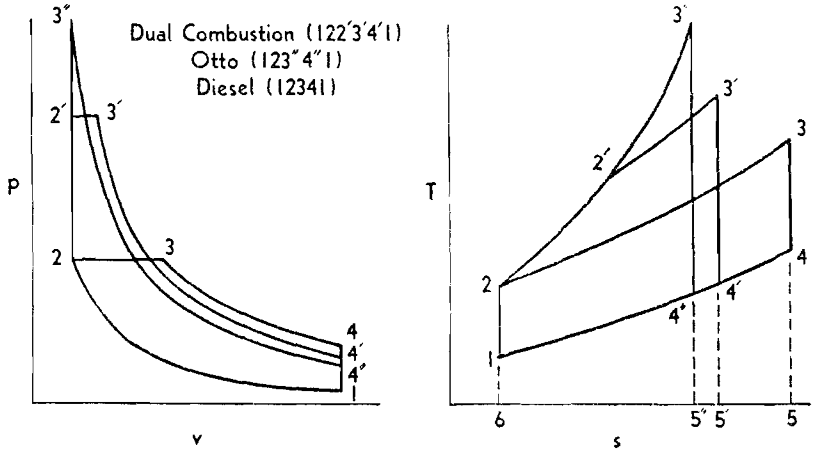


FIG. 3-6. Comparison of theoretical cycles with the same compression ratio and the same amount of heat supplied.

of heat rejected from each cycle is represented by the appropriate area under the line 4 to 1 on the T-s diagram. Since the heat supplied is the same for each cycle, the cycle. from the thermal efficiency equation

$$\eta_t = 1 - \frac{q_R}{q_S},$$

which has the least quantity of heat rejected, will have the highest thermal efficiency. The smallest quantity of heat rejected is in the Otto cycle area 4″ 5″ 6 1 4″. Even though the Otto cycle has the highest efficiency for this set of conditions, it must be noted that the peak pressure (p_3'') and peak temperature (T_3'') for the Otto cycle are far greater than they are in the other two cycles. Even though this is a theoretical comparison, it shows that the cycle which permits the greatest expansion of the working medium after the completion of the heat supplied process will have the highest thermal efficiency.

In engine design, the maximum pressure is often the limiting factor. In Fig. 3-7, the Otto and Diesel cycles are compared on a basis of the same maximum pressure and same quantity of heat supplied. The heat supplied areas 2 3 5 6 2 and 2′ 3′ 5′ 62′ must be equal. Since the heat supplied is the same, the cycle having the least amount of heat rejected will have the highest thermal efficiency. This is the Diesel cycle with the amount of heat rejected shown as the area 4 5 6 1 4. It should

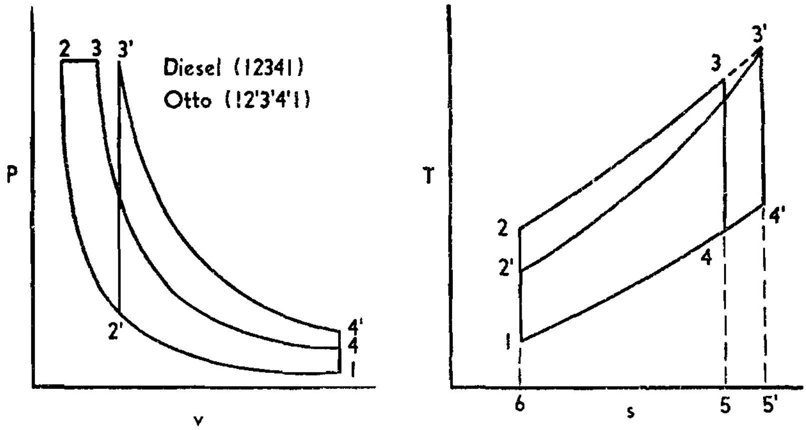


FIG. 3-7. Comparison of Otto and Diesel cycles with the same maximum pressure and the same amount of heat supplied.

be noted that under these conditions the compression ratio for the Diesel cycle is much higher than the compression ratio for the Otto cycle which is the usual case with diesel and spark ignition engines. Also, it is shown, once again, that the cycle which permits the greatest expansion of the working medium after completion of the heat supplied process will have the highest thermal efficiency.

3-8. Problem on Use of Air Tables. (Table 3-1). An Otto cycle (Fig. 3-2) with air as its working medium (air cycle analysis) has the following given data:

(a) $$T_1 = 520° \text{ F abs}$$

(b) $$p_1 = 14.7 \text{ psia}$$

(c) $$r = \frac{v_1}{v_2} = 7.657$$

(d) $$q_S = u_3 - u_2 = 756.1 \text{ Btu/lb.}$$

The problem is to determine the properties (p, v, T, u, h, p_r, v_r) at the state points 1, 2, 3, and 4 in Fig. 3-2.

State		1	2	q_s	3	4
p	psia	**14.7**	249		996	70.58
v	ft³/lb	13.09	1.710		1.710	13.09
T	°F abs	**520**	1150		4600	2496
u	Btu/lb	−6.87	104.43	**756.13**	860.56	378.51
h	Btu/lb	28.77	183.26		1175.86	549.60
v_r		5192	678.1		9.087	69.54
p_r		2.504	42.40		12655	897.3

Given quantities are in bold face in the table.

(a) Enter the Air Tables with $T_1 = 520°$ F abs. Read directly the value u_1, h_1, v_{r1}, p_{r1}. (Values as read are shown in the above Table).

(b) Compute v_1 using the equation of state

$$v_1 = \frac{RT_1}{p_1} = \frac{(53.3)(520)}{(14.7)(144)} = 13.09 \text{ ft}^3/\text{lb}.$$

(c) $r = \dfrac{v_1}{v_2} = \dfrac{v_{r1}}{v_{r2}} = 7.657$

(d) Compute v_{r2} from the isentropic relationship[3]

$$v_{r2} = \frac{v_{r1}}{r} = \frac{5192}{7.657} = 678.1$$

(e) Enter Air Tables with v_{r2}. Interpolate from the values p_{r2}, T_2, u_2, h_2. (Values determined are shown in the above table).

(f) Compute v_2 from r and v_1

$$v_2 = \frac{v_1}{r} = \frac{13.09}{7.657} = 1.710 \text{ ft}^3/\text{lb}$$

(g) Compute p_2 from the isentropic relationship

$$p_2 = p_1\left(\frac{p_{r2}}{p_{r1}}\right) = 14.7\left(\frac{42.4}{2.504}\right) = 249 \text{ psia}$$

or p_2 from the equation of state

$$p_2 = \frac{RT_2}{v_2} = \frac{(53.3)(1150)}{(1.710)(144)} = 249 \text{ psia}.$$

[3] The equation $v_1/v_2 = v_{r1}/v_{r2}$ may be used only for an isentropic process.

(h) It should be emphasized here that the Air Tables are based on isentropic processes only. However, if any one property at any state point is known, the Air Tables may be entered with that property in order to find the other properties at that point.

(i) Compute: $u_3 = u_2 + q_S = 104.43 + 756.13 = 860.56$ Btu/lb

(j) Enter Air Tables with u_3. Interpolate for the values T_3, h_3, v_{r3}, p_{r3}. (Values determined are shown in the above Table).

(k) $v_2 = v_3$.

(l) Compute

$$p_3 = \frac{RT_3}{v_3} = \frac{(53.3)(4600)}{(1.710)(144)} = 996 \text{ psia.}$$

Note that (p_2/p_3) does not equal (p_{r2}/p_{r3})

(m) Compute v_{r4} from the isentropic relationship

$$v_{r4} = (v_{r3})(r) = (9.087)(7.657) = 69.54$$

(n) Enter Air Tables with v_{r4}. Interpolate for values T_4, p_{r4}, u_4, h_4. (Values determined are shown in above Table).

(o) $v_4 = v_1$

(p) Compute p_4 from the isentropic relationship.

$$p_4 = p_3\left(\frac{p_{r4}}{p_{r3}}\right) = 996\left(\frac{897.3}{12655}\right) = 70.6 \text{ psia}$$

or

$$p_4 = \frac{RT_4}{v_4} = \frac{(53.3)(2496)}{(13.09)(144)} = 70.58 \text{ psia.}$$

Having determined values of the internal energy for an Otto cycle, the heat rejected, the net work, and the thermal efficiency may be calculated.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

3-1. E. F. Obert, *Internal Combustion Engines*, International Textbook Co., Scranton, 1950.

3-2. C. F. Taylor and E. S. Taylor, *Internal Combustion Engines*, International Textbook Co., Scranton, 1948.

3-3. A. P. Fraas, *Combustion Engines*, McGraw-Hill Book Co., 1948.

3-4. J. H. Keenan and J. Kaye, *Thermodynamic Properties of Air*, John Wiley and Sons, Inc., New York, 1945.

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3-8. J. K. Salisbury, *Kent's Mechanical Engineers' Handbook*, John Wiley and Sons, Inc., New York, 1950.

3-9. L. C. Lichty, *Internal Combustion Engines*, McGraw-Hill Book Co., Inc., New York, 1951.

EXERCISES

3-1. Differentiate between the ideal cycle and (a) the air cycle, (b) the fuel-air cycle, (c) the actual cycle.

3-2. An Otto cycle, operating with a perfect gas as the working medium, has a compression ratio of 8 to 1. ($c_v = 0.171$, $c_p = 0.24$, $k = 1.4$).

(a) Sketch the cycle on *p-v* and *T-s* coordinates and fill in the unknown properties in the following table.

State		1	2	3	4
p	psia	15			41.5
v	ft³/lb	13.1			
T	°F abs	530		3380	1470

Calculate the following for an ideal cycle:

(b) Heat supplied, Btu/lb.
(c) Heat rejected, Btu/lb.
(d) Net work, Btu/lb.
(e) Thermal efficiency, %.
(f) Expansion ratio.

Ans: (a) $v_2 = v_3 = 1.64$ ft³/lb; $T_2 = 1220°$ F abs; $p_2 = 276$ psia; $p_3 = 763$ psia; $v_4 = 13.1$ ft³/lb; (b) 370 Btu/lb, (c) 160.5 Btu/lb, (d) 209.5 Btu/lb, (e) 56.6%, (f) 8 to 1.

3-3. An Otto cycle, operating with air as the working medium, has 370 Btu/lb of heat supplied. Values of the properties at change of process points as determined from the Air Tables are shown below:

State		1	2	3	4
p	psia	15	268		42.5
v	ft³/lb	13.1			13.1
T	°F abs	530	1189	2964	1508
u	Btu/lb	−5	112		173
h	Btu/lb	31	193		276

(a) Sketch the cycle on *p-v* and *T-s* coordinates and fill in the unknown values in the above table.

Calculate the following for an air cycle:

(b) Compression ratio.
(c) Heat rejected, Btu/lb.
(d) Net work, Btu/lb.

(e) Thermal efficiency, %.
(f) Expansion ratio.

Ans: (a) $v_2 = v_3 = 1.64$ ft³/lb; $p_2 = 670$ psia; $u_3 = 482$ Btu/lb; $h_3 = 685$ Btu/lb, (b) 8 to 1, (c) 178 Btu/lb, (d) 192 Btu/lb, (e) 51.9%, (f) 8 to 1.

3-4. A Diesel cycle is operating with a perfect gas as the working medium. The compression ratio is 15 to 1 and the heat supplied is 392 Btu/lb. ($c_v = 0.171$, $c_p = 0.24$, $k = 1.4$).

(a) Sketch the cycle on *p-v* and *T-s* coordinates and fill in the unknown properties in the following table:

State		1	2	3	4
p	psia	15			40.5
v	ft³/lb	13.1		1.78	13.1
T	°F abs	530	1562		1438

Calculate the following for an ideal cycle:
(b) Heat rejected, Btu/lb.
(c) Work in, Btu/lb.
(d) Net work, Btu/lb.
(e) Thermal efficiency, %.
(f) Cut-off ratio.
(g) Expansion ratio.

Ans: (a) $v_2 = 0.873$ ft³/lb; $p_2 = p_3 = 664$ psia; $T_3 = 3195°$ F abs. (b) 155 Btu/lb, (c) 177 Btu/lb, (d) 237 Btu/lb, (e) 60.5%, (f) 2.04 to 1, (g) 7.36 to 1.

3-5. A Diesel cycle has air as its working medium. Values of the properties at change of process points as determined from the Air Tables are shown below:

State		1	2	3	4
p	psia	15			40.9
v	ft³/lb	13.1	0.772	1.465	13.1
T	° F abs	530	1556	2950	
u	Btu/lb	−5	182	479	160
h	Btu/lb	31	289	681	259

(a) Sketch the cycle on *p-v* and *T-s* coordinates and fill in the unknown properties in the above table.

Calculate the following for an air cycle:
(b) Compression ratio.
(c) Cutoff ratio.
(d) Heat supplied, Btu/lb.
(e) Heat rejected, Btu/lb.
(f) Net work, Btu/lb.
(g) Thermal efficiency, %.

Ans: (a) $p_2 = p_3 = 745$ psia; $T_4 = 1443°$ F abs; (b) 17 to 1, (c) 1.9 to 1, (d) 392 Btu/lb, (e) 165 Btu/lb, (f) 227 Btu/lb, (g) 58%.

3-6. Derive an expression for the work out of a Diesel cycle in terms of internal energy and enthalpy.
Ans: $wk_{out} = (h_3 - h_2) - (u_4 - u_1)$.

3-7. Given the following Air Table analysis for a dual combustion cycle:

	State	1	2	3	4	5
p	psia	14.6	205.5	718	718	80.6
v	ft³/lb	13.8	2.06	2.06	2.55	13.8
T	° Fabs	545	1143	4000	4950	3005
u	Btu/lb	−2.6	103.1	719.4	943.8	491.3
h	Btu/lb	34.8	181.5	993.6	1283.1	697.2

(a) Sketch the p-v and T-s diagrams.
Calculate the following for an air cycle:
(b) Heat supplied, Btu/lb.
(c) Heat rejected, Btu/lb.
(d) Net work, Btu./lb.
(e) Thermal efficiency, %.
(f) Cut-off ratio.
(g) Expansion ratio.

Ans: (b) 905.8 Btu/lb, (c) 493.9 Btu/lb, (d) 411.9 Btu/lb, (e) 45.5%, (f) 1.24 to 1, (g) 5.41 to 1.

CHAPTER IV

ENGINE POWER

The production of power is the primary reason for an engine's existence. In order to determine how well an engine is producing, or in order to compare the output of one engine with that of other engines, it is necessary to have some method of measuring the power produced and expressing the results in standard engineering terms or parameters. A *parameter* is an expression in which a number of variables are combined into a factor which can be used as an indicator of performance.

4-1. Basic Power Measurements. In general, and as indicated in Article 1-8, the energy flow and energy losses through the engine are expressed as three distinct categories of power. They are indicated horsepower (ihp), friction horsepower (fhp), and brake horsepower (bhp). Ihp may be computed from the measurement of forces in the cylinder and bhp may be computed from the measurement of forces at the delivery point of the engine. The fhp is generally determined by driving the "non-firing" engine with an outside source of power. The quantity of power so required is considered to be the fhp. It may also be calculated as the difference between ihp and bhp, if these latter two are known.

$$\text{ihp} = \text{bhp} + \text{fhp} \quad \text{or,} \quad \text{fhp} = \text{ihp} - \text{bhp}.$$

The following articles will develop the usually employed formulae for the computation of power.

4-2. Indicated Mean Effective Pressure (imep). It was stated in Article 4-1 that ihp may be computed from a measurement of some factor indicative of the forces developed in the cylinder, namely, the pressure of the expanding gases.

During the study of cycles and the p-V diagrams connected therewith, it was seen that the pressures in the cylinder vary throughout the cycle. Such a continuous variation does not readily lend itself to simple mathematical analysis in the computation of ihp. If a constant pressure for one cycle is utilized, the computation is far less difficult, and the results are the same.

As the piston moves back and forth between TDC and BDC (Fig. 4-1), the process lines on the p-V diagram indicate the successive states of the working fluid through the cycle. The indicated net work of the cycle is represented by the area 1234 enclosed by the process lines for that cycle. If the area of rectangle $AA'B'B$ equals area 1234, the vertical distance between the horizontal lines AA' and BB' represents the **indicated mean effective pressure (imep).** It is a mean value expressed in pounds per square inch (lb/in^2) which, when multiplied by the piston displacement (PD), gives the same indicated net work as is actually produced with the varying pressures.

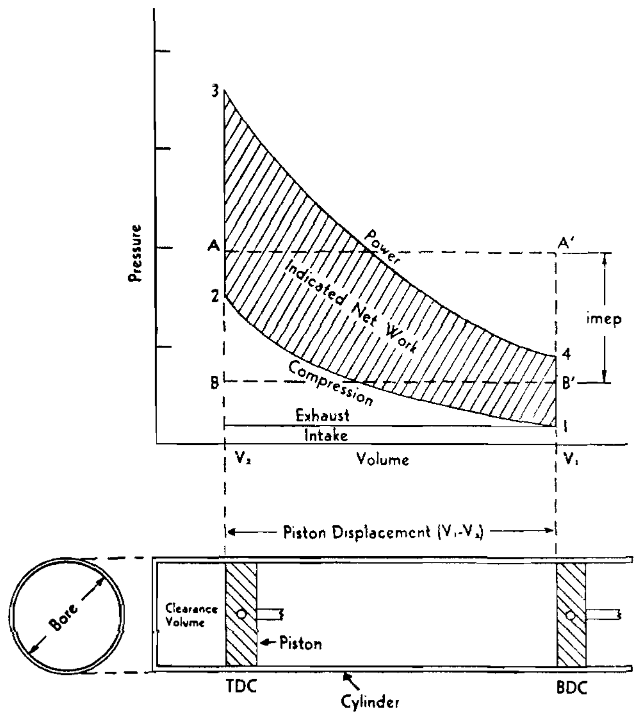


FIG. 4-1. Pressure-volume diagram for an ideal four-stroke cycle engine, showing imep and relation of p-V diagram to actual cylinder (not to scale).

Thus,

$$(\text{imep}) \times (\text{P.D.}) = \text{net wk of cycle} \tag{4-1}$$

or,

$$(\text{imep}) \times (V_1 - V_2) = \text{net wk of cycle}$$

therefore,

$$\text{imep} = \frac{\text{net wk of cycle}}{V_1 - V_2}. \tag{4-2}$$

With the units generally employed, equation 4-2 becomes,

$$\text{imep} = \frac{\text{net wk/cycle}\left(\dfrac{\text{Btu}}{\text{lb fluid}}\right) \times (778)\left(\dfrac{\text{ft-lb}}{\text{Btu}}\right)}{(v_1 - v_2)\left(\dfrac{\text{ft}^3}{\text{lb fluid}}\right) \times (144)\left(\dfrac{\text{in}^2}{\text{ft}^2}\right)}$$

$$= \frac{(\textbf{net wk/cycle})(\mathbf{778})}{(\boldsymbol{v_1 - v_2})(\mathbf{144})}\left(\frac{\textbf{lb}}{\textbf{in}^2}\right). \tag{4-3}$$

On an actual engine, the p-V diagram (called the **indicator diagram**) is obtained by a mechanical or electrical instrument attached to the cylinder. The area enclosed by the actual cycle on the indicator card may be measured by a planimeter. The value of the area measured, when divided by the piston displacement, results in the mean ordinate, or imep.

4-3. Indicated Horsepower (ihp). Power is defined as the rate of doing work. In the analysis of the *ideal* and *air* cycles (Chapter III), the net work is usually expressed in Btu per *pound* of air. This may be converted to power by multiplying by the weight rate of flow of air through the engine in *pounds* per unit time. Since the net work obtained from the p-V diagram is the net work produced in the cylinder as measured by an indicator diagram, the power based thereon is termed indicated power, or in terms of horsepower, **indicated horsepower (ihp).**

$$\text{Power} = (w_a) \times (\text{net wk})$$

or.

$$\text{ihp} = \frac{(w_a) \times (\text{indicated net wk})}{2545}$$

where w_a = weight rate of flow of air $\left(\dfrac{\text{lbs air}}{\text{hour}}\right)$

2545 = conversion factor $\left(\dfrac{\text{Btu}}{\text{hp-hr}}\right)$

and indicated net work is expressed as Btu/lb air.

The total heat supplied during a given period of time may be expressed as:

$$Q_S = (w_a) \times (q_S)\left(\frac{\text{Btu}}{\text{hr}}\right)$$

where w_a = weight rate of flow of air $\left(\dfrac{\textbf{lbs air}}{\textbf{hour}}\right)$

and $\qquad q_S$ = heat supplied per pound of air $\left(\frac{\text{Btu}}{\text{lb air}}\right)$.

In the *actual* cycle, the energy supplied to the engine is obtained through the liberation of the chemical energy in the fuel as the fuel is burned. The heat supplied during a given period of time is:

$$Q_S{}^{1} = (w_f) \times (HV)\left(\frac{\text{Btu}}{\text{hour}}\right)$$

where $\qquad w_f$ = weight rate of flow of fuel $\left(\frac{\text{lbs fuel}}{\text{hour}}\right)$

and $\qquad HV$ = heating value of fuel $\left(\frac{\text{Btu}}{\text{lb fuel}}\right)$.

In terms of potential horsepower supplied by the fuel, this becomes:

$$\text{hp}_S = \frac{(w_f) \times (HV)}{2545}.$$

The ratio of the indicated horsepower to the potential horsepower available in the fuel supplied is termed the indicated thermal efficiency (η_t).

$$\eta_t = \frac{\text{ihp}}{\text{hp}_S}.$$

In working with actual engines, it is often desirable to compute ihp from a given imep and given engine operating conditions. The necessary formula may be developed from the equation of net work based on the imep and piston displacement.

From equation 4-1

Indicated net work = (imep) (Piston displacement).

[1] Note that Q_s may be obtained from the following approximate expression:

$$(w_f) \times (HV) = (w_a) \times (q_S)$$

or

$$q_S = (F/A) \times (HV).$$

By definition

$$\text{Indicated Power} = \frac{\text{indicated net work}}{\text{time}}$$

$$= \frac{\text{(imep) (Piston displacement)}}{\text{time}}$$

and

$$\text{ihp} = \frac{p_i\left(\frac{\text{lb}}{\text{in}^2}\right)D\left(\frac{\text{in}^3}{\text{power stroke}}\right)N\left(\frac{\text{rev}}{\text{min}}\right)}{33{,}000\left(\frac{\text{ft-lb}}{\text{hp-min}}\right)n\left(\frac{\text{rev}}{\text{power stroke}}\right)12\left(\frac{\text{in}}{\text{ft}}\right)}$$

$$= \frac{p_i\,DN}{(33{,}000)(n)(12)}\ \text{(hp)} \qquad (4\text{-}4)$$

where p_i = imep (lb/in²)

D = **total** piston displacement of engine $= (V_1 - V_2) \times$ (number cylinders in engine) = (stroke) $\times$ (area of top of piston) $\times$ (number of cylinders in engine) $= \left(\frac{\text{in}^3}{\text{power stroke}}\right)$

N = engine speed $\left(\frac{\text{rev}}{\text{min}}\right)$

n = conversion factor to reduce N to number of power strokes per minute. The numerical value of n is 2 for a 4-stroke cycle engine and 1 for a 2-stroke cycle engine

$$\left(\frac{\text{rev}}{\text{power stroke}}\right).$$

In the commonly used form, equation 4-4 is expanded to become

$$\text{ihp} = \frac{p_i\left(\frac{\text{lb}}{\text{in}^2}\right)L\left(\frac{\text{in}}{\text{power stroke}}\right)A\left(\frac{\text{in}^2}{\text{cylinder}}\right)N\left(\frac{\text{rev}}{\text{min}}\right)C\ \text{(no. of cylinders)}}{33{,}000\left(\frac{\text{ft-lb}}{\text{hp-min}}\right)n\left(\frac{\text{rev}}{\text{power stroke}}\right)12\left(\frac{\text{in}}{\text{ft}}\right)}$$

or,

$$\text{ihp} = \frac{p_i\,LANC}{(33{,}000)(n)(12)}\ \text{(hp)} \qquad (4\text{-}5)$$

where p_i = imep $\left(\frac{\text{lb}}{\text{in}^2}\right)$ $\qquad$ N = engine speed $\left(\frac{\text{rev}}{\text{min}}\right)$

L = stroke (in) $\qquad$ A = area of piston $\left(\frac{\text{in}^2}{\text{cylinder}}\right)$

n = conversion factor to reduce N to power strokes per minute $= \left(\dfrac{\text{rev}}{\text{power stroke}}\right)$.

C = number of cylinders

Example Problem 4-1:

A nine cylinder 4-stroke cycle aircraft engine with a 6⅛″ bore and 6⅞″ stroke delivers 675 bhp at 1900 rpm while burning 67.5 lbs of fuel in 10 minutes. The A/F ratio is 15:1 and the ihp produced is determined to be 900 hp.

Find:

(a) imep
(b) fhp

Solution:

(a)
$$\text{ihp} = \frac{p_i\,LANC}{(33{,}000)(n)(12)}$$
$$p_i = \frac{(33{,}000)(n)(12)(\text{ihp})}{LANC}$$
$$= \frac{(33{,}000)(2)(12)(900)}{(6.875)\left(\frac{\pi}{4}\right)(6.125)^2(1900)(9)} = 206.0 \text{ lb/in}$$

(b)
$$\text{fhp} = \text{ihp} - \text{bhp}$$
$$= 900 - 675 = 225 \text{ hp.}$$

4-4. Brake Horsepower (bhp). Ihp is based on indicated net work and is thus a measure of the forces developed within the cylinder. Of more practical interest is the rotational force available at the delivery point of the engine crankshaft (termed the **driveshaft**), and the power connected therewith. This power is interchangeably referred to as brake horsepower, shaft horsepower, or delivered horsepower. In general, only the term **brake horsepower (bhp)** will be used in this book to indicate the horsepower actually delivered by the engine.

The bhp is usually measured by attaching a power absorption device to the driveshaft of the engine. Such a device sets up measurable forces counteracting the forces delivered by the engine, and the determined value of these measured forces is indicative of the forces being delivered.

Some of the various types of measuring devices are:

(a) The **prony brake**—Figure 4-3 is a schematic drawing of a prony brake. A wheel, attached rigidly to the engine driveshaft, is enveloped by an adjustable friction band. An arm, rigidly attached to the friction band, is free to move through a limited arc, and rests on a weighing scale. As the driveshaft and attached wheel rotate, the friction between the surrounding band and the wheel tends to rotate the arm, causing a force to be applied to the weighing scales. The rotating force of the engine can then be evaluated. This type of device is limited to low speed engines, but is convenient to use as a basis for the development of the bhp formula, and this will be accomplished in a succeeding paragraph of this article.

(b) The **fan brake.**—A fan arrangement, attached to the engine driveshaft, is utilized to provide a counteracting force due to the resistance offered to the fan by the air. The power required to drive the fan under various conditions may be separately calibrated. This device is neither easily adjusted nor extremely accurate, and is generally limited to use on long duration tests of engines.

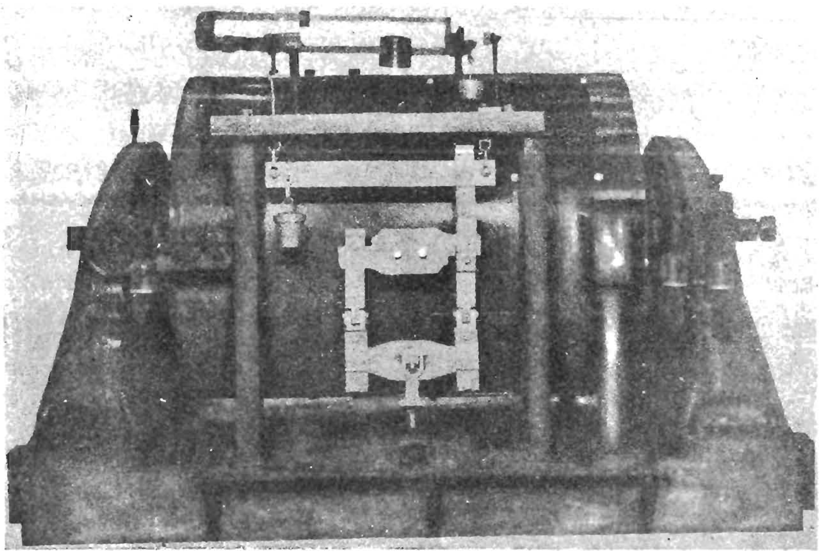


FIG. 4-2. Picture of a typical electric dynamometer (courtesy of Westinghouse Electric Corporation).

(c) The **water brake**—A cupped or vaned disc, attached to the engine driveshaft, is mounted in a cupped or vaned casing containing a supply of water. As the disc rotates, the resistance of the water tends to rotate the casing, which is free to move through a limited arc. An arm attached to the casing then applies a force to a weighing scale, and the turning force produced by the engine can be evaluated. This type of brake is particularly desirable for testing under conditions of heavy load and high speed.

(d) The **eddy current dynamometer**—In this type of brake, an electric current is usually passed through a metal disc attached to, and rotating with, the driveshaft of the engine. The disc is enclosed in a metal casing which is free to rotate through a limited arc. Eddy currents are set up in the non-electrified casing which create a force tending to rotate the casing in the same direction as the disc is rotating. A force is applied to a weighing scale by an arm attached to the casing, and the rotating force of the engine can thus be evaluated.

(e) The **electric dynamometer**—A direct current electric generator is often used as a power measuring device by connecting it to the driveshaft of the engine, measuring the generator electrical output, and correcting it for generator efficiency. The generator efficiency, however, varies with such factors as load, speed, and temperature, and accurate

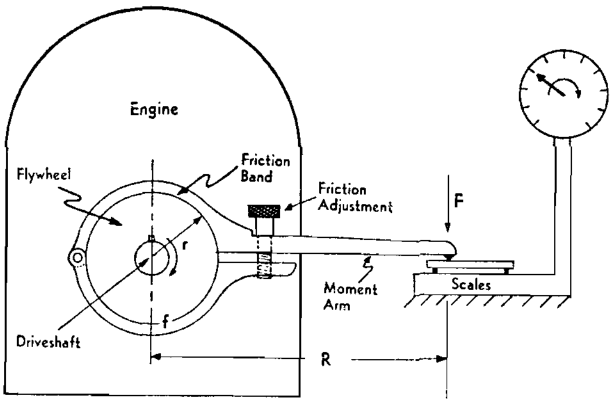


FIG. 4-3. Schematic diagram showing adaptation of prony brake for power measurement.

results are not readily obtained. By designing the stator of the generator so that it is free to rotate through a limited arc, and attaching it to a weighing device, the force produced by the engine can be evaluated more accurately. Such an arrangement is termed an electric dynamometer. The electric generator and electric dynamometer have the advantage that they can be used to motor the engine and thus estimate fhp, as will be described later, or they can be used to start the engine. The electric dynamometer is the most satisfactory measuring device, but is also the most expensive type of apparatus.

By using the geometry of a simple prony brake as the basis, a formula can now be developed for computing the bhp delivered by an engine. Work has been defined as the product of a force acting through a distance. If the driveshaft of the engine turns through one revolution, any point on the periphery of the rigidly attached wheel moves through a distance equal to $2\pi r$ (Fig. 4-3). During this movement, a friction force (f) is acting against the wheel. The force (f) is thus acting through the distance $2\pi r$, and producing work. Thus,

$$\text{Work during one revolution} = (\text{distance})(\text{force}) = (2\pi r)(f).$$

The turning moment (rf) produced by the driveshaft is opposed by a moment equal to the product of the length of the moment arm (R) and the force (F) measured by the scale.

$$rf = RF$$

$$\text{Work during one revolution} = 2\pi RF$$

$$\text{Power} = \frac{\text{Work}}{\text{Time}} = 2\pi RFN$$

where N = revolutions per minute of the driveshaft

$$\text{therefore, bhp} = \frac{2\pi R(\text{ft/rev})F(\text{lb})N\left(\frac{\text{rev}}{\text{min}}\right)}{(33{,}000)\left(\frac{\text{ft-lb}}{\text{hp-min}}\right)} = \frac{2\pi RFN}{33{,}000}(\text{hp}). \qquad (4\text{-}6)$$

It should be noted that the N factor in equation 4-6 is not modified by the number of power strokes per revolution of the engine, as is the case in equations 4-4, 4-5, and 4-7. The friction force is acting during *every* revolution of the crankshaft, regardless of whether or not that revolution contains a power stroke.

Equation 4-6 could have been developed using an electric dynamometer as the basis, by substituting the flux resistance of the dynamometer for the frictional resistance of the prony brake.

The product of the moment arm (R) and the measured force (F) is termed the torque of the engine, and is usually expressed in lb-ft. **Torque (T)** is the uniform or fluctuating turning moment, or twist, exerted by a tangential force acting at a distance from the axis of rotation. For an engine operating at a given speed and delivering a given horsepower, the torque must be a fixed amount, or the product of (F) and (R) must be constant ($T = FR$). In such a case, if (R) is decreased, (F) will have to be increased proportionately, and vice versa. In practice, the length of the moment arm (R) of the measuring equipment is so designed that the value of the constants 2π, 33,000, and the constant (R) combine to give a round number (i.e., in even hundreds or thousands) in order to simplify computations.

Torque is the capacity of an engine to do work, while power is the rate *at which an engine does work.* A simple example is that of a tractor pulling a given load. The torque developed will determine whether or not the tractor is *capable* of pulling the load, and the power delivered will determine how *fast* the load can be pulled.

Example Problem 4-2:

A ten cylinder, 2-stroke cycle diesel engine with an $8\frac{1}{2}''$ bore and a $20''$ stroke delivers 1800 bhp at 800 rpm.

Find:

(a) Torque

Solution:

(a)
$$\text{bhp} = \frac{2\pi RFN}{33{,}000} = \frac{2\pi TN}{33{,}000}$$
$$T = \frac{33{,}000\ (\text{bhp})}{2\pi N}$$
$$= \frac{(33{,}000)(1800)}{(2\pi)(800)} = 11{,}820\ \text{lb-ft.}$$

4-5. Friction Horsepower (fhp). In article 1-8, it was indicated that some of the power which is produced in the cylinder is not delivered at the driveshaft. This power, which is the difference between the power produced (ihp) and the useful power delivered (bhp), is termed **friction horsepower** (fhp).

$$\text{ihp} - \text{bhp} = \text{fhp}.$$

Some of the fhp is utilized to overcome friction in the pistons, bearings, cams, and a multitude of other moving parts, and some is required to overcome the "pumping losses" concerned with the induction of the fresh charge and the expulsion of the exhaust gases. Any one of these sources of friction loss is usually not large in itself, but the total effect can sometimes be considerable.

Due to the multiplicity of sources of the friction losses and the variations in their values with changing operating conditions, fhp is difficult to measure accurately. The total fhp can be determined by measuring the bhp, computing the ihp from the engine indicator card as mentioned in Article 4-2, and subtracting the former from the latter. This procedure requires expensive equipment and entails much time and effort. It is generally more practical and convenient to approximate fhp by use of the electric dynamometer. The bhp is measured by the dynamometer while the engine is "firing," and then the engine is shut off and the dynamometer used as a motor to drive the engine under, as nearly as possible, the same operating conditions. The amount of power required to "motor" the engine is then the estimated fhp. Certain errors in the friction losses arise with this procedure, but the effect of some errors apparently counteracts the effect of others. The resultant effect is reasonably close to the actual conditions existing, hence this procedure is widely used throughout industry. In fact, the standard procedure for determining ihp is to combine the fhp and bhp so measured, rather than to compute ihp through the use of indicator card diagrams taken at the cylinder.

4-6. Brake Mean Effective Pressure (bmep). Imep may be considered to consist of fmep and bmep, two hypothetical pressures. Fmep is that portion of imep which is required to overcome friction losses, and bmep is the portion which produces the useful power delivered by the engine.

$$\text{imep} = \text{bmep} + \text{fmep}.$$

Since bmep is that portion of imep which goes into the development of useful power, it has the same relationship to bhp as imep has to ihp,

or
$$\frac{\text{bmep}}{\text{imep}} = \frac{\text{bhp}}{\text{ihp}}\,.$$

Equation 4-5 was developed as a means of computing ihp when imep is determined from an engine indicator card.

$$\text{ihp} = \frac{p_i\,LANC}{(33{,}000)(n)(12)}\,(\text{hp}). \tag{4-5}$$

For a given engine operating under given conditions, L, A, N, C, and n are constants. Since bhp and bmep have the same relationship to one another as do ihp and imep, equation 4-5 can be written as

$$\text{bhp} = \frac{p_b\,LANC}{(33{,}000)(n)(12)}\,(\text{hp}) \tag{4-7}$$

where p_b = brake mean effective pressure (lb/in^2).

And due to the same relationship, the mechanical efficiency (η_m) of the engine can be expressed as the ratio of bmep to imep.

$$\eta_m = \frac{\text{bhp}}{\text{ihp}} = \frac{\text{bmep}}{\text{imep}}\,. \tag{4-8}$$

It should be noted that for a given engine operating under given conditions, the torque developed is proportional to the bmep. This relationship may be obtained by equating formulas 4-6 and 4-7, and noting that $T = FR$.

Bmep is very useful in comparing engines or in establishing engine operating limits.

4-7. Specific Fuel Consumption (sfc). One of the most important parameters used in the comparison of engines, and one which is based on the power produced or delivered, is the specific fuel consumption (sfc). **Specific fuel consumption** is the ratio of the number of pounds of fuel used by the engine, per hour, to the horsepower produced or delivered by the engine.

$$\text{sfc} = \frac{w_f}{\text{hp}} = \frac{\text{lb fuel}}{\text{hp-hr}} \tag{4-9}$$

where w_f = lb fuel/hr.

When sfc is based on the ihp produced, it is termed **indicated specific fuel consumption** (isfc).

$$\text{isfc} = \frac{w_f}{\text{ihp}} = \frac{\text{lb fuel}}{\text{ihp-hr}} \cdot \tag{4-10}$$

When sfc is based on the bhp delivered, it is termed **brake specific fuel consumption** (bsfc)

$$\text{bsfc} = \frac{w_f}{\text{bhp}} = \frac{\text{lb fuel}}{\text{bhp-hr}} \cdot \tag{4-11}$$

Example Problem 4-3:

A six cylinder, 2-stroke cycle marine diesel engine with a $4\frac{1}{4}''$ bore and $5''$ stroke delivers 225 bhp at 2100 rpm while burning 112.2 lbs of fuel per hour. Ihp is 275 hp.

Find:

(a) Torque
(b) bmep
(c) bsfc
(d) imep
(e) isfc
(f) mechanical efficiency
(g) fhp

Solution:

(a) $$\text{bhp} = \frac{2\pi RFN}{33{,}000}$$

and since $T = RF$

then $$T = \frac{(33{,}000)(\text{bhp})}{2\pi N} = \frac{(33{,}000)(225)}{(2\pi)(2100)} = 564 \text{ lb-ft.}$$

(b) $$\text{bhp} = \frac{p_b\,LANC}{(33{,}000)(n)(12)}$$

$$p_b = \frac{(33{,}000)(n)(12)(\text{bhp})}{LANC} = \frac{(33{,}000)(1)(12)(225)}{(5)\left(\frac{\pi}{4}\right)(4.5)^2(2100)(6)} = 89\ \text{lb/in}\ .$$

(c) $$\text{bsfc} = \frac{\text{lb fuel}}{\text{bhp-hr}} = \frac{112.2}{225} = 0.499\ \text{lb fuel/bhp-hr}$$

(d) $$\frac{\text{imep}}{\text{bmep}} = \frac{\text{ihp}}{\text{bhp}}$$

$$\text{imep} = (89)\frac{(275)}{(225)} = 109\ \text{lb/in}^2.$$

(e) $$\text{isfc} = \frac{\text{lb fuel}}{\text{ihp-hr}} = \frac{112.2}{275} = 0.408\ \text{lb fuel/ihp-hr}$$

(f) $$\eta_m = \frac{\text{bhp}}{\text{ihp}} = \frac{225}{275} = 0.82 \text{ or } 82 \text{ per cent}$$

(g) $$\text{fhp} = \text{ihp} - \text{bhp} = 275 - 225 = 50\ \text{hp}.$$

4-8. Interrelationship of Some Problem Variables. Many of the formulae used in this book can be expressed in a variety of forms. To memorize all possible variations of a basic formula would be placing an unnecessary burden on the mind of the student. The logical method is for the student to learn and understand the basic formulae. If the particular arrangement of the basic formula is not in the form desired for the solution of a problem, whether it be a practice problem or an actual laboratory exercise, it can easily be rearranged to suit the occasion. But *the student must know the basic formulae and understand the interrelationship of the variables concerned.* Part (b) of the solution to Example Problem 4-3 is a simple illustration. Equation 4-5 is considered a basic formula which the student should know. By substituting bhp for ihp, and p_b for p_i, and rearranging, the formula necessary for the solution is readily obtained in a desirable form. It is not necessary to memorize this formula since it can be so easily determined by knowing equation 4-5 and understanding the variables involved.

The relationship between air, fuel, and air-fuel ratio is another common example. An actual engine test set up might readily allow for an accurate measurement of the rates of air flow and fuel flow, from which A/F ratio can be easily determined. If the arrangement were such that the rate of fuel flow and the A/F ratio were given, it would be an easy matter to compute the rate of air flow. It is necessary, however, to understand what A/F ratio is, and the relationship of the variables involved.

The above are only examples of a very few of the approaches to a problem, many more of which could be cited. The point to remember is that it is not necessary for the student to clutter up his mind with several variations of the same basic formula. If the student learns the basic formulae and understands the relationship of the variables concerned, the problems in the classroom, in the laboratory, or in the field can be solved with a minimum of effort. The following example problems, and the problems at the end of this chapter, should serve to illustrate these principles.

Example Problem 4-4:

An internal combustion engine develops 200 hp to overcome friction and delivers 1200 bhp. Air consumption is 180 lbs per minute. A/F ratio is 15 to 1.

Find:

(a) ihp
(b) mechanical efficiency
(c) isfc

Solution:

(a) $$\text{ihp} = \text{bhp} + \text{fhp} = 1200 + 200 = 1400 \text{ hp}$$

(b) $$\eta_m = \frac{\text{bhp}}{\text{ihp}} = \frac{1200}{1400} = 0.8575 \text{ or } 85.8 \text{ per cent}$$

(c) $$\text{sfc} = \frac{w_f}{\text{ihp}}$$

$$A/F = \frac{w_a}{w_f}$$

therefore $$w_f = \frac{w_a}{A/F} = \frac{(180)(60)}{15} = 720 \frac{\text{lb fuel}}{\text{hr}}$$

$$\text{sfc} = \frac{720}{1400} = 0.515 \text{ lb fuel/ihp-hr.}$$

Example Problem 4-5:

An engine operating at a given load produces 190 ihp and the mechanical efficiency is known to be 85 per cent. The engine uses 26.2 lb of gasoline during a 20 minute test run, with an air rate of 18.5 lb/min. The heating value of the gasoline is 19,000 Btu/lb.

Find:

(a) A/F ratio
(b) indicated thermal efficiency
(c) bsfc
(d) fhp

Solution:

(a) $w_f = 3(26.2) = 78.6$ lb fuel/hr

$w_a = (18.5)(60) = 1110.0$ lbs air/hr

$$A/F = \frac{1110.0}{78.6} = 14.12 \text{ to } 1$$

(b) $$\text{Heat supplied} = \frac{26.2}{20}(19{,}000) = 24{,}870 \text{ Btu/min}$$

$$\text{Net power} = (\text{ihp})(33{,}000)\left(\frac{1}{778}\right)$$

$$\eta_t = \frac{\text{Net work}}{\text{Heat supplied}} = \frac{(190)(33{,}000)}{(778)(24{,}870)} = 0.324 \text{ or } 32.4 \text{ per cent}$$

c) $$\eta_m = \frac{\text{bhp}}{\text{ihp}}$$

$$\text{bhp} = (\eta_m)(\text{ihp}) = (0.85)(190) = 161.6 \text{ hp}$$

$$\text{bsfc} = \frac{w_f}{\text{bhp}} = \frac{78.6}{161.6} = 0.487 \text{ lb fuel/bhp-hr}$$

(d) $$\text{fhp} = \text{ihp} - \text{bhp}$$

$$= 190 - 161.6 = 28.4 \text{ hp.}$$

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

4-1. S. J. Young and R. W. J. Pryer, "*The Testing of Internal Combustion Engines,*" D. Van Nostrand Company, Inc.

4-2. Lester C. Lichty, "*Internal Combustion Engines,*" McGraw-Hill Book Company, Inc.

4-3. Edward F. Obert, "*Internal Combustion Engines Analysis and Practice,*" International Textbook Company.

4-4. Arthur P. Fraas, "*Combustion Engines,*" McGraw-Hill Book Company, Inc.

EXERCISES

4-1. What is imep? bmep? fmep?

4-2. Differentiate between ihp, bhp, and fhp.

4-3. What is torque?

4-4. How is fhp generally determined?

4-5. A 4-stroke cycle CI engine is delivering 600 bhp while running at 2400 rpm. Total engine piston displacement is 1650 cubic inches and the A/F ratio is 22 to 1. The engine uses 69.5 pounds of air per minute.

Find: (a) bmep
(b) Torque
(c) bsfc

Ans: (a) 120 lb/in² (b) 1311 lb-ft (c) 0.315 lb fuel/bhp-hr

4-6. An engine is using 9.1 pounds of air per minute while operating at 1200 rpm. The engine requires 0.352 pound of fuel per hour to produce 1 ihp. The

A/F ratio is 14 to 1, indicated thermal efficiency is 38 per cent, and mechanical efficiency is 82 per cent.

Find: (a) bhp (b) Heating value of fuel used
Ans: (a) 90.9 hp (b) 19,025 Btu/lb fuel

4-7. A twelve cylinder 4-stroke cycle marine gasoline engine, with a $6\frac{3}{8}$ inch bore and $6\frac{1}{2}$ inch stroke, delivers 1200 bhp at 2400 rpm while burning 126.8 gallons of fuel per hour. The fuel weighs 6.2 pounds per gallon. The engine produces 1500 ihp.

Find: (a) bmep (b) bsfc (c) imep (d) η_m
Ans: (a) 159 lb/in² (b) 0.655 lb fuel/bhp-hr (c) 198.8 lb/in² (d) 80%

4-8. An engine with an A/F ratio of 13.5 to 1, using a fuel with a heating value of 19,000 Btu per pound, delivers 450 bhp. It uses 90 pounds of fuel during a 30 minute run. From test data, the indicated thermal efficiency is found to be 39 per cent.

Find: (a) Mechanical efficiency
(b) Amount of heat rejected per hour
(c) bsfc
(d) Air flow (lb/min)
Ans: (a) 85.9% (b) 2,087,000 Btu/hr (c) 0.4 lb fuel/bhp-hr (d) 40.5 lb air/min

4-9. A 2-stroke cycle CI engine has 14 cylinders and a total engine piston displacement of 4500 cubic inches. While operating at a speed of 1620 rpm, it produces a continuous torque reading of 5960 lb-ft. At this speed, the engine is known to have a brake specific fuel consumption of 0.61 lb/bhp-hr.

Find: (a) bhp
(b) bmep
(c) Time to use 5000 gallons of fuel (specific gravity = 0.84)
Ans: (a) 1840 hp (b) 100 lb/in² (c) 31.2 hr.

CHAPTER V

FUELS

During the study of cycles in Chapter III, frequent mention was made of the term "heat supplied," which, in an actual engine, is obtained from the combustion of fuel. The chemical combination of the fuel with the oxygen in the combustion chamber produces heat, which the engine converts into a mechanical form of energy. Since the heat supplied to an actual engine is obtained from the fuel used, it is desirable to have a fundamental knowledge of fuel types and their characteristics. Furthermore, it should be understood that the use of a particular fuel in internal combustion engines is determined not only by its physical and chemical characteristics, but also by its availability and cost of production.

5-1. Introduction. Internal combustion engines use fuels in either the solid, gaseous, or liquid state.

The use of solid fuels presents problems of complicated injection systems as well as difficulties associated with solid residue or ash. Solid fuels consequently find little practical application at present, although considerable experimental work is being conducted on the use of this type of fuel.

Gaseous fuels create problems concerned with the storage and handling of large volumes, thus entailing the use of large and bulky storage tanks. This seriously restricts the use of such fuels on mobile equipment. Such fuels, however, do reduce or eliminate many of the problems of starting and distribution that are encountered with liquid fuels. Consequently, gaseous fuels do find considerable use in stationary power plants located near an abundant supply of this type of fuel. Some gaseous fuels can be liquefied under pressure, thereby reducing the storage tank problem, but this arrangement is presently rather costly to find very wide usage.

Most internal combustion engines, therefore, utilize liquid fuels which are derivatives of petroleum. In some areas where shortages of petroleum exist, liquid fuel similar to that derived from petroleum is processed from coal. Such a procedure is expensive, but as petroleum reserves become depleted, this type of processed fuel will undoubtedly become more important. The three principle commercial types of liquid fuels are:

(1) *Benzol* is a by-product of high temperature coal carburization and consists principally of benzene (C_6H_6) and touluene (C_7H_8). It has

a high anti-knock quality but a lower heating value than gasoline, and in this country is used to a very limited extent as a blending agent for gasoline. In certain other areas of the world where petroleum supplies are less abundant, benzol is often used as a basic fuel.

(2) *Alcohol,* which is somewhat similar to the derivatives of petroleum but contains oxygen atoms in the molecule. When used as a fuel, at the present time, it is usually blended with gasoline. It has good anti-knock qualities, but has a lower heating value than gasoline and is presently more expensive to produce. Furthermore, when blended with gasoline, it tends to absorb water from the atmosphere, causing separation of the gasoline and alcohol into two layers at a certain water content in the mixture and at a given temperature. However, alcohol can be manufactured from grain and certain waste products, and as petroleum supplies become depleted, it will possibly become an increasingly important fuel.

(3) *Refined products of petroleum,* then, are the major source of liquid fuels at the present time. It is this type of fuel, such as gasoline, kerosene, and diesel oil, to which attention will be primarily directed in this book.

5-2. Petroleum Base Liquid Fuels. Crude petroleum consists of a mixture of large numbers of hydrocarbon compounds differing widely in molecular structure, plus varying small amounts of sulphur, oxygen, nitrogen, and impurities such as water and sand. The specific characteristics of a particular fuel vary widely with its composition. For purposes of comparison, it is desirable to arrange these hydrocarbon compounds into families based on the hydrogen and carbon arrangement within the molecule. Due to the inherent complexity of petroleum base fuels, however, it is difficult to assign a particular fuel entirely to one family. Most petroleum base fuels, therefore, will tend to exhibit the characteristics of that type of hydrocarbon which forms a major portion of the fuel.

TABLE 5-1

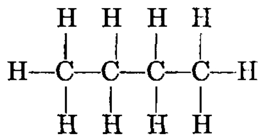
PRIMARY HYDROCARBON FAMILIES IN PETROLEUM

Family	*General Formula*	*Molecular Arrangement*
Paraffin	C_nH_{2n+2}	Chain
Olefin	C_nH_{2n}	Chain
Diolefin	C_nH_{2n-2}	Chain
Naphthene	C_nH_{2n}	Ring
Aromatic	C_nH_{2n-6}	Ring

FUELS

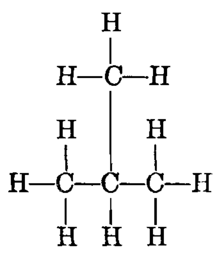
Table 5-1 shows the basic families of fuels, their general formulae, and their molecular structure. A brief discussion of the structure and characteristics of each family follows.

(1) *Paraffins*—The normal paraffin hydrocarbons consist of a straight chain (open chain) molecular structure. A schematic diagram of the presumed molecular arrangement of a typical member of this family, butane, is presented herewith.

Butane 

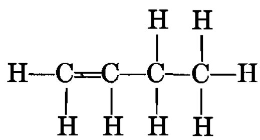
It should be noted that each carbon atom contains four attaching bonds, while each hydrogen atom has one attaching bond. The number of bonds which an atom contains is termed the *valence* of that atom, and is indicative of the atom's combining capacity. Thus carbon has a valence of four, and hydrogen a valence of one. In normal paraffins, the valence of each carbon atom is fully utilized in combining, by a single bond, with other carbon atoms and with hydrogen atoms. *Straight chain paraffins* are, therefore, termed saturated compounds and are characteristically very stable.

Another variation of the paraffin family consists of an open chain structure with an attached branch, and is usually termed a *branch chain paraffin.* Isobutane, shown below, is an example of this type. It has the same general chemical formula and molecular weight as butane, but a different molecular structure and different physical characteristics, and hence is called an *isomer* of butane.

Isobutane 

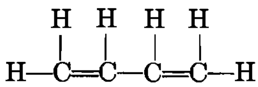
This is also a saturated compound and is very stable. The branch chain paraffins have good anti-knock qualities when used as SI engine fuels.

(2) *Olefins* are chain compounds similar to paraffins, but are unsaturated because they contain one double carbon to carbon bond. A typical example is butene.

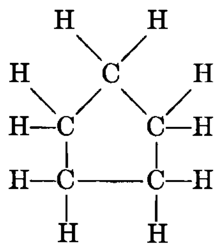
Butene 

Olefins are not as stable as the single bond paraffins due to the presence of the double bond.

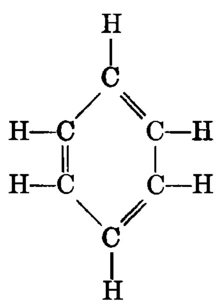
(3) *Diolefins* are essentially olefins with two double bonds. They are unsaturated and rather unstable, tending to form gum deposits during storage. A typical diolefin is butadiene.

Butadiene 

(4) *Naphthenes* have the same general formula as olefins, but have a ring structure. They are often termed cyclo-paraffins, are saturated, and tend to be stable. Cyclopentane is a typical naphthene.

Cyclopentane 

(5) *Aromatics* are ring structure compounds based on the benzene ring, shown below. While the double bonds indicate unsaturation, a peculiar nature of these bonds causes this family to be more stable than the other unsaturated families. A typical example of this family, and the basic structural element of all members, is benzene.

Benzene 

A few of the general characteristics exhibited by these families due to their molecular structure are summarized below:

(1) The anti-knock quality of a fuel when used in a SI engine appears to be poorest in the normal paraffins and improve generally in the order in which the families are presented above. The aromatics offer the best resistance to detonation in SI engines.
(2) The suitability of these fuels for CI engines is in the inverse order of their suitability for SI engines. For CI engines, the normal paraffins are the better fuels, and aromatics are the least desirable.
(3) In general, as the number of atoms in the molecular structure increases, the boiling point temperature rises. Thus fuels with fewer atoms in the molecule tend to be more volatile.
(4) As the proportion of hydrogen atoms to carbon atoms in the molecule increases, the heating value generally increases. This is because hydrogen has a greater heating value than carbon. Paraffins thus tend to have the greatest heating value and aromatics the least heating value of the above listed families of fuels.

It is interesting to note that in this country, the petroleum from the eastern area contains relatively large portions of the paraffin type compounds, and its derivatives are sometimes termed *paraffin-base* fuels. Petroleum from the western area tends to run somewhat lower in paraffin types with a corresponding increase in the naphthene types. Due to the asphalt residue left after distillation, fuels derived from western petroleum are sometimes referred to as *asphalt-base* fuels. Petroleum from the mid-west tends to be a compromise between eastern and western petroleum, and the derivatives are termed *mixed-base* fuels.

5-3. Petroleum Refining. Crude petroleum is rarely used as fuel for internal combustion engines. It is processed or refined so as to produce desirable commercial products. While the modern refinery is a very complex chemical processing plant, it is nevertheless based on the simple fact that the constituents of crude petroleum have different boiling points varying roughly with their molecular weight.

Fractional distillation is the basic refining process used to separate the crude petroleum into more desirable products or fractions. The number of detailed systems used for accomplishing this separation is nearly as large as the number of plants involved, but the basic principle of all the systems is much the same. The process most widely used today entails the use of a *fractionating tower*. As the petroleum is taken from the ground, the liquid contains a certain amount of "wet" gas. By passing the petroleum through a separator, a product known as *natural gasoline* is absorbed from the wet gas and removed. The resultant "dry" gas is known as *natural gas*. The liquid petroleum then goes to a still where it is vaporized by raising the temperature to about 600°F, and the vapor passes to the fractionating tower. As it rises through the tower, it is forced to pass along a laborious path consisting of a labyrinth-like arrangement of plates, or similar obstacles, which direct the vapor through trays of liquid fuel at different temperatures. These trays are located at definite predetermined levels in the tower, and the temperature of the absorbing liquids in each tray is gradually lowered in proceeding from the lower to the upper levels of the tower. The compounds with higher boiling points are separated out at lower levels while those with lower boiling points are removed at higher levels. The condensed fractions in each tray are tapped off continuously. In general terms, the top fraction is called *straight run gasoline*, and the other fractions in descending order through the tower are *naphtha, kerosene, gas oils, lubricating stocks,* and *residue*. Each of these fractions covers a certain boiling point range, and each may be further refined by again fractionating within a narrow range of boiling points.

The yield of some of the petroleum products from the fractional distillation process does not always coincide with the commercial demand for such products. Economic necessity usually dictates the need for conversion of some of the products in small demand into products for which the demand is greater. To cope with this situation, various processes can be used to convert some of these fractions to compounds for which there is a greater need. Some of these processes are briefly mentioned herewith:

(1) *Cracking* consists of breaking down large and complex molecules into lighter and simpler compounds with lower boiling points. *Thermal cracking* subjects the heavy hydrocarbons to rather high temperatures and pressures. *Catalytic cracking* is accomplished at somewhat lower temperatures and pressures, but in the presence of a catalyst, and generally produces a fuel with higher anti-knock qualities.

(2) *Hydrogenation* differs from the cracking process in that hydrogen atoms are added to certain hydrocarbons, under high pressure and temperature, to produce more desirable compounds. It is often used to convert unstable to stable compounds.

(3) *Polymerization* brings together light, unsaturated gases of one family, in the presence of a catalyst, to produce a liquid.

(4) *Alkylation* combines light gases of different families in the presence of a catalyst. Generally an olefin is combined with a paraffin in this process to give a branch chain paraffin.

(5) *Isomerization* changes the relative position of the atoms within the molecule of a hydrocarbon without changing its molecular formula. It produces isomers of the original hydrocarbon.
(6) *Cyclization* essentially joins together the ends of a straight chain molecule to form a ring compound of the naphthene family.
(7) *Aromatization* is a process similar to cyclization except that the product is an aromatic compound.
(8) *Reforming* is a type of cracking process in which naphtha or straight run gasoline is converted into gasolines of higher octane rating.
(9) *Blending* is a process of mixing certain refinery products to obtain a commercial product of desired quality.

5-4. Heating Value of Fuels. The energy content, or **heating value (HV)** of a fuel varies with the type of fuel and approximately in accordance with the hydrogen to carbon ratio. It is expressed in Btu per pound. Since hydrogen has a HV of about 61,000 Btu per pound while that of carbon is only about 14,500 Btu per pound, the greater the proportion of hydrogen in a fuel, the greater its HV.

Water vapor is one of the products of combustion of hydrocarbon fuels. The heating value of the fuel depends upon the state of the water at the end of combustion, i.e., whether vapor or liquid. As a result, two heating values are used. The **higher heating value (HHV)** includes the energy made available by condensing the water vapor, while the **lower heating value (LHV)** does not include this energy. The temperature of the exhaust gas in a combustion engine is higher than that at which water vapor condenses. As a result, it is not possible, in an internal combustion engine, to utilize the energy obtained through condensation of the water vapor and, hence, the LHV is ordinarily used in computing engine performance.

5-5. Ratings of SI Engine Fuels. Hydrocarbon fuels used in SI engines have a tendency, when engine operating conditions become severe, to cause engine knock. (This phenomenon will be discussed in Chapter VIII.) The severity of operation is a function of such factors as engine load, speed, spark advance, A/F ratio, and temperature in the latter stages of combustion. For given engine operating conditions, the temperature towards the end of combustion is a function of compression ratio. As a result, a given fuel will have an increasing tendency to knock with increasing compression ratio.

Fuels differ widely in their ability to resist knock, and much effort has been expended to determine a satisfactory standard for comparing the anti-knock qualities of the various fuels. Fuel knock rating specifications presently employed require the use of standard engines operating under prescribed conditions. The rating of a particular fuel is accomplished by comparing its performance with that of a standard reference fuel which is usually a combination of iso-octane and normal heptane, or iso-octane plus tetraethyl lead.

Iso-octane, being a very good anti-knock fuel, is arbitrarily assigned a rating of 100 octane number. Normal heptane, on the other hand, has very poor anti-knock qualities and is given a rating of 0 octane number. By mixing these two fuels in varying proportions (by volume), any number of blends may be obtained which will have anti-knock qualities varying from poor to good, or from 0 to 100 octane number. The percentage, by volume, of iso-octane in a mixture of iso-octane and normal heptane, which exactly matches the knocking intensity of a given fuel, in a standard engine under given standard operating conditions, is termed the *octane number rating* of that fuel. Thus, if a mixture of 50 per cent iso-octane and 50 per cent normal heptane matches the fuel under test, this fuel is assigned an octane number rating of 50. If a fuel matches in knocking intensity a mixture of 75 per cent iso-octane and 25 per cent normal heptane, this fuel would be assigned an octane number rating of 75. Thus, *the octane number rating is an expression which indicates the ability of a fuel to resist knock in a SI engine*. Since a given fuel can exhibit different anti-knock qualities in different engines and under different operating conditions, octane number rating may require determination under more than one condition to be precise, and is not as exact a standard as is desired. But considering the many variables involved, it is as good an indicator as has yet been devised, and does provide a reasonably good standard reference.

Iso-octane and normal heptane are known as *primary reference fuels.* It is sometimes more economical to calibrate other less expensive but stable fuels against the primary fuels and use these to match the fuel under test. Such fuels are known as *secondary reference fuels.*

The higher the octane number rating of a fuel, the greater will be its resistance to knock, and the higher will be the compression ratio which may be used without knocking. Since the power output and specific fuel consumption are functions of compression ratio, it can be said that these are also functions of the octane number rating. This fact indicates the extreme importance of the octane number rating in fuels for SI engines.

Since the establishment of the octane scale, fuels superior in anti-knock qualities to iso-octane have become increasingly important. It has been known for some time that the addition of certain soluble compounds of lead to gasoline will produce marked effects in reducing knock. The addition of various amounts of one of these compounds (tetraethyl lead) to iso-octane produces fuels of greater anti-knock quality than iso-octane alone. The anti-knock quality of fuels above 100 octane number is, in this country, *measured* in terms of milliliters (cubic centimeters) of tetraethyl lead per U. S. gallon of iso-octane.

As shown in Fig. 5-1, the anti-knock effectiveness of tetraethyl lead, for the same quantity of lead added, decreases as the total content of lead in the fuel increases. In other words, one milliliter of lead added to

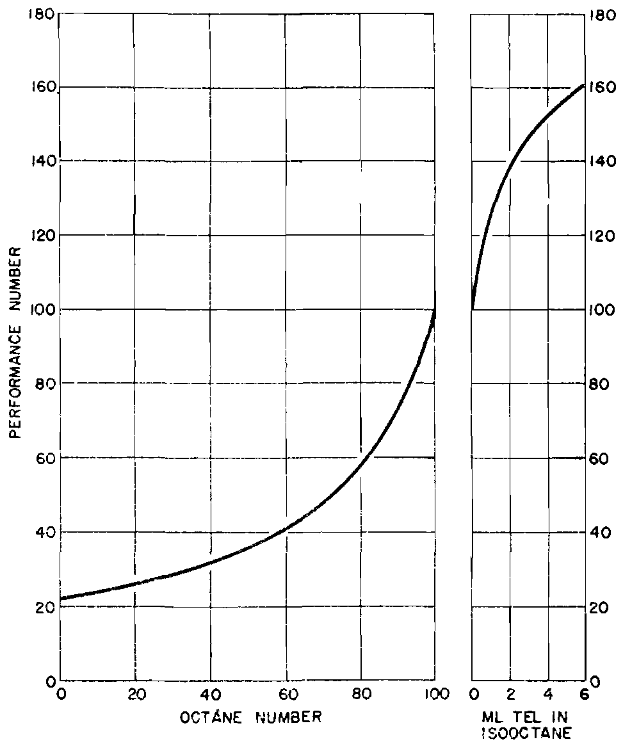


FIG. 5-1. Performance number equivalents for octane numbers and tetraethyl lead in iso-octane (courtesy of Ethyl Corporation).

pure iso-octane will have considerably greater anti-knock effect in comparison to the case when the same quantity of lead is added to iso-octane already containing 4 milliliters of lead. Furthermore, each octane number at the higher range of the octane scale will produce greater anti-knock effect compared to the same unit at the lower end of the scale. For instance, one octane number increase between, say 90 and 91, will produce greater anti-knock effect than a similar increase between, say 30 and 31 octane number. Because of this non-linear variation, a new scale was devised which *expresses* the approximate relative engine performance, and the units of this scale are known as **performance numbers** (PN).

The performance number (PN) of a fuel is based on the indicated power produced in a supercharged standard engine operating under prescribed conditions. It is indicative of the maximum power which may be obtained with that fuel, without knock, relative to the maximum power which may be obtained using iso-octane, also without knock. Iso-octane is arbitrarily assigned a PN of 100. A supercharged engine using a fuel rated at 130 PN, for instance, can produce approximately 1.3 times as much power (without knock) as it can develop with a 100 PN fuel, namely, iso-octane (without knock).

Usually, basic knock ratings are obtained in terms of octane number or iso-octane plus tetraethyl lead and the performance number calculated, or obtained from a graph such as Fig. 5-1.

The knock rating of a fuel of anti-knock quality below that of iso-octane is usually designated by octane number, but may be expressed in terms of an equivalent *PN* as indicated by Fig. 5-1. Fuels superior to iso-octane in anti-knock quality are correctly designated in terms of PN. Iso-octane may be designated as either 100 octane number or 100 PN.

Gasoline ratings are often expressed in terms of a symbol such as 115/145. This means that the performance number of this gasoline when knock-rated with a lean A/F ratio is 115, and when with a rich A/F ratio is 145.

5-6. Important Qualities of SI Engine Fuels. The fuel used in most present day SI engines is gasoline. Gasoline is usually a blend of several refinery products containing paraffins, naphthenes, and aromatics in varying proportions. The particular blend depends upon the desired characteristics of the fuel. Some of the more important qualities to be considered in the selection of a gasoline for use in a SI engine are:

(1) Volatility
(2) Gum deposits
(3) Sulphur content
(4) Anti-knock quality

A discussion of each of these qualities follows.

(1) *Volatility.* Since gasoline is a mixture of many hydrocarbons with different boiling points, the constituents will boil off at a wide range of temperatures. The usual practice for measuring the volatility characteristics of a fuel is a distilling method standardized by the

American Society for Testing Materials (ASTM). The graphical results of this test are generally referred to as ASTM distillation curves. Figure 5-2 is a typical diagram of such a curve. From the distillation curve for a particular gasoline, a determination can be made as to whether or not that gasoline will meet the volatility requirements of an engine.

Volatility is one of the most important properties of a gasoline,

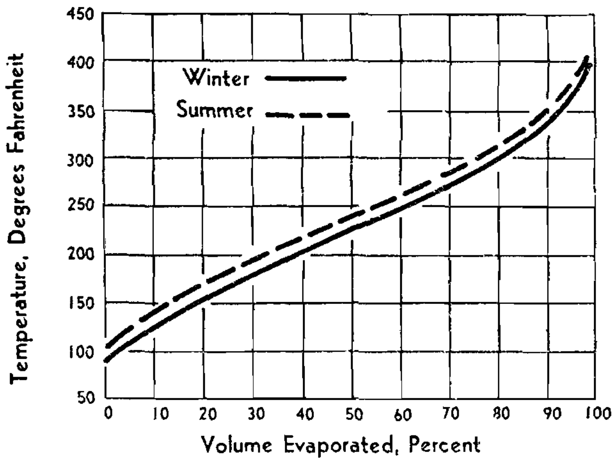


FIG. 5-2. ASTM distillation curves of typical winter and summer grade regular gasoline (courtesy of Ethyl Corporation).

since it exerts important effects on several phases of the operation and maintenance of the engine. The more important aspects of volatility, discussed in conjunction with the distillation curve of Fig. 5-2, are:

(a) *Starting and warmup*—For ease in starting, it is necessary to have some of the gasoline vaporize at the starting temperatures. It is thus *desirable* that the portion of the distillation curve between about 0 per cent and 10 per cent boiled off have relatively low boiling temperatures. As the engine warms up, the temperatures will gradually increase to the operating temperature. For best warm up, low distillation temperatures are *desirable* throughout the range of the distillation curve.

(b) *Operating range performance, acceleration, and distribution*—It is *desirable* to have low distillation temperatures in the engine operating range, in order to obtain good vaporization of the gasoline. Better vaporization tends to produce both more uniform distribution of fuel to the cylinders and better acceleration characteristics by reducing the quantity of liquid droplets in the intake manifold. This latter subject will be discussed in more detail in Chapter VI.

(c) *Crankcase dilution*—Liquid gasoline in the cylinder is undesirable since it washes away oil from the cylinder walls. This loss of oil impairs lubrication and tends to cause damage to the engine through increased friction between the piston rings and the cylinder. The liquid gasoline may also work down into and dilute the lubricating oil supply, resulting in the possibility of further lubricating troubles through weakening of the oil film between rubbing surfaces. To prevent these undesirable possibilities, the upper portion of the distillation curve should exhibit sufficiently low distillation temperatures to insure that all of the gasoline in the cylinder will be vaporized.

(d) *Vapor lock characteristics*—If the distillation temperatures of the gasoline are too low, the gasoline tends to vaporize in the fuel system. Too great a rate of vaporization can upset the carburetor metering, or even stop the fuel flow to the engine, by setting up a vapor lock in the fuel passages. This characteristic makes it *desirable* to have *high* boiling off temperatures *throughout* the distillation range. Obviously, this requirement is not consistent with the characteristics desired in (a), (b), and (c), above, and any particular gasoline must be a compromise of desired distillation temperatures. The vapor lock characteristic of a gasoline is especially important in aircraft where high altitudes lower the boiling points of the fuels.

(e) *Winter and summer gasolines*—Because of the higher atmospheric temperatures encountered during the summer months, commercial refiners usually reduce the volatility of automotive gasoline intended for warm weather consumption. In cold weather, however, the distillation curve for the gasoline must be lowered, particularly in the 0 per cent to 10 per cent range, to facilitate engine starting.

(2) *Sulphur content*. Most gasoline contains some sulphur which tends to form corrosive compounds that may attack various parts of the engine with injurious results. Consequently, gasoline specifications limit the permissible quantity of sulphur which may be present.

(3) *Gum deposits*. Certain unsaturated hydrocarbons have an inclination to oxidize during storage and form a product known as gum. The gum in the fuel, in turn, tends to cause undesirable de-

posits on intake valves, piston rings, and other engine parts, as well as clogging of carburetor jets. Gasoline specifications, therefore, limit both the gum content of the fuel and its tendency to form gum during storage.

(4) *Anti-knock quality.* The anti-knock rating of SI engine fuels was covered in Article 5-5, and the subject of knocking will be presented in Chapter VIII. In general, and with all other factors equal, the best SI engine fuel will be that having the highest octane rating or PN, since higher compression ratios may be used, and thus the engine thermal efficiency and the power output will be greater. For a given engine, however, the use of a fuel of higher anti-knock rating than that necessary to prevent detonation under the operating conditions of that engine, is an economic waste.

5-7. Qualities and Ratings of CI Engine Fuels. Most CI engine fuels are obtained from the fractions of crude petroleum in the kerosene and gas oil range. These fuels are heavier and more viscous than the gasoline used in the SI engine. Some of the important characteristics of CI engine fuels are:

(1) *Ignition quality.* Knocking is also encountered in CI engines, with an effect similar to that in SI engines, although it is due to a different phenomenon. Knock in the CI engine is due to sudden ignition and abnormally rapid combustion of accumulated fuel in the combustion chamber. Such a situation occurs because of an ignition lag in the combustion of the fuel between the time of injection and the actual burning. (This subject will be covered in detail in Chapter XII.) As the ignition lag increases, the amount of fuel accumulated in the combustion chamber, before combustion commences, also increases. When combustion actually takes place, abnormal amounts of energy are suddenly released, causing an excessive rate of pressure rise which results in an audible knock.

CI engine knock can be controlled by decreasing ignition lag. The shorter the ignition lag, the less tendency to knock. Good CI engine fuel, therefore, will ignite more readily, that is, it will have a short ignition lag. Furthermore, ignition lag affects the starting, warmup, and the production of exhaust smoke in CI engines.

The property of ignition lag is generally measured in terms of cetane number. Cetane, a straight chain paraffin with good ignition quality, is arbitrarily assigned a rating of 100 cetane number. It is mixed with alpha-methyl-naphthalene, a hydrocarbon with poor ignition quality, which is assigned 0 cetane number. The mixture is matched with a fuel under test in a standard engine running under prescribed conditions. The **cetane number** of a fuel, then, is defined as the per cent by volume of cetane in a mixture of cetane and alpha-methyl-naphthalene that produces the same ignition lag as the fuel being tested, in the same engine and under the same operating conditions.

The straight chain paraffins are the most suitable for CI engines. The other families, as presented in Article 5-2, are less suitable in descending order, with the aromatics being least suited to CI engines. Note that this is opposite to the suitability of the families as SI engine fuels.

(2) *Volatility.* The fuel should be sufficiently volatile in the operating temperature range to produce good mixing and combustion, and thus reduce objectionable smoke and odor in the exhaust gases. Figure 5-3 is a representative distillation curve of a typical diesel fuel.

(3) *Viscosity.* CI engine fuels are inherently more viscous than SI engine fuels. They should, however, be able to flow through the fuel system and strainers under the lowest operating temperatures to which the engine will be subjected.

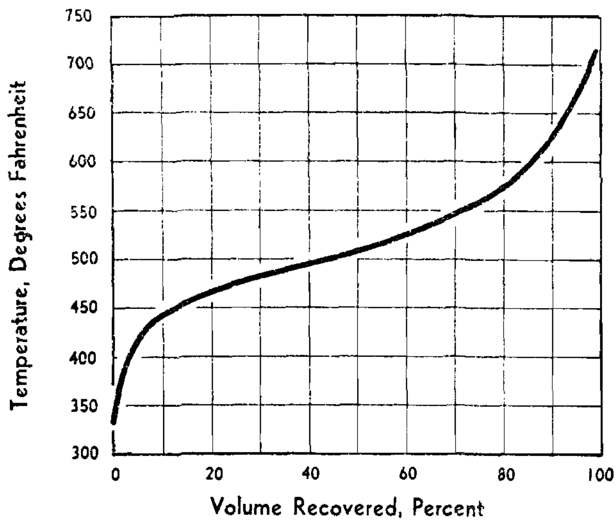


FIG. 5-3. ASTM distillation curve of a typical diesel fuel (courtesy of Ethyl Corporation).

(4) *Impurities.* CI engine fuels have a tendency to contain more solid particles than SI engine fuels. These particles must be re-

duced to a minimum to prevent excessive engine wear. Also, as in SI engine fuels, the sulphur content must be limited to prevent the formation of corrosive compounds.

(5) *Flash point.* The flash point should be sufficiently high to prevent fire hazard.

5-8. Gas Turbine Fuels. Most present day gas turbines operate on liquid petroleum based fuels, namely, gasoline, or kerosene, or a blend of these two. Some of the more important qualities to be considered in the selection of a fuel for use in a gas turbine are:

(1) *Volatility.* Of particular importance is the volatility of the fuel, since this quality exerts a major effect on starting and combustion efficiency, particularly at low temperatures and under adverse conditions. Especially in aircraft gas turbines, where there is often the possibility of "blowout" of the flame, the volatility of the fuel must be conducive to a quick and successful restart. On the other hand, the more volatile fuels are conducive to vapor lock and to excessive loss of fuel during flight due to evaporation of certain of the lighter hydrocarbons. This latter point can be of serious consequence in limiting the range of the aircraft gas turbine, which has a relatively high fuel consumption rate. Furthermore, highly volatile fuels are more susceptible to fire in a crash, although they normally have less tendency to explode if the fuel tanks are penetrated by bullets.

(2) *Energy content.* The heating value of the fuel is another important factor, particularly in connection with the aircraft gas turbine. This type of engine has relatively high fuel consumption, and greater heating value means greater range, or the same range with a smaller fuel load.

(3) *Impurities.* At the present time, there appears to be little or no injurious effect on gas turbines as a result of products of combustion from sulphur in the fuels. There is some indication that some of the sulphur compounds in the fuel do attack certain of the fuel system parts, and most specifications limit the sulphur content of the fuel. Likewise, gum and solid particles in the fuel must be kept to a minimum to prevent fuel system difficulties.

(4) *Combustion products.* The fuel must have a minimum tendency to form solids on combustion. Such solids tend to deposit on the combustion chambers and turbine blades and vanes, causing a loss in efficiency. These deposits may even break off, and damage the airfoil design of the turbine blades.

(5) *Lubricating properties.* With some fuel systems, the use of high speed and high capacity fuel pumps demand that the fuel provide a certain amount of lubrication to the friction surfaces of the pumps in contact with the fuel.
(6) *Availability.* Cost and economic availability are also to be considered in the selection of the fuel to be used. In fact, the latter is a most important factor in the choice of standard aircraft military gas turbine fuels.

As stated above, most gas turbines operate on gasoline, kerosene, or a blend of the two. A short discussion of a few of the fuels presently used for military aircraft gas turbines appears desirable. The general types are:

(1) *Kerosene.* This fuel has been widely used with aircraft gas turbines. It is not as volatile as gasoline, and thereby is less apt to produce vapor lock or loss of range due to evaporation. It is inferior to gasoline in combustion efficiency and its potential supply is limited in comparison with other types. The portion of kerosene which may be processed from a barrel of crude varies from about 5 per cent to 20 per cent depending upon the rigidity of the specifications.
(2) *Gasoline.* Although not as widely used as kerosene, gasoline has become an important military aircraft gas turbine fuel. Due to high volatility, it tends to produce vapor lock, and has a relatively high rate of loss due to evaporation. Its combustion efficiency is higher than kerosene, especially at low temperatures, but its lubrication properties are poorer. Approximately 40 to 50 per cent of a barrel of crude may be refined into gasoline suitable for gas turbine fuels, indicating its greater availaility than kerosene.
(3) *Blends of gasoline and kerosene.* In order to obtain the greatest possible portion of satisfactory gas turbine fuel from crude oil, certain blends of gasoline and kerosene have been studied in this country. The more important of these blends are designated JP-3, JP-4 and JP-5.[1]

    JP-3 is considered as a high-vapor pressure, JP-4 as a low-vapor pressure and JP-5 as a high flash point fuel.

    The advantages and disadvantages of these blended fuels are, of course, a compromise between those of the major constituents, gasoline and kerosene.

While the above discussion indicates that there are several basic gas

[1] *JP* indicates jet power fuels.

turbine fuels of widely varying characteristics, it is in no way intended to convey the impression (so wrongly held by many) that this type of engine will successfully operate on any and all forms of liquid petroleum and its derivatives. A gas turbine can be designed to operate on any of the above listed fuels, but not to satisfactorily operate on all of them interchangeably. The gas turbine is a sensitive piece of machinery which is designed to provide optimum performance on a given fuel, and usually suffers when operating on fuels of distinctly different characteristics.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

5-1. Harold Moore, "*Liquid Fuels for Internal Combustion Engines,*" D. Van Nostrand Company, Inc.

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5-3. H. S. Bell, "*American Petroleum Refining,*" D. Van Nostrand Company, Inc.

5-4. David T. Day, editor-in-chief, "*A Handbook of the Petroleum Industry,*" John Wiley and Sons, Inc.

5-5. William A. Gruse, "*Petroleum and Its Products,*" McGraw-Hill Book Company, Inc.

5-6. Robert T. Haslam and Robert P. Russell, "*Fuels and Their Combustion,*" McGraw-Hill Book Company, Inc.

5-7. "*Aviation Fuels and Their Effects on Engine Performance,*" Ethyl Corporation.

5-8. Robert Schlaifer, "*Development of Aircraft Engines*" and S. D. Heron, "*Development of Aviation Fuels,*" Harvard University Graduate School of Business Administration, Boston, 1950.

EXERCISES

5-1. What are the principal objections to the use of solid and gaseous fuels for internal combustion engines at the present time?

5-2. What are the three main commercial types of liquid fuels for internal combustion engines?

5-3. What factor in the composition of petroleum makes it suitable for fractional distillation?

5-4. What are five primary hydrocarbon families found in petroleum? Which are chain type? Which are ring type?

5-5. State the essential difference between the molecular structure of a stable and an unstable hydrocarbon.

5-6. Which of the primary families tends to be the better SI engine fuel? CI engine fuel?

5-7. What determines the HV of a particular hydrocarbon?

5-8. Briefly describe the refining process known as fractional distillation. What is "cracking"? What is "blending"?

5-9. What is meant by LHV?

FUELS

5-10. Define "octane number rating." What does it indicate?

5-11. On what basis are SI fuels compared when they are better than iso-octane in anti-knock characteristics?

5-12. Name four important qualities of SI engine fuels.

5-13. What volatility characteristics are desirable in a SI engine fuel for starting and warmup? In the operating range? To prevent crankcase dilution? To prevent vapor lock?

5-14. Why is anti-knock rating of a SI engine fuel of such great importance?

5-15. Define "cetane number." What does it indicate?

5-16. How does the viscosity of CI engine fuels compare with that of SI engine fuels?

5-17. On what types of fuels may gas turbines operate?

5-18 What property of gasoline causes it to be objectionable as a gas turbine fuel?

CHAPTER VI

CARBURETION

In both Article 1-8 and the introductory remarks to Chapter V, it was mentioned that the combustion of fuel with oxygen in the combustion chamber provides the energy necessary to drive the piston. In a SI engine, the liquid fuel and the air (which contains the necessary oxygen) are generally mixed prior to their arrival in the combustion chamber. This mixing process, including the theory involved, and its practical application, is generally termed **carburetion.**

6-1. The Induction System. The system responsible for preparing the correct mixture of air and fuel, and directing this mixture to each of the cylinders is often termed the **induction system.** Such a system is illustrated schematically in Fig. 6-1.

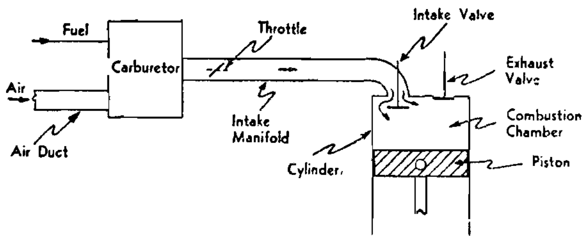


FIG. 6-1. Diagram of induction system (not to scale).

The **carburetor** is the focal point of the induction system. Liquid fuel is supplied to the carburetor from the **fuel system,** which consists of the fuel supply tanks and the necessary fuel pumps, lines, and filters. Air is drawn into the carburetor from the atmosphere by the action of the engine pistons on the intake stroke. Most automobile engines induct the air directly into the carburetor through an air cleaner. Aircraft engine installations, however, often require an elaborate ducting system, and many include a supercharger. The subject of supercharging will be discussed in Chapter X.

Most carburetors are designed to use gasoline as the fuel, but some are also built to utilize different types of liquid fuel such as alcohol or kerosene. This text is concerned only with the gasoline type of carburetor.

The **intake manifold** is the ducting or piping through which the fuel and air mixture travels from the carburetor to the cylinder. The **throttle,** located in the carburetor, regulates the quantity of mixture entering the cylinder.

6-2. Chemically Correct Air-Fuel Ratio. Prior to a study of the requirements for proportioning and mixing the fuel and air in the carburetor (termed carburetion), it is desirable to present a brief analysis of one method for determining the **chemically correct,** or **stoichiometric,** air-fuel ratio.

Since the products of combustion of hydrocarbons are known through analysis obtained by an Orsat apparatus or other means, it is possible to balance the chemical reaction equation and thus predict the amount of air necessary for complete combustion of a particular hydrocarbon. The products of combustion, when complete burning of the hydrogen and carbon in the fuel takes place, are carbon dioxide, water, and nitrogen. The oxygen necessary for combustion is taken from the air, and since air contains a large proportion of nitrogen, this gas is also inducted. Nitrogen being an inert gas, however, does not enter into the combustion process and thus appears as uncombined nitrogen in the products of combustion.

Consider the reaction of carbon. In order to burn it we must supply oxygen. The oxygen comes from the air and thus nitrogen will be introduced but will not affect the reaction. When the carbon unites with the oxygen, the products of combustion will be carbon dioxide and nitrogen. For example:

$$C + O_2 + N_2 \rightarrow CO_2 + N_2.$$

The left side of this equation consists of the carbon plus the oxygen and nitrogen. The right side contains the products of combustion and includes the uncombined nitrogen.

In balancing combustion equations such as that shown above, it should be kept in mind that atoms are neither made nor destroyed during the combustion process. The atoms that existed before combustion must appear in the same numbers after combustion takes place. The atoms, however, may be *rearranged* within the molecules, thereby forming different compounds.

Now consider the combustion of hydrogen.

$$H_2 + \tfrac{1}{2}O_2 + N_2 \rightarrow H_2O + N_2.$$

In this case, the products are water and nitrogen, and again, the number of atoms on the left side of the equation is equal to the number on the right side.

Octane fuel, for example, is represented by a formula of C_8H_{18}. This fuel combines with oxygen to produce CO_2 and H_2O. Therefore, the products of combustion, when burned in air will consist of CO_2, H_2O, and N_2. Since uncombined N_2 always appears in the same quantity on both sides of the equation, we can disregard it for convenience. As a result, we have the following:

$$(\quad)C_8H_{18} + (\quad)O_2 \rightarrow (\quad)CO_2 + (\quad)H_2O.$$

It can be seen that eight atoms of carbon and eighteen atoms of hydrogen appear on the left side but not on the right. This inequality must be corrected. Also, the oxygen atoms must be equalized on both sides of the equation. Hence:

$$(1)C_8H_{18} + (12.5)O_2 \rightarrow (8)CO_2 + (9)H_2O.$$

The air-fuel ratio can now be readily calculated from the left side of the equation in the following manner. By multiplying the number of atoms of each of the constituents by its respective molecular weight, the following expression will result:

$$C_8H_{18} + 12.5O_2 \rightarrow 8CO_2 + 9H_2O$$

$$[(8 \times 12) + (18 \times 1)] + (12.5 \times 32) \rightarrow 8(12 + 32) + 9(2 + 16)$$

$$114 \text{ lbs } C_8H_{18} + 400 \text{ lbs } O_2 \rightarrow \ldots\ldots\ldots\ldots\ldots$$

$$1 \text{ lb fuel} + 3.509 \text{ lbs oxygen} \rightarrow \ldots\ldots\ldots\ldots\ldots$$

Thus, one pound of fuel (octane) requires 3.509 pounds of oxygen for complete combustion. Air consists of 23.2 per cent of oxygen and 76.8 per cent of nitrogen *by weight.* Since air is used as the source of oxygen supply, the equivalent amount of air corresponding to 3.509 pounds of oxygen can be obtained by the following division:

$$\frac{3.509}{0.232} = 15.12 \text{ lbs air/lb fuel.}$$

For complete combustion, therefore, there must be 15.12 pounds of air for every pound of C_8H_{18} fuel. The A/F ratio in this case is 15.12 to 1. The F/A ratio is the reciprocal of the A/F ratio, or is 1 divided by 15.12, i.e., 0.0661. The quantity of fuel calculated by this method is an ideal amount necessary to react completely with all of the oxygen. As a result, this A/F ratio is known as the *chemically correct,* or *stoichiometric* A/F ratio for this type of fuel, octane. This chemically correct mixture will vary slightly in numerical value between different hydrocarbons, but can always be computed from the chemical combustion equation *for a particular fuel.*

The chemically correct A/F ratio is not necessarily that at which an engine will operate. In fact, it will be shown later that the actual operating A/F ratio is not likely to be this stoichiometric ratio.

There is, however, a limited range of A/F ratios, in a homogeneous mixture, within which combustion in a SI engine will occur. Outside of this range, the ratio is either too rich or too lean to sustain flame propagation. This range of useful A/F ratios runs from approximately 20/1 (lean) to 8/1 (rich), as indicated in Fig. 6-2.

The carburetor should provide an A/F ratio in accordance with engine operating requirements, and this ratio must be within the combustible range.

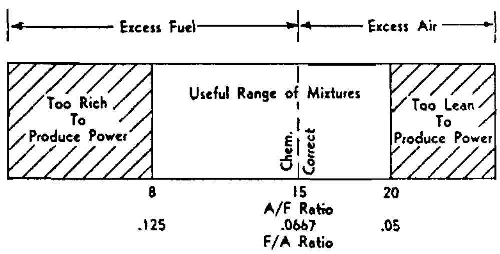


FIG. 6-2. Diagram of useful air-fuel mixture range of gasoline (after diagram by Pratt and Whitney Aircraft).

6-3. Theoretical Carburetor Considerations. A carburetor is a mechanical device designed to fulfill the following functions:

(1) Meter the liquid fuel in such quantities as to produce the A/F *ratio required to meet engine operating conditions.*
(2) Atomize the fuel, and mix it homogeneously with the air.

Due to engine operating characteristics, the A/F ratio required by the engine will vary somewhat over the engine operating range. An important point to remember is that the *carburetor must be designed so that it will provide, as nearly as possible, the A/F ratios which the* **engine requires.**

From the standpoint of both design and operation, it would simplify matters if operating engines could utilize a carburetor providing a constant economical A/F ratio throughout the operating range of the engine. If it were possible to use such a carburetor, it might operate at a chemically correct A/F ratio, as represented by the dashed line Y-Y' in Fig. 6-4. Engine operating *requirements* imposed upon the carburetor, however, prohibit such a simple solution.

Now consider an actual engine operating at full throttle and constant speed. Under these conditions, the A/F ratio will affect both the power output and the brake specific fuel consumption, as indicated by the typical curves shown in Fig. 6-3.

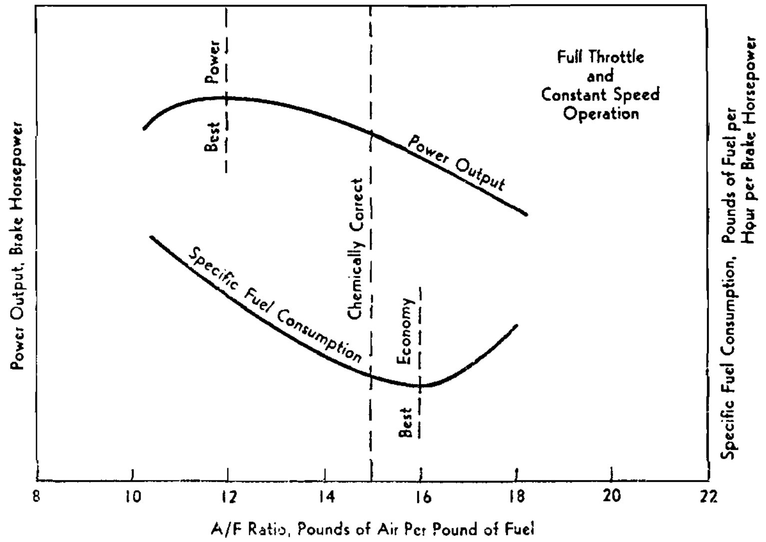


FIG. 6-3. Effect of A/F ratio on power output and specific fuel consumption for a SI engine.

The mixture corresponding to the maximum point on the bhp curve (Fig. 6-3) is called the **best power mixture.** The mixture corresponding to the minimum point on the bsfc curve is called the **best economy mixture.** Note that the best power mixture is richer than the chemically correct mixture, and the best economy mixture is leaner than the chemically correct.

Figure 6-3 is based on full throttle operation. The A/F ratios for best power and best economy at part throttle are not strictly the same as at full throttle. Let us assume, for the moment, however, that the A/F ratios for best power and best economy are constant over the full range of throttle operation. With this assumption, and disregarding other influencing factors, the ideal fuel metering device would be merely a two position carburetor. Such a carburetor could be set for the best power mixture when maximum performance is desired, and for the best economy mixture when the primary consideration is fuel economy. These two settings are indicated in Fig. 6-4 by the solid horizontal lines X-X' and Z-Z', respectively. Actual engine requirements, however, again preclude the use of such a simple and convenient arrangement. These requirements will be discussed in the succeeding article.

6-4. Engine Air-Fuel Mixture Requirements. Actual air-fuel mixture requirements in an operating engine vary considerably from the ideal conditions upon which the simple hypothetical carburetors of the previous article were based. Under actual conditions of successful operation, the engine *requires* the carburetor to provide mixtures which follow the general shape of the curve *ABCD* in Fig. 6-4. This figure shows a representative aircraft engine carburetor performance curve.

As indicated in Fig. 6-4. there are three general ranges of throttle operation. In each of these, the engine requirements differ. As a result,

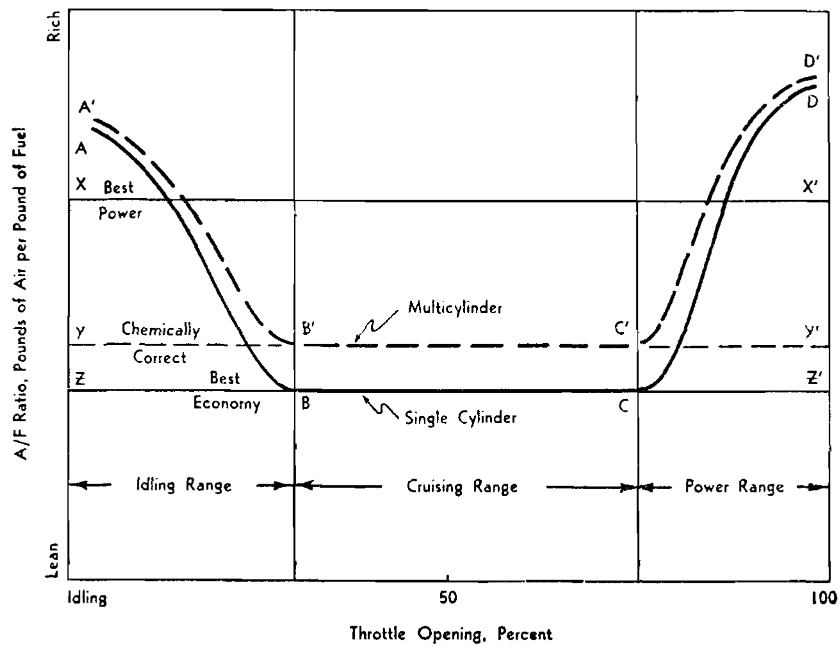


FIG. 6-4. Illustration of carburetor performance necessary to fulfill engine requirements.

the carburetor must modify the A/F ratio to satisfy these demands. These ranges are:

(1) Idling (mixture must be enriched)
(2) Cruising (mixture must be leaned)
(3) High Power (mixture must be enriched)

(1) *Idling Range.* An idling engine is one operating at no load and with nearly closed throttle. Under idling conditions, the engine requires a rich mixture, as indicated by point *A* in Fig. 6-4. This is due to the existing pressure conditions within the combustion

chamber and intake manifold which cause exhaust gas dilution of the fresh charge. The pressures indicated in Fig. 6-5 are representative values which exist during idling. Actually, the exhaust gas pressure at the end of the exhaust stroke does not vary greatly from the value indicated in Fig. 6-5, regardless of the throttle position. Since the clearance volume is constant, the mass of exhaust gas in the cylinder at the end of the exhaust stroke tends to remain fairly constant throughout the throttle range. The amount of fresh charge brought in during idling, however, is much less than

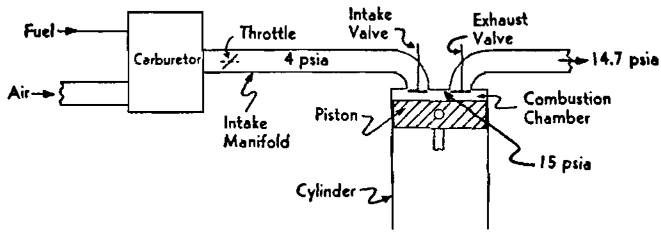


FIG. 6-5. Schematic diagram of combustion chamber and induction system at start of intake stroke, showing representative pressures encountered under idling conditions.

during full throttle operation, due to the restriction imposed by the throttle. The result is a much larger proportion of exhaust gas being mixed with the fresh charge, under idling conditions. Furthermore, the nearly closed throttle restriction tends to keep the pressure in the intake manifold considerably below atmospheric. When the intake valve opens, the pressure differential between the combustion chamber and the intake manifold results in "backward" initial flow of exhaust gases into the intake manifold. As the piston proceeds down on the intake stroke, these exhaust gases are drawn back into the cylinder, along with the fresh charge. As a result, the final mixture of fuel and air in the combustion chamber is diluted by exhaust gas. The presence of this exhaust gas tends to obstruct the contact of fuel and air particles—a requirement necessary for combustion. Such obstruction results in poor combustion and, as a result, in loss of power. It is, therefore, necessary to provide more fuel particles by richening the air-fuel mixture. This richening increases the probability of contact between fuel and air particles, and thus improves combustion.

As the throttle is gradually opened from A to B, Fig. 6-4, the pressure differential indicated in Fig. 6-5 becomes smaller, and the exhaust gas dilution of the fresh charge diminishes. Mixture requirements then proceed along line AB, Fig. 6-4, to a leaner A/F ratio.

(2) *Cruising Range.* In the cruising range from B to C, Fig. 6-4, the exhaust gas dilution problem is relatively insignificant. The primary interest lies in obtaining the maximum fuel economy. Consequently, in this range, it is desirable that the carburetor provide the engine with the best economy mixture.

(3) *Power Range.* During high power operation, the engine requires a richer mixture, as indicated by the line CD, Fig. 6-4, for the following reasons:

(a) *Provide best power*—Since high power is desired, it is logical to transfer the economy settings of the cruising range to that mixture which will produce the greatest power, or a setting in the vicinity of the best power mixture, usually slightly richer.

(b) *Prevent overheating of exhaust valve area.*—At high power, the increased mass of gas passing through the cylinder results in the necessity of transferring greater quantities of heat away from critical areas such as those around the exhaust valve. Enrichening the mixture reduces the flame temperature and the cylinder temperature, thereby reducing the cooling problem and lessening the tendency to damage exhaust valves at high power. In the cruising range, the mass of charge is smaller, and the tendency to burn the exhaust valve is not as great.

(c) *Inhibit detonation in aircraft engines.*—Detonation in an aircraft engine is a very serious matter, since it can cause considerable damage to the engine in a matter of seconds. (The subject of detonation will be discussed in Chapter VIII). Enrichening the mixture beyond chemically correct reduces the flame temperature and thereby tends to reduce detonation. By further enrichening beyond the best power setting, excess fuel is introduced which, in vaporizing, tends to further lower the temperature in the combustion chamber and thereby assist in reducing the possibility of detonation.

In an automobile engine, detonation warnings are produced in the form of an audible "knock" or "ping," and the operator can make the engine operating conditions less stringent by "letting up" on the accelerator or shifting to a lower gear ratio. Furthermore, automobile engines generally operate well below full power and a complicated and expensive system for enrichment for this purpose is not economically feasible, although some means of richening at high power is usually used. For aircraft engine installations, however, the complication and expense is justified and necessary to increase permissible take-off power.

Figure 6-4, then, is more representative of the aircraft engine requirements for the carburetor. Automobile engine requirements are similar in the idling and cruising ranges, but tend to be relatively lower, or less rich, in the power range (*C* to *D* in Fig. 6-6). A more representative engine requirement curve for automobiles is shown in Fig. 6-6. That portion

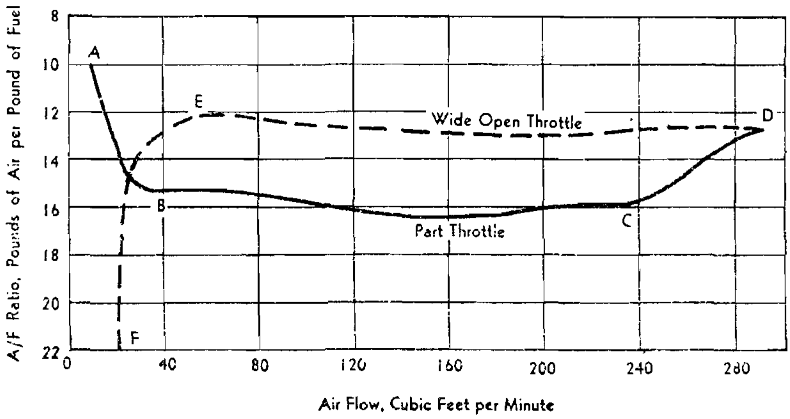


FIG. 6-6. Typical performance curve of an automobile carburetor (courtesy of Carter Carburetor Corporation).

of the curve from *D* to *E* indicates the requirements after the throttle is full open and the load is further increased.

6-5. Distribution. The engine requirements discussed in Article 6-4, and depicted by *ABCD* in Fig. 6-4, were based on a single cylinder engine. In multi-cylinder engines, where the carburetor must supply the air-fuel mixture to each of several cylinders, the problem of distribution becomes important.

Complete atomization and vaporization of the fuel by the carburetor is difficult to obtain. The mixture passing through the intake manifold generally contains a certain amount of liquid in droplet form. These droplets have greater inertia than the gaseous mixture. Consequently, whenever the direction of flow is changed abruptly, the droplets tend to continue in their original direction of movement, as indicated in

Fig. 6-7. As a result, there is a variation in the A/F ratio between cylinders, depending upon the cylinder location and the manifold design.

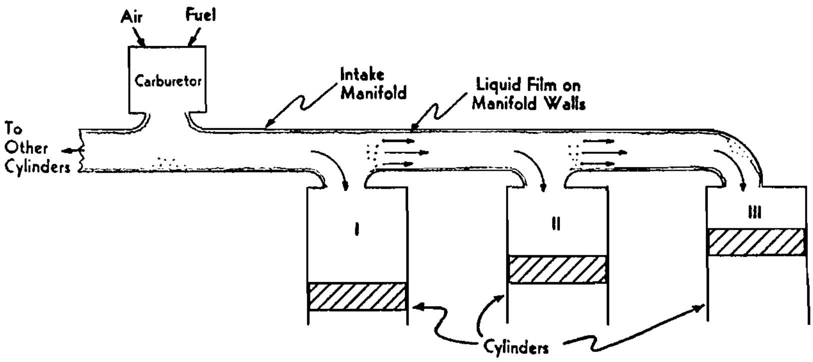


FIG. 6-7. Schematic diagram of intake manifold of a multi-cylinder engine, indicating the flow tendency of liquid fuel droplets.

In addition to the fuel droplets, there exists a thin film of liquid fuel adhering to the inner walls of the intake manifold. This liquid film also contributes to uneven distribution.

A partial solution of this problem is to heat the mixture in the intake manifold, causing the less volatile parts of the fuel to vaporize and thus reducing the number of liquid droplets present. This heating causes the mixture to expand, however, thereby reducing the mass of charge, with consequent reduction in power output. Some engines are designed to utilize this principle by heating certain sections of the intake manifold with the exhaust gas.

Another partial remedy is to enrich the over-all air-fuel mixture so that the leanest cylinder receives the required A/F ratio. As a result, some of the cylinders receive a richer A/F ratio than necessary. Nevertheless, multi-cylinder engines require this over-all enrichment, as indicated by the curve $A'B'C'D'$, Fig. 6-4.

6-6. Acceleration. In the preceding article, it was mentioned that there is a tendency for some non-vaporized liquid droplets to form a thin liquid film along the inner wall of the intake manifold. Under normal steady operating conditions, this film moves slowly along the walls to the cylinders, and forms a portion of the fuel in the air-fuel mixture of the various cylinders. Should the throttle suddenly be opened wide, such as during acceleration, the gaseous charge of fuel and air moves rapidly into the cylinders. The liquid film, due to its greater inertia, is unable to change its rate of movement as rapidly as is the vapor. Consequently, the mixture reaching the cylinder becomes momentarily leaner than required for the new throttle setting. In order to compensate for this tendency of the carburetor, during acceleration, to fail momentarily to supply a sufficiently rich mixture, a mechanical accelerating device is provided. It is usually connected directly to the throttle mechanism. When the throttle is opened rapidly, an extra charge of liquid fuel is forced into the intake manifold to insure that the A/F ratio actually received by the cylinders will be as rich as required by the engine during the acceleration period. This enriched mixture must not only counteract the temporary leaning out due to retarded flow of a liquid fuel film, but also to supply additional fuel necessary for acceleration. This device, usually termed an **accelerating pump,** fulfills these requirements. It is so constructed and actuated that the amount of fuel furnished is dependent upon the rapidity with which the throttle is opened. During a very gradual opening of the throttle, little or no excess fuel will be provided by the accelerating pump.

6-7. Simple Float Type Carburetor. The air-fuel mixture requirements of an operating engine, as fulfilled by a carburetor, have been presented in the preceding articles of this chapter. It is now desirable to see how they are mechanically obtained.

Figure 6-8 is a schematic diagram of a simple, down draft, float type carburetor with both an idling enrichment jet and a choke valve.

The fuel supply to the float chamber is controlled by the action of the float and the attached fuel supply valve. If the amount of fuel in the float chamber falls below the designed level, the float lowers, thereby opening the fuel supply valve. When the designed level has been reached, the float closes the fuel supply valve, thus stopping additional fuel flow from the supply system.

Air from the atmosphere is drawn through the venturi by the action of the pistons on the intake stroke. As the air passes through the venturi, its velocity is increased and the pressure in the venturi throat is decreased. Because of this fact, and since the float chamber is vented to the atmosphere, a pressure differential exists between the float chamber and the tip of the fuel discharge nozzle. This differential causes the fuel to discharge into the air stream in an amount dependent upon the magnitude of this pressure difference. Since the pressure drop in the venturi is dependent upon the rate of air flow, and since the fuel flow is dependent upon the pressure drop in the venturi, the A/F ratio provided by the carburetor is theoretically constant.

During idling, however, the nearly closed throttle causes a reduction in the mass of air flowing through the venturi. At such low rates of air flow, the pressure differential between the float chamber and the fuel discharge nozzle becomes very small. It is insufficient, in fact, to cause fuel to flow over the restraining lips of the fuel discharge nozzle. To compensate for this fact, and to provide the engine with the required rich *A/F* ratio (Article 6-4), an idling jet is added.

The idling jet is located in the wall of the air system, on the downstream side of, and adjacent to the edge of the nearly closed throttle valve. The pistons, descending on the intake stroke, cause a reduction in pressure at the idling jet. Because of the pressure differentials now existing between the float chamber and the idling jet, as well as between the idling air bleed and the idling jet, an air-fuel mixture is

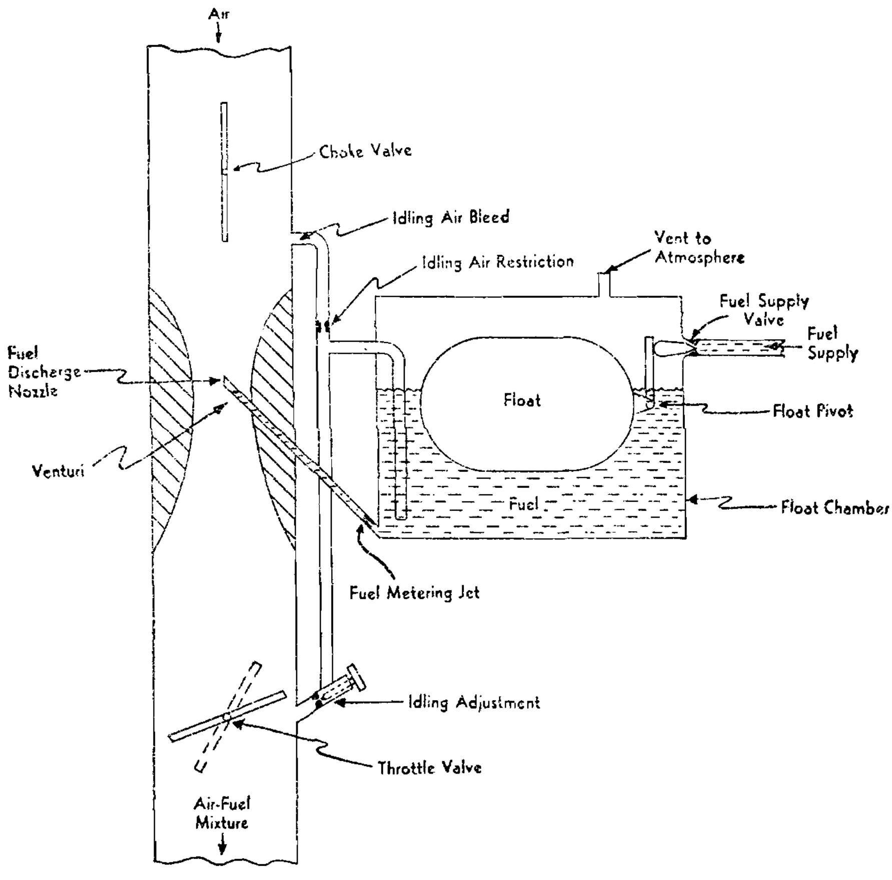


FIG. 6-8. Schematic diagram of a simple float type carburetor with idling system and choke valve.

provided to the engine. The desired air-fuel mixture is regulated by the idling adjustment valve.

An accelerating pump (not shown) has been explained in principle in Article 6-6.

During starting and warmup in cold weather, it is necessary to provide an extra rich mixture to ensure that enough fuel is available, in vaporized form, for combustion. This is accomplished by inserting a choke valve in the air intake system on the upstream side of the venturi. When the choke valve is nearly closed (engine "choked"), a vacuum is created in the area around the fuel discharge nozzle. The pressure differential between the float chamber and the venturi area forces additional fuel into the air stream, thereby fulfilling the extra rich engine requirements.

The simplest type of float carburetor would be that pictured in Fig. 6-8, less the idling jet and choke valve. While it appears that the proportions of fuel and air provided to the engine as the throttle is opened should remain constant, such is not the case. Actually, a simple carburetor will provide an increasingly richer mixture as the throttle is opened, because the density of the air tends to decrease as the rate of air flow increases. In order to correct for this fault, a number of compensating devices are often used. Some of these include the auxiliary air valve, the compensating jet, the air-bleed jet, and the variable orifice jet. Such devices affect the curve BC or $B'C'$ of Fig. 6-4. The idling jet is a major factor in shaping the curve AB or $A'B'$ of Fig. 6-4. In addition, various so-called "economizer" systems are used to enrich the mixture as the throttle approaches the open position, and these affect the portion of the curves CD and $C'D'$ in Fig. 6-4.

Various combinations of these several types of compensating devices are employed on the many carburetors in use today. The number and type of compensating devices used on any particular carburetor is determined by the rigidity of the requirements set forth by the engine and which must be met by the carburetor. A large number of properly chosen compensating devices will tend to cause the carburetor to reproduce more closely the curve required by the engine. However, increasing the number of compensating devices increases the carburetor complexity, with consequent higher costs and increased maintenance problems. Carburetors designed for a particular engine are, therefore, a compromise between these conflicting factors.

Many carburetor problems encountered with modern high powered aircraft are not readily alleviated with the conventional float type carburetor. The float mechanism is adversely affected by the plane attitude, plane maneuvers, and rough air. In addition, under certain atmospheric conditions, the float type carburetor is particularly susceptible to the formation of "ice" in the throttle area, with detrimental results. These many undesirable characteristics of the float type carburetor have led to the development of diaphragm types and pressure-injection types of carburetors for high powered aircraft engines.

The scope of this book does not permit further discussion of either the carburetor compensating devices, or these special aircraft carburetor types.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

6-1. Stanwood W. Sparrow, "Relation of Fuel-Air Ratio to Engine Performance," NACA Technical Report 189, 1924.

6-2. Stanwood W. Sparrow, "The Effect of Changes in Compression Ratio Upon Engine Performance," NACA Technical Report 205, 1924.

6-3. Percival S. Tice, "Metering Characteristics of Carburetors," NACA Technical Report 49, 1918.

6-4. Robert H. Thorner, "*Aircraft Carburetion*," John Wiley and Sons, Inc.

6-5. Lester C. Lichty, "*Internal Combustion Engines*," McGraw-Hill Book Company, Inc.

EXERCISES

6-1. What is meant by "carburetion"?

6-2. What is the induction system?

6-3. What is meant by the chemically correct A/F ratio?

6-4. Balance the combustion equation for the following fuels burning in air, and compute the chemically correct A/F ratio for each of the fuels listed:

(a) Butane, C_4H_{10}
(b) Butene, C_4H_8
(c) Heptadiene, C_7H_{12}
(d) Cyclopentane, C_5H_{10}
(e) Benzene, C_6H_6
(f) Heptane, C_7H_{16}
(g) Dodecane (kerosene), $C_{12}H_{26}$
(h) Ethyl alcohol, C_2H_6O
(i) Methyl alcohol, CH_4O

Ans: (a) 15.48 (b) 14.80 (c) 14.40 (d) 14.81 (e) 13.29 (f) 15.20 (g) 15.03 (h) 9.01 (i) 6.48

6-5. What basic functions must the carburetor fulfill?

6-6. Is it the carburetor or the engine which determines the A/F mixture requirements?

6-7. What is meant by the best power mixture? Best economy mixture?

6-8. What is the major reason for richening the air-fuel mixture in the idling range?

6-9. In what primary factor relating to the air-fuel mixture is the operator interested during cruising?

6-10. Why must the air-fuel mixture be richened during high power operation?

6-11. What factor tends to produce valve burning during high power operation? What method is generally used to overcome this tendency?

6-12. Why is valve burning less of a problem in the cruising range?

6-13. How does richening of the air-fuel mixture assist in preventing detonation in the high power range of operation?

6-14. What condition existing in the intake manifold of a multicylinder SI engine tends to produce poor distribution to the cylinders? Why does this condition cause poor distribution? What remedies are utilized to provide the engine with the necessary *A/F* ratios?

6-15. What condition existing in the intake manifold of a SI engine creates a momentarily lean *A/F* ratio to the cylinder during acceleration? How is this undesirable situation alleviated?

6-16. In a simple float type carburetor, explain how the air-fuel mixture is provided to the engine during cruising conditions. During idling.

6-17. Why is a "choke" used on a carburetor, and how does it operate?

6-18. Why are float type carburetors not suitable for modern high-powered aircraft?

CHAPTER VII

SPARK IGNITION

The ignition system is one of the auxiliary systems that is required for the operation of spark ignition engines. There are many problems, both mechanical and electrical, that are associated with this system which produces the ignition of the charge within the combustion chamber of an engine. A detailed treatment of the electrical problems is beyond the scope of this text.

Therefore, a general basic discussion of the ignition systems in current use, as well as some of their mechanical problems, is presented in this chapter.

7-1. Ignition System Requirements. An ignition system must provide the following basic requirements:

(1) A *source of electric energy.*
(2) A means for boosting the low voltage from the source to the very high potential required to produce a high-tension arc across the spark plug gap that ignites the combustible mixture.
(3) A means for timing and distributing the high voltage, i.e., distribute the high potential to each spark plug at the exact instant it is required in every cycle for each cylinder.

The basic source of the electrical energy is either a battery, a generator, or a magneto. The battery and generator normally provide a 6 to 12 volt potential direct current, while the magneto provides an alternating current of higher voltage. The relatively low voltage produced by the three different types of electric sources must be boosted to a very high potential, 10,000 to 20,000 volts, in order to overcome the resistance of the spark gap and to release enough energy to initiate a self-propagating flame front within the combustible mixture. The low voltage from the source is raised in the secondary circuit by means of an ignition coil, breaker points, and condenser (Figs. 7-1 and 7-2). The timing and distribution of the high potential to the proper spark plug at the exact instant it is required within each cylinder is accomplished by means of the distributor and breaker points.[1]

The two basic ignition systems in current use, *battery ignition system* and *magneto ignition system*, both fulfill the requirements for satisfactory operation. The basic difference in the two systems is the source of the low voltage.

[1] See Articles 7-4 and 10-5 for spark timing.

SPARK IGNITION

7-2. Battery Ignition System. The majority of spark ignition engines (other than aircraft engines), i.e., engines below 500 hp, utilize the battery ignition system. A typical automotive battery ignition system is illustrated in Fig. 7-1. The components of such a system are a battery, ignition switch, ignition coil, breaker points condenser, distribu-

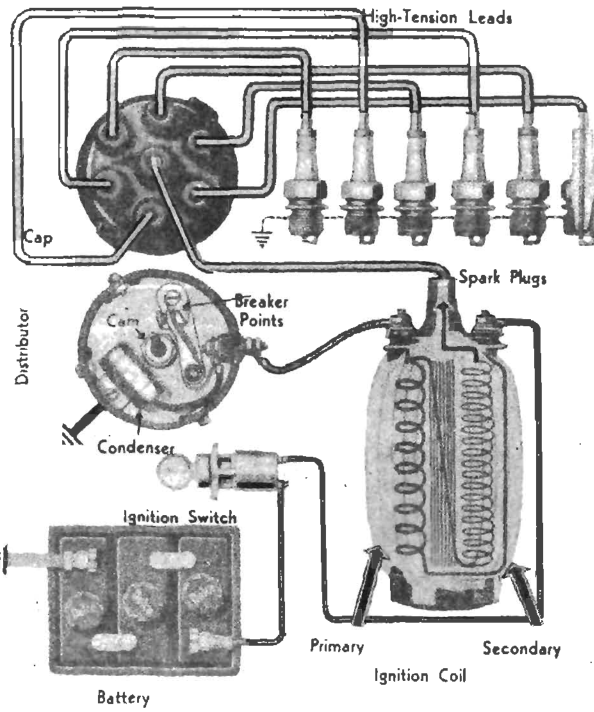


FIG. 7-1. Typical automotive engine ignition system (Courtesy of Delco-Remy Division, General Motors Corp.).

tor, and spark plugs. The breaker points, condenser, distributor rotor, and the spark advance mechanisms[2] are usually housed in the ignition distributor as shown in Fig. 7-2. The breaker points are actuated by a shaft driven at half engine speed for a four-stroke cycle engine. The distributor rotor is directly connected to the same shaft.

A circuit diagram of a battery-ignition system is shown in Fig. 7-3. The system has a primary circuit of low-voltage current and a secondary circuit for the high-voltage current. The primary circuit consists of the battery, ammeter, ignition switch, primary coil winding, con-

[2] Article 7-5

denser, and breaker points. The primary coil winding usually has approximately 240 turns of relatively heavy copper wire wound around the soft iron core of the ignition coil. The secondary circuit contains the secondary coil windings, the lead to the distributor rotor, the distributor, spark plug leads, and the spark plug. The secondary winding consists of about 21,000 turns of small, well insulated copper wire.

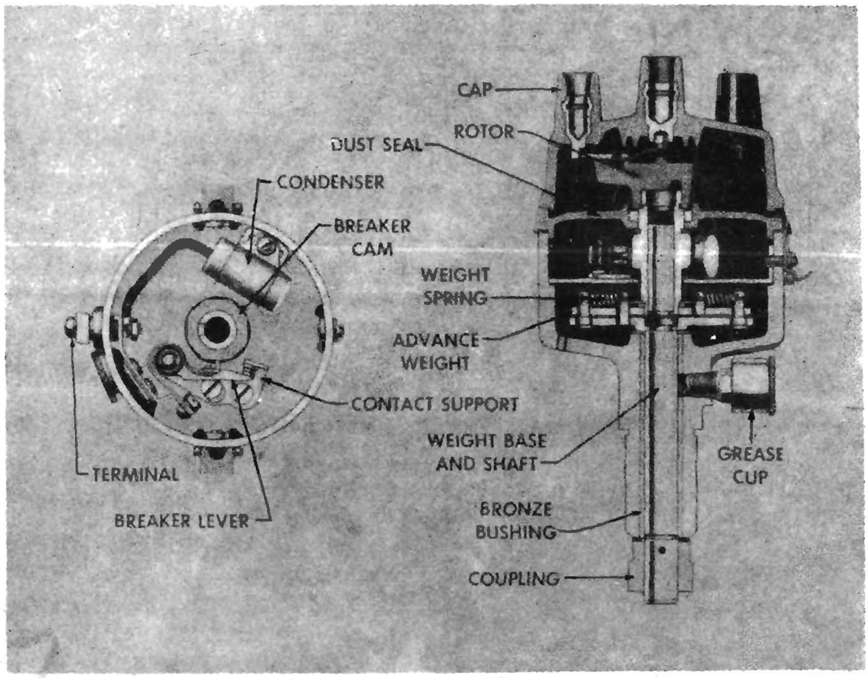


FIG. 7-2. Ignition distributor housing (Courtesy of Delco-Remy Division, General Motors Corp.).

When the ignition switch and the breaker points are closed a low-voltage current flows from the battery through the primary circuit and builds up a magnetic field around the soft iron core of the ignition coil. When the breaker points are opened by the action of the cam on the distributor shaft, the primary circuit is broken and the magnetic field begins to collapse.

An induced current from the collapsing magnetic field flows in the same direction in the primary circuit as the battery current and charges the condenser which acts as a reservoir for a flowing current. Due to a rapidly collapsing magnetic field, high voltage is induced in the primary (it might be as high as 250 volts) and even higher in the secondary (10,000 to 20,000 volts). The high voltage in the secondary passes through the distributor rotor to one of the spark plug leads and into the spark plug. As soon as sufficient voltage is built up in the secondary to overcome the resistance of a spark plug gap, the spark arcs across the gap and the ignition of the combustible charge in the cylinder takes place.

The induced current in the primary, as it was pointed out above, flows in the same direction as it did before the breaker points opened up and charges the condenser. The increasing potential of the condenser retards and finally stops the flow of current in the primary circuit, and then rapidly "backfires" or discharges again through the primary, but in the direction opposite to the original flow of current. This rapid discharge of condenser produces directional oscillation in the current flow in the primary circuit. This oscillation is weakened with every succeeding reversal in the current flow until the original potentials and direction of current flow in the primary circuit are established. Now, the system is ready for another firing cycle.

It should be remembered, however, that the condenser discharges back into the primary circuit after the spark has occurred at the gap

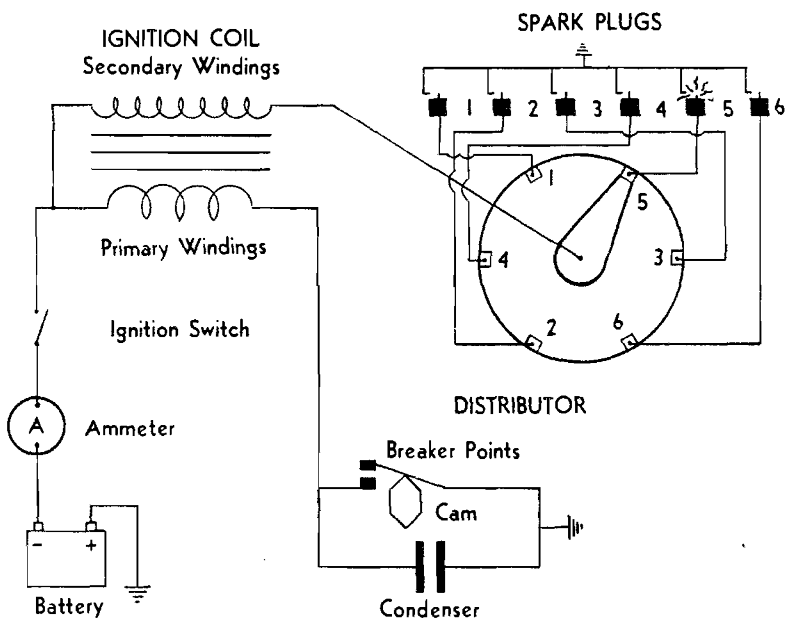


FIG. 7-3. Circuit diagram for battery ignition system.

of a spark plug. The discharge of condenser by itself does not produce the spark, but only hastens the collapse of the magnetic field around the soft iron core.

The condenser, which has a capacitance range from 0.15 to 0.25 mf in the automotive system, not only assists in the collapse of the magnetic field, but also prevents arcing at the breaker points by providing a place for the induced current to flow in the primary circuit. If the condenser is too small or too large, the breaker points will arc and excessive pitting will result. The breaker mechanism is shown in Fig. 7-2.

The breaker points and the distributor must be carefully synchronized with the crankshaft of the engine to give the proper timing of the spark in each of the cylinders.[3] The breaker is often referred to as the timer, since the time or point in the cycle that the spark occurs depends upon the time of opening of the breaker points. The distributor rotor finger must be in contact with the proper spark plug lead when the high voltage is induced in the secondary circuit by the opening of the breaker points.

The spark plug leads are called the ignition harness. Since the leads carry a very high potential, a special insulation is required to prevent a short. Even with the special insulation, these leads are subject to breakdowns which result in high-tension short circuits and to leakage that lowers the voltage available at the spark plug. Also, the leads should be shielded to aid in the prevention of radio interference.

A generator, starting motor, and lighting system are integral with the battery ignition system in an automotive engine. They are interconnected as shown in the schematic wiring diagram of Fig. 7-4. The battery supplies current to the starting motor and may supply current to the lighting circuit. The generator supplies current to the ignition system and the lighting system, and charges the battery when the engine is running. The cutout in the voltage regulator automatically connects the generator to the circuit when the voltage it produces is higher than the battery voltage, and disconnects the generator when its potential is lower than that of the battery. When the ignition switch is closed before the engine is started, the contact points on the cutout are separated by the tension of the spring on the armature of the cutout. The generator is then disconnected from the battery ignition circuit and the battery cannot discharge through the generator. After the engine starts, current flows from the generator through the cutout solenoid and the resistance to the ground. When engine speed becomes

[3] Articles 7-5 and 10-5.

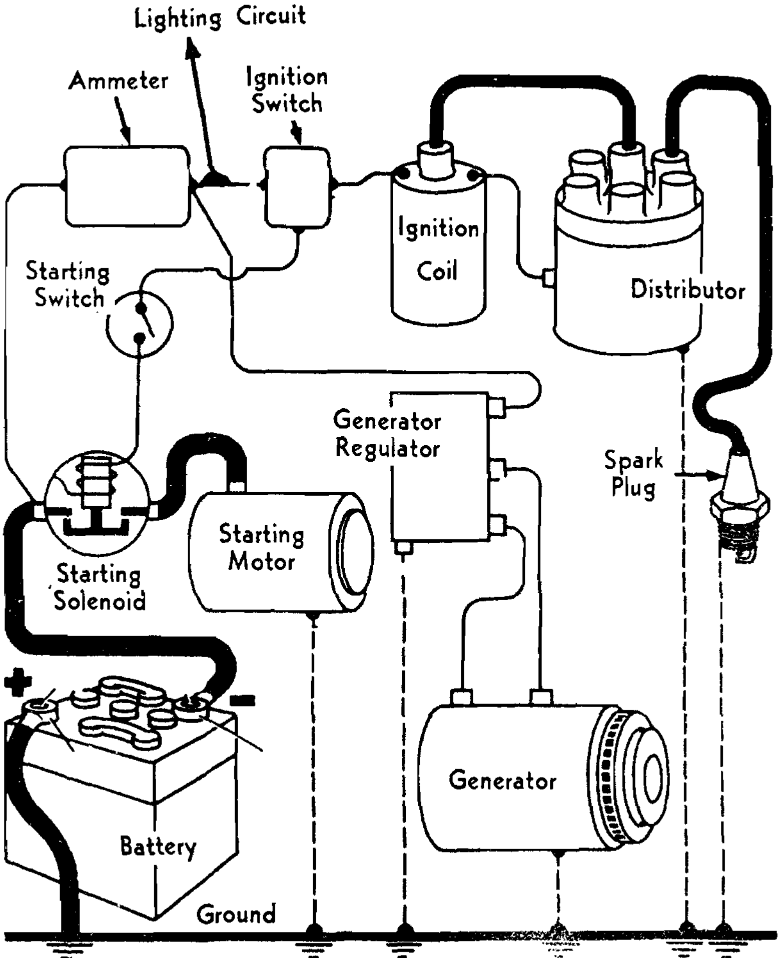


FIG. 7-4. Schematic wiring diagram of a generator, a starting motor, and a lighting circuit connected to a battery ignition system (Courtesy of Chrysler Corp.).

high enough to cause the generator voltage to reach a predetermined value higher than the battery voltage, the cutout is closed and generator supplies current to the battery ignition and lighting circuits. Thus, the voltage supplied to the system by the generator is controlled by the voltage regulator.[4]

[4] Ref. 7-4.

7-3. Magneto Ignition System. For many applications of the spark ignition engine, a lighter and more compact source of electrical energy than the battery is desirable. The higher powered, high speed spark ignition engines, such as aircraft engines, use a compact magneto or magnetos to generate the high potential required by the spark plugs. The basic components of a magneto ignition system are illustrated in Figs. 7-5 and 7-6. Figure 7-5 is a simplified magneto ignition system

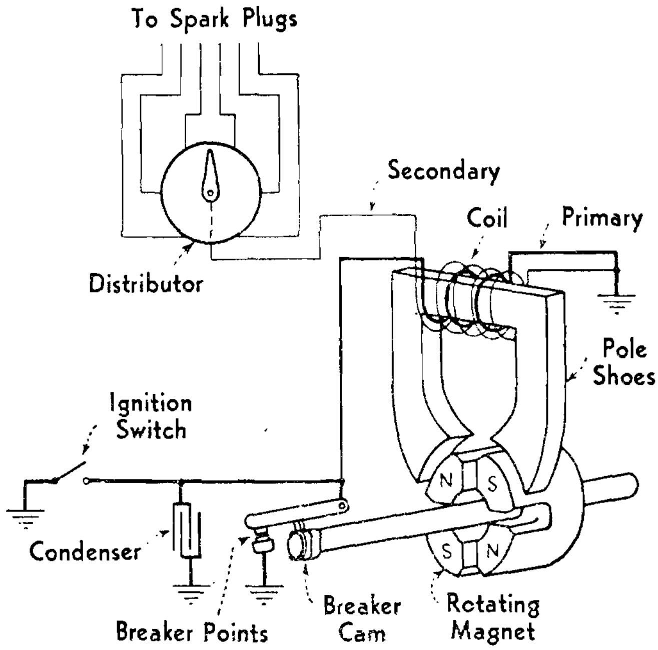


FIG. 7-5. Schematic wiring diagram of a magneto ignition system (Courtesy of General Motors Corp.).

and Fig. 7-6 shows a typical dual magneto ignition system commonly used in relatively large aircraft engines.

The basic components of the magneto ignition system consist of a magneto, breaker points, condenser, ignition switch, distributor, spark plug leads, and spark plugs. The battery and ignition coil of the battery system have been replaced by a compact magneto. The magneto may have a rotating coil and a stationary permanent magnet or it may have rotating magnets and a stationary coil. The magnetos shown contain a four-pole magnet, two-pole shoes, and the primary and secondary coils. As the rotating magnet turns, the direction of the magnetic flux through the soft iron core of the coil reverses direction. As the magnetic flux alternately builds up and breaks down (four times for each revolution of the rotor with a four-pole magnet), a voltage is induced in the primary and secondary coils. The change in flux, however, is not rapid enough to induce the high potential required to overcome the resistance of the air gap in the spark plug. Therefore, a means of creating a more rapid breakdown of the magnetic flux is required if current is to flow in the secondary coil.

An increase in the rate of the magnetic flux breakdown is accomplished by means of breaker points and a condenser. The breaker points are actuated by a cam attached to the rotor of the magneto. When the breaker points are closed, current flows in the primary circuit. When this current approaches its maximum, the breaker points are opened by the action of the cam and the condenser is charged. This is followed by a rapid discharge of the condenser which produces an almost instantaneous breakdown of the magnetic flux. The very rapid

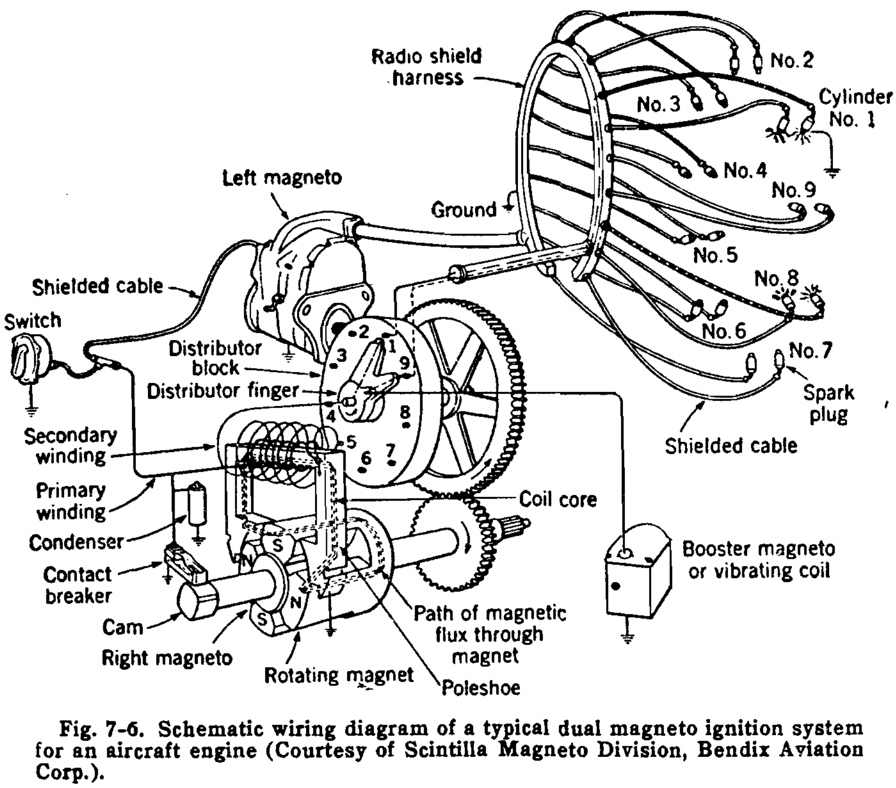


Fig. 7-6. Schematic wiring diagram of a typical dual magneto ignition system for an aircraft engine (Courtesy of Scintilla Magneto Division, Bendix Aviation Corp.).

breakdown, then, induces a high voltage, 10,000 to 20,000 volts, in the secondary circuit which is more than ample to overcome the resistance of the air gap between the spark plug electrodes and ignite the charge in the combustion chamber. The breaking of the primary circuit, the motion of the rotor, and the rotation of the distributor arm must all be timed accurately in order to produce the spark in each cylinder at the proper time.

When the ignition switch in the magneto system is closed, the primary circuit is short-circuited. This prevents the rapid breakdown of the magnetic flux and prevents a high potential from being induced in the secondary circuit. However, in order to stop the engine the flow of current is not cut in the primary circuit, as is the case when the ignition switch is opened in the battery ignition system (Fig. 7-3). In the magneto ignition system, the engine is stopped by closing the ignition switch and grounding the primary circuit.

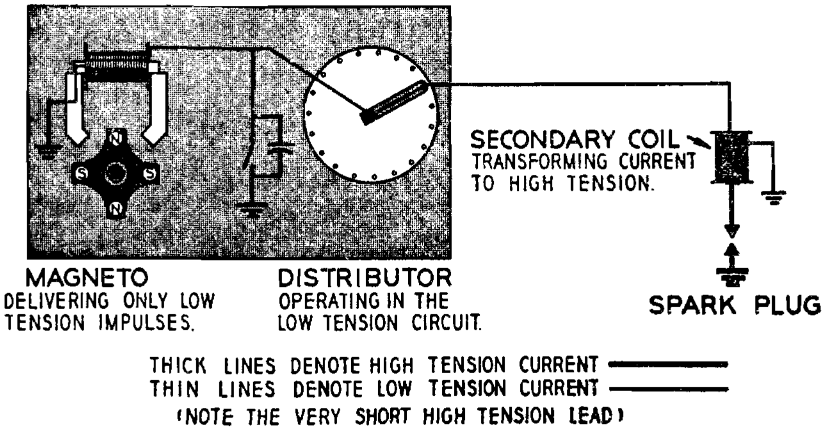


FIG. 7-7. Low-tension distribution system (Courtesy of Bendix Aviation Corp.).

The primary voltage in a magneto system increases with an increase in the speed of the engine, and, therefore, the resultant potential in the secondary increases with an increase in speed. This is an advantage over the battery system where the opposite characteristic is found. In general, the higher the engine speed the greater the resistance in the air gap and the greater the potential required. However, the variation of the induced voltage with speed has a disadvantage in that trouble may be experienced in starting an engine with a magneto ignition system. One method of overcoming the low induced voltage during the starting of an engine is to connect a booster magneto to the system as shown in Fig. 7-6. The booster insures a good spark for starting; and the spark may be retarded from the normal firing time to reduce the possibility of a "kick" or reversal of the engine when cranking. Another method is to use high tension vibrating coil to which primary current is supplied by means of a battery. Also, either batteries or auxiliary generators as an outside source, may be connected to the circuit to provide energy for the starting motor and for the initial spark.

The high-tension ignition system used in aircraft engines requires extensive shielding to prevent radio interference. Also, at higher altitudes where the density of the surrounding air is low, the high-tension ignition harness is subject to leakage which reduces the potential available at the spark gap and may cause the engine to misfire. This may be partially offset by pressurizing the ignition harness, i.e., dry air under a few pounds pressure is sealed in the harness. Another method being used in aircraft engines is to have a low-tension distribution system which confines the high-tension voltage to localized areas around each spark plug. The low-tension distribution system is made by changing the secondary winding of the magneto to limit the induced voltage to around 350 volts and to substitute a brush contact for the distributor finger. This relatively low induced voltage then flows through the ignition harness to the spark plug where the high potential required at the air gap is obtained by means of a small step-up transformer (Fig. 7-7). Even though the low-tension ignition harness, also, has to be shielded, the radio interference and the leakage are reduced with this system.

7-4. Spark Plugs. The spark plug conducts the high potential from the ignition harness into the combustion chamber. It provides the proper gap across which the high potential discharges to ignite the combustible mixture within the combustion chamber. The spark plug also must provide suitable insulation between the two electrodes to prevent a short circuit.

Typical sectionalized spark plugs are shown in Fig. 7-8. A spark plug consists essentially of a terminal, a steel shell, an insulator, and two electrodes. The center electrode connected to the terminal is well insulated with porcelain, mica, or other ceramic materials. The other electrode is grounded to the cylinder through the steel shell of the plug. The electrodes are usually made of high-nickel alloy to withstand the severe erosion and corrosion to which they are subjected in the combustion chamber.

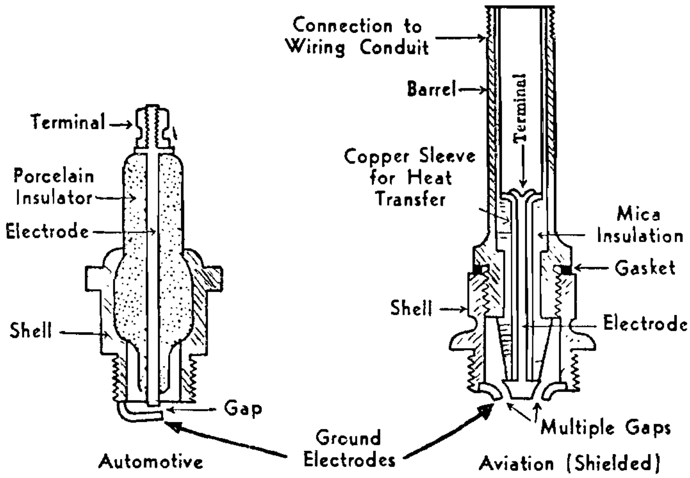


FIG. 7-8. Typical sectionalized spark plugs (*Elements of Applied Energy*, F. T. Morse, Copyright 1947, D. Van Nostrand Co., Inc.).

The center electrode and insulator are exposed to the combustion process. This results in the insulators having a tendency to crack from the high thermal and mechanical stresses. Some insulators are also seriously affected by moisture and by abnormal surface deposits. Since the center electrode and insulator are subjected to the high temperature of the combustion chamber, the heat must flow from the insulator to the steel shell, which is in contact with the relatively cool cylinder head in order to cool the electrodes and prevent pre-ignition.

Spark plugs are usually classified as *hot plugs or cold plugs* depending on their relative operating temperature range. The operating temperature is governed by the amount of heat transferred which in turn depends on the length of the heat transfer path and on the amount of surface area exposed to the combustion chamber. A **cold plug** has a short heat transfer·path and a small area exposed to the combustion chamber as compared to the hot plug, as shown in Fig. 7-9. The type of spark plug used depends on the particular engine requirements. Each manufacturer must determine which type of plug, cold or hot, is best suited to his engine. A spark plug may run at a satisfactory temperature at cruising speeds while at idling speeds it may run so cool that abnormal deposits will foul the electrodes. These deposits may be soft dull carbon from incomplete combustion or hard shiny carbon from excess lubricating oil that passes the piston rings and reaches the combustion chamber. The carbon deposits from incomplete combustion will burn off at temperatures above 650° F, while the excess oil carbon deposits require a temperature above 1000° F. If a spark plug runs hot enough at idling speeds to prevent carbon deposits, it may run too hot at high speeds and cause pre-ignition.[5] If a spark plug runs at a temperature above 1500° F pre-ignition usually results. A compromise must be made in order to obtain the proper spark plug which will operate satisfactorily throughout the entire engine operating range. Even using the spark plugs designated in an engine operating manual, the spark plugs may still be a major source of engine trouble, such as fouling, pre-ignition, and misfiring.

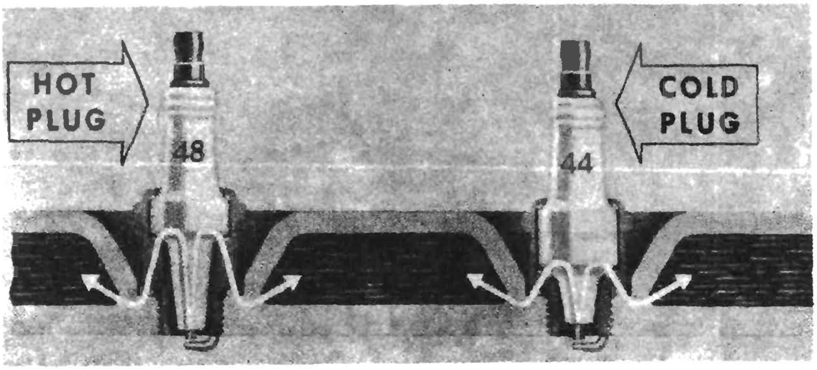


FIG. 7-9. Heat transfer path of hot and cold spark plugs (Courtesy of General Motors Corp.).

7-5. Ignition Timing. The ignition system has to be timed so as to insure that the high tension spark occurs at the proper instant within each cylinder in order to produce optimum power and economy from the engine under a particular set of operating conditions. Since combustion does not take place instantaneously, the spark should occur before the end of the compression stroke in order that approximately half of the pressure rise takes place before TDC (top dead center), (Fig. 8-3), in order to obtain the best power and the lowest specific fuel consumption. When the spark occurs before TDC, it is known as an advanced spark and is given in terms of crank angle degrees before TDC.

The optimum spark setting to produce the best power with the minimum fuel consumption varies with different operating conditions. In general the spark setting is dependent on the combustion time which in turn varies with the design of the cylinder and the operating condi-

[5] See Article 8-5 for pre-ignition.

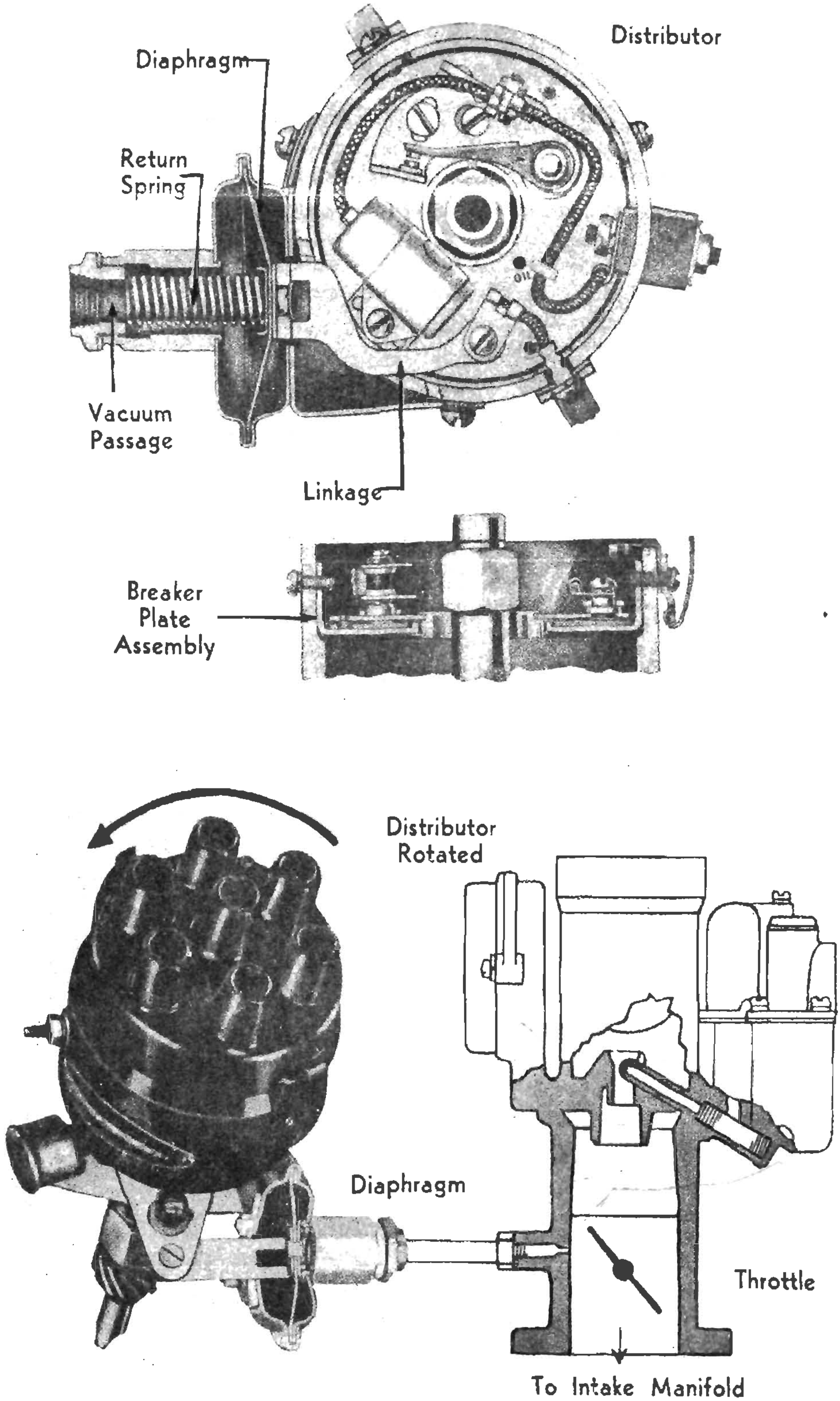


FIG. 7-10. Vacuum advance mechanism (Courtesy of Delco-Remy Division, General Motors Corp.).

tions. Therefore, some of the major factors affecting the optimum spark setting are as follows:

(1) *Type of fuel.* The rate at which each type of fuel will burn varies with the fuel. A slow burning fuel must have a greater spark advance than a fast burning fuel in order that half the pressure rise will occur before TDC.

(2) *Engine speed.* An increase in engine speed will decrease the actual time of combustion but not as much as the speed increases. The combustion time then increases in terms of crank angle degrees, and, therefore, the optimum spark advance must increase with speed.

(3) *Air-fuel ratio.* In general, the lower the air-fuel ratio the faster will

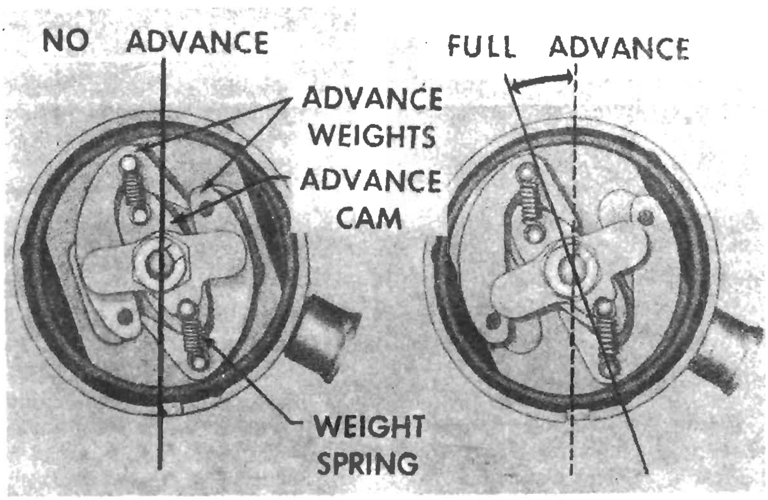


FIG. 7-11. Centrifugal advance mechanism (Courtesy of Delco-Remy Division, General Motors Corp.).

be the rate of burning. As the air-fuel ratio is lowered, the optimum spark advance must be retarded, i.e., the number of degrees of crank angle before TDC is decreased and the spark occurs closer to TDC.

(4) *Part load conditions.* In carbureted-spark ignition engines, part load operation is obtained by throttling the amount of the charge or mixture going to the engine. This produces a greater dilution of the smaller incoming charge with the gases of combustion. This, coupled with the higher air-fuel ratios required for part load running, produces an increase in the combustion time. Part load operation, then, requires an increase in the optimum spark advance.

The optimum spark setting must be regulated to account for changes in the load and speed of the engine. Most automobile engines are equipped with a mechanism that is integral with the distributor and provides an automatic regulation of the optimum spark advance to account for changes in the load and speed. The two mechanisms are the **vacuum advance** and the **centrifugal advance.**

The vacuum advance antomatically compensates for the part load throttling of an engine. As the throttle is closed, the spark must be advanced for optimum performance. This is accomplished by a diaphragm mechanism as shown in Fig. 7-10. As the throttle is closed, the diaphragm moves away from the distributor due to the low pressure in the manifold and the spark is advanced through a linkage mechanism. As the throttle is opened the vacuum advance tends to retard the spark, i.e., cause the spark to occur closer to TDC. Most engines require a rich mixture and a retarded spark when in the idling range. This is accomplished by having the inlet to the vacuum line pass into the high pressure area on the upstream side of the throttle, when the throttle is almost closed as it is in the idle position.

The centrifugal advance automatically compensates for changes in the speed of the engine. This is accomplished by attaching the breaker cam to weights (Figs. 7-2 and 7-11). When the engine speed increases, the weights move due to centrifugal force and advance the angular position of the cam relative to the drive shaft. In this new position, the breaker points open earlier causing an advance in the spark setting.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

7-1. L. C. Lichty, *Internal-Combustion Engines*, McGraw-Hill Book Co., Inc., New York, 1951.

7-2. E. F. Obert, *Internal-Combustion Engines*, International Textbook Co., Scranton, 1950.

7-3. A. P. Fraas, *Combustion Engines*, McGraw-Hill Book Co., Inc., New York, 1948.

7-4. Department of Marine Engineering, *Internal-Combustion Engines*, Dept. of Marine Engineering, U. S. Naval Academy, Annapolis, 1947.

7-5. H. L. Hartzell, "Post-War Automotive Practices on Ignition Performance," *SAE Journal*, Vol. 53, No. 7, July 1945.

7-6. C. F. Taylor and E. S. Taylor, *Internal-Combustion Engines*, International Textbook Co., Scranton, 1948.

EXERCISES

7-1. Name three sources of electrical energy for ignition systems.

7-2. Name the two basic types of ignition systems.

SPARK IGNITION

7-3. Which ignition system usually has generator interconnected with it?

7-4. What are the functions of the primary circuit and what are its component parts?

7-5. What is the function of the secondary circuit and what are its component parts?

7-6. Which carries the higher potential, the primary or the secondary circuit?

7-7. Draw a schematic circuit diagram of a battery ignition system and label all the components.

7-8. How is the high potential in the secondary circuit produced?

7-9. What are the functions of the condenser?

7-10. What tends to prevent the arcing of the breaker points?

7-11. How are the breaker points actuated?

7-12. How fast does the rotor of the distributor turn as compared to the engine speed for a four-stroke cycle engine? a two-stroke cycle engine?

7-13. What supplies electrical energy to the starting motor?

7-14. What are the functions of a generator when connected to a battery ignition system?

7-15. Does the generator supply current to the system at all times? If not, when and how is the current supplied?

7-16. Describe the operation of the distributor?

7-17. What is the usual voltage of automobile batteries?

7-18. What are two advantages of the magneto ignition system as compared to the battery ignition system?

7-19. Describe the operation of the magneto.

7-20. Does the magneto produce a rapid enough change in flux to induce the required high potential at the spark gap? If not, how is the rapid change in flux accomplished?

7-21. How does the magneto ignition system differ from the battery ignition system?

7-22. What is the function of the ignition switch? Describe how this is accomplished in the battery ignition system and in the magneto ignition system.

7-23. Which has the most windings, the primary or the secondary coil? Why?

7-24. What are two means of lowering the leakage in the ignition system of an aircraft engine?

7-25. How does the low-tension distribution system differ from the high-tension distribution system? What are its advantages?

7-26. What are the two basic differences in the design of the hot and the cold spark plugs?

7-27. What are the causes of carbon deposits?

7-28. How can a spark plug cause pre-ignition?

7-29. How is it possible to remove some of the carbon deposits from a fouled spark plug while the engine is running?

7-30. What are four major factors that affect the optimum spark advance?

7-31. What two mechanisms help to change the spark advance automaticallv? How is this accomplished?

CHAPTER VIII

COMBUSTION IN THE SI ENGINE

In Article 1-8, it was stated that the energy which drives the reciprocating engine is obtained through burning of the fuel in the combustion chamber. Chapter V contained a discussion of the various types of fuels, while Chapter VI covered the subject of mixing these fuels with air to provide a combustible mixture to the combustion chamber. The subject of the combustion of this mixture in a SI engine will be dealt with in this chapter.

8-1. General Combustion Theory. Combustion may be considered as a relatively rapid chemical combination of the hydrogen and carbon in the fuel with the oxygen in the air, resulting in liberation of energy in the form of heat. The released energy is utilized to drive the internal combustion engine.

The theory of combustion is a very complex subject. In spite of the tremendous amount of research which has been, and is being conducted on the process, a good deal concerning the phenomenon is as yet unknown. In fact, much of the present information consists of individual theories, borne out to some extent by experimental data, but often at variance with one another. A detailed presentation of these theories is beyond the scope of this book, but enough general information will be presented to permit the student to understand the combustion effects as they apply to SI engine operation.

The conditions necessary for combustion are the presence of a combustible mixture and some means of initiating the combustion process. In a SI engine, the combustible mixture is generally supplied by the carburetor, and a spark plug is utilized to initiate the combustion.

A chemical combustion equation for a particular hydrocarbon, which relates the original constituents to the products of combustion, can be rather easily written out. This was accomplished for one hydrocarbon in Article 6-2 and is repeated here:

$$C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O.$$

Such an equation indicates that the combustion process is a simple and direct combination of the atoms involved. It is now a well established fact that such is a definite oversimplification of the actual process. Apparently, a very complicated series of intermediate chain type reactions takes place in transposing the original fuel and air into the final products. Just what happens is a matter of much conjecture.

The essence of most theories, however, is that there are two general phases in the combustion process, the "*preparation*" phase, and the "*actual burning*" phase. During the "preparation" phase, the spark sets off a local reaction resulting in the formation of some intermediate reactors. The intermediate reactors, in turn, apparently both prepare the mixture for, and initiate, the actual burning.

The intermediate substances, and their behavior, are so little understood at the present time that no further discussion concerning them will be attempted—it is sufficient to the needs of this book to know that such intermediate complicated reactions do exist.

8-2. Normal Combustion and Flame Front Propagation. Combustion in the SI engine may be roughly divided into two general types, normal and abnormal. A study will first be made of the normal combustion process and the variables associated therewith.

In the ideal and air cycles, heat is assumed to be added instantaneously. Likewise, in the fuel-air cycle, the burning is assumed to be instantaneous. In an actual SI cycle, however, combustion occurs over a finite period of time. It normally begins at the spark plug and progresses through the combustible mixture with a rather definite flame front separating the unburned charge from the products of combustion.

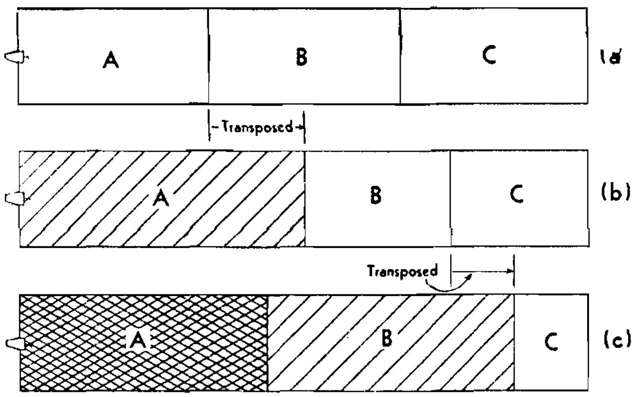


FIG. 8-1. Diagram of flame front "transposition rate" (not to scale).

The rate of motion of the flame front across the combustion chamber is determined by the "reaction rate" and the "transposition rate." The **reaction rate** is the result of a purely chemical combination process in which the flame "eats" its way into the unburned charge; it is comparable, for example, to a forest fire progressing through the combustible underbrush under no wind conditions. The **transposition rate** is the result of a physical movement of the flame front relative to the cylinder walls, and is due to the pressure differentials set up between the burning gases and the other gases in the combustion chamber. Figure 8-1 will help to illustrate this latter action.

Figure 8-1(a) is assumed to be a combustion chamber hypothetically divided into three equal volumes and masses of combustible mixture. Assume that the flame proceeds across the chamber from left to right. If the mass of mixture in section (A) is completely burned, it will expand and compress sections (B) and (C) into smaller volumes of in-

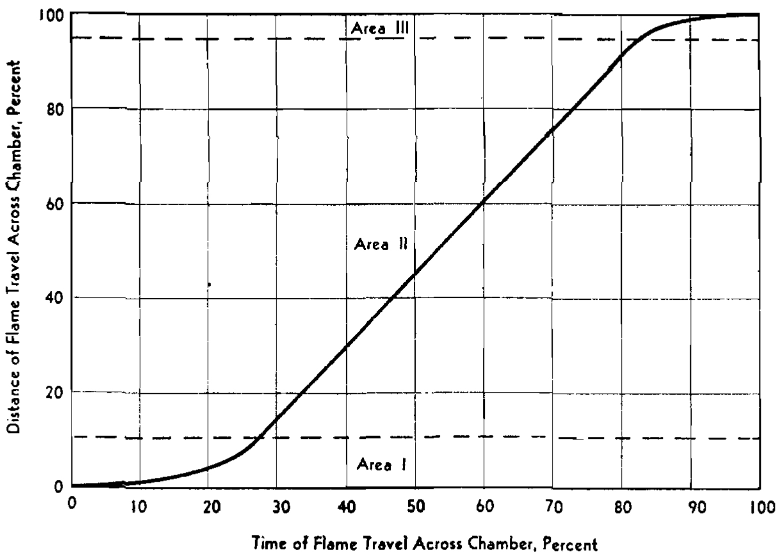


FIG. 8-2. Distance of flame travel across chamber versus time (after Taylor and Taylor, *The Internal Combustion Engine*, International Textbook Company).

creased density, as illustrated in Fig. 8-1(b). The flame front has progressed across section (A) due to the "reaction rate" but has been moved to the right even further due to the "transposition rate." Now suppose the flame front progresses through section (B). This section will expand and compress section (C) into a still smaller volume, and will also compress section (A) to a lesser extent. The flame front will again be transposed to the right. If the number of sections is assumed to be infinite, the net result is a rather smoothly progressing flame front moved forward not only by the reaction effect, but also by the transposition effects of the expanding burning gases.

Analysis of actual photographic flame front traces indicates that the flame velocity across the chamber follows a pattern similar to that indicated in Fig. 8-2. The flame travel appears to pass through three rather distinct stages.[1]

Initially (Area I, Fig. 8-2), the flame front progresses relatively slowly due primarily to a low "transposition rate" and low turbulence. Since there is a relatively small mass of charge burned at the start, there is very little transposition of the flame front. It is thus propagated almost entirely by the reaction rate, resulting in a slower advance. Also, since the spark plug is necessarily located in a quiescent layer of gas that is close to the cylinder wall, the lack of turbulence reduces the "reaction rate" and further lowers flame speed.

As the flame front proceeds into more turbulent areas (leaves the quiescent zone) and commences to consume a greater mass of mixture, it progresses more rapidly and at a rather constant rate (Area II of Fig. 8-2). The average velocity of the flame during this period of travel is often referred to as the "flame velocity" or "flame speed."

Toward the end of flame travel, the volume of unburned charge is reduced appreciably, and the "transposition rate" again becomes negligible, thereby reducing flame speed. Also, the "reaction rate" is again reduced since the flame is entering a zone of relatively low turbulence.

As stated in Article 8-1, the actual reactions taking place during combustion are questionable. It appears, however, that the heat liberated by the burning portion of the flame front "prepares" the adjacent portion of the unburned charge for the combustion reaction.

8-3. Factors Affecting Flame Speed. The subject of flame speed is of importance primarily for two reasons:

(1) Its influence on the rate of pressure rise in the combustion chamber, a subject which will be dealt with in Article 8-4 and
(2) Its effect in connection with certain types of abnormal combustion, a subject which will be discussed in Article 8-5.

There are many operating variables which affect, in varying degree, the flame speed. The most important of these, however, appear to be turbulence and the A/F ratio, which will be discussed below:

(1) *Turbulence*—The most important single factor affecting flame speed is turbulence. The flame speed increases with increasing

[1] C. Fayette Taylor and Edward S. Taylor, *The Internal Combustion Engine*. International Textbook Company.

turbulence. This, apparently, is due to the additional physical intermingling of the burning and unburned particles at the flame front, which expedites reaction by increasing the rate of contact. A turbulence consisting of many minute "swirls" appears to increase reaction and produce higher flame speed than that made up of larger and fewer "swirls." Increasing engine speed generally increases turbulence and, therefore, exerts considerable effect on flame speed.

(2) *Air-fuel ratio*—Another variable which affects the flame speed is the A/F ratio. The highest flame speeds will be obtained with an A/F ratio somewhat richer than chemically correct. If the A/F ratio is increased or decreased from this value, the flame speed is reduced.

(3) *Other factors*—The flame speed is also affected by such variables as the pressure and temperature of the entering charge, humidity, amount of residual gas, spark timing, and compression ratio. The effect of these variables is not generally large, however, and will not be covered in this discussion.

8-4. Rate of Pressure Rise. The rate at which the pressure rises in the cylinder, during the combustion process, is of primary interest to both the designer and operator. This *rate of pressure rise* exerts considerable influence on the peak pressures encountered, the power produced, and the smoothness with which the forces are transmitted to the piston. The rate of pressure rise is dependent upon the *mass rate of combustion of the mixture* in the cylinder.

A convenient and generally used graphical presentation depicting the pressure rise in the cylinder is known as the **pressure-crank angle** or the **pressure-time** (p-t) diagram. Such a diagram is shown in Fig. 8-3. In addition to showing the rate of pressure rise, the p-t diagram indicates, more clearly than the p-V diagram, the events occurring near TDC.

Figure 8-3 shows the relationship between pressure and crank angle for three different rates of combustion, namely, a high, a normal, and a low rate. Note that with the lower rates of combustion, it becomes necessary to initiate burning at an earlier point on the compression stroke because of the longer time necessary to complete combustion. Also, note that higher rates of pressure rise, as a result of the higher rates of combustion, generally produce higher peak pressures at a point closer to TDC, which is a generally desirable feature. Higher peak pressures closer to TDC produce a greater force acting through a larger portion of the power stroke, and hence, increase the power output. Practical operating considerations, however, place a limit on the rate of pressure rise. If the rate is too high, the forces exerted on the piston tend to become "impactive," causing rough, or "jerky" operation. Also, if the peak pressures become excessive, they tend to create a situation conducive to an undesirable occurrence known as detonation. A compromise between these opposing factors is obviously necessary. This is accomplished by designing and operating the engine in such a manner that approximately one-half of the pressure rise has

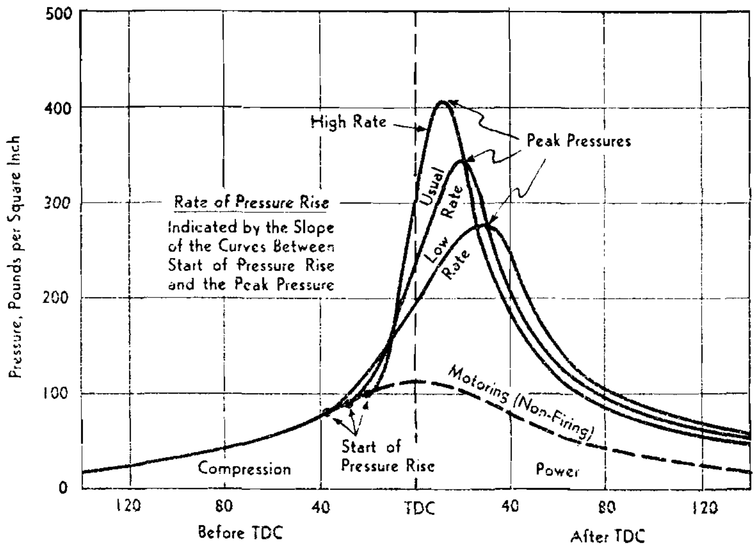


FIG. 8-3. Illustration of various combustion rates.

taken place as the piston reaches TDC. The result is a peak pressure reasonably close to the beginning of the power stroke, yet maintaining smooth engine operation.

8-5. Abnormal Combustion. The discussion thus far has been concerned with normal combustion. Under certain engine operating conditions, however, the air-fuel mixture has inherent characteristics conducive to various forms of abnormal combustion. The more important types of abnormal combustion, to be discussed in this article, are preignition and detonation.

(1) *Preignition*—If some portion of the boundary of the combustion chamber, such as a spark plug, exhaust valve, or carbon particle, becomes overheated under certain operating conditions, it is possible for this part to act in the same manner as the regular spark, and ignite

the adjacent fresh combustible charge. An entirely distinct flame front is thus produced, and the process is termed **preignition.** Such a condition is undesirable since combustion becomes both erratic and uncontrollable. Moreover, preignition tends to raise the temperatures and pressures in the chamber which cause the temperature of the "hot spot" to rise further, and encourage still earlier preignition on succeeding cycles. The cumulative effect not only tends to raise peak pressures and encourage the possibility of detonation, but also tends to cause the peak pressures to occur progressively earlier in the cycle. In fact, preignition may advance these peak pressures to such a point that they occur before the piston reaches TDC on the compression stroke. In such a case, the peak pressure in those cylinders which are preigniting will oppose piston movement during the last part of the compression stroke, thus decreasing total output as well as causing rough engine operation. Preignition may also cause damage, through burning, to those engine parts which are subjected to the extreme temperatures.

(2) *Detonation*—A combustible mixture of fuel and air, under certain conditions of temperature, pressure, and density, has the faculty of igniting without the assistance of an initiating flame or spark. Such an event is known as **auto-ignition.** It is comparable to the more familiar layman's term of "spontaneous combustion."

In SI engines, the main "actor" in the auto-ignition phenomenon is *the last portion of the unburned charge* in the combustion chamber. As the normal flame front proceeds across the chamber, it raises the pressure and temperature of the remaining portion of the unburned charge. Under certain conditions of pressure, temperature, and density of the unburned charge, this charge may auto-ignite and burn almost instantaneously, thus releasing energy at a much greater rate than during the normal combustion process. The extremely rapid release of energy causes pressure differentials of considerable magnitude in the combustion chamber which give rise to radical vibrations of the gaseous products, producing an audible knock. This condition is known as **detonation.**

Detonation is a most important aspect in the operation of SI engines, since it is *the major factor limiting the compression ratio* of an engine. The violent pressure fluctuations accompanying detonation can cause severe damage to the engine, and *sustained detonation cannot be tolerated.* In the operation of automobile engines, sonic warnings are given in the form of an audible "pinging" sound, and operating conditions can be eased to prevent damage. In a SI aircraft engine, however, the engine noises override any detonation sounds and thus increase the inherent danger of detonation damage.

Although little is actually known concerning the processes which occur during detonation, it is desirable to follow through the events which are believed to take place.

It is known that, in order to auto-ignite, the last unburned portion of the charge must reach and remain for a definite amount of time above

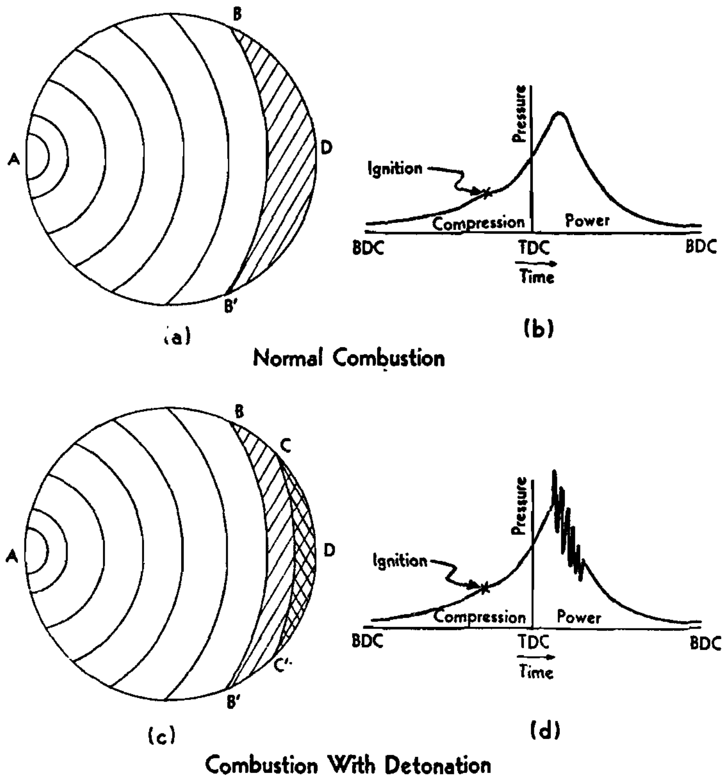


FIG. 8-4. Schematic presentation of the principles of normal and detonating combustion processes.

a certain critical temperature which is dependent upon conditions of pressure and density of the unburned charge. Once these conditions are reached, a "preparation" phase (similar to that described in Article 8-1) commences, followed by the "actual burning" phase. The "preparation" phase is known as the **ignition delay.**

Figure 8-4(a) represents a normal flame front travelling across a combustion chamber from A toward D, and increasing the pressure, temperature, and density of the unburned charge (area $BB'D$). If this unburned charge does not reach its critical temperature for auto-ignition, it will not auto-ignite, and the flame front BB' will proceed on through the unburned charge to point D in an orderly manner. The *p-t* diagram for such normal combustion is illustrated in Fig. 8-4(b).

Refer now to Fig. 8-4(c). If the unburned charge (area $BB'D$) reaches and remains above its critical conditions for auto-ignition, there is a possibility of detonation. In essence, a "race" develops between the flame front and the ignition delay. If the flame front can proceed from BB' to D and consume the unburned charge in a normal manner, prior to completion of the ignition delay period, there will be no detonation. If, however, the flame front is able to proceed only as far as, say CC', during the ignition delay period, then the remaining portion of the unburned charge (area $CC'D$) will detonate. Figure 8-4(d) is a *p-t* diagram showing the extreme pressure fluctuations occurring during detonation. Note that *detonation occurs, in a SI engine, near the end of combustion.*

In summary, if the unburned charge does not reach its critical temperature, there can be no detonation. If the ignition delay period is longer than the time required for the flame front to burn through the unburned charge, there can be no detonation. But if the critical temperature is reached and maintained, and the ignition delay is shorter than the time it takes for the flame front to burn through the unburned charge, then the charge will detonate. Consequently, in order to inhibit detonation, *a high critical temperature for auto-ignition, and a long ignition delay, are desirable qualities in SI engine fuels.*

8-6. Variables Affecting Detonation. As it was shown in Chapter V, the use of fuels with different anti-knock qualities exerts considerable effect on the detonation problem. As a result, the following discussion is a comparison on the basis of a given grade of fuel.

From the discussion of detonation in Article 8-5, four major factors appear to be involved in the tendency to produce or prevent detonation. These are temperature, pressure, time, and the density of the unburned charge. Since the effects of temperature, pressure, and density are closely interrelated, these will be consolidated into one group in the following discussion of variables affecting detonation. The time effect will be considered in the second group.

(1) *Temperature, pressure, and density factors*—In general, any action which tends to reduce the temperature of the unburned charge will tend to prevent detonation by reducing the possibility for the

charge to reach its critical temperature for auto-ignition. Likewise, any action which reduces the density of the charge tends to reduce the possibility of detonation by providing lower energy release. The following specific actions tend to reduce the possibility of detonation:

(a) Lowering the compression ratio lowers the pressures and temperatures.
(b) Reducing the mass of inducted charge by throttling or by reducing the amount of supercharging.
(c) Lowering the inlet temperature of the mixture.
(d) Maintaining the lowest possible temperature of the combustion chamber walls.
(e) Retarding spark timing so that the peak pressures are reached farther down on the power stroke and are thus of lower magnitude.
(f) Using excessively rich or lean mixtures to reduce flame temperature.

(2) *Time factors*—In general, any action which tends to increase the normal flame speed, or the length of the ignition delay period, will tend to reduce detonation. Such action will assist the normal flame front in "devouring" the last portion of the unburned charge before it can auto-ignite. The following specific actions tend, in most cases, to reduce the tendency to detonate:

(a) Increasing turbulence and thus increasing flame speed.
(b) Increasing engine speed (rpm) and thus increasing turbulence.
(c) Decreasing the distance which the flame front has to travel, thereby shortening the time required for the flame front to traverse the combustion chamber.
(d) Using a rich A/F ratio to obtain maximum flame speed.

Table 8-1 presents a general summary of the effect which the above listed variables exert on detonation. It indicates the apparent major effect on the unburned charge and the action which can be taken to reduce detonation. This table also shows the variables which the engine *operator* is usually able to control.

8-7. Combustion Chamber Design. Combustion chambers are usually designed with every possible attempt made to meet the following general objectives:

(1) To regulate the rate of pressure rise such that the greatest force is applied to the piston as closely after TDC on the power stroke as possible, with a gradual decrease in the force on the piston during

Variable	Major Effect on Unburned Charge	Action to be Taken To Reduce Detonation	Can Operator Usually Control
Compression Ratio	Temp. and p	Reduce	No
Mass Inducted	p	Reduce	Yes
Inlet Temperature	Temp.	Reduce	In Some Cases
Chamber Wall Temperature	Temp.	Reduce	Not Ordinarily
Spark Timing	Temp. and p	Retard	In Some Cases
A/F Ratio	Temp. and Time	Make Very Rich	In Some Cases
Turbulence	*Time*	*Increase*	Somewhat (Through Engine Speed)
Engine Speed	Time	Increase	Yes
Distance of Flame Travel	Time	Reduce	No

TABLE 8-1. Summary of variables affecting detonation in a SI engine.

the power stroke. The forces must be applied to the piston smoothly, however, thus placing a limit on the rate of pressure rise, as well as the position of the peak pressure with respect to TDC.

(2) To prevent the possibility of detonation at all times.

In the attempt to obtain these objectives, most present day combustion chambers are designed with the following factors in mind:

(1) To achieve the highest possible flame front velocity through the creation of high turbulence of the minute "swirl" type.

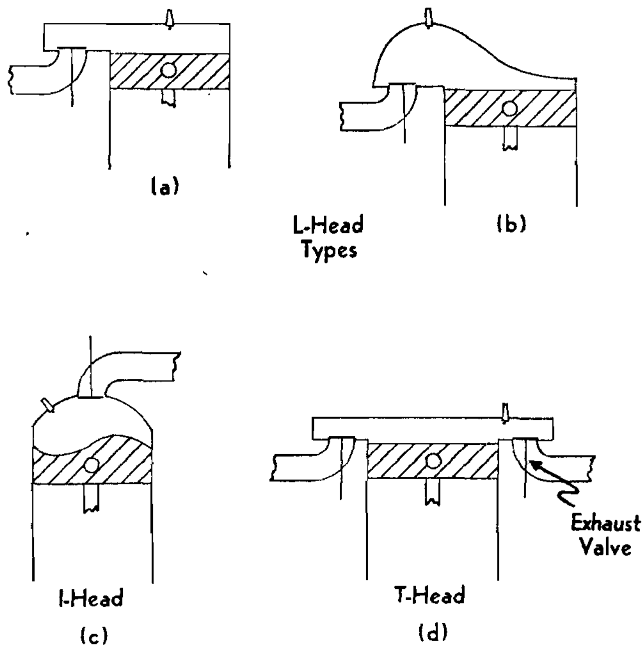


FIG. 8-5. Examples of typical combustion chambers (not to scale).

(2) To burn the largest mass of charge as soon as possible after ignition (consistent with a smooth application of force), with progressive reduction in the mass of the charge burned toward the end of combustion.

(3) To reduce the possibility of detonation by:

  (a) Reducing the temperature of the last portion of the charge to burn, through application of a high surface to volume ratio in that part of the combustion chamber where this portion burns. Such a ratio increases the heat transfer to the combustion chamber walls and thereby tends to reduce the temperature of the final unburned charge.

  (b) Reducing the distance for the flame to travel by centrally locating the spark plug or, in some engines, by using dual spark plugs.

Figure 8-5 shows a few representative types of combustion chambers, of which there are many more variations. A great majority of present day SI engines utilize some variation of the chambers indicated in (b) or (c) of Fig. 8-5, namely, an L-head or an I-head type. Note that these chambers are designed to obtain the objectives outlined above, namely, a high combustion rate at the start, a high surface to volume ratio near the end of burning, and a rather centrally located spark plug.

8-8. SI Eng. Fuel Injection. While most present day SI engines utilize a carburetor in which the fuel and air are integrally mixed, there are certain types which employ variations in the method of providing the mixture. Some aircraft engines, for instance, are equipped with a carburetor which meters the fuel, but the fuel is not actually injected into the air stream at the carburetor. Rather, it is injected in the area of the impeller of a built-in supercharger. Other types of SI engines go even further, and actually inject the fuel into the combustion chamber, but still utilize a spark plug to initiate combustion.

A variation of this latter type of engine has been of particular interest in recent years, and is generally referred to as a "stratified charge engine." An interesting and widely publicized engine of this general type is the Texaco Combustion Process engine.[1] In Article 8-5, it was stated that time is entailed in the factors affecting detonation. The Texaco Combustion Process engine is so designed that the combustible mixture is localized in an area near the spark plug, and *is not present in the chamber long enough to produce auto-ignition* and knock at the conditions of engine pressure and temperature existing. Figure 8-6 illustrates the method utilized to accomplish this result. The intake valve is shrouded so that the air in the combustion chamber is caused to assume a swirling motion. The fuel is injected into the chamber in a direction converging with the swirling air. A spark plug is located a short distance from, and downstream of, the fuel nozzle. The first portion of the fuel which is injected is mixed with the swirling air and carried toward the spark plug, where it is ignited. The remainder of the fuel is injected continuously into the swirling air, forming a patch of combustible mixture in the vicinity of the spark plug (area 2), and a nearly stationary flame front immediately following (area 3). Fresh combustible mixture is formed continuously and fed into the flame front, and the combustion products are carried away by the swirling air (area 4). Thus, the flame front remains essentially stationary and the combustible mixture is brought to it. Note that a combustible mixture exists only in areas 2 and 3 of the chamber. This arrangement insures that the flame front devours the unburned mixture before the mixture can auto-ignite, and thereby elim-

[1] E. M. Barber, Blake Reynolds, and W. T. Tierney, *The Elimination of Combustion Knock—Texaco Combustion Process*, paper presented at June 1950 meeting of The Society of Automotive Engineers.

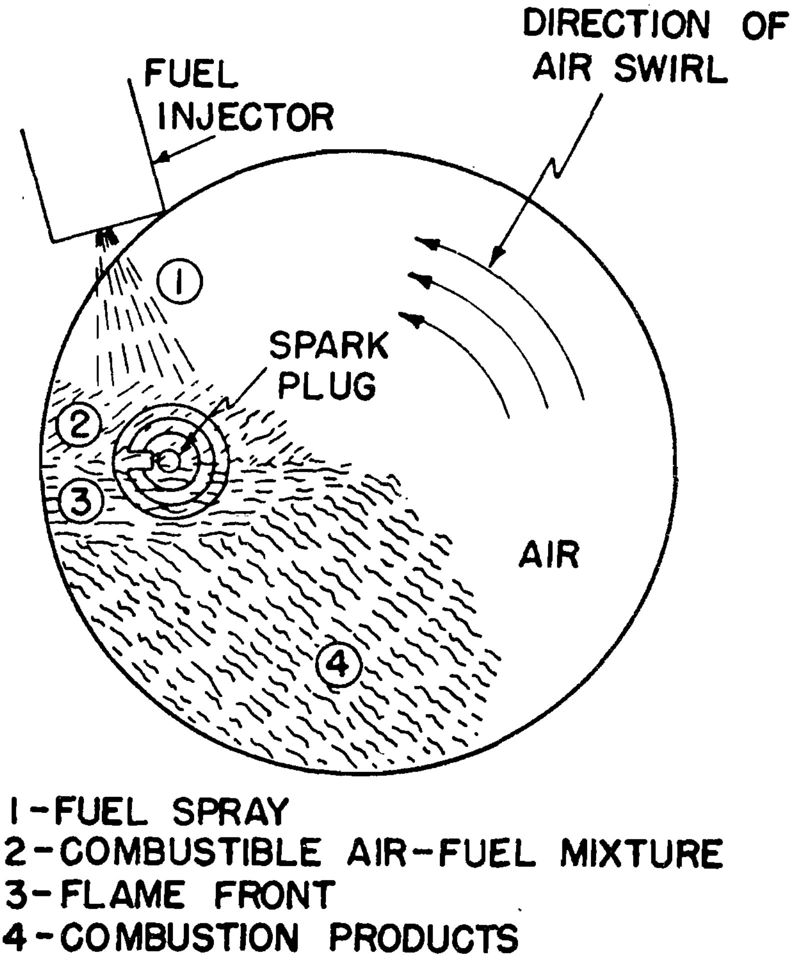


FIG. 8-6. Schematic Diagram of Texaco Combustion Process (E. M. Barber, Blake Reynolds, and W. T. Tierney, *The Elimination of Combustion Knock—Texaco Combustion Process*, paper presented at June 1950 meeting of The Society of Automotive Engineers).

inates detonation. While considerable research and development remains to be done on this type of engine, it promises many advantages over the conventional SI engine, as follows:

(1) The compression ratios may be increased without regard for the octane number of the fuel.

(2) Supercharging may be employed without regard for the octane number of the fuel.

(3) The ability to burn lean over-all mixtures results in favorable part load fuel economy.

(4) The ability to control the load by the mixture strength, without resort to air throttling, makes it attractive for use with two-stroke cycle engines.

(5) The engine may utilize fuels of any octane number and a wide range of boiling points. The specifications for the fuel can be considerably relaxed, and the yield of this fuel from crude petroleum, therefore, made proportionately greater than the present combined yield of motor gasoline, aviation gasoline, and diesel fuel, thus conserving petroleum resources.

In general, this type of engine appears to combine most of the best characteristics of the Otto and Diesel cycle engines without introducing the fuel quality requirements of either, and should have a promising future.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

8-1. A. M. Rothrock and A. E. Bierman, "The Knocking Characteristics of Fuels in Relation to Maximum Permissible Performance of Aircraft Engines," NACA Technical Report 655, 1939.

8-2. R. C. Spencer, "Preignition Characteristics of Several Fuels under Simulated Engine Conditions," NACA Technical Report 710, 1941.

8-3. H. Lowell Olsen and Cearcy D. Miller, "The Interdependence of Various Types of Autoignition and Knock," NACA Technical Report 912, 1948.

8-4. Cearcy D. Miller, "Relation Between Spark-Ignition Engine Knock, Detonation Waves, and Autoignition as Shown by High-Speed Photography," NACA Technical Report 855, 1946.

8-5. Charles F. Marvin, Jr., and Robert D. Best, "Flame Movement and Pressure Development in an Engine Cylinder," NACA Technical Report 399, 1931.

8-6. Charles F. Marvin, Jr., "Combustion Time in the Engine Cylinder and its Effect on Engine Performance," NACA Technical Report 276, 1927.

EXERCISES

8-1. What is meant by combustion as applied to hydrocarbon fuels?

8-2. What are the general conditions necessary for combustion?

8-3. What are the two general phases entailed in the combustion process?

8-4. What two rates enter into the normal flame front propagation? How does each of these rates advance the flame front?

8-5. Why is the flame front progress relatively slow at the beginning and near the conclusion of combustion, and faster during the intermediate portion?

8-6. Why is flame speed important?

8-7. What is the most important single factor affecting flame speeds? Why is this factor apparently so important?

8-8. How does the flame speed vary with changes in *A/F* ratio?

8-9. What is meant by the rate of pressure rise? What immediate single factor determines this rate?

8-10. At what point in the cycle is it desirable to locate the peak pressure? Why?

8-11. What is preignition? Auto-ignition? Detonation?

8-12. Why is detonation such a detrimental factor in engine operation?

8-13. During what portion of the combustion process does detonation occur in a SI engine?

8-14. What qualities are desired in a SI engine fuel in order to inhibit detonation?

8-15. What four major factors enter into the tendency to produce or prevent detonation?

8-16. What action can be taken with regard to the following variables, in order to reduce the possibility of detonation in a SI engine? Why?

(a) Compression ratio
(b) Mass of charge inducted
(c) Mixture inlet temperature
(d) Combustion chamber wall temperature
(e) Spark timing
(f) Turbulence
(g) Engine speed
(h) Distance of flame travel

8-17. Does the anti-knock quality of a fuel have any effect on detonation in an engine? In what way?

8-18. What are the general objectives in combustion chamber design?

8-19. In general, how are most present day combustion chambers designed to fulfill these objectives?

CHAPTER IX

COOLING

During the process of converting thermal energy to mechanical energy, high temperatures are produced in the cylinders of the engine as a result of the combustion process. A large portion of the heat from the gases of combustion is transferred to the cylinder head and walls, piston, and valves. Unless this excess heat is carried away and these parts are adequately cooled, the engine will be damaged. A cooling system must be provided not only to prevent damage to the vital parts of the engine, but the temperature of these components must be maintained within certain limits in order to obtain maximum performance from the engine. Adequate cooling is then a fundamental problem associated with reciprocating internal combustion engines.

This chapter deals with some of the fundamental cooling problems and the cooling systems in general use. A detailed discussion of the theory of heat transmission is beyond the scope of this text.

9-1. Engine Cooling Problem. Normal combustion of a hydrocarbon fuel in a reciprocating internal combustion engine produces a peak temperature between 3000 to 5000° F. A large quantity of the heat produced by the combustion is absorbed by the cylinder walls, cylinder head, piston, and valves, resulting in an increase in the temperature of these parts. If the absorbed heat is not properly dissipated by a coolant, liquid or air, the surfaces within the cylinder become overheated, causing difficulties that may result in the damage and even complete failure of the engine.

If the temperature at the base of the cylinder head is permitted to rise to 400° F or higher, the lubrication between the piston and the cylinder wall may break down resulting in a scuffing of the piston and a sticking of the rings. If the temperature goes above 500° F around the valve body, the valve guides may scuff due to a lubrication breakdown and the valves themselves will become excessively hot. This may result in the burning of the valves and seats. Furthermore, above 500° F, aluminum alloys lose their strength, resulting in the possibility of a rupture or break in these parts. In addition to the above difficulties, the overheating of the surfaces within the cylinder of a spark ignition engine will result in an increase in the tendency of the charge to detonate.[1] Overheating may also produce hot spots that could cause preignition. The cylinder temperature then is a controlling factor in the

[1] Article 8-6.

operation of an engine and must be maintained below a certain maximum allowable limit.

A well designed cooling system should provide adequate cooling, but not excessive cooling. While too much cooling is not as harmful to the engine as overheating, it is undesirable for several reasons. First, until the engine is heated to a certain minimum temperature, the fuel is not

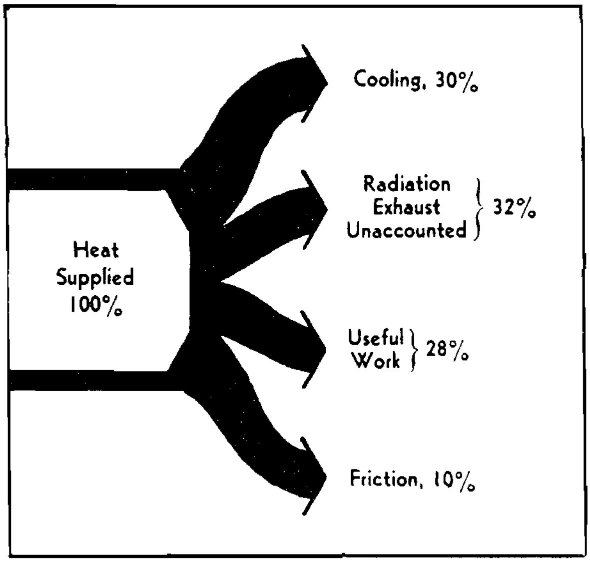


FIG. 9-1. Typical heat distribution for a reciprocating internal combustion engine.

vaporized sufficiently to give a homogeneous mixture with the air. This results in poor combustion and fuel economy, plus a possibility of crankcase dilution of the oil. Also, the cool engine with poor vaporization of the fuel has a tendency to cut out and run erratically. Secondly, as the engine temperature is decreased below the set minimum there will be an increase in the friction horsepower due to the high viscosity of the lubricating oil. A third difficulty from overcooling is that it may change the valve clearance and settings. The valves are set by the manufacturer to operate properly above a certain fixed temperature which is the lower limit of the normal operating temperature range of the engine.

Heat balance tests on reciprocating internal combustion engines show that between 25 and 30 per cent (Fig. 9-1) of the total heat supplied to the engine by the combustion of the fuel passes through the cylinder walls and is absorbed by the cooling medium (Also see Figs. 10-1 and 13-1 for typical curves of an engine heat balance test). The quantity of heat absorbed by the cooling medium depends upon the engine design, size, operating conditions, and the type of cooling medium. The cooling system, however, is designed to account for these factors. The function of the cooling system, then, is (1) to absorb and dissipate the excess heat from the engine in order to prevent damage to the engine and (2) to maintain a sufficiently high operating temperature in order to assure smooth and efficient operation of the engine.

9-2. Heat Transfer. A quantity of heat flows from the gases of combustion to the walls of the cylinder and cylinder head and through these metal parts to the coolant, where it is absorbed and dissipated. The rate of the heat flow through the metal walls is expressed by Fourier's Law which may be stated as

$$Q = \frac{kA\Delta t}{L} \qquad \text{(Btu/hr)} \tag{9-1}$$

where

Q = rate of heat transfer, Btu per hr
k = average thermal conductivity, Btu per hr-°F-ft
Δt = temperature difference between the hot and cold surfaces, ° F
L = length of the heat flow path, ft
A = surface heat transfer area, sq ft.

The rate of heat transfer is directly proportional to the heat transfer area, the temperature difference between the two surfaces, and the heat conductivity of the metal. It is also inversely proportional to the length of the heat flow path. Since the heat conductivity (k) of a metal wall is high, the metal parts offer little resistance to the flow of heat. The principal restrictions to this flow are the two surface or fluid films, one on each side of the metal wall. One film restricts the flow of heat from the hot gases to the combustion chamber walls, while the other hinders the flow of heat from the metal walls to the coolant.

Since the thickness of the fluid film cannot be measured in most cases, the conductivity (k) and the length of the heat flow path (L) in equation (9-1) are combined and replaced by a surface or film coefficient, h (Btu per hr-sq ft-° F). The equation for the rate of heat flow through fluid films is then restated as

$$Q = hA\Delta t \qquad \text{(Btu/hr)} \tag{9-2}$$

Film Coefficient. The film coefficient (h), which is the most important factor in equation (9-2), is a complex quantity that is dependent on many factors. In general, it depends on the length of the heat transfer path and the conductivity of the fluid. The thickness of the film surface, i.e., the length of the heat transfer path, is a function mainly of the velocity of the fluid flowing over the surface (Fig. 9-2) and the type of flow, i.e., the type and amount of turbulence. An increase in the piston speed will produce an increase in the velocity of the gases of combustion and an increase in turbulence, which tends to sweep away the stagnant film layer. This results in a rapid increase in the film coefficient and consequently an increase in the rate of heat trans-

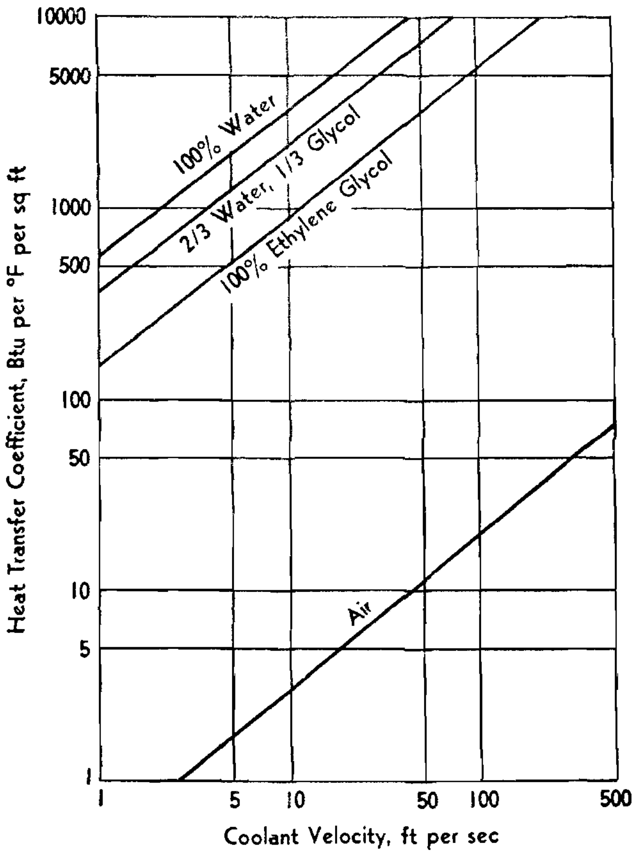


FIG. 9-2. Effect of coolant velocity on the film coefficient for flow through smooth tubes one inch in diameter. (By permission from *Combustion Engines*, by A. P. Fraas. Copyright 1951. McGraw-Hill Book Co., Inc.)

fer. Even though the rate of heat transfer increases with speed, the percentage of heat lost relative to the total heat supplied will have little change (Fig. 10-1). Another important factor in the determination of the film coefficient is the density of the gases within the cylinder. The density within a spark ignition engine will increase as the load is increased resulting in an increase in the rate of heat flow. The density of the charge of a compression ignition engine varies only a small amount with changes in load, and therefore the heat transfer rate from this factor remains nearly constant.

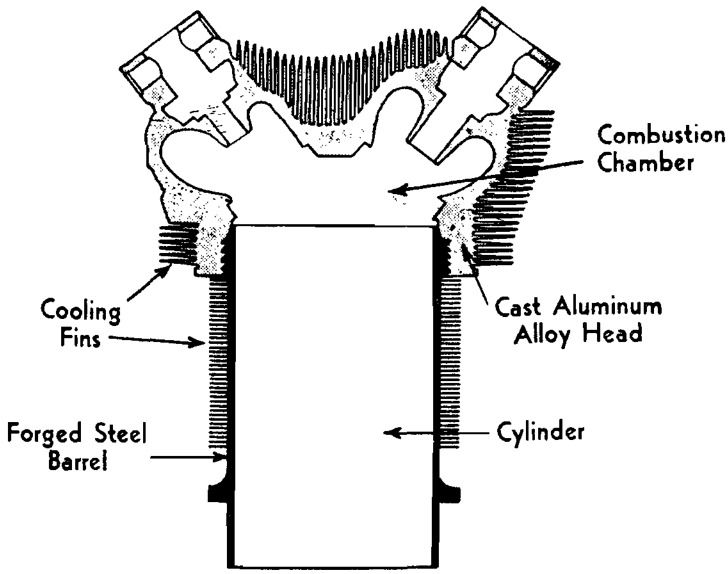


FIG. 9-3. Cylinder and cylinder head of an air cooled aircraft engine.

The specific heat, conductivity, and viscosity of the fluid also play an important role in the determination of the complex property, the film coefficient. These factors are particularly important as far as the coolant is concerned, for the factors vary widely depending upon the type of coolant, liquid or air, being employed to cool the engine. Figure 9-2 gives the film coefficients of water, a mixture of glycol and water, ethylene glycol, and air as a function of the coolant velocities. Water has the highest film coefficient at any particular velocity. Ethylene glycol,[2] which is used as an anti-freeze, has a film coefficient about one-fourth as great as water due to its relatively high viscosity. Air has a film coefficient approximately one one-thousandth as great as

[2] Trade names "Prestone," "Zerex," etc.

water, due basically to the low density of air. To obtain as satisfactory cooling with glycol and with a mixture of glycol and water as with water alone, higher coolant velocities must be employed. With air as the coolant, not only must higher coolant velocities be maintained but much greater cooling surface areas must be employed. This is accomplished by the utilization of cooling fins such as those used on cylinders and cylinder heads of radial air-cooled aircraft engines[3] (Fig. 9-3).

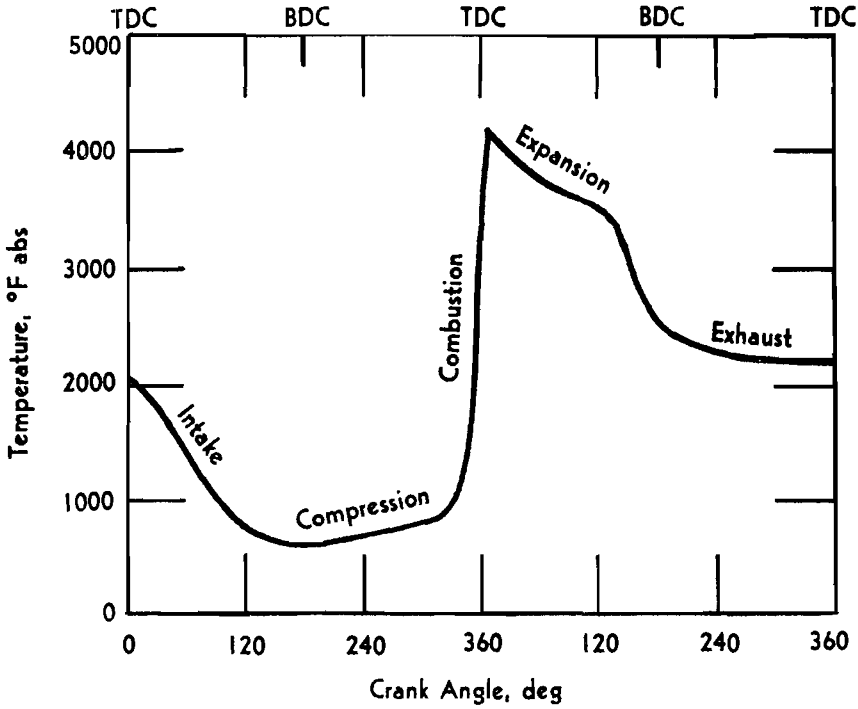


FIG. 9-4. Gas temperature variation in a cylinder (5.4 to actual comp. ratio; C_8H_{18}+59.5 air+residual gases). (By permission from *Internal Combustion Engines*, by L. C. Lichty. Copyright 1951. McGraw-Hill Book Co., Inc.)

Temperature. The temperature differences, (Δt) equations (9-1) and (9-2), also affect the rate of heat transfer through the cylinder walls and surface films. The temperature differences are governed by the temperatures of the gases within the cylinder, the walls, and the coolant. The temperature of the gases within the cylinder varies widely for the different processes during each cycle of operation. This is illustrated in Fig. 9-4. At the beginning of the intake stroke (induction

[3] Reference 9-1.

process), the temperature within the cylinder is that of the residual gases remaining from the previous cycle. During the intake stroke, the cool incoming charge is heated by the residual gases and the cylinder wall, i.e., the cool incoming charge rapidly decreases the temperature within the cylinder. The temperature rises during the compression process and increases very rapidly to the maximum peak temperature during the combustion of the fuel-air mixture. The expansion of the gases during the power stroke (expansion process) produces a pronounced decrease in the tempertaure. There is a rapid drop in the temperature when the exhaust valve is opened and a small drop during the remainder of the exhaust stroke. The peak temperature within the cylinder is greatly affected by a change in fuel-air ratio. The heat released from the combustion of a fuel-air mixture increases rapidly with an increase in the fuel-air ratio until it reaches a maximum at a ratio slightly richer than stoichiometric. Since the fuel-air ratio of a CI engine is always lower than stoichiometric, the gas temperatures increase linearly with an increase in the load resulting in an increase in the rate of heat transfer.

Area. The heat transfer area, equations (9-1) and (9-2), remains constant for any given engine. Although the period of time the area is exposed to the hot gases decreases with an increase in the speed, this is compensated for by the increase in the number of cycles per unit time. Consequently, the proportion of time that an area is exposed to the hot gases of combustion is practically independent of the rpm. The surface area within a cylinder depends upon the design of the cylinder and cylinder head. The greater the surface area the higher will be the rate of heat transfer.

There are two basic types of cooling systems employed in reciprocating internal combustion engines to absorb and dissipate the heat from the hot cylinders. The two systems in current use are *liquid cooled* and *air cooled*. Liquid cooled systems usually permit a better temperature control and a quicker warm up than is possible in an air cooled engine. The liquid cooled system, however, requires a replacement of the coolant when evaporation or boiling takes place. In general, it is heavier and requires more maintenance but permits a more even temperature control than the air cooled system.

9-3. Liquid Cooled Systems. Water is the cooling medium generally used in a liquid cooled system. However, other liquids or a combination of other liquids and water (anti-freeze solutions[4]) may be employed in the system to prevent freezing of the coolant.

[4] See Article 9-5 for anti-freeze solutions.

In a liquid cooled system, water or other solutions flow through the water jackets around the cylinders and cylinder heads and absorb the heat from the hot metal walls. The water is then cooled in a radiator, cooling tower, or cooler, and recirculated through the engine jackets.

The method of circulation of the coolant gives rise to three basic types of liquid cooled systems:

(1) Thermosiphon or natural circulation.
(2) Forced or pump circulation.
(3) Evaporative cooling.

Thermosiphon System. The system is designed so that the water may circulate naturally because of the difference in density of water at different temperatures. When the water is heated, its density de-

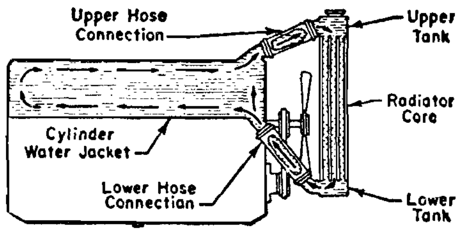


FIG. 9-5. Thermosiphon or natural circulation cooling system. (By permission of American Technical Society from *Automotive Engines*, 1951.)

creases and it tends to rise, while the colder molecules tend to sink. Circulation then is obtained as the water heated in the water jacket tends to rise and the water cooled in the radiator tends to sink, as shown in Fig. 9-5. However, even under the most favorable conditions the rate of circulation is slow. This makes the system unsatisfactory for heavy-duty engines that require high rates of heat transfer. This system may be used on small engines where the loads are light and only a relatively simple cooling system is necessary.

Forced Circulation System. The forced or pump circulating cooling system, Fig. 9-6, is similar in construction to the thermosiphon system except that a pump, which is usually a centrifugal type, is inserted in the system. The pump forces the water to flow through the system at a comparatively high rate. A header is usually employed to provide equal distribution of water to all of the cylinders. The header is supplemented by tubes or ducts which give high rate flow around critical sections of the engine, such as the exhaust valve seats. This type of system is employed on most diesel and automotive spark ignition engines.

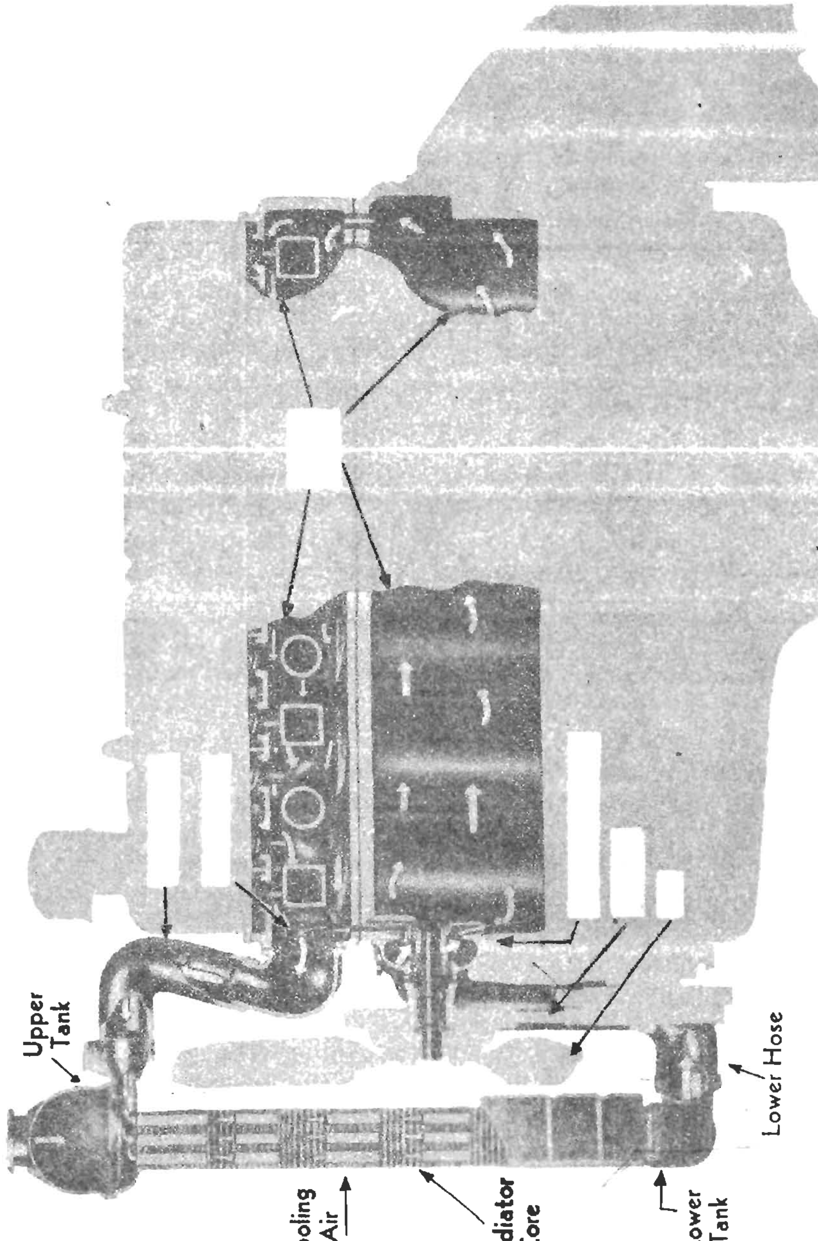
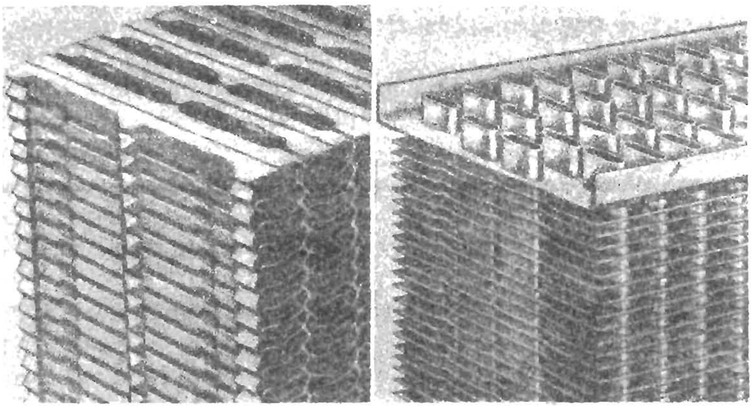


FIG. 9-6. Pump or forced circulation cooling system. (Courtesy of Ford News Bureau.)

The forced circulation cooling system can be divided into a number of component parts: *pump, water jacket, radiator, fan, and thermostat.*

(1) *Pump.* In automotive systems, the centrifugal pump (or pumps) is usually attached at some point on the front end of the cylinder block and is driven by the engine through a friction belt. Water from the bottom of the radiator (Fig. 9-6) enters the pump from the lower hose connection. After being circulated through the water jackets, the water leaves the cylinder block through the upper hose connection and enters the radiator, where it is cooled by the flow of cool air through the radiator core.



(a) (b)

FIG. 9-7. Radiator core construction; (a) cellular type and (b) tubular type with horizontal fins. (Courtesy McCord Corporation.)

(2) *Water jacket.* The water passages between the double walls of the cylinders and the cylinder heads are called the water jackets. The water jackets are usually cast as an integral part of the cylinder block and head. The jackets should cover the entire length of stroke in order to avoid unequal expansion of the cylinder and to prevent a breakdown of the lubricating oil film by excessive temperatures. Headers should be provided to furnish equal distribution of the cooling water to all cylinders. Extra cooling is usually provided around the hotter sections of the engine by means of special tubes and ducts.

(3) *Radiators.* The purpose of the radiator is to cool the water received from the engine. The radiator is a heat exchanger in which the water is cooled by the forced flow of atmospheric air around the pipes or tubes carrying the water.

The radiator consists of an upper and lower tank. Between the tanks is the core which divides the water into thin streams. In passing through the core, the heat from the water is transferred to the metal walls and to the air stream which is forced through by the fan. Typical radiator core construction is shown in Fig. 9-7.

(4) *Thermostat.* In most pump systems, a thermostat is located in the upper hose connection as shown in Fig. 9-6. The thermostatic valves, usually two to an engine, automatically maintain the normal minimum water temperature and permit a quick warm-up of the engine after starting. The opening and closing of the thermostat is controlled by the temperature of the water in the cooling system. During the warm-up period the thermostatic valve is closed, and the water pump circulates the coolant through the water jacket only. When the normal operating temperature is reached, the thermostatic valve opens and permits the water to circulate through the radiator. The normal operating-temperature range maintained varies from 140 to 170° F in the automotive engine. The thermostatic valves are usually set to start to open at 140° F.

(5) *Fan.* After the engine has reached a temperature of approximately 160° F and the thermostat is wide open, the coolant temperature is determined by the quantity and temperature of the air flowing through the radiator core. At low vehicle speeds with high loads, or at idling, the cooling air must be forced through the radiator core in order to provide sufficient cooling. This is accomplished by a belt driven fan.

As the road speed is increased, a quantity of air is forced through the radiator core by the ram effect produced by the motion of the vehicle through the air. At high speeds sufficient cooling air may be forced through so that the high output of the fan is not required. However, in the current automotive engines the fan can not be disconnected so that it continues to absorb power from the engine even when it is not required. At high engine speeds, the fan may absorb as much as ten per cent of the engine output.

Marine Pump System. In the many marine diesel and spark ignition engines, the radiator is replaced by a cooler as shown in Fig. 9-8. The fresh water leaving the engine is forced through the cooler by a fresh water pump. The salt water absorbs the heat from the fresh water in the cooler. The system shown is a forced or pump liquid-cooling system.

Evaporative Cooling. When the coolant is close to its boiling point, a high temperature gradient does not exist across the film and the heat transfer rate is low. The liquid immediately in contact with the hot metal may begin to boil. In vaporizing, the liquid abstracts much more heat from the metal surface than would otherwise be possible. In evaporating, one pound of water at atmospheric pressure will absorb

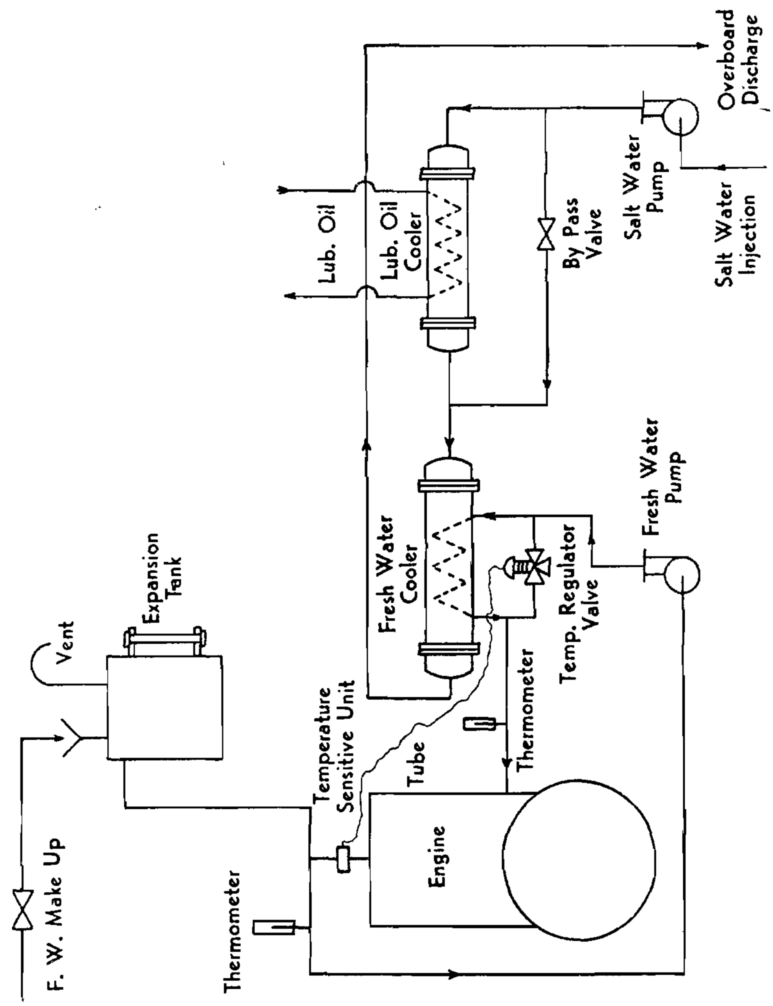


FIG. 9-8. Pump, liquid-cooling system for marine engines.

970 Btu in comparison to the absorption rate of one Btu per pound of water with an increase in the temperature of water one degree Fahrenheit. Thus, if the coolant is close to the boiling point, an extremely high rate of heat transfer may be obtained with a relatively small temperature gradient. When the liquid next to the surface vaporizes, the small vapor bubbles that are formed leave the metal surface and mix with the coolant where, in most cases, they are recondensed. The high latent heat of vaporization of water then provides a very high heat transfer rate in evaporative cooling systems.

The simplest form of evaporative cooling is the vertical hopper system. The cylinder and head are enclosed in a large water jacket that is open at the top to the atmosphere. The water has to be periodically replaced.

Another evaporative system maintains the water jacket under pressure by means of a throttle which allows only steam to leave the jacket. The steam leaving the jacket is condensed in a radiator and returned to the engine jacket by a pump. This system has a relatively complex system of controls.

The evaporative cooling system is not used widely and is adapted principally to relatively small engines. For large engines it is desirable to maintain lower temperatures in the water jackets to avoid high thermal stresses within the engine.

9-4. Air Cooling System. The film coefficient of air is low as compared to those of liquids as shown in Fig. 9-2. The best means of improving the rate of heat transfer, therefore, is to increase the surface heat transfer area. This is accomplished in the air cooled system by the liberal use of cooling fins on both the cylinder and the cylinder head. This is illustrated in Figs. 9-3 and 9-9.

The amount of cooling surface area of the fins required to provide adequate cooling depends on the type, size, and power output of the engine, and the conditions under which the engine is to operate. The fin surface area may be increased by the use of either deeper or more closely spaced fins, or both. Most air cooled engines employ fin spacings from 4 to 12 per inch. Machined fins, such as on a forged cylinder barrel, may have 12 spacings to the inch, while cast fins, such as in a cylinder head, may have only 4 to 6 spacings to the inch. The height of the fin depends on the manufacturing process, the strength of the material, and the durability desired. From the practical standpoint, the fins vary in height from less than one inch to two inches.

In the higher output engines, directed air flow is required to provide proper and uniform cooling. If the air flow is not directed, the front of

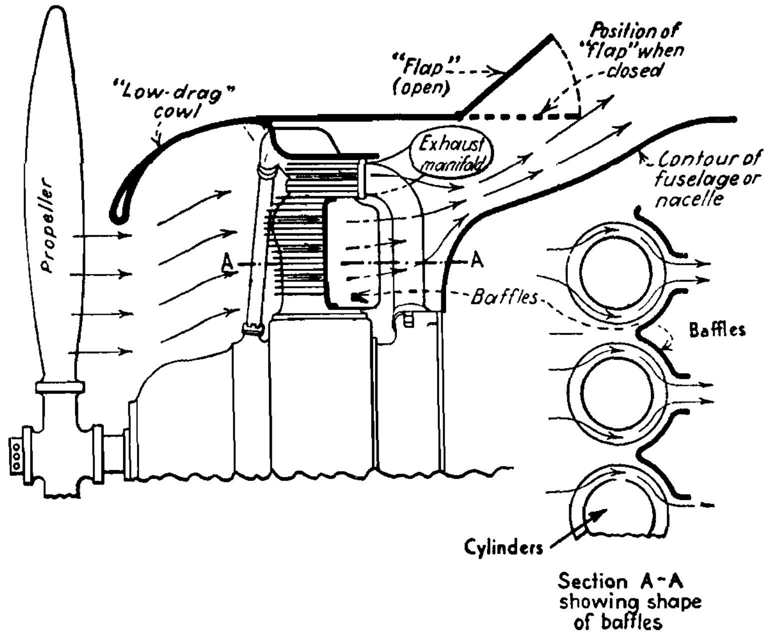


FIG. 9-9. Flow of cooling air through a radial engine. (By permission from *The Airplane and Its Engine* by C. H. Chatfield. Copyright 1940. McGraw-Hill Book Co., Inc.)

the cylinder placed in an air stream will be satisfactorily cooled while the down stream side of the cylinder may receive inadequate cooling. To force the air to flow between the fins on the down stream side, it is necessary to employ baffles as shown in Fig. 9-9. In order that air may flow through the baffles, which are similar to orifices, a pressure drop across the baffle is required. This method of cooling has been termed "pressure baffling" due to the pressure drop.

Even though some small low-powered engines obtain adequate circulation of the cooling air by convection, most engines require a forced circulation. With motorcycle engines and aircraft engines, the motion of the vehicle through the air is sufficient to induce an adequate air flow over the cylinders to cool the engine. The propeller aids in the forced circulation of the aircraft engine. However, for high powered aircraft engines, a cooling fan may be required on the propeller hub to insure a sufficient air flow around the cylinders. In applications other than motorcycles and airplanes, a cooling fan or blower is essential.

The controlling factor in the determination as to whether or not an aircraft engine is being properly cooled is the cylinder head temperature. This may be controlled to some extent by the opening and closing of the cowl flaps (Fig. 9-9) which regulate the quantity of air flowing between the cowling and the engine. A sheet metal cowling encloses the aircraft engine and provides not only better air circulation for cooling but decreases the profile drag of the engine installation. The cowling is usually designed to provide adequate cooling with the cowl flaps closed when the engine is operating at normal cruising conditions. When additional cooling air is required for high power engine output or for an engine idling on the ground, the cowl flaps are opened.

9-5. Antifreeze Solution. Special attention must be given to the liquid cooling system in climates where the temperature drops below 32° F. Water expands approximately ten per cent in volume when it freezes. An expansion of the water in freezing usually results in a damaged radiator, a cracked cylinder block or cylinder head.

The principal antifreezes employed to prevent the freezing of the coolant and damage to the engine in cold weather operation are: (1) methyl, ethyl, and isopropyl alcohols, (2) a solution of alcohol and water, (3) ethylene glycol (Prestone, Zerex, etc.), and (4) a solution of water and ethylene glycol. The per cent by volume of antifreeze, alcohol or ethylene glycol, used in solution with water is dependent on the atmospheric temperatures that will be encountered. If the temperature drops below the freezing point of an antifreeze solution, i.e., one containing more than fifteen per cent alcohol or ethylene glycol, the solution will not freeze solid but will form a slush. The water particles freeze while the alcohol or ethylene glycol remains liquid producing the slush. The slush while it will not crack the cylinder block may clog the small passages of the radiator and water jacket sufficiently to prevent the proper circulation of the coolant. This may result in localized overheating of the engine. Also, if alcohol is used, it may boil off leaving the engine coolant subject to future freezing.

When ethylene glycol is added to water, the boiling point of the solution is raised. The alcohols on the other hand when mixed with water will lower the boiling point of the solution so that alcohol is subjected to a boiling off and must be periodically checked and replaced. When water or alcohol are confined under pressure greater than atmosphere, a higher temperature must be reached before the water or solution of water and alcohol will boil. A simple pressurized system incorporated in some automotive engines makes use of this fact to prevent the boiling off of the coolant.

In a pressurized system, the atmospheric vent or tube in the radiator is separated from the radiator proper by a pressure relief valve. The valve is usually incorporated in the radiator cap as shown in Fig. 9-10. As the engine is warmed up, a pressure is created in the cooling system. The pressure relief valve is generally set to open at a pressure from four to seven pounds in order to prevent an excessive pressure building up in the cooling system. At these pressures, the boiling point of the

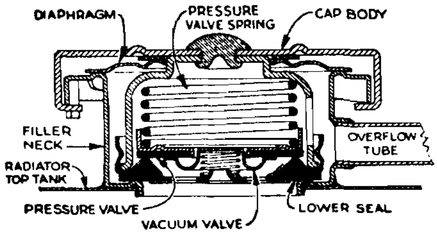


FIG. 9-10. Pressure relief valve in radiator cap. (Courtesy of Stant Manufacturing Co.)

coolant is raised 15 to 20° F which tends to reduce the loss of alcohol through evaporation

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

9-1. A. P. Fraas, *Combustion Engines*, McGraw-Hill Book Co., New York, 1948.

9-2. L. C. Lichty, *Internal Combustion Engines*, McGraw-Hill Book Co., New York, 1951.

9-3. I. Frazee and E. L. Bedell, *Automotive Fundamentals*, American Technical Society, Chicago, 1950.

9-4. I. Frazee, E. L. Bedell, E. Vack, *Automotive Engines*, American Technical Society, Chicago, 1951.

9-5. F. T. Morse, *Elements of Applied Energy*, D. Van Nostrand Co., New York, 1947.

EXERCISES

9-1. Why must a reciprocating internal combustion engine be cooled?

9-2. What happens if the engine temperature is permitted to rise to 400° F? 500° F?

9-3. Will overheating of the cylinder of a SI engine increase or decrease the tendency to detonate?

9-4. Why is overcooling an engine harmful?

9-5. Write the equation for the rate of heat transfer through a metal wall? through a film?

9-6. What is the film coefficient?

9-7. Upon what does the film coefficient in general depend?

9-8. If the speed of an engine is increased, will the heat transfer rate increase or decrease? Why?

9-9. Which has the highest film coefficient, water or air? Why?

9-10. How does the temperature within the cylinder vary during a cycle?

COOLING

9-11. What is the effect of a change in fuel-air ratio on the temperature within a cylinder?

9-12. What type of coolant is usually employed in a liquid cooled system?

9-13. Name three types of liquid cooled systems.

9-14. How is the circulation accomplished in a thermosiphon system?

9-15. Can the thermosiphon system be used in a high output engine? Why?

9-16. What are the differences between the forced and the thermosiphon systems?

9-17. Describe the operation of the thermostat.

9-18. What are the advantages of using a thermostat in the system?

9-19. What is the purpose of the fan? Is it required at all times? Why?

9-20. How does a marine diesel engine cooling system differ from one used in an automotive engine?

9-21. What is evaporative cooling?

9-22. Why are fins required in an air cooled system?

9-23. What is the purpose of baffles? Cowl flaps? Cowling?

9-24. What is the purpose of pressurizing a liquid cooled system? How is it accomplished?

9-25. Name two main types of antifreeze solutions.

CHAPTER X

SPARK IGNITION ENGINE PERFORMANCE

The preceding chapters have been concerned with a general study of certain subject matter which enters into the makeup of a SI engine. Chapter V presents a discussion of the source of energy used to drive the engine, while Chapter VI discusses certain aspects regarding the conversion of the chemical energy of the fuel into heat, and the problems associated therewith. Chapter III entails a discussion of certain theoretical cycles that represent the ultimate which the actual cycle approaches. All of these studies have been preparatory to an understanding of the most important phase, namely, the engine performance.

10-1. Heat Balance. In Article 1-8, the flow of energy through a reciprocating engine was covered briefly. A certain quantity of energy is provided to the engine in the fuel. Another, and smaller quantity of energy arrives at the driveshaft and can be utilized to do useful work. Between the input and output, there is an amount of energy that escapes in a non-usable form, and which is called the energy loss. It is the purpose of the designer and operator to reduce this loss to a minimum, thereby collecting the maximum amount of energy at the driveshaft in comparison with the amount invested with the introduction of the fuel. The accounting of the energy supplied, the energy losses, and the useful energy output may be expressed by what engineers commonly call a *heat balance,* which is merely a determination of the disposition of the energy which is provided to the engine in the fuel. In the combustion chamber area, heat loss to the coolant and to the exhaust is predominant. From the piston to the driveshaft, the pumping losses and losses due to friction reduce the amount of energy arriving at the driveshaft. Throughout the engine, unaccountable losses reduce further the useful energy realized. The "balance sheet" of energy disposition can be determined for a given engine and illustrated in a diagram such as Fig. 10-1, which is a typical heat balance for a SI engine. The figure shows the areas which must be reduced, and that which must be increased, to obtain better engine performance.

10-2. Engine Performance. Engine performance is really a relative term. It usually means how well an engine is "putting out" or doing its job with relation to the energy fed in, or, how effectively it provides useful energy in relation to some other comparable engine. The succeeding articles will be devoted to a study of engine performance, the bases upon which this performance is compared, and the manner in which it is affected by operating variables. They will be presented in the following sequence:

(1) A discussion of the processes and events occurring in the combustion chamber area of an actual engine and their effect on engine output, and

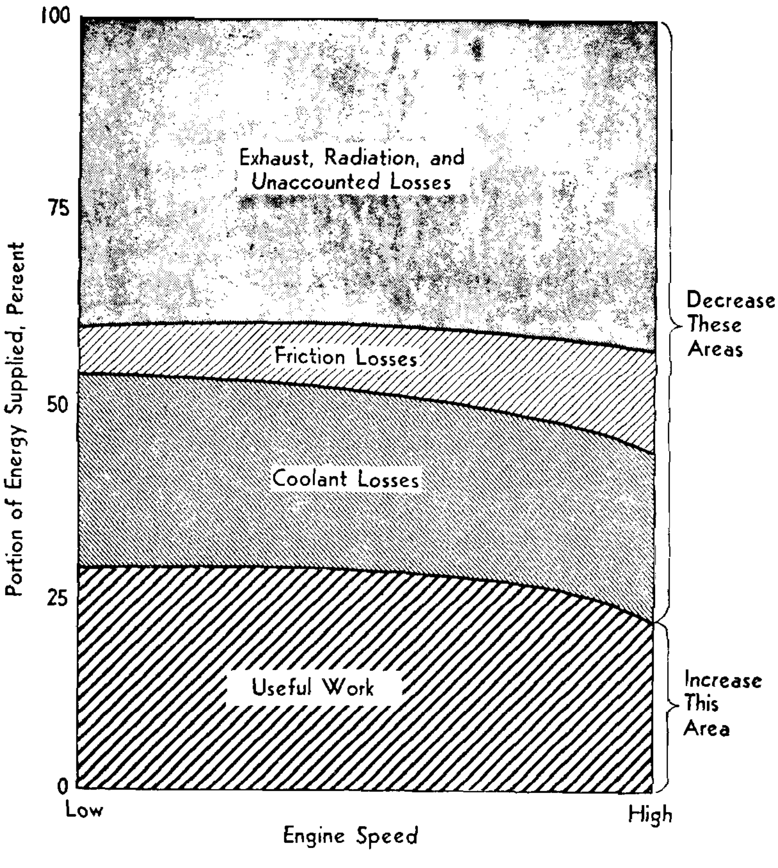


FIG. 10-1. Typical heat balance diagram for a SI engine. (Full throttle.)

(2) A presentation of the performance curves of an operating engine, their interrelationship, and how they are affected by engine operating variables.

10-3. Variations Between the Air Cycle and the Actual Cycle. In Chapter III, three approximations to conditions existing in the actual engine were discussed, that is, the ideal cycle, the air cycle, the fuel-air cycle, and the actual cycle. Most of the work on the analysis of the various cycles in Chapter III was based on the *air* cycle. The *actual* cycle shows the conditions existing in the cylinder of an actual engine, and will be the basis of the discussion in this chapter. Since the actual cycle entails several variations from the air cycle, the more important differences will be reviewed at this point in conjunction with Fig. 10-2:

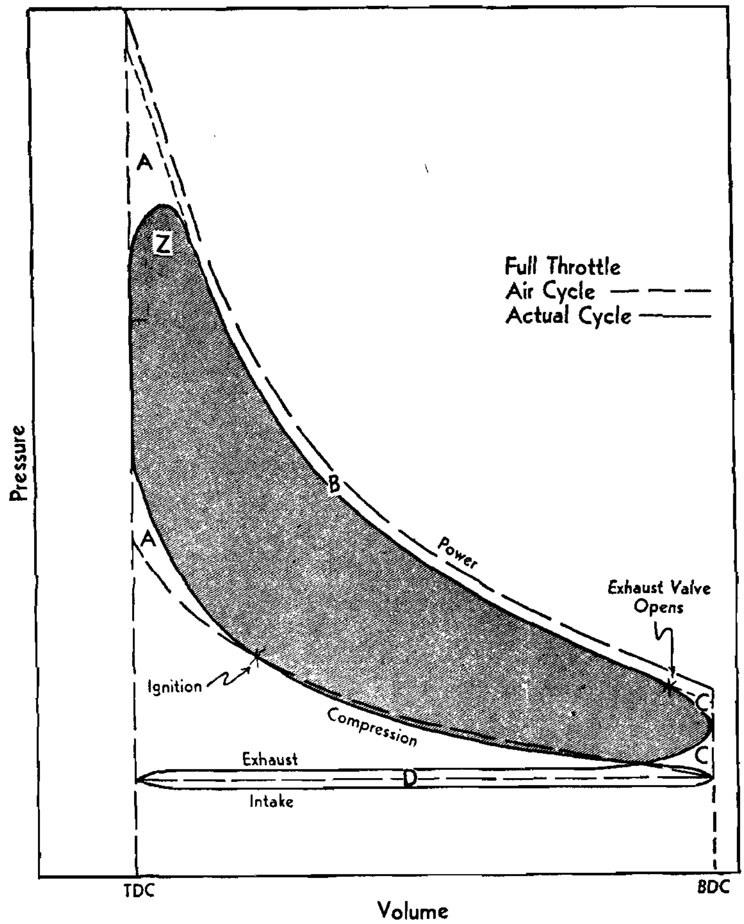


FIG. 10-2. Comparison of air cycle p-V diagram and actual cycle indicator diagram (not to scale).

(1) *Variation of specific heat*—In both the air cycle and the actual cycle, the specific heats of the working substance are assumed to vary with temperature. Since the difference between the specific heats at constant pressure and constant volume is constant ($c_p - c_v = \frac{R}{J}$), an

increase in the specific heats means a decrease in their ratio[1] (k), and a reduction in the maximum temperature in the cycle.[2] This results in a decrease in the thermal efficiency and a reduction in net work, and in this, both cycles are alike. However, in the actual cycle, the products of combustion have higher specific heats than air in the temperature ranges involved. Note the behavior of CO_2 in Fig. 2-5. Therefore, the maximum temperature and the value of k are reduced further, thus resulting in additional decrease in thermal efficiency and net work for the actual cycle.

(2) *Dissociation*—At the high temperatures involved, the products of combustion tend to break up, or dissociate, into other compounds such as CO, H_2, and O_2 in an endothermic reaction. This results in a reduction in available energy and tends to further reduce the thermal efficiency and net work of the actual cycle.

(3) *Non-instantaneous burning*—Addition of heat, in the air cycle, was assumed to be instantaneous. In the actual cycle, the burning requires a finite time. This factor causes a reduction in the net area of the p-V diagram, with consequent loss in net work (Area A of Fig. 10-2).

(4) *Heat losses*—No heat loss was assumed in the air cycle. The actual cycle entails heat losses in the cylinder area, with further reduction in the net work (Area B of Fig. 10-2).

(5) *Exhaust gas effect*—All of the products of combustion in an actual engine are not exhausted during the exhaust stroke, primarily because of the clearance volume. The succeeding fresh charge is consequently heated and diluted by the remaining exhaust gas, thereby tending to lower the net work of the cycle due to the smaller mass admitted.

(6) *Valve timing and blow-down losses*—Instantaneous rejection of heat was assumed in the air cycle. It is not possible to obtain such rejection in the actual cycle due to the inertia of the exhaust valves and exhaust gases. Actual exhaust valve opening commences prior to the time at which the piston reaches BDC. This effect causes a further loss in the net work of the actual cycle (Area C of Fig. 10-2).

(7) *Pumping loop*—In the actual cycle, the pressure in the combustion chamber is different during the intake stroke from that during the exhaust stroke. This variation produces a negative work area, or pumping loop, on the p-V diagram (Area D of Fig. 10-2). The negative work comprises a part of the fhp, and thus reduces engine output.

[1] See Fig. 2-5.

[2] The reduction in maximum temperature of the cycle can be shown through an expression of $q_s = c_v(T_3 - T_2)$, from which $T_3 = T_2 + \frac{q_s}{c_v}$. The same reasoning may be also applied to the constant pressure process.

The total effect of the above listed variations is to reduce the net work area of the actual cycle from that of the air cycle, as indicated in Figure 10-2. Consequently, the output of the actual engine will be lower than a hypothetical engine operating on the air cycle.

It should be noted that combustion in the actual cycle tends to occur partially at constant volume and partially at constant pressure. This latter tendency is indicated in the vicinity of point Z in Fig. 10-2. Thus, the actual SI engine cycle has a tendency toward the dual combustion cycle, rather than to follow strictly the Otto cycle.

10-4. The Indicator Diagram. The p-V diagram measured on an actual engine is referred to as an *indicator diagram* (or *indicator card*) and is essentially a record of the pressures existing in the cylinder at various positions of the piston throughout the engine cycle. It serves two important functions. Its *area* may be measured by a planimeter and the imep determined. From the imep, the total piston displacement, and the number of power strokes per given time, the power which is produced in the engine cylinder may be determined (Chapter IV). Also, the *shape* of the indicator diagram is an indication of the manner in which the various processes affecting the power produced in the cylinder are taking place.

As pointed out in Article 4-5, the standard practice in industry is to measure bhp on a "firing" engine, estimate fhp by measuring the power required to move the "non-firing" engine, and obtain ihp by adding these values. The use of the indicator diagram for determining ihp is generally limited to laboratory research, or in the testing of new engine designs. Such a diagram, however, assists materially in an explanation of many of the processes occurring in the combustion chamber area, and will be used here for that purpose.

The instruments used to record the indicator diagram are usually referred to as *engine indicators.* While there are several types of these instruments, they may be divided generally into two classes—mechanical and electrical.

In the mechanical system, a small cylinder and piston are attached to the engine cylinder, such that the pressures in the engine cylinder force the small piston to move against the force of a calibrated spring. The movement of the small piston is proportional to the pressures in the cylinder, and is transmitted through a magnifying linkage to a stylus. A sheet of paper is wrapped around a small drum which is caused to oscillate in accordance with the strokes of the engine piston. The stylus moves up and down parallel to the axis of the oscillating drum, producing a record of engine cylinder pressures against the engine piston position, or cylinder volume. This record is a p-v diagram for the cylinder, or an indicator card. If the drum is caused to

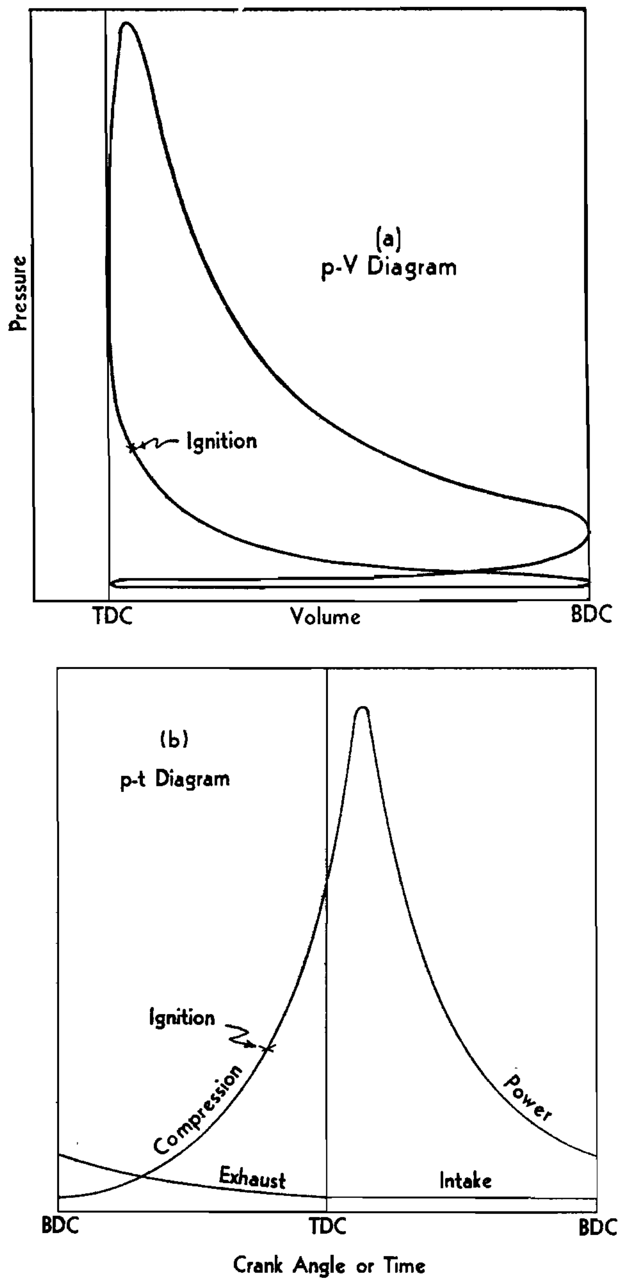


FIG. 10-3. Comparison of general shape of p-V and p-t diagrams (not to scale).

rotate at constant velocity in accordance with the crankshaft rotation, the record produced will be a pressure-time (*p-t*) diagram or a pressure-crank angle diagram. An indicator of the type described is similar to the type used with steam engines, and is useful only with large, slow speed CI engines. It is not satisfactory for use with small engines because it radically affects the engine compression ratio, and the inertia of the mechanical linkages is too great for satisfactory use with engines of high speed. Certain high speed mechanical indicators have been developed which partially reduce some of these disadvantages.

A number of various types of electrical indicators are in use at the present time. These instruments overcome most of the faults of the mechanical type. In general, they all employ a pressure pickup element which is attached to the engine cylinder in such a manner as to communicate with the combustion chamber gas. Changes in combustion chamber pressure are transmitted through an amplifier to the vertical plates of an oscillograph. The pressure pickup element contains a quartz crystal, strain gage, electromagnetic system, or other such device, which is affected by the pressure in the engine cylinder and transmits responses to the oscillograph accordingly. The horizontal plates of the oscillograph are electrically related to the crankshaft rotation. The resulting picture on the oscillograph is, thus, a *p-t* diagram. This picture may be photographed and the image is referred to as an oscillogram. A *p-V* diagram may be constructed from the data appearing on the oscillogram.

Figure 10-3 shows the general difference in appearance of the *p-V* diagram (indicator diagram) and the *p-t* diagram (pressure-crank angle diagram).

Each type of diagram has certain advantages over the other. The *p-V* diagram is useful in computing the net work per cycle, in computing the imep, and in denoting the manner in which various processes are occurring, particularly during the intake and exhaust strokes. The *p-t* diagram is most useful as an indication of the events occuring near TDC, especially the rate of pressure rise during combustion and the effects associated herewith.

Figure 10-4 illustrates typical full throttle and part throttle indicator diagrams for a four-stroke cycle SI engine.

There are two distinct areas, or loops (*A* and *B*) enclosed by the process lines during one complete cycle of operation. In Chapter II it was shown that the area under a process line on the *p-V* diagram indicated the work done during that process. Also, in Chapter III, it was indicated that the work done during the compression and exhaust strokes was negative, while that done during the power and intake strokes was positive. A comparison of the areas under the four process lines (Fig. 10-4), with proper application of positive and negative signs, will show that area A represents positive work, while area B represents negative work. Area A is a measure of the *indicated net work* produced in the cylinder. Area B is usually termed the *pumping loop*, and is a measure of the pumping losses encountered during the cycle. These pumping losses, which are entailed with exhausting the burned gases and bringing in the fresh charge, constitute a portion of the work required by the engine to overcome friction, and thus constitute a part of the basis for the computation of friction horsepower.

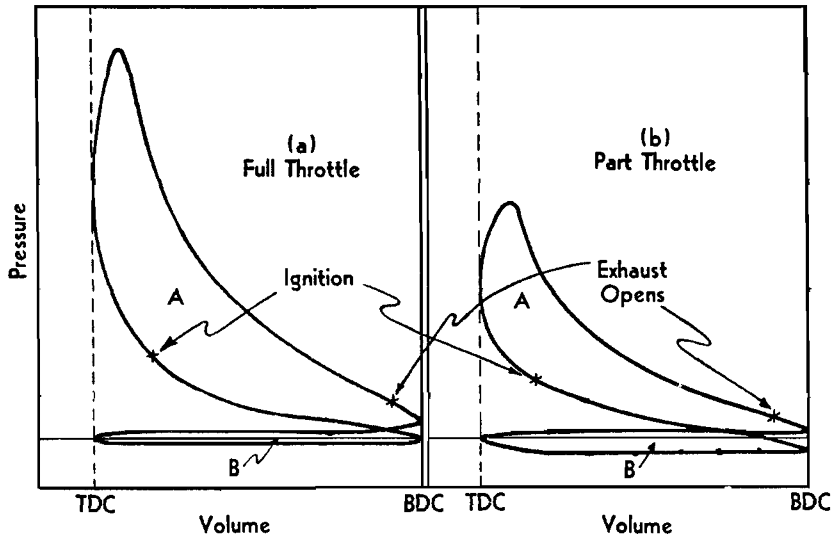


FIG. 10-4. Comparison of typical indicator diagrams for a SI engine at full throttle and part throttle operation (not to scale).

It should be noted that the ignition point occurs prior to TDC on the compression stroke. This timing of the spark is necessitated by the finite time interval required for combustion. The most desirable setting for given operating conditions is generally determined by experiment, and is usually that setting which will produce the maximum power output of the given engine, without roughness or detonation. As discussed in Article 8-4, the spark is generally timed so that approximately one-half the pressure rise has taken place as the piston reaches TDC.

It should be noted further that the exhaust valve opens prior to the time the piston reaches BDC on the power stroke. This condition is necessitated by the inertia of both the exhaust gases and the exhaust valve. At the high speeds involved, it takes a finite length of time to get the exhaust valve open and the flow of gases started. If the exhaust valve did not start to open until BDC, the pressures in the cylinder would be considerably above atmospheric during the first portion of the exhaust stroke, thus increasing the size of the negative loop (B) and thereby decreasing the work output of the engine. Opening the exhaust valve early reduces the pressures near the end of the power stroke and thus causes some loss of useful work on this stroke. But the over-all effect of opening the valve prior to the time the piston reaches BDC generally results in an increase in the over-all work of the engine.

Two important differences in the indicator diagrams (a) and (b) of Fig. 10-4 are readily evident. The area of loop A is greater at full throttle than at part throttle, and the pumping loop (B) is greater at part throttle than at full throttle.

The cause of the first difference is due to the mass of charge admitted. A full (open) throttle offers much less restriction to the flow of the air-fuel mixture and a greater mass of mixture, therefore, enters the combustion chamber, causing loop A to be greater at full throttle. At part throttle, the restriction to flow offered by the throttle reduces the mass of fresh charge entering the combustion chamber, and therefore reduces the area of loop A and the work output.

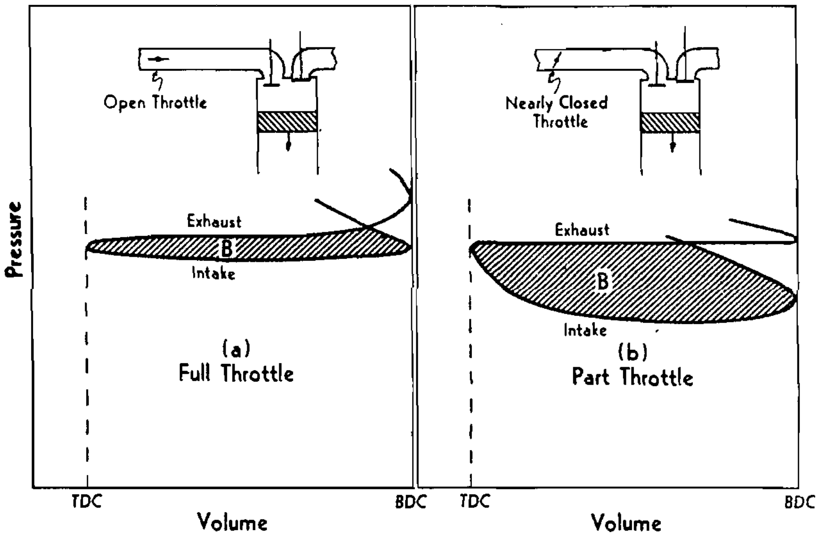


FIG. 10-5. Comparison of full throttle and part throttle pumping loops of indicator diagram (not to scale).

The effect of part throttle on the pumping loop (B) is illustrated in Fig. 10-5, which is an expanded diagram of the pumping loops for full throttle and part throttle of Fig. 10-4. The restriction to flow of the charge by the throttle is one of the major factors affecting the size of the pumping loop. At full throttle, Fig. 10-5(a), the throttle offers very little restriction to the flow of charge into the cylinder. With a well designed induction system, as the piston descends on the intake stroke, the pressure in the cylinder remains relatively close to atmospheric. When the throttle is partly closed, however, a considerable restriction to the flow of the incoming charge is offered in the intake system. When the piston descends on the intake stroke, the incoming charge is unable to flow into the chamber as rapidly as the volume of the chamber is increasing due to the descending piston. Consequently, the pressure in the combustion chamber is lowered. This causes a reduction in the cylinder pressures during the intake stroke, an increase in the size of the pumping loop as indicated in Fig. 10-5(b), and an increase in the negative work represented by the pumping loop. Thus, the friction work is increased, and the over-all output of the engine is decreased. It can be seen, therefore, that both the valve timing and the throttle opening exert considerable influence on the pumping loop, and consequently on the fhp and the over-all engine output.

10-5. Indicator Diagram Diagnosis. In Article 10-4, it was stated

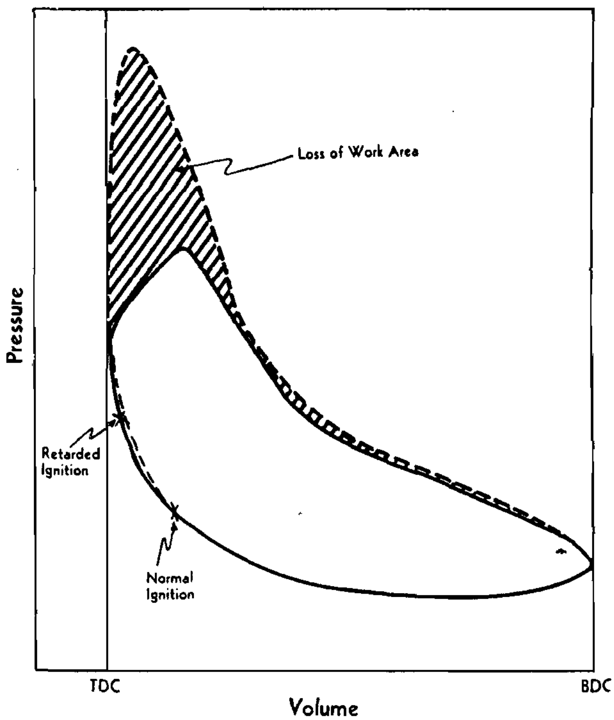


FIG. 10-6. Effect of retarded ignition on indicator diagram (not to scale).

that the shape of the indicator diagram produced an indication of the manner in which the various processes affecting the power produced in the cylinder are taking place. Some of the more important variables which affect the shape of the indicator diagram, and thus the indicated net work, will be discussed in this article. In each case, a schematic indicator diagram will be used to show the trend in the shape of the indicated net work loop as the variable under consideration is changed. In order to facilitate comparison with the "normal" loop of Fig. 10-4(a), this latter will be superimposed on each diagram in dotted lines.

(1) *Ignition timing*—Fig. 10-6 indicates the effect of a late spark (retarded ignition). Due to the finite burning time, peak pressures are

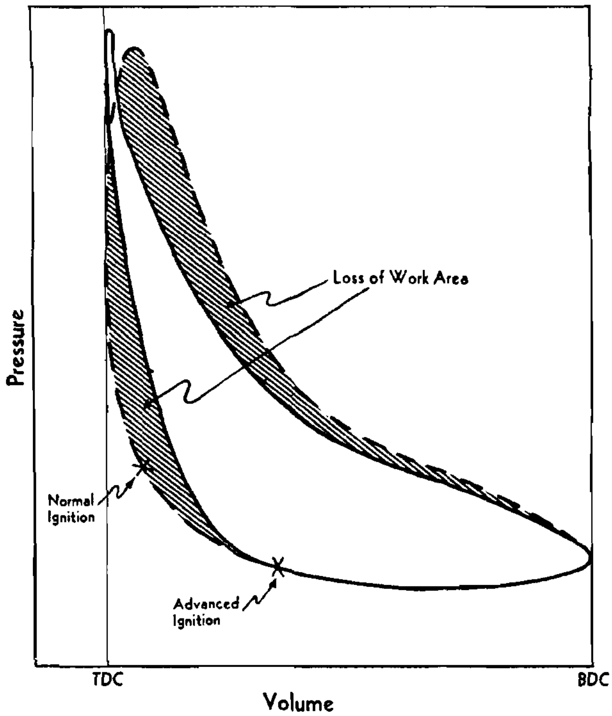


FIG. 10-7. Effect of advanced ignition on indicator diagram (not to scale).

not reached until the piston has progressed considerably toward BDC on the power stroke resulting in lower peak pressures. The indicated net work area is thereby reduced and the work produced in the cylinder is therefore lowered. A slow combustion rate with normal spark setting would produce essentially the same effect.

If the spark is set to occur early (advanced), the indicator diagram tends toward the shape shown in Fig. 10-7. Again, the indicated net work area is reduced and the work produced is lowered.

(2) *Dilution of the fresh charge*—If the fresh charge is diluted by exhaust gas, there is less energy available for liberation, and the rate of flame propagation is slower. The result is lower pressures during the power stroke and less indicated net work, as indicated in Fig. 10-8.

(3) *High exhaust back pressure*—If the exhaust system is improperly designed such that the valve is too small or other restrictions are im-

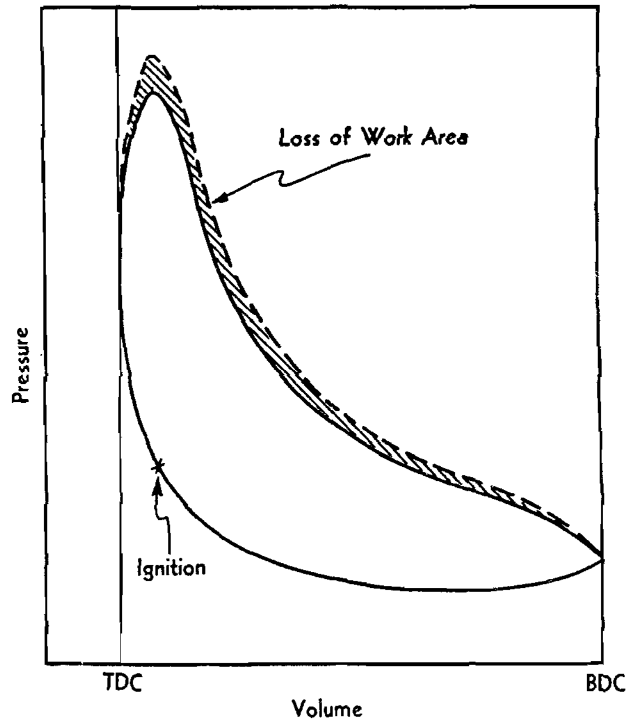


FIG 10-8. Effect of diluted charge on indicator diagram (not to scale).

posed, or if the exhaust valve is opened too near BDC on the power stroke, the result is a tendency for the pressures during the exhaust stroke to be greater. This results in an increase in the size of the pumping loop, Fig. 10-9, with consequent reduction in the over-all work of the engine.

(4) *Improper intake system design*—If restrictions are imposed upon the flow of the combustible charge through the intake system by undersized intake valves or other impediments, the effect is a reduction in pressure in the cylinder during the intake stroke. This effect is similar to that produced with part throttle. The result is an increase in

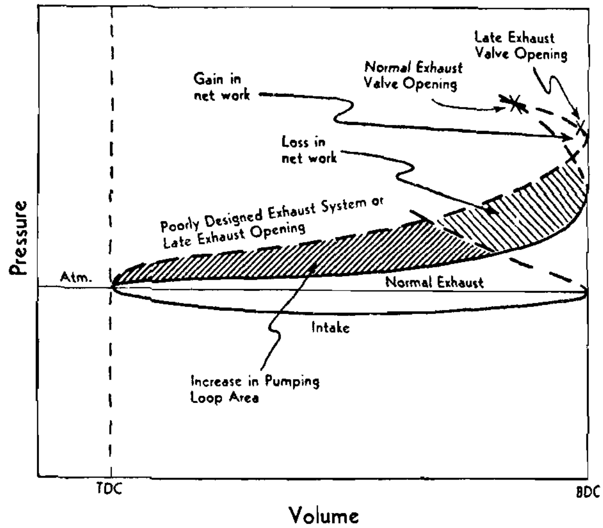


FIG. 10-9. Effect of high exhaust back pressure on indicator diagram (not to scale).

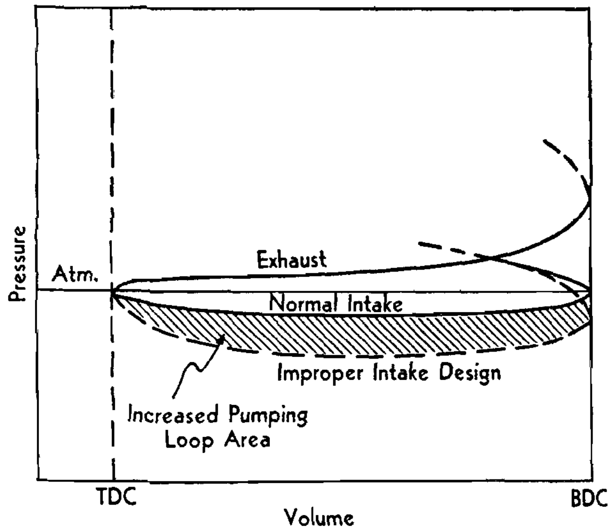


FIG. 10-10. Effect of improper intake system design on indicator diagram (not to scale).

the pumping loop area, Fig. 10-10, and an increase in fhp, with consequent reduction in overall engine output.

10-6. Air Consumption of a SI Engine. The output of a SI engine depends largely on the quantity of energy liberated during the combustion of the fuel and the air. The volume of air utilized by the engine is many times the volume of the fuel used. Since the useful air-fuel mixture range is restricted, the output of the engine is pretty much limited by the amount of air which can be inducted. Thus, the SI engine may be considered to be essentially an "air pump." One of the most important factors concerning the output of a SI engine, therefore, is the induction of the greatest possible amount of air. More air inducted permits the useful addition of more fuel, thus increasing the energy available to produce work. Consequently, designers are continuously confronted with the problem of obtaining the most efficient induction of air in a SI engine.

10-7. Volumetric Efficiency. Volumetric efficiency is a measure of the success with which the air supply, and thus the charge, is inducted into the engine. *It is defined as the ratio of the actual weight of air drawn into the engine during a given period of time to the theoretical weight which should have been drawn in during that same period of time, based upon the total piston displacement of the engine, and the temperature and pressure of the surrounding atmosphere.*

$$\eta_v = \frac{w_{\text{act}}\left(\dfrac{\text{lb air}}{\text{unit time}}\right)}{w_{\text{theor}}\left(\dfrac{\text{lb air}}{\text{unit time}}\right)} \tag{10-1}$$

where,

$$w_{\text{theor}} = \frac{N\left(\dfrac{\text{rev}}{\text{min}}\right)\text{ Total Piston Displacement}\left(\dfrac{\text{in}^3}{\text{intake stroke}}\right)}{n\left(\dfrac{\text{rev}}{\text{intake stroke}}\right)1728\left(\dfrac{\text{in}^3}{\text{ft}^3}\right)v\left(\dfrac{\text{ft}^3}{\text{lb air}}\right)} = \frac{\text{lb air}}{\text{min}}$$

The actual weight is a measured quantity. The theoretical weight is computed from the geometry of the cylinder, the number of cylinders, and the speed of the engine, in conjunction with the density of the surrounding atmosphere.

The above definition was given for the unsupercharged engine. In the case of the supercharged engine, however, the theoretical weight of air should be calculated at the conditions of pressure and temperature prevailing in the intake manifold.

The volumetric efficiency is affected by many variables, some of the more important being:

(1) *The density of the fresh charge after arrival in the cylinder*—As the fresh charge arrives in the hot cylinder, heat is transferred to it from the hot chamber walls and the hot residual exhaust gases, raising its temperature. This results in a decrease in the mass of fresh charge admitted and a reduction in volumetric efficiency. The volumetric efficiency is increased by low temperatures (provided there are no heat transfer effects) and high pressures in the fresh charge, since density is thereby increased, and a greater weight of charge can be inducted into a given volume.

(2) *The pressure and temperature of the exhaust products in the clearance volume*—As the piston starts down on the intake stroke, these products tend to expand and occupy a portion of the piston displacement greater than the clearance volume, thus reducing the space available to the incoming charge. In addition, these exhaust products tend to raise the temperature of the fresh charge, thereby decreasing its density and further reducing volumetric efficiency.

(3) *The design of the intake and exhaust manifolds*—The exhaust manifold should be so designed as to enable the exhaust products to escape readily, while the intake manifold should be of such design as to bring in the maximum possible fresh charge. This entails construction such that minimum restriction is offered to the fresh charge flowing into the cylinder, as well as to the exhaust products being forced out.

(4) *The timing of the intake and exhaust valves*—Valve timing is the regulation of the points in the cycle at which the valves are set to open and close. Since the valves require a finite period of time in which to open or close without abruptness, a slight "lead" time is necessary for proper operation. The design of the valve operating cam provides for the smooth transition from one position to the other, while the cam setting determines the timing of the valve.

The effect of the *intake valve* timing on the engine air capacity is indicated by its effect on the air inducted per cylinder per cycle, i.e., the mass of air taken in to one cylinder during one suction stroke. Figure 10-11 shows representative intake valve timing for both a low speed and a high speed SI engine. In order to understand the effect of the intake valve timing on the air inducted per cylinder per cycle, it is desirable to follow through the intake process, in conjunction with the figure.

While the intake valve should open, theoretically, at TDC, most SI engines utilize an intake valve opening which occurs a few degrees prior to the arrival of the piston at TDC on the exhaust stroke. This is to insure that the valve will be fully open and the fresh charge starting to flow into the cylinder as soon as possible after TDC. In Fig. 10-11, the intake valve starts to open 10° before TDC. For the low speed engine, the intake valve closes 10° after BDC, and for the high speed engine, 60° after BDC.

As the piston descends on the intake stroke, the fresh charge is drawn in through the intake port and valve. When the piston reaches BDC and starts to ascend on the compression stroke, the inertia of the entering fresh charge tends to cause it to continue to move into the cylinder. At low engine speeds, the charge is moving into the cylinder relatively slowly, and its inertia is relatively low. If the intake valve were to remain open much beyond BDC, the up-moving piston on the compression stroke would tend to force some of the charge, already in the cylinder, back into the intake manifold, with consequent reduction in volumetric efficiency. Hence, the intake valve is closed relatively early after BDC for a slow speed engine. High speed engines, however, bring the charge in through the intake manifold at greater speeds, and the charge has greater inertia. As the piston starts up on the compression stroke, there is a "ram" effect produced by the incoming mix-

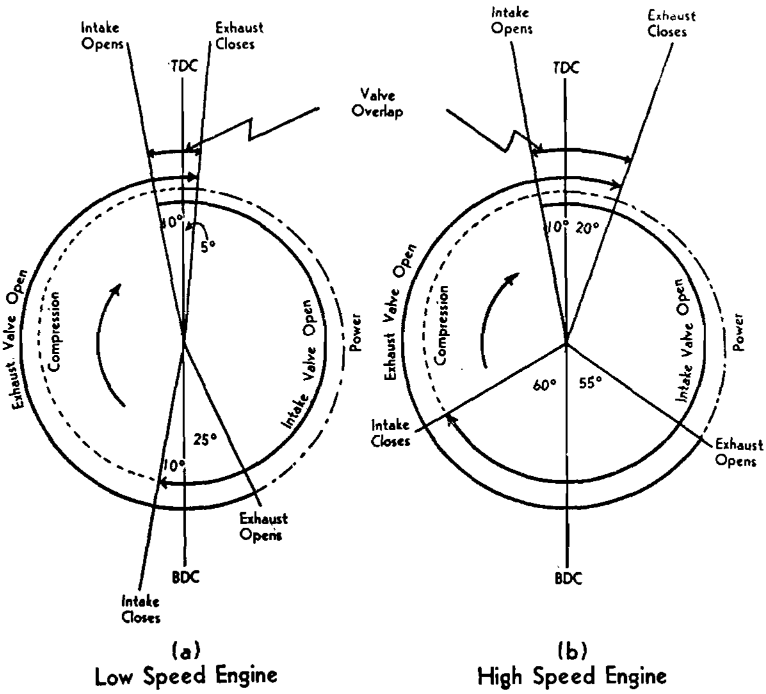


FIG. 10-11. Representative valve timing diagrams for low and high speed four-stroke cycle SI engines.

ture which tends to pack more charge into the cylinder. In the high speed engine, therefore, the intake valve closing is delayed for a greater period of time after BDC in order to take advantage of this "ram" and induct the maximum quantity of charge.

For either a low speed or a high speed engine operating in its range of speeds, there is some point at which the charge per cylinder per cycle becomes a maximum, for a particular valve setting. If the revolutions of the low speed engine are increased beyond this point, the intake valve in effect closes too soon, and the charge per cylinder per cycle is reduced. If the revolutions of the high speed engine are increased beyond this maximum, the higher speed of flow of the incoming charge through the intake manifold is accompanied by rapidly rising losses due to fluid friction. These losses can become greater than the benefit of the "ram," and the charge per cylinder per cycle falls off.

The chosen intake valve setting for an engine operating over a range of speeds must necessarily be a compromise between the best setting for the low speed end of the range and the best setting for the high speed end.

The timing of the *exhaust valve* also affects the volumetric efficiency. As explained in Article 10-4, the exhaust valve usually opens prior to the time when the piston reaches BDC on the power stroke. This reduces the work done by the expanding gases during the power stroke, but decreases the work necessary to expel the burned products during the exhaust stroke, and results in an over-all gain in output.

During the exhaust stroke, the piston forces the burned gases out the exhaust at high velocity. If the closing of the exhaust valve is delayed beyond TDC, the inertia of the exhaust gases tends to scavenge the cylinder by carrying out a greater mass of the gas left in the clearance volume, and results in increased volumetric efficiency. Consequently, the exhaust valve is often set to close a few degrees after TDC on the exhaust stroke, as indicated in Fig. 10-11.

It should be noted that it is quite possible for both the intake and exhaust valves to be open, or partially open, at the same time. This is termed *valve overlap.* This overlap, of course, must not be excessive enough to allow the burned gases to be sucked into the intake manifold, or the fresh charge to escape through the exhaust valve.

The necessity for valve overlap and the timing of valves so that they are opened and closed at positions other than TDC or BDC, as the ideal case requires, was explained above, taking into consideration only the dynamic effects of gas flow. One must realize, however, that the presence of a mechanical problem also is involved in actuating the valves, and hence, it has an influence in the timing of the valves.

The valve can not be lifted instantaneously to a desired height, but must be opened gradually due to acceleration problems involved. If the sudden change in acceleration from positive to negative values are encountered in design of a cam, the cam follower may lose the contact with the cam and then be forced back to close contact by the valve spring, resulting in a blow against the cam. This type of action must be avoided and, hence, cam contours are so designed as to produce gradual and smooth changes in directional acceleration. As a result, the opening of the valve must commence ahead of the time at which it is fully opened. The same reasoning applies for the closing time. It can be seen, therefore, that the timing of valves depends on dynamic and mechanical considerations.

Both the intake and exhaust valves are usually timed to give the most satisfactory results for the average operating conditions of the particular engine, and the settings are determined by actual engine running of the experimental model.

10-8. Torque, Indicated Horsepower, and Air Consumption. In the preceding article, the air inducted by one cylinder during one cycle was discussed. It was mentioned that there is a certain speed, within the speed range of a particular engine, at which this charge per cylinder per cycle will be greatest. At this point, the greatest mass of charge can be packed into the cylinder, and the greatest force can therefore be exerted on the piston. For all practical purposes, the torque, or engine capacity to do work, will also be greatest at this point. Thus, *there is some engine speed at which the charge per cylinder per cycle is a maximum, and at* ***approximately*** *this same speed, the torque of the engine will be a maximum.*

As the speed of the engine is increased above this speed for maximum charge per cylinder per cycle, the quantity of charge will drop off. However, since the engine speed is increased, more charges will be inducted during a given period of time. The air consumption of the engine (or the quantity of air swallowed during a given period of time) will continue to rise. In fact, air consumption will continue to increase with increased engine speed until some point is reached where the charge per cylinder per stroke is dropping off more rapidly than the number of strokes per time is increasing. The maximum air consumption point is usually not reached during the usual operating range in most engines. Air consumption, then, brings in a factor of time. Increased air consumption means that increased quantities of fuel can be added during a given period of time. This means that the power output, which also includes a time factor, can be thus increased. In fact, *the ihp produced in the cylinder is almost directly proportional to the engine air consumption.*

The relationship between air charge per cylinder per cycle and torque, as well as air consumption and ihp, is illustrated in Fig. 10-12. Note that the maximum torque occurs at a lower speed than the maximum ihp.

10-9. Engine Performance Curves. Engine performance curves are a convenient graphical presentation of an engine's performance. They are constructed from data obtained during actual test runs of the engine (Chapter IV), and are particularly useful in comparing the performance of one engine with that of another.

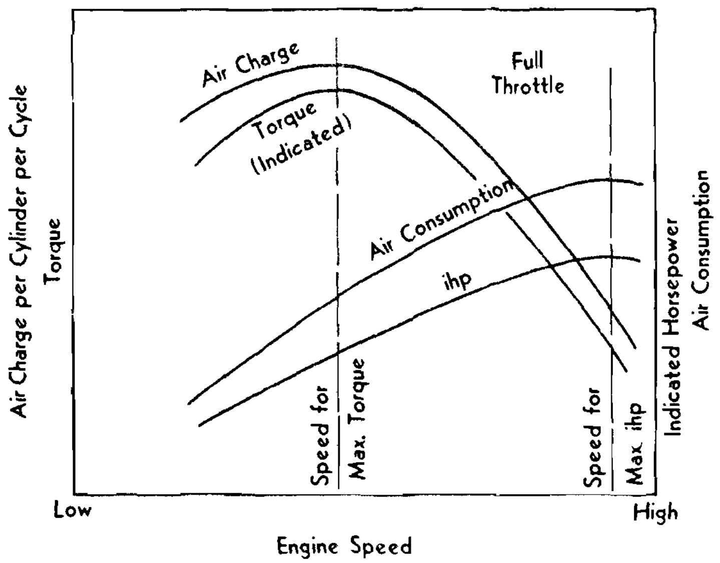


FIG. 10-12. Representative curves of torque, air charge per cylinder per cycle, air consumption, and ihp plotted against engine speed (not to scale).

Figure 10-13 shows some of the more important performance factors for a typical SI engine. In this figure, torque, ihp, bhp, and fhp are plotted against engine speed throughout the operating range of the engine, at full throttle and variable load.

The difference between the ihp produced in the cylinder, and the bhp realized at the driveshaft, is the fhp. At low engine speeds, the fhp is relatively low, and bhp is nearly as large as ihp. As engine speed increases, however, fhp increases at a continuously greater rate. At engine speeds above the usual operating range, fhp increases very rapidly. Also, at these higher speeds, ihp will reach a maximum and then fall off. At some point, ihp and fhp will be equal, and bhp will then drop to zero.

Note that the torque reaches a maximum at approximately 1800 rpm on this engine, while the ihp has not yet reached a maximum at the highest operating speed of 3600 rpm.

Figure 10-14 shows fuel consumption and bsfc plotted against engine

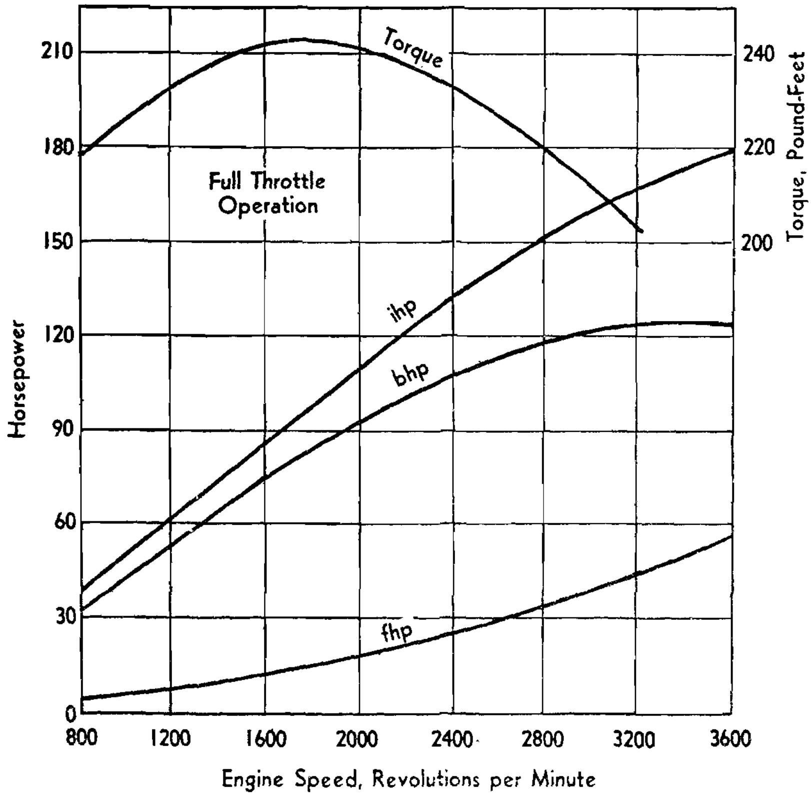


FIG. 10-13. Typical SI engine performance curves (data courtesy of Bendix Aviation Corporation, Stromberg Aircraft Carburetor Department).

speed, for the same engine operating under the same conditions. The quantity of fuel consumed increases with engine speed. The bsfc, on the other hand, drops as speed is increased in the low speed range, nearly levels off at medium speeds, and increases in the high speed range. At low speeds, the heat loss to the combustion chamber walls is proportionately greater, and combustion efficiency is poorer, resulting in higher fuel consumption for the power produced. At the high speeds, the fhp is increasing at a rapid rate, resulting in a slower increase in bhp than in fuel consumption, with a consequent increase in bsfc.

The bsfc curve of Fig. 10-14 is for full throttle, variable speed operation. At any one speed, it represents the bsfc which will result when the engine is carrying its *maximum load* at that speed. By reducing throttle

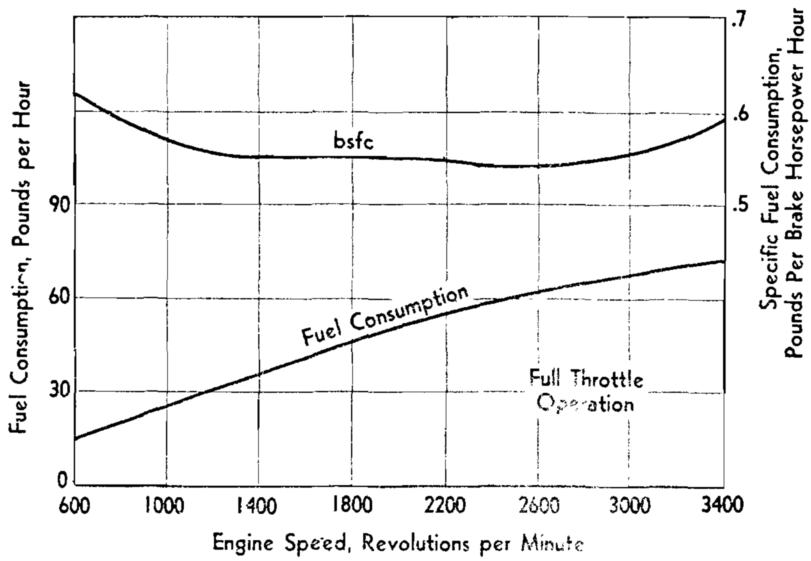


FIG. 10-14. Typical fuel consumption curves for a SI engine under full throttle, variable load conditions (data courtesy of Bendix Aviation Corporation, Stromberg Aircraft Carburetor Department).

opening and load, that same speed may be obtained, but at loads less than the maximum. A family of curves for various speeds can be obtained, each showing the effect on bsfc of varying the load at constant speed. Under these conditions of constant speed and variable load, and at a constant A/F ratio, the bsfc will rise consistently and rapidly as the load (and throttle opening) is decreased. Figure 10-15 illustrates the general shape of the curve for any given rpm. The reason for the rapid increase in bsfc with the reduction in throttle opening is that the fhp is remaining essentially constant, while the ihp is being reduced. The bhp drops more rapidly than fuel consumption, and bsfc rises.

Performance curves can be constructed for other operating factors such as imep, bmep, air consumption, etc. However, the curves presented are typical, and are among the more important. Probably the most important of these are the curves of torque, bhp, and bsfc plotted against engine speed at full throttle operation. These three curves are the ones most generally published by engine companies with the descriptive literature on their engine models. Such a plot would look similar to Fig. 10-16. If one is looking for an engine of a particular capacity, he can readily determine which engine models approximate his needs by referring to these curves as published by the various

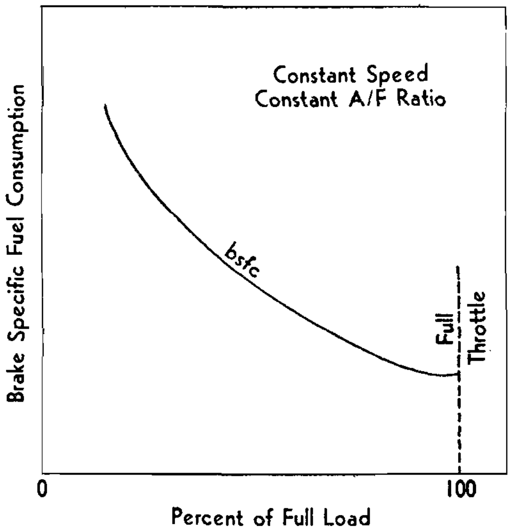


FIG. 10-15. Representative curve of brake specific fuel consumption under constant speed and variable load conditions (not to scale).

manufacturers. Once the models which suit the requirements have been located, more comprehensive performance curves may then be studied to determine which engine is the most desirable for selection.

10-10. Variables Affecting Engine Performance. In the preceding article, engine performance curves were discussed. The shape of these curves, or the engine performance, is determined by the regulation of the many design and operating variables which have been covered in previous chapters. Some of the more important of these variables will be briefly discussed and summarized at this point:

(1) *Combustion rate and spark timing*—The spark should be timed and the combustion rate controlled such that the maximum pressures occur as close as possible to the beginning of the power stroke, consistent with a smooth running engine. As a general rule, the spark

timing and combustion rate are regulated such that approximately one half the pressure rise due to combustion has occurred as the piston reaches TDC on the compression stroke.

(2) *Air-fuel ratio*—This ratio must be set to fulfill engine require-

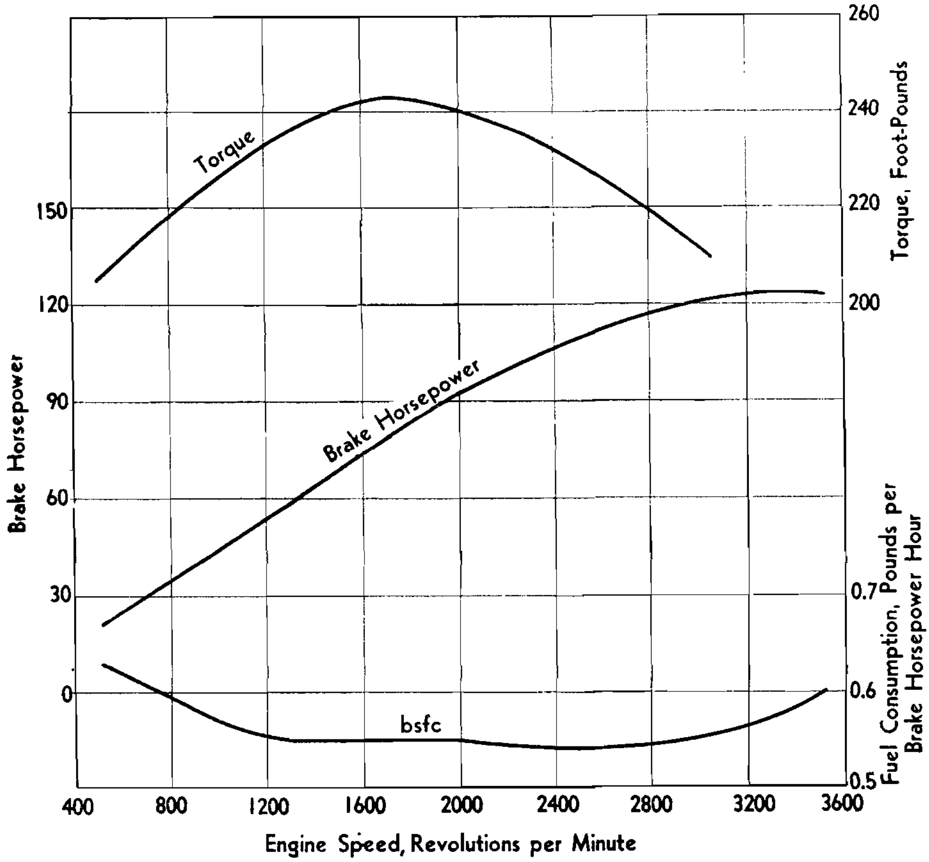


FIG. 10-16. Typical performance curves for a SI engine (data courtesy of Bendix Aviation Corporation, Stromberg Carburetor Department).

ments. Consistent with these requirements, however, it is usually set as close as possible to best economy proportions during normal cruising speeds, and as close as possible to best power proportions when maximum performance is required.

(3) *Compression ratio*—An increase in compression ratio increases the thermal efficiency, and is, therefore, generally advantageous. The compression ratio in most SI engines is limited by detonation, and the use of economically feasible anti-knock quality fuels. Increasing com-

pression ratio also increases the friction of the engine, particularly between piston rings and the cylinder walls, and there is a point at which further increase in compression ratio would be unprofitable, though this point appears to be rather high.

(4) *Engine speed*—At low speeds, a greater length of time is available for heat transfer to the cylinder walls, with consequent greater proportion of heat loss. Up to a certain point, higher speeds produce greater air consumption and therefore greater ihp. Higher speeds, however, are accompanied by rapidly increasing fhp and by greater inertia in the moving parts. Consequently, the engine speed range must be a compromise, although most present day designs appear to favor the higher speeds.

(5) *Weight of inducted charge*—The greater the mass of the charge inducted, the higher will be the power produced. For a given engine, the geometry is fixed, and it is desirable to induct a charge with the maximum possible density and at the highest possible volumetric efficiency.

(6) *Heat losses*—Figure 10-1 illustrated the large percentage of available energy which escaped in a non-usable form through heat losses. Any method which can be utilized to prevent excessive heat loss and cause this energy to leave the engine in a usable form will tend to increase engine performance. Higher coolant temperatures, for instance, provide a smaller temperature gradient around combustion chamber walls, and a reduction in heat loss, but are limited by the possibility of damage to engine parts.

10-11. Methods of Improving Engine Performance. The engineer is always interested, obviously, in methods through which engine performance may be improved. By referring to Fig. 1-8, it can be seen that there are two general areas in which methods can be utilized to improve performance, as follows:

(1) The energy put into the engine at the start may be increased, and/or
(2) The efficiency with which the fuel energy is converted to mechanical energy may be increased (Areas *A* and *B* of Fig. 1-8).

Energy supply may be increased by increasing the mass of charge entering the combustion chamber. Supercharging is one method of accomplishing this, and will be discussed in the following articles of this chapter. Larger piston displacement is another solution, but is limited by engine weight and cooling problems. Improvement in volumetric efficiency would also increase the mass of charge. Higher *engine speeds may be utilized, but these result in increased friction* losses, and above a certain point, in lowered volumetric efficiency. Improvements in fuels resulting in greater usable energy content without detonation would also help.

The use of higher compression ratios would increase the efficiency of conversion of the energy in the fuel into useful mechanical energy. This requires the development of economically feasible higher antiknock quality fuels. Even with such fuels, as pointed out in the previous article, there appears to be a limit to the advantages of compression ratio increases. Another solution would be to reduce the losses between the air cycle and the actual cycle (Article 10-3), and thereby increase the proportion of energy which can be mechanically utilized.

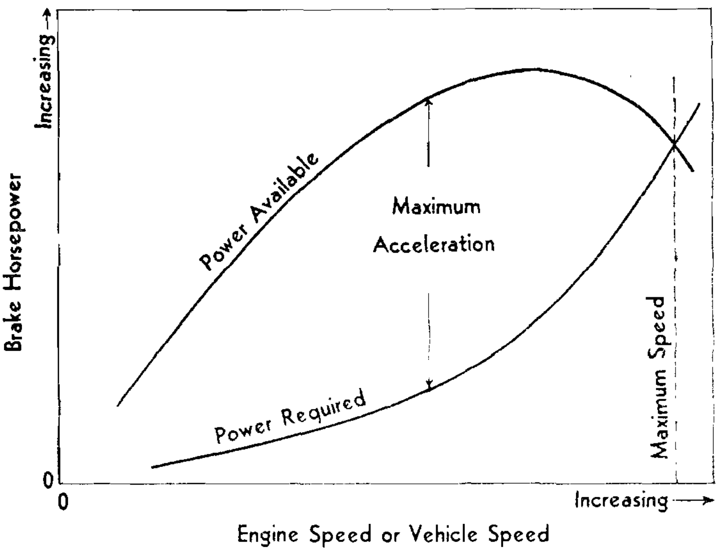


FIG. 10-17. Representative power available and power required curves (not to scale).

Also, it is possible to take advantage of the kinetic energy in the exhaust gas to increase the engine output through use of turbine or turbines. In this case, the exhaust gas from engine cylinders drives a turbine which is connected to the engine crankshaft, thus increasing engine output. Engines having this type of power booster are known as *turbo compound engines* which will be brought out in Article 10-16.

10-12. Power Required and Power Available Curves. The engine performance curves discussed in Article 10-9 represent the engine output. If the engine is utilized to power some vehicle, the bhp curve represents the power which is *available* in that engine to drive the vehicle, at various engine speeds. The vehicle will *require* a certain amount of power to drive it at various operating speeds. The power available curve, therefore, is a function of engine performance, while power required curve is a function of vehicle requirements for power to operate at various speeds. Vehicle in this case might be an automobile, speed boat, ship, aircraft, or anything that moves in the air, on the ground, on the surface, or under the surface of water. Methods used to obtain the power required curve are involved and are not only beyond the scope of this book, but also are not within the province of this text. Suffice it to say here that by plotting the *power available* curve and the *power required* curve on the same graph, against vehicle or engine speed, an estimate can be made of some of the performance characteristics of the vehicle. Figure 10-17 represents the general shape of the curves of power required and power available for a given set of operating conditions. The maximum speed will be obtained where the curves intersect. The maximum acceleration will be obtained at the speed producing the greatest difference between the power available and the power required curves, as shown by a vertical line on Fig. 10-17.

10-13. Supercharging. As pointed out in previous articles, the engine is essentially an air pump. Increasing the air consumption permits greater quantities of fuel to be added, and results in a greater potential output. The ihp produced is almost directly proportional to the engine air consumption. While bhp is not so closely related to air consumption, it is, nevertheless, dependent upon the mass of air consumed. It is desirable, then, that the engine take in the greatest possible mass of air.

Three possible methods which might be utilized to increase the air consumption of an engine are:

(1) Increasing the piston displacement, but this increases the size and weight of the engine, and introduces additional cooling problems.

(2) Running the engine at higher speeds, which results in increased fluid and mechanical friction losses, and imposes greater inertia stresses on engine parts.

(3) Increasing the density of the charge, such that a greater mass of charge is inducted into the same volume or same total piston displacement.

The last method of increasing the air capacity of an engine is widely used, and is termed *supercharging*. Of the three possible methods for increasing air capacity listed, supercharging accomplishes the purpose in the most practical manner. Also, existing designs of engines need be altered only relatively slightly to withstand the higher pressures involved.

The apparatus used to increase the air density is known as a *supercharger*. It is merely a compressor which provides a denser charge to the engine, thereby enabling the consumption of a greater mass of charge with the same total piston displacement. For ground installations, it is used to produce a *gain* in the power output of the engine. For aircraft installations, in addition to producing a gain in power output at sea level, it also enables the engine to *maintain* a higher power output as altitude is increased.

During the process of compressing the charge, the supercharger produces the following effects:

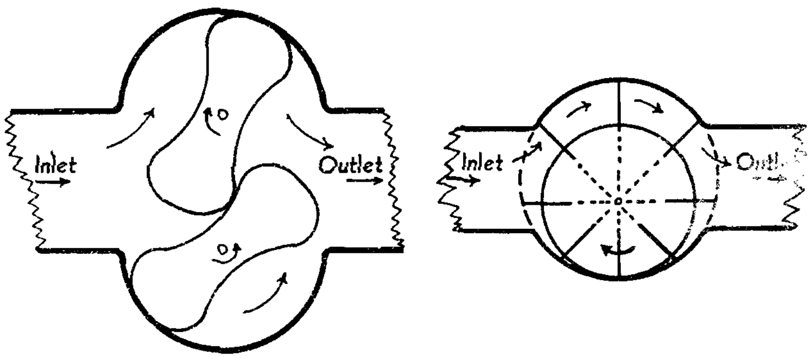
(1) Provides better mixing of the air-fuel mixture. The turbulent effect created by the supercharger assists in additional mixing of the fuel and air particles. The arrangement of certain types of superchargers, particularly the centrifugal type, also encourages more even distribution of the charge to the cylinders.

(2) The temperature of the charge is raised as it is compressed, resulting in a higher temperature within the cylinders. This is partially beneficial in that it helps to produce better vaporization of the fuel, but detrimental in that it tends to lessen the density of the charge. The increase in temperature of the charge also affects the detonation of the fuel. Supercharging tends to increase the possibility of detonation in a SI engine and lessen the possibility in a CI engine.

(3) Power is required to drive the supercharger. This is usually taken from the engine and thereby removes, from over-all engine output, some of the gain in power obtained through supercharging.

10-14. Types of Compressors Used for Supercharging. There are three basic types of compressors that may be used as superchargers, namely, the positive displacement type, the axial flow type, and the centrifugal flow type.

Positive displacement superchargers may be further divided into the *piston and cylinder*, the *rotary*, and the "*screw*" types. In the piston and cylinder arrangement, a piston compresses air in a cylinder in much the same manner as it compresses the air in a CI engine. In the rotary type, the air may be compressed by a meshing "gear" arrangement (exemplified by a Roots blower), or by a rotating vane element. These are illustrated in Fig. 10-18(a) and (b). In both of these rotary types, a volume of air is taken from the intake and discharged at the outlet end. The air is compressed as it is forced against the higher pressures at the outlet side of the compressor. The "screw" arrangement traps air between intermeshing helical shaped "gears" and forces it axially toward the outlet end. The "gears" are in some cases designed so that the volume of the pocket of entrapped air is reduced as it proceeds through the compressor axially, thus producing compression of the air. (An example of this type, the Lysholm compressor, is illustrated in Fig. 16-29.) Positive displacement superchargers are used with many reciprocating engines in stationary plants, vehicles, and marine in-



(a)
Rotary (Roots)

(b)
Vane

FIG. 10-18. Schematic diagrams of two positive displacement types of compressors (not to scale).

stallations. The piston and cylinder arrangement is generally limited to use on large, low speed CI engines.

The *axial flow compressor* consists of several stages of alternating fixed and moving blades which compress the air as it moves axially along the compressor (see Fig. 16-31). While it is seldom used to supercharge reciprocating engines, it is widely used as the compressor unit of gas turbines, and will be discussed in Chapter XVI.

The *centrifugal compressor* is widely used as the supercharger for reciprocating engines, as well as the compressor for gas turbines. It is found in both stationary plants and in the power plants for vehicles. It is almost exclusively used as the supercharger with reciprocating power plants for aircraft, because it is relatively light and compact, and produces continuous flow rather than pulsating flow as in some positive displacement types. In this type, Fig. 10-19, the mixture

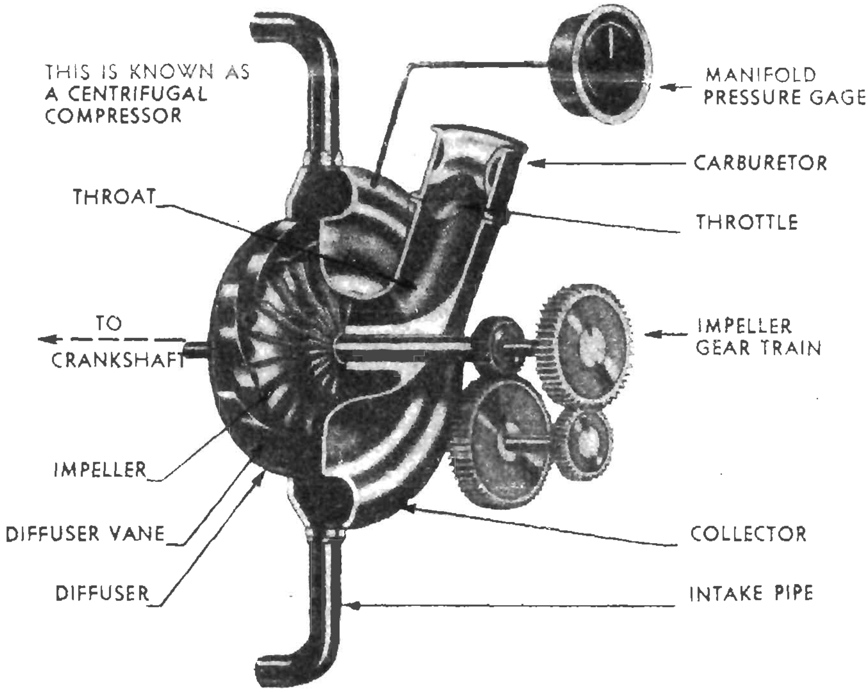


FIG. 10-19. Diagram of a centrifugal supercharger (courtesy of Pratt and Whitney Aircraft Division, United Aircraft Corporation).

enters the rotating impeller close to, and in a direction parallel to, the shaft. It leaves the impeller radially, and passes through a diffusor into a collector ring, from which it is directed through the intake pipe to the cylinders. For the purpose of this course, the *intake manifold* will be considered as consisting of the collector ring and the intake pipe. The impeller imparts a high velocity to the mixture. In passing through the diffuser, the velocity is reduced and the pressure increased, thereby accomplishing the compression and increasing the density of the charge.

In general, compressors are designed to give maximum efficiency at some chosen speed. This designed speed for maximum efficiency varies widely between the various types of compressors. However, comparing the types on a basis of their designed speed for maximum efficiency shows that the axial flow type generally reaches higher efficiencies than the centrifugal flow type. The efficiencies of the various positive displacement compressors vary widely, depending upon the type. The range of maximum efficiencies may extend from above the axial flow efficiency to below that of the centrifugal flow type, as indicated by the "band" in Fig. 10-20. As the compressor speed is varied with respect to that producing optimum performance, the efficiency of axial flow compressors drops off rapidly, whereas, that of the centrifugal and positive displacement types drops off to a lesser degree, in that order. This general trend is illustrated in Fig. 10-20, which indicates that over a range of operating speeds, the positive displacement type would tend to give the best over-all efficiency. The centrifugal type is the next most desirable for a range of operating speeds, whereas, the axial flow gives the poorest relative results.

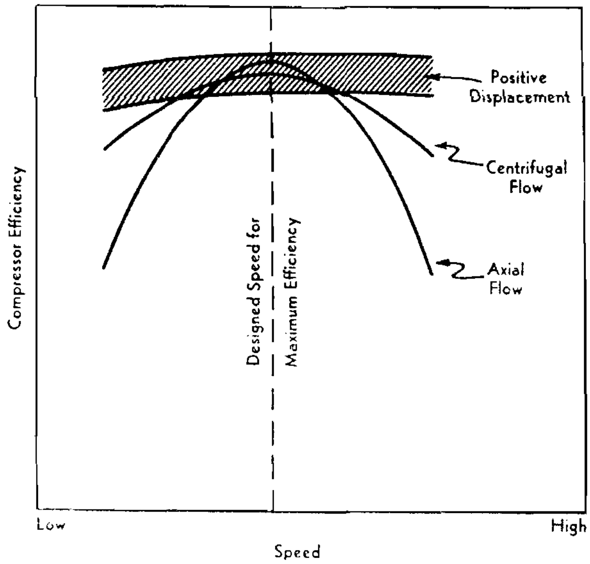


FIG. 10-20. Illustration of relative efficiencies of various compressor types (not to scale).

In addition to efficiency, another important parameter used to designate supercharger capability or performance is termed the *pressure ratio* (r_p). It is the ratio of the pressure at the outlet end of the supercharger to the pressure at the inlet end.

10-15. Aircraft Supercharger Operation. The supercharger is particularly well suited for use with aircraft power plants, because of the following reasons:

(1) Aircraft operate over a wide range of altitudes. As the altitude increases the density of the air decreases. With no supercharger, the

density of the charge, and thus the power output, will decrease as the altitude is increased. A supercharger may be used to compress the charge and maintain sea level pressures of the incoming mixture up to an altitude within the compression limit of the supercharger.

(2) The supercharger provides the desired increase in charge density with a relatively small over-all weight increase, a particularly important factor in aircraft design. By using proper grades of anti-knock quality fuels, and with relatively few changes to strengthen certain engine parts, a supercharger can be used to increase greatly the output of a production model with no radical change in its basic design, or increase in piston displacement. With a supercharger, a relatively light engine can be used for all normal operations, but when high power is needed, such as during take-off, the supercharger provides the engine with the mass of air necessary to produce it.

Superchargers are necessary for high powered aircraft operation. Since they are so readily adapted to, and widely used in, this field, the following discussion will be based on aircraft supercharger operation. Many of the principles which follow apply equally well, however, to other supercharger installations.

For a particular engine operating under given conditions, the pressure in the intake manifold is an indication of the mass of the charge flowing into the cylinders, and hence of power produced. In aircraft installations, the pressure at the supercharger collector ring of supercharged engines, or at the intake manifold of unsupercharged engines, is measured, and is termed *manifold pressure.* Variations in manifold pressure give an indication of the variations in power output.

In order to understand the effect of supercharging on aircraft engine operation, it is desirable to note first what happens to an unsupercharged engine as the altitude is increased. Then a supercharger will be added to the same hypothetical basic engine, and the effect on engine operation shown. The various configurations will be compared on the basis of the same engine speed and the same anti-knock quality fuel. *Numerical values of altitude and power are hypothetical and are used merely for comparative purposes.*

(1) *Unsupercharged engine*—Assume an unsupercharged aircraft engine whose output is 700 hp at sea level. With full throttle and constant engine speed, the manifold pressure and power output of the engine will gradually be reduced as altitude increases, since the density of the atmosphere decreases. The variation of power output with altitude of this engine, which will be designated by the letter A, is indicated in Fig. 10-21.

(2) *Supercharged engine*—Now, let the same engine be strengthened as necessary to handle the higher pressures, and equipped with a centrifugal type supercharger that rotates at a 6 to 1 impeller to engine speed ratio (Engine *B*). Assume that the engine now produces 950 hp at sea level with full throttle, operating on a fuel with high enough

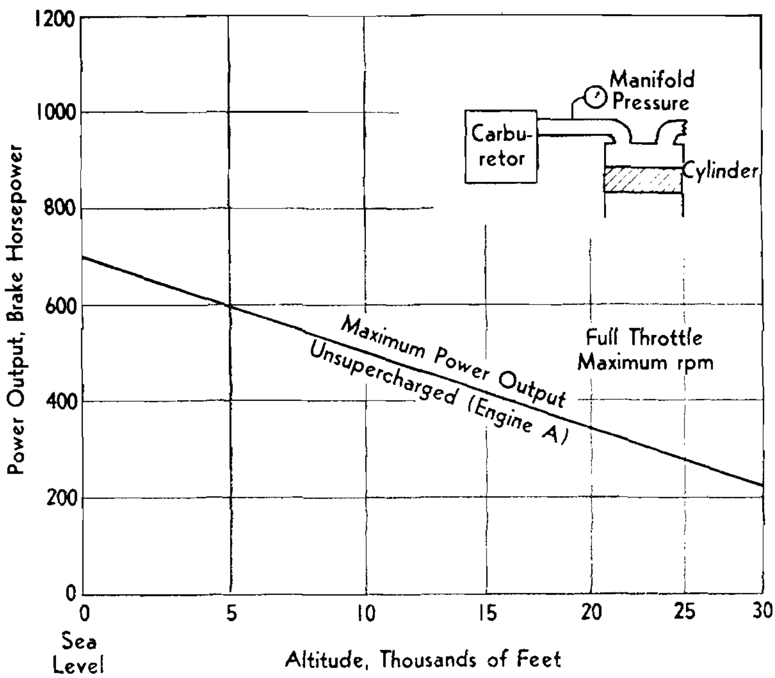


FIG. 10-21. Maximum power output versus altitude—unsupercharged engine.

anti-knock rating so that detonation does not occur. The gain due to supercharging amounts to 250 hp at sea level, but some of this, say 50 to 100 hp, will be required to drive the supercharger. Assume that this installation requires 50 hp. The net gain, therefore, is 200 hp and the power delivered by the engine at sea level is now 900 hp. This power will be reduced as altitude is increased, as indicated in Fig. 10-22. Area I represents the net increase in power output of the supercharged over the unsupercharged engine at all altitudes.

From the above, it appears that a larger or higher speed impeller may be used to give a still greater pumping capacity, and thereby further increase the power output of this engine. This is essentially true, except that the permissible power output is limited by the following factors:

(1) *The temperature rise of the charge as it passes through the supercharger.* An increase in the capacity of the supercharger results in a greater increase in temperature of the charge delivered to the cylinder. In the SI engine used in aircraft, this encourages the tendency of the fuel to detonate. The capacity of the supercharger, for a given anti-

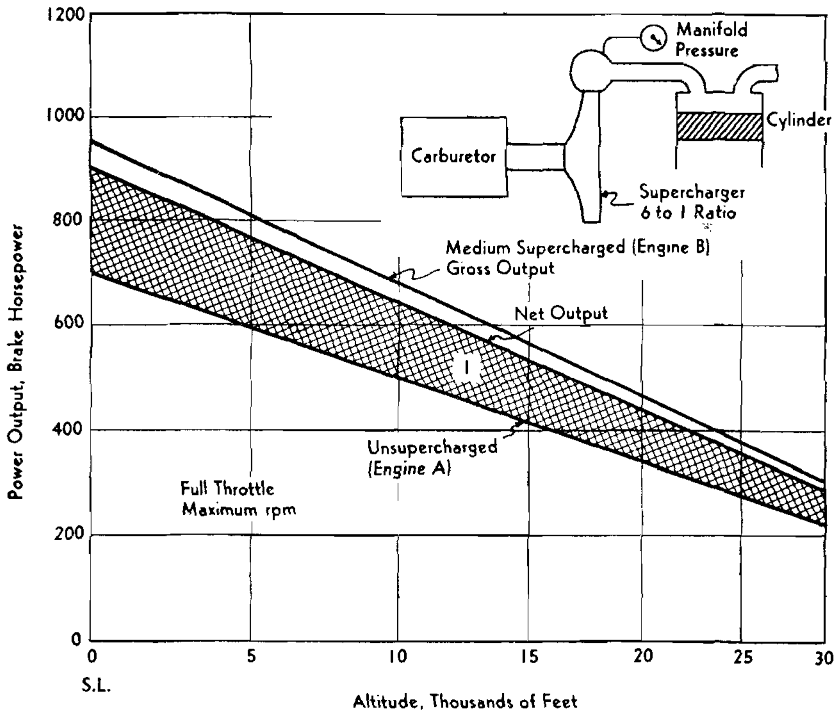


FIG. 10-22. Maximum power versus altitude—medium supercharged engine (6 to 1 impeller to engine speed ratio).

knock quality of the fuel, is limited to an extent necessary to prevent detonation. Higher anti-knock quality fuels, of course, allow the use of greater supercharger capacity.

(2) *The engine structural strength characteristics.* Even with the best possible anti-knock quality fuels, a point can be reached where it is not practically feasible to further strengthen engine parts. A supercharger capacity which results in pressures exceeding the strength limit of the engine will result in engine failure.

(3) *The power required to drive the supercharger.* The supercharger, in

aircraft, is driven by the engine, and lowers the over-all engine output accordingly. The power required to deliver a given mass of charge varies nearly as the square of the impeller tip speed. Increasing the speed of the impeller, or its size, increases the power absorbed by the supercharger. A point might be reached at which the power required to drive the supercharger would become exhorbitant in relation to engine output. Usually, however, the limiting temperature rise will be reached prior to this point.

(4) *The size and configuration of the supercharger installation.* This must not be so great as to produce excessive drag of the plane.

Now, to continue with the discussion concerned with supercharging of the basic engine, assume that a higher capacity supercharger is added to the same strengthened engine *B*. By retaining the same size impeller, and increasing the impeller to engine speed ratio to 9 to 1, a greater air consumption can be attained. With this supercharger, and disregarding the above limitations, suppose the engine will now deliver 1150 hp at sea level with wide open throttle. The larger capacity supercharger will absorb more horsepower, say 100 to 200 hp at sea level. Assuming this installation requires 100 hp, the net output of the engine at sea level would then be 1050 hp, with a gradual decrease in power output as the altitude is increased. This is indicated by the line *WYZ* in Fig. 10-23. The higher capacity of the supercharger, however, produces greater pressure and temperature rises, such that at full throttle this engine exceeds the structural or detonation limits outlined above. In order not to exceed these limitations, assume that the engine is restricted to a maximum output of 800 hp. The engine is still *potentially* able to deliver 1050 hp at sea level, but the limitations will allow utilization of not more than 800 hp. With these restrictions, this engine *C* cannot be operated at full throttle at sea level or low altitudes. This is indicated by the part throttle operating line *X Y* in Fig. 10-23, which represents the maximum power which can be safely utilized at altitudes up to about 8000 feet. The potential power, which is not available because of the limitations of fuel quality or engine structure, is indicated by the combined areas I and II. Note that this installation is restricted to power outputs below those of engine *B* from sea level to about 3500 feet altitude, as indicated by area II. Above this altitude, however, greater power output is available with engine *C* than with engine *B*, as indicated by the dotted area III.

Higher anti-knock quality of the fuel, or higher strength limitations, would raise the allowable power output at low levels, and raise line *X Y* of Fig. 10-23 accordingly. Most high powered aircraft installations tend to be limited at low altitudes and exhibit characteristics similar to engine *C*. What determines a medium (engine *B*) or a highly (engine *C*) supercharged engine is, of course, a matter of relative values. Any supercharged engine has a certain potential power output at sea level and at a given altitude. As the amount of supercharging increases,

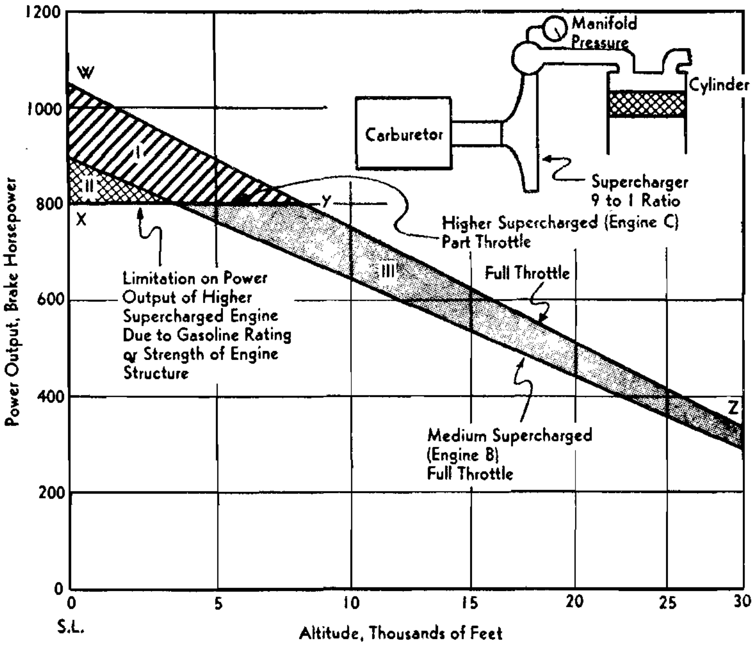


FIG. 10-23. Maximum power versus altitude—medium (6 to 1 ratio) and higher (9 to 1) supercharged engines.

however, the possibility of encountering limitations at low altitudes, because of fuel quality or engine strength, increases.

Figure 10-24 presents a comparison of the performance of the unsupercharged engine *A*, the medium supercharged engine *B*, and the limited higher supercharged engine *C*. Note that both supercharged engines may deliver greater power at all altitudes than the unsupercharged engine. As indicated by area I, the medium supercharged engine gives a greater available output than the higher supercharged engine at any specific altitude from sea level to about 3500 feet. This is due to the increased temperature rise through compression in the higher supercharged engine. The maximum usable power in each engine must be low enough so that the cylinder pressures existing, in combination with the temperature rise, do not exceed the detonation limits. Since the temperature rise with a higher degree of supercharging is greater, the maximum cylinder pressure and *corresponding power output* of the higher supercharged engine must be reduced at low altitudes. At any specific altitude above 3500 feet, however, the limited supercharged engine has a greater amount of available power, as shown by area II.

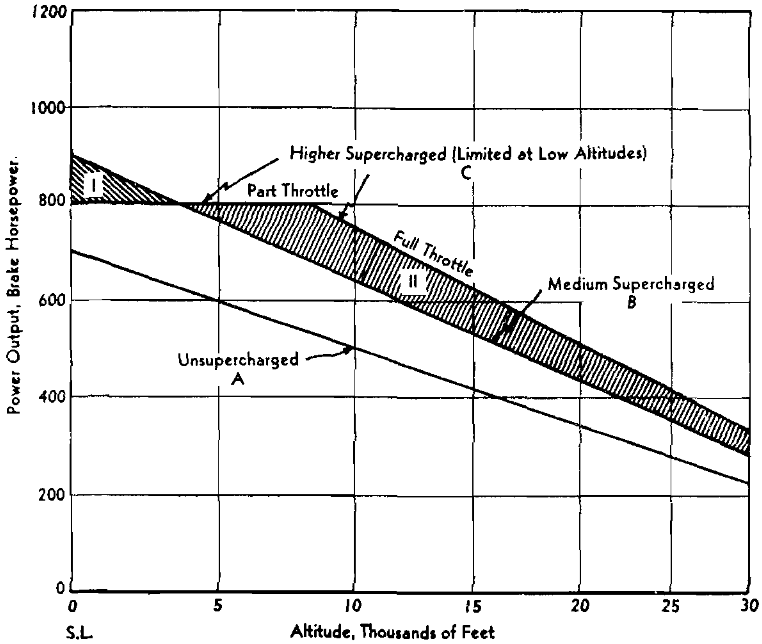


FIG. 10-24. Maximum power versus altitude—unsupercharged, medium supercharged, and higher supercharged engines.

The choice of whether or not to supercharge, or how much to supercharge, depends upon the requirements of the aircraft in which the power plant is to be installed. The unsupercharged engine might well be satisfactory in a training plane. The power plant would be lighter and occupy a smaller space. For a plane in which the greatest power output is desired from sea level to 3500 feet (Fig. 10-24), medium supercharging would appear to be the best solution of the examples given. If the plane is to operate primarily above 3500 feet, maximum performance would be obtained by using the limited higher supercharged engine, and sacrificing some power below 3500 feet.

The above discussion has been concerned with a given size impeller operating at one impeller to engine speed ratio. Such an installation is called a *single stage, single speed supercharger*. By using a higher impeller to engine gear ratio than 9 to 1, the engine could maintain its

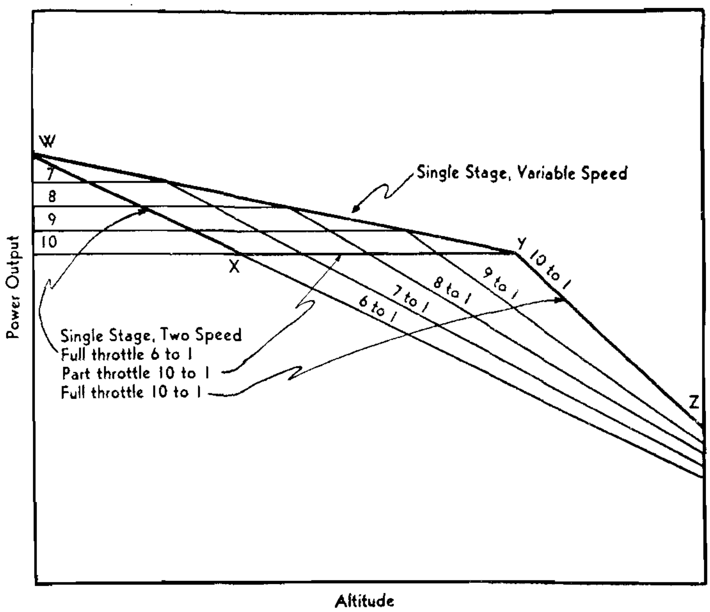


FIG. 10-25. Maximum power versus altitude—single stage, variable speed supercharger and single stage, two speed supercharger (not to scale).

limited power output to higher altitudes, but the utilizable output at sea level would be further restricted. Figure 10-25 illustrates the effect of increasing the impeller to engine speed ratio. Note that with higher ratios, the power output is greater at higher altitudes, but the power output which can be utilized at sea level and the lower altitudes is reduced. With a single stage and one fixed speed ratio, a sacrifice must be made to either altitude performance or sea level operation. To partly offset this deficiency, the single stage, two speed and single stage, variable speed superchargers were developed.

In the *single stage, two speed supercharger*, the same impeller can be driven at either of two different impeller to engine speed ratios. Assume that both 6 to 1 and 10 to 1 ratios are available. From sea level to 4000 feet, the impeller would be placed in the 6 to 1 ratio, while above 4000 feet, it would be shifted to the 10 to 1 ratio. The performance curve for such an installation is indicated by the path *WXYZ* of Fig. 10-25.

A more refined version is the *single stage, variable speed supercharger.* Such an installation, in essence, produces the optimum performance corresponding to an infinite number of impeller to engine speed ratios. The most desirable ratio for a given altitude can be utilized such that the output varies with altitude as indicated by the heavy line *WYZ* of Fig. 10-25.

A single stage supercharger becomes prohibitive in size and weight for high altitude planes. *Two stage superchargers* are, therefore, used for high altitude aircraft. Two superchargers are used in tandem, and the charge compressed in two stages. Such an arrangement produces

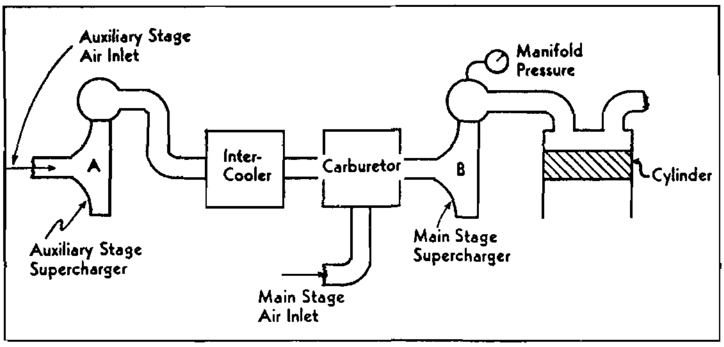


FIG. 10-26. Schematic diagram of one typical two-stage supercharger arrangement.

the necessary compression without the excessive size or speed of the impeller which would be required for a single stage supercharger of the same capacity. It also provides a convenient arrangement for the use of an intercooler between stages to assist in keeping the temperature of the charge from exceeding the detonation limits due to compression. One typical two stage arrangement is shown in Fig. 10-26. At low altitudes, only the main stage (*B*) is utilized, and the air enters through the main stage air inlet. At some altitude where the main stage no longer has sufficient capacity to provide the mass of air required, the auxiliary stage is cut in, the main stage air inlet closed, and the air is inducted through the auxiliary air inlet. The auxiliary supercharger then compresses the air, which passes through the intercooler where its temperature is reduced, and then flows into the main stage compressor where it is compressed further. The general performance curve for such an installation is shown in Fig. 10-27 by the solid line *VWXYZ*.

The auxiliary stage may be two-speed, and the installation known as a *two stage, two speed supercharger.* The performance curve for this arrangement is shown by the dashed line in Fig. 10-27. Or, both stages may utilize a variable speed supercharger and the installation termed a *two stage, variable speed supercharger,* producing results as indicated by the line *STU* of Fig. 10-27.

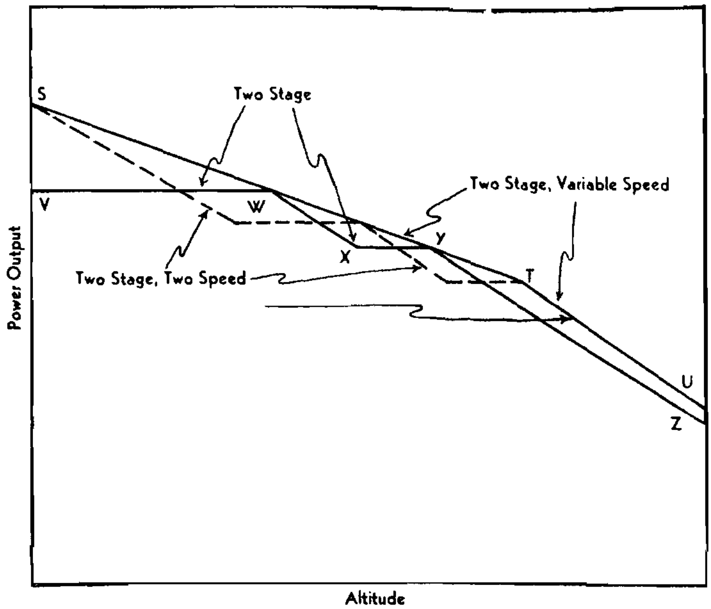


FIG. 10-27. Relative performance of two stage, two stage two speed, and two stage variable speed superchargers (not to scale).

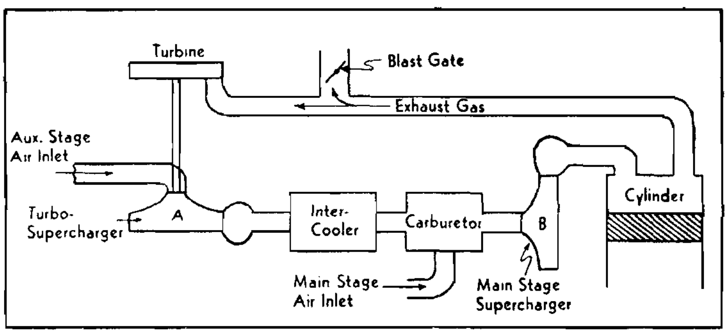


FIG. 10-28. Schematic diagram of a turbosupercharger arrangement.

A *turbosupercharger* installation is often used for high altitude aircraft. Figure 10-28 represents a two stage system in which the auxiliary stage is driven by energy remaining in the exhaust gas. At low altitudes, the auxiliary stage is not utilized and the exhaust gases are passed to the atmosphere through an open "blast gate." When it becomes necessary to utilize the auxiliary stage at altitude, the blast gate is gradually

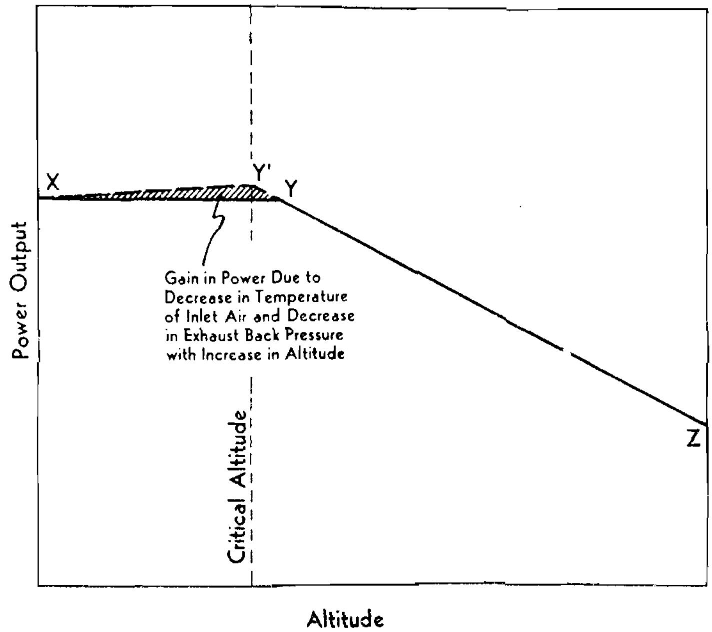


FIG. 10-29. Effect of increased altitude on part throttle power output (not to scale).

closed, forcing some of the exhaust gases through a turbine wheel which drives the auxiliary stage. This stage is thus a variable speed supercharger, whose capacity is increased by increasing the flow of exhaust gases through the turbine by reducing the blast gate opening. When the blast gate is fully closed, the maximum capacity of the supercharger has been reached.

A turbosupercharger produces a slight adverse effect by increasing the exhaust back pressure in the engine cylinders. This effect is partially overcome at altitude, however, since the pressure differential across the turbine blading increases, thus increasing the capacity of the auxiliary stage.

The horizontal part throttle operating line in the above performances of the supercharger was based on the assumption of a single limiting temperature at all altitudes. Actually, the temperature at the atmosphere changes from altitude to altitude, and normally the temperature decreases as altitude increases (up to altitude covered in this discussion). Due to the decreasing temperature, the charge entering the cylinders tends to have a slightly greater density, as altitude increases, than that upon which the above performances were based. In other words, although the *over-all* density of the atmosphere decreases with altitude, the lowered temperatures cause this decrease to be of a lower magnitude than that upon which the above curves were computed. Also, the *over-all* decrease in atmospheric density produces a reduction in exhaust back pressure with altitude. These two factors cause a gradual increase in bhp, with altitude, above that presumed in the above examples, as illustrated in Fig. 10-29. The solid line XYZ represents the maximum power output for the single stage supercharger (engine C of Fig. 10-23) disregarding the decrease in temperature and back pressure with altitude. The actual power output of the engine, considering these effects, will follow the curve $X\ Y'\ YZ$. The altitude (Y') at which the maximum usable power output is reached at full throttle is called the *critical altitude* for that particular installation and engine speed. An installation with more than one stage or one speed ratio will have more than one critical altitude.

10-16. Turbo Compound Engines. A turbo compound engine is a reciprocating type of engine whose power is increased through employment of gas turbines that are driven by the exhaust gases emanating from the engine cylinders and that are connected through suitable means to the engine crankshaft, Fig. 10-30.

In the normal internal combustion engine, it is desirable to obtain the greatest expansion of gas charge in the cylinder per power stroke in order to obtain the maximum work per unit charge. However, this expansion is limited by practical considerations such as permissible piston speed, piston stroke, pumping losses, heat losses, etc., and the requirement of scavanging the cylinder to permit entrance of the following fresh charge. As a result, the exhaust gases of the normal engine contain a considerable amount of useful residual energy which is not used.

The Wright-Aeronautical Division of Curtiss-Wright Corporation determined that a desirable way to recover some of this unused energy in the exhaust gases is to employ blowdown type of gas turbine. This type of turbine was selected in preference to the pressure type of turbine on the basis of the magnitude of engine back pressure that is caused by the use of these turbines, for the range of operating altitude obtained by the turbo compound engine. The blowdown type of turbine produced less back pressure than the pressure type.

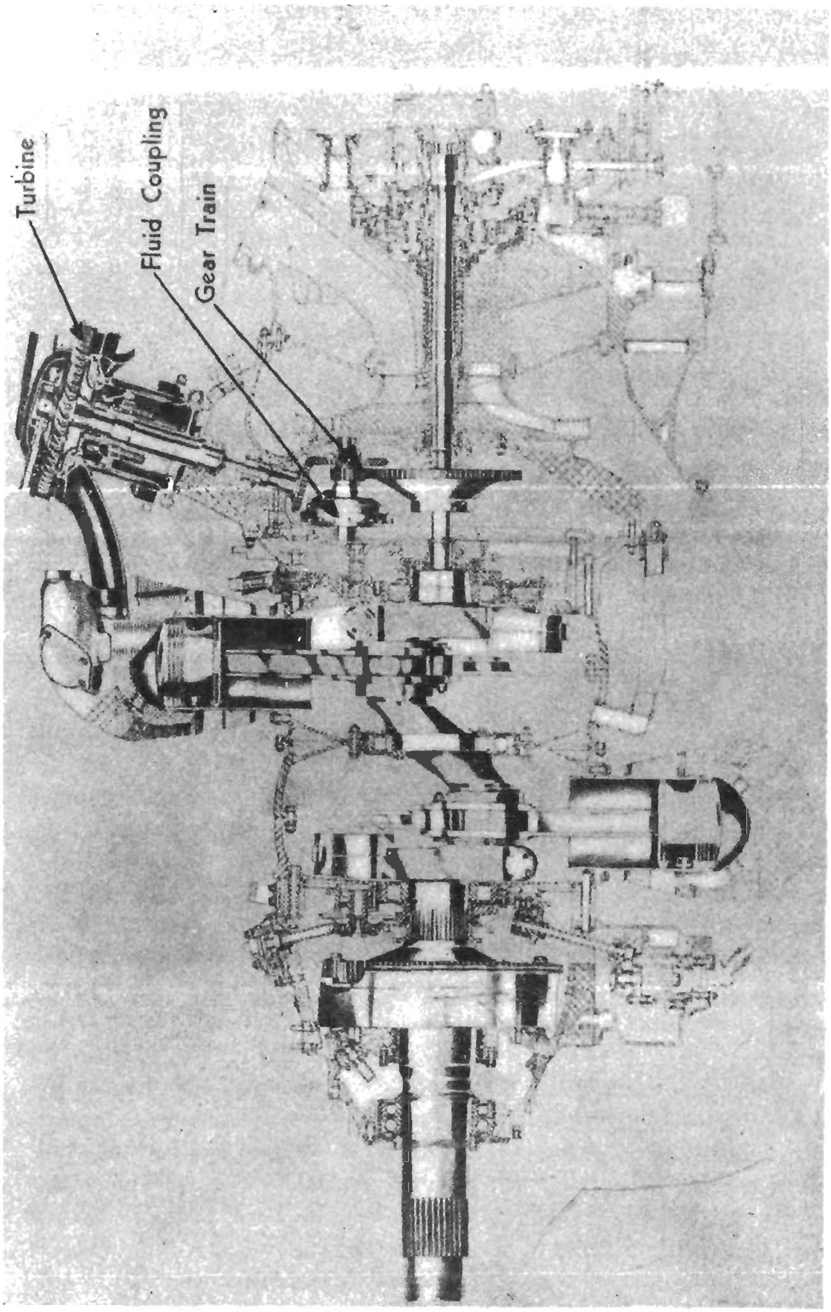


FIG. 10-30. Cross section of turbo compound engine (courtesy of Wright Aeronautical Division Curtiss-Wright Corporation)

It is not intended, in this chapter, to explain the operating principles of the gas turbine. This will be accomplished in Chapter XVI. It is in order, however, to explain the principal difference between a blowdown and the pressure turbines.

The *blowdown* or *impulse* type gas turbine extracts some of the kinetic energy contained in the exhaust gas as it is deflected through blading of the turbine wheel, while *pressure* type gas turbine gains the energy from the exhaust gas through expansion of this gas in the nozzle and the turbine blades.

The Wright turbo compound air-cooled radial engine has 18 cylinders arranged in a double row with 3350 cu in. displacement. The exhaust gases from the cylinders drive three impulse turbines which are located in the rear of the engine, and the power from the turbines is transferred to the crankshaft through a gear train and the three fluid couplings. In addition to the three turbines, this engine also has a two-speed gear driven supercharger that is used for altitude operation.

Due to the employment of the power recovery turbines, the power of this engine was increased by 18 per cent at take-off, and by 10 per cent at cruise power over the basic engine brake horsepower when calculations were made at sea level conditions. The increase in power percentage was even greater when calculations were based at critical altitude conditions. These were 26 per cent for take-off and 19 per cent increase at cruise power over the basic engine horsepower.

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

10-1. Arthur P. Fraas, "*Combustion Engines,*" McGraw-Hill Book Company, Inc.

10-2. Edward F. Obert, "*Internal Combustion Engines Analysis and Practice,*" International Textbook Company.

10-3. Lester C. Lichty, "*Internal Combustion Engines,*" McGraw-Hill Book Company, Inc.

10-4. "*Aviation Fuels and Their Effects on Engine Performance,*" Ethyl Corporation.

EXERCISES

10-1. What is meant by a "heat balance," and what does it show?

10-2. What is meant by engine performance?

10-3. How does the actual cycle differ, in general, from the air cycle?

10-4. What is the pumping loop and what does it represent?

10-5. What are the uses of the p-V diagram? The p-t diagram? Make a sketch of each and indicate point of ignition and points of valve opening and closing.

10-6. What is the effect of part throttle operation on the pumping loop, and why?

10-7. Why is air consumption so important in a SI engine?

10-8. What is volumetric efficiency? Name some important variables which affect this efficiency.

10-9. Why does the intake valve usually close later in the cycle in a high speed engine than in a low speed engine?

10-10. What is meant by valve overlap, and in what part of the cycle does it occur?

10-11. How are maximum torque and the charge per cylinder per cycle related?

10-12. How are ihp and air consumption related?

10-13. Why are maximum torque and maximum ihp reached at different engine speeds?

10-14. How does fhp vary with engine speed?

10-15. Draw typical curves for torque, bhp, and bsfc, plotted against engine speed. Explain why each of these curves attains this general configuration.

10-16. How would a bmep curve compare with the torque curve when plotted against engine speed?

10-17. How does the bsfc curve vary with load at any given constant engine speed?

10-18. Name and discuss six important operating variables which affect engine performance.

10-19. What two *general* methods may be used to improve engine performance?

10-20. What are power available and power required curves?

10-21. What is mean tby supercharging?

10-22. Why is supercharging used with ground installation? With aircraft?

10-23. Name some additional effects of supercharging.

10-24. What general types of compressors exist, and how do they compare as to maximum efficiency and stability range?

10-25. What is meant by the pressure ratio (r_p) of a compressor?

10-26. How does the power output of an unsupercharged aircraft engine vary with altitude? Why?

10-27. What is meant by manifold pressure, and at what point is it measured?

10-28. What limitations are placed on the amount of supercharging which can be used?

10-29. Why is a medium supercharged engine often able to give a greater available power output at low altitudes than a higher supercharged engine?

10-30. Does an unsupercharged, a medium supercharged, or a higher supercharged engine give a greater power output at a given altitude? Why?

10-31. Explain the difference between the following supercharger installations:

(a) single stage, single speed
(b) single stage, two speed
(c) single stage, variable speed
(d) two stage
(e) two stage, two speed
(f) turbosupercharger

10-32. A four-stroke cycle automobile engine is tested while running at 3600 rpm. The inlet air temperature is 60° F and the pressure is 14.7 psia.

The engine has eight in-line cylinders with a total piston displacement of 248.1 in^3. The A/F ratio is 14:1. The bsfc is 0.62 lb/bhp-hr. Dynamometer readings show a power output of 115 bhp. Find the volumetric efficiency.

Ans. 84.5%

10-33. An aircraft with a single stage supercharged engine is flying at 23,000 feet. At this altitude, the manifold pressure is 27 inches of mercury absolute, the absolute air pressure is 12 inches Hg at 20° F, and the carburetor discharge pressure is 10 inches Hg at 10° F. Make a schematic sketch of this installation. Assuming ideal air ($k = 1.4$) with no friction or heat transfer in the supercharger, compute the following:

(a) pressure ratio (r_p) of the supercharger
(b) temperature of the working fluid in the intake manifold.

Ans. (a) 2.7 (b) 164° F.

CHAPTER XI

THE COMPRESSION IGNITION ENGINE AND FUEL INJECTION

Many investigators have contributed to the development of the CI engine. The work of Dr. Rudoph Diesel in the field, however, was instrumental in producing a workable engine. He crystallized the various ideas of a number of his predecessors into a plan that resulted in a patent issued to him in Germany during 1892. Due to the efforts of Dr. Diesel in this field, CI engines are commonly referred to as diesel engines. It is of interest to note that the first CI engine built to successfully operate commercially was installed in a St. Louis brewery in 1898, and that Dr. Diesel delivered a lecture at the U. S. Naval Academy in 1912.

The following three chapters will be concerned solely with the CI engine. This chapter will present a review of the operating principles, an analysis of the outstanding features, and a study of the means used to deliver the fuel to the combustion chamber. In Chapter XII, a study will be made of the CI engine combustion process, the variables that affect the conversion of energy in the fuel into heat, and the means used to secure efficient utilization of the energy in the fuel. Chapter XIII will present a study of CI engine performance.

No attempt will be made to include all of the many designs of CI engines. Fundamental theory, however, applies to all designs, and sufficient theory will be included to give an understanding of the operation of this type of engine.

11-1. General Information Pertaining to the CI Engine. At the present time, CI engine fuels are less expensive than those used in SI engines. Furthermore, since CI engine fuels have a higher specific gravity than gasoline, and since fuel is usually sold on the volume basis, more pounds of fuel per gallon are obtained in purchasing CI engine fuel. Although CI engine fuels contain fewer Btu's per pound than SI engine fuels, the cost per Btu is usually less, due to the greater number of pounds per gallon. Consequently, this fact, plus the inherently higher thermal efficiencies obtainable in CI engines, make an attractive case for this type of power plant for many uses.

Compression ignition engines are widely used in a variety of applications. In the automotive field, they are used mostly on trucks and busses. Many small boats, both pleasure and commercial, employ this type of engine. Some large tankers and ocean liners also use the CI engine for propulsion. The railroads use such engines for switchers, as well as freight and passenger train locomotives. Construction machinery and earth-moving equipment largely make use of CI engines. Even aircraft have been powered with CI engines (Junkers Jumo 205), although such an application is not generally used at this time.

Compression ignition engines are built in various sizes, weights, operating speeds and power output. The range of speeds of American made engines varies from about 100 to about 3000 rpm, and the range of power is between about 3 and 10,000 bhp. The largest CI engine built to date develops 22,500 bhp. This engine is an 8 cylinder, 2-stroke cycle, double-acting type of engine built by Burmeister and Wain of Copenhagen, Denmark.

11-2. Characteristics of the CI Engine. Since the following three chapters will be dealing exclusively with CI engines, it would be well to review the basic operating principle of this type of engine.

The air necessary for combustion in the CI engine enters the engine generally through an air cleaner and is distributed to various cylinders, either by an intake manifold in a four-stroke cycle engine, or by air passages built in the engine block, in a two-stroke cycle engine. Air only is compressed in the cylinder. Before the piston reaches TDC on the compression stroke, however, liquid fuel is introduced into the combustion chamber and is ignited due to the high temperature of the compressed air. It can be seen, therefore, that the CI engine operates in a different manner from the SI engine. As a result, certain major differences exist between these two types of engines. These were stated briefly in Article 1-7 and are as follows:

(1) Basic operating cycle
(2) Method of introduction of fuel
(3) Method of ignition
(4) Compression ratio and thermal efficiency
(5) Engine weight

The differences in the basic operating cycles were discussed in Chapter III, and will not be discussed further.

Fuel is placed in the cylinder of the CI engine at the appropriate time by a rather complex fuel injection system. This system meters, injects, and distributes liquid fuel at a pressure considerably higher than that existing in the cylinder. High injection pressures are necessary not only to overcome the compression pressure, but also to atomize the fuel and distribute it in a desired manner.

Compression ignition engines operate at compression ratios which are considerably higher than those used by SI engines. These ratios vary from about 12 to about 20 to 1 as compared to the range of ratios from about 5 to 10.5 to 1 for the SI engine. High compression ratios of CI engines are necessary in order to compress the air to the point producing a temperature necessary to promote satisfactory combustion. This air temperature, of course, must be higher than the ignition temperature of the fuel used. As a result, shortly after the injection of fuel takes place, it commences to burn.

As stated in Chapter III, an increase in compression ratio increases the thermal efficiency of the engine, and for *given* compression ratio the CI engine cycle is less efficient than the SI engine cycle. It should be noted, however, that actual CI engines operate at considerably higher compression ratios than SI engines and, consequently, have higher thermal efficiencies.

High compression ratios result in greater forces acting on the piston, wrist pin, connecting rod, bearings, crankshaft, etc. In order to counteract these enlarged forces, it is necessary to strengthen engine component parts and, hence, a heavier engine results.

In some CI engines, however, utilizing special combustion chambers, such as the Energy Cell type (see Fig. 12-8), the peak pressure rise is absorbed by the cell and, hence, the main combustion chamber pressures are lower than those encountered in the usual CI engines. This fact enables the use of the same crankshafts and connecting rods in these CI engines as those used on similar displacement SI engines.

In the CI engine, for a given speed, and irrespective of load, an approximately constant supply of air enters the cylinder. The power produced is changed by adjusting the amount of fuel injected into the cylinder. The CI engine, therefore, is often termed a constant air supply engine.

11-3. Types of CI Engines. There are two basic types of CI engines, namely, those operating on the four-stroke cycle and those operating on the two-stroke cycle principle. As stated previously, in a four-stroke cycle engine, the cycle of events is accomplished in four strokes of piston travel or in two engine revolutions. On the other hand, in a two-stroke cycle engine, the cycle of events takes place in two piston strokes or in one engine revolution.

The four-stroke cycle CI engine is similar in construction to the SI engine as far as piston, cylinder, and valve arrangement are concerned. This type of engine may have one or two intake valves and one or two exhaust valves. In most of the CI engines, the piston, connecting rod, and crankshaft are of heavier construction than the corresponding parts in the SI engine.

A typical p-V diagram of a four-stroke cycle unsupercharged engine is shown in Fig. 11-1.

On the p-V diagram, note the timing of valve events and also the time with respect to the compression stroke when injection of the fuel begins.

The two-stroke cycle CI engine differs from its counterpart in the SI field not only in the construction of component parts, but also in the mode of supplying the air for combustion. This type of engine is extensively used in this country as well as abroad, because it is possible to obtain greater power output per unit engine weight at a given speed, than from a comparable four-stroke cycle engine. The two-stroke cycle

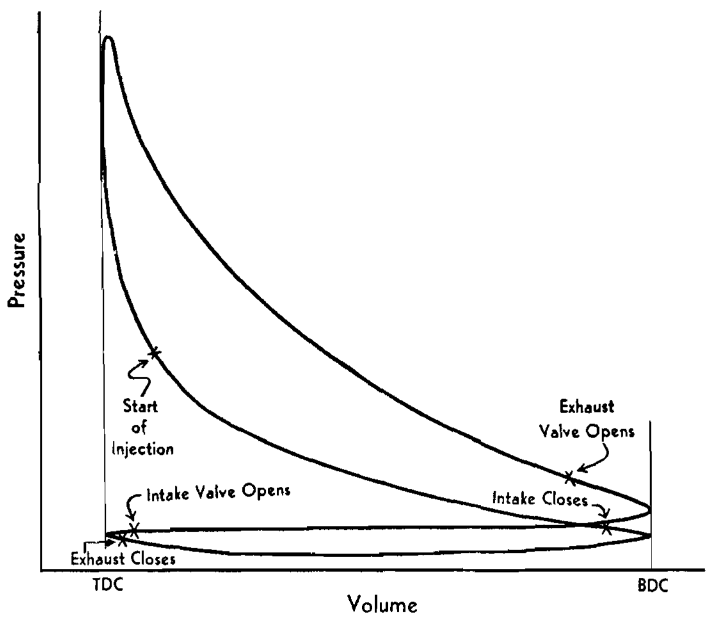


FIG. 11-1. Typical p-V diagram for a four-stroke cycle unsupercharged CI engine (not to scale).

engine lends itself to a number of piston, valve, and port arrangements. The most popular of these are shown in Fig. 11-3(b), (d), and (e). A typical two-stroke cycle unsupercharged engine p-V diagram is shown in Fig. 11-2 and designated by letters *bcdb*.

Note on this p-V diagram the position of port opening and closing in this cycle as compared to the valve events in the four-stroke cycle as shown in Fig. 11-1.

Since one cycle in the two-stroke engine is completed in only one revolution, the allowable time for the expulsion of the burned gas and induction of the air is considerably shortened, and, hence, the natural scavenging of this type of engine is poor. In order to overcome this deficiency, scavenging blowers are provided which supply the engines with air at pressures from about 2 to 5 pounds per square inch above atmospheric pressure. This compressed air hastens the removal of the burned gases and also provides the fresh air charge for the following cycle. The scavenging blower should not be confused with the supercharger, which provides air at considerably higher pressures for the primary purpose of increasing power output, although some of this air is used for scavenging purposes.

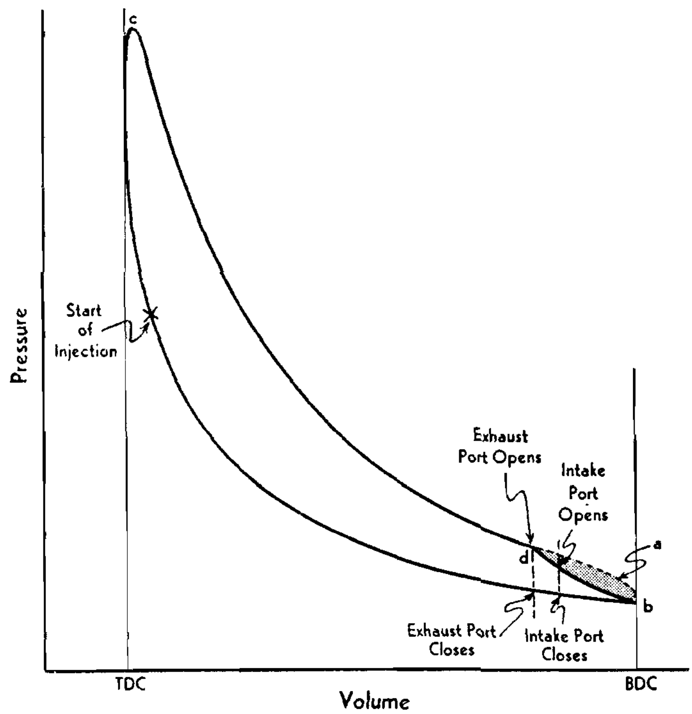


FIG. 11-2. Typical p-V diagram for a two-stroke cycle unsupercharged CI engine (not to scale).

Both the two-stroke and the four-stroke CI engine may be supercharged in order to produce more power for a given engine weight. Supercharging may result in a power increase up to about 50 percent above that produced by the unsupercharged engine. The principles of supercharging were presented in Chapter X for the SI engine, and apply equally well to the CI engine.

Compression ignition engines are built in nearly all of the cylinder arrangements described in Article 1-3. The most common arrangements, however, are vertical "in-line" and upright "V."

These engines may be further grouped by the action of the piston or pistons within the cylinders. The three major types, shown in Fig. 11-3, are as follows:

(1) Single-acting piston
(2) Double-acting piston
(3) Opposed piston

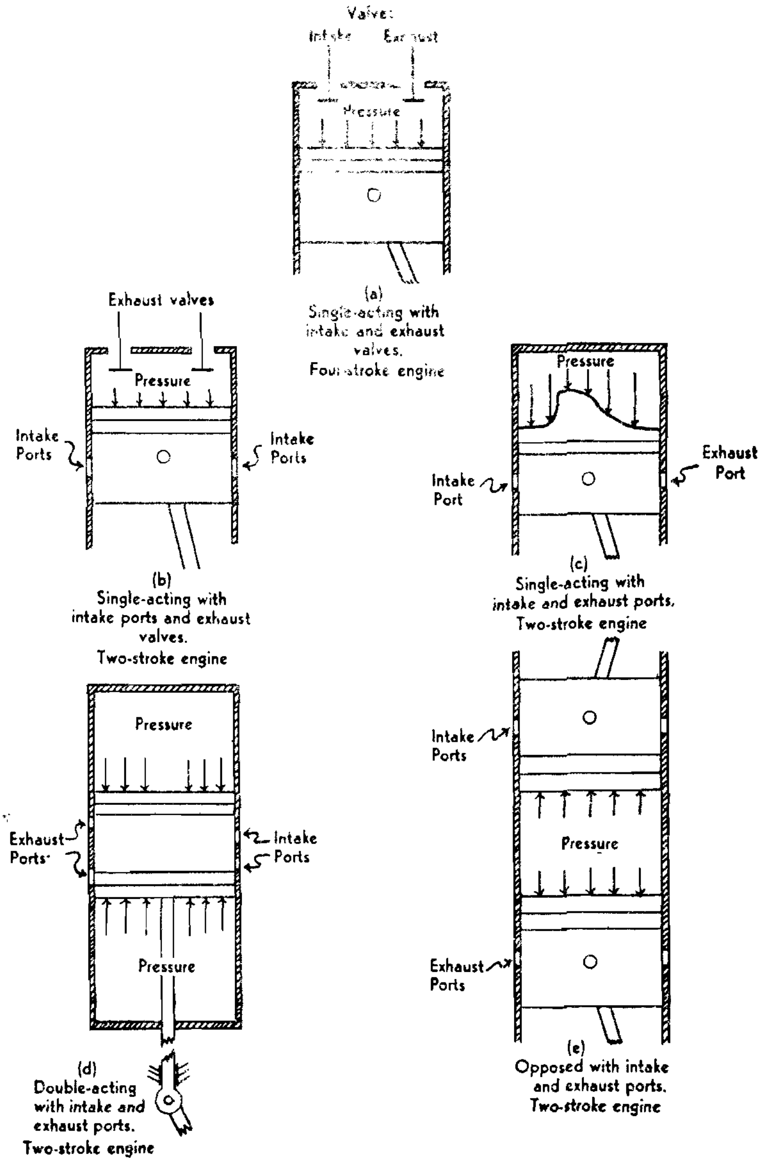


FIG. 11-3. Typical piston, valve, and port arrangements used in CI engines.

(1) *Single-acting piston type*—In the single acting type, the combustion pressures that affect the movement of the piston are exerted on the top of the piston, which together with the cylinder walls and the top of the cylinder form a combustion chamber. The following port and valve arrangements may be used with the single-acting type of engine:

(a) Intake and exhaust valve or valves, Fig. 11-3(a).
(b) Intake ports and exhaust valve or valves, Fig. 11-3(b).
(c) Intake ports and exhaust ports, Fig. 11-3(c).

(2) *Double-acting piston type*—In the double-acting type, Fig. 11-3(d), both ends of the piston are subject to pressure from a burning and expanding gas. The lower part of the cylinder as well as the bottom of the piston are closed, thus forming two combustion chambers, one in each end of the cylinder. The piston is driven down by the pressure created by the expanding gas in the upper combustion chamber, and is driven upward by the pressure created by the expanding gas in the lower combustion chamber. Note that the total force exerted on the piston by the gases in the lower combustion chamber is less than that exerted by the gases in the upper chamber. This is due to the fact that the piston area, in the lower chamber, is smaller by the amount equivalent to the crossectional area of the piston rod.

(3) *Opposed piston type*—In the opposed piston type, Fig. 11-3(e), two pistons operate in the same cylinder. The force created by the expanding gas is exerted only on the tops of the pistons (facing each other), which, along with the cylinder, form a single combustion chamber. On compression, the pistons travel toward each other while on the expansion stroke, they are forced away from each other by the expanding gas. Two sets of ports are provided for this type of engine—one to let the fresh air in, and the other to allow the burned gas to escape.

11-4. Fuel Supply and Injection Systems. One of the most important parts of the CI engine is its injection system. Engine performance depends on a proper functioning of this injection system, which must supply, meter, inject, and atomize the fuel. In fact, it plays such an important part that it can be considered the "heart" of the CI engine.

Injection systems are manufactured with great accuracy, especially the parts that actually meter and inject the fuel. Some of the tolerances between moving parts are very small—of the order of a hundred-thousandth of an inch. Such closely fitting parts require special attention during their manufacture and, as a result, injection systems are a costly item.

There are a number of injection systems in use today. Each of the

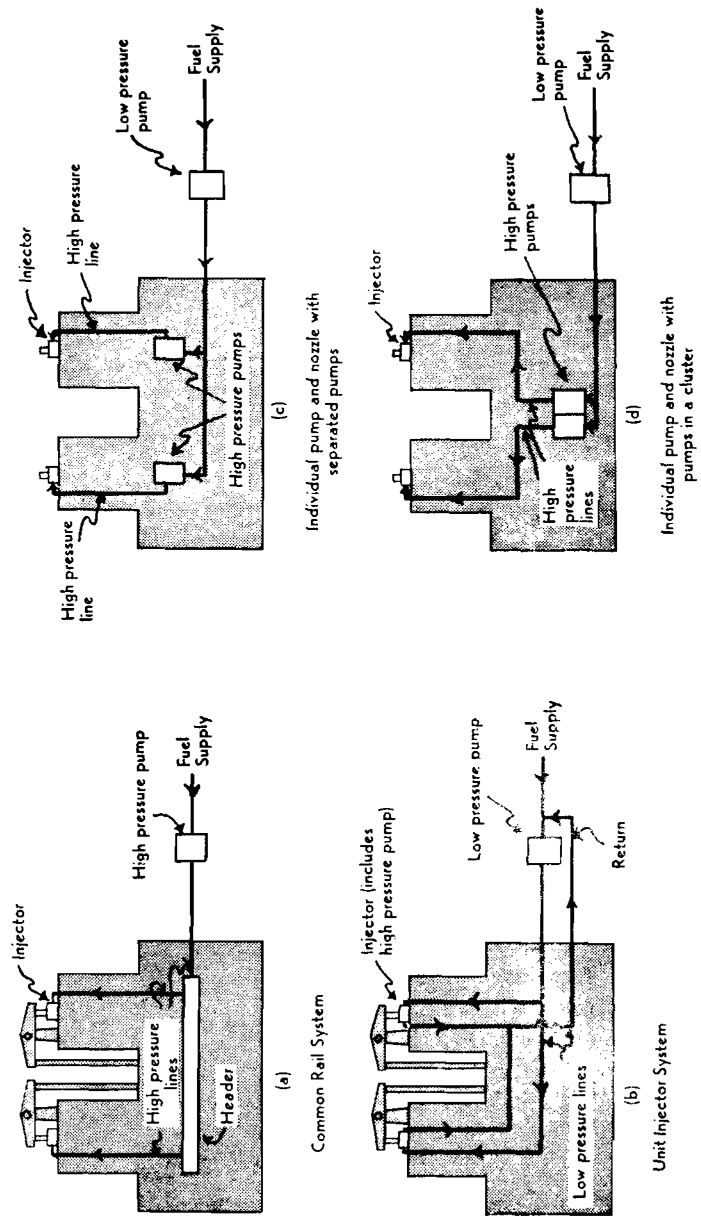


FIG. 11-4. Injection systems with pump and nozzle arrangements commonly used in CI engines.

systems has its own peculiarities which make it perform better from one point of view or another. Before adapting any one of these systems to a given engine, it is necessary to study the characteristics of an injection system in view of the engine requirements and engine design.

Injection systems may be divided into two general types, as follows:

(1) Air injection, and
(2) Solid injection

(1) *Air injection*—In the air injection system, fuel is forced into the cylinder by means of compressed air. This system is little used universally, at present, because it requires a multi-stage air compressor, which both increases engine weight and reduces brake horsepower. One advantage that is claimed for the air injection system is good mixing of fuel with the air, with resultant higher mep. Another is the ability to utilize fuels of high viscosity which are less expensive than those used by the engines having solid injection systems.

(2) *Solid injection*—The solid injection system consists essentially of two main parts: one is a high pressure pump, and the second is a check valve that allows fuel to enter the cylinder, but prevents the cylinder gases from backing up into the fuel system. A mechanical force actuates the positive displacement pump, which injects the fuel into the combustion chamber. Depending upon the location of the fuel pumps and injectors, upon the grouping and method of actuating the pumps, and upon the method used to meter the fuel, solid injection systems may be further classified as follows:

(a) common rail
(b) unit injection
(c) individual pump and nozzle

(a) *Common rail*—In the common rail system, Fig. 11-4(a), a pump supplies fuel, under high pressure, to a fuel header. The high pressure in the header forces the fuel to each of the nozzles located in the cylinders. At the proper time, a mechanically operated (by means of a push rod and rocker arm) valve allows the fuel to enter the cylinder through the nozzle. The pressure in the fuel header must be that for which the injector system was designed, that is, one able to penetrate and disperse the fuel in the combustion chamber. The amount of fuel entering the cylinder is regulated by varying the length of the push rod stroke.

(b) *Unit injector*—The unit injector system, Figure 11-4(b), is one in which the pump and the nozzle are combined in one housing. Each cylinder is provided with one of these unit injectors. Fuel is brought up to the injector by a low pressure pump, where, at the proper time, a rocker arm actuates the plunger and thus injects the fuel into the cylinder. The amount of fuel injected is regulated by the effective stroke of the plunger. Additional details of operation of this type of injector will be given in Article 11-5.

(c) *Individual pump and nozzle*—The individual pump and nozzle system is shown in Fig. 11-4(c) and (d). In this system, each cylinder is provided with one pump and one injector. This type differs from the unit injector in that the pump and injector are separated from each other, that is, the injector is located on the cylinder, while the pump is on the side of the engine. Each pump may be placed close to the cylinder which it supplies, as in Fig. 11-4(c), or they may be arranged in a cluster, as in Fig. 11-4(d). The high pressure pump plunger is actuated by a cam, and produces the fuel pressure necessary to open the injector valve at the correct time. Again, the amount of fuel injected depends on the effective stroke of the plunger. Further details of this type of pump will be covered in Article 11-5.

The most popular injector systems used today are the "unit injector" and the "individual pump and nozzle" type.

Injector pressures used to operate injectors range from 2,000 to about 20,000 psi. The usual range, however, is from about 2,000 to about 4,000 psi.

11-5. Typical Solid Injection Systems. There are many and varied injection systems in use today. All of these systems are designed with an aim to fulfill the following ideal requirements:

(1) The system must be able to meter the fuel accurately. This is very important, since quantity injected per cylinder and per cycle is indeed small, and metering errors become proportionately large.

(2) The fuel must be properly atomized.

(3) The fuel must enter the cylinder at the desired moment. That is, it must reach the cylinder at the time required by the engine to produce maximum performance. Injection must begin and end without lag. "After dribbling" is highly undesirable.

(4) Since the fuel in the CI engine continues to be injected during combustion, and since the rate of pressure rise may be controlled by the rate of fuel injection, this rate of injection must be properly regulated by the injection system.

(5) The fuel must penetrate into desired areas of the combustion chamber and be properly dispersed and distributed.

There are many injection system designs, but no attempt will be made to cover all of them. Only three typical systems will be discussed herewith, namely, (1) the American Bosch system, (2) the Unit Injector system, and (3) the Ex-Cell-O system.

While the pumping elements of these injection systems vary widely in their physical configuration, they all operate on the same basic principle. This principle may be simply illustrated by referring to Fig. 11-5(a) and (b). In both illustrations, the volume enclosed by the barrel and plunger is completely filled with a liquid, and A is a spring loaded check valve which opens only when a given pressure is reached. Another valve B, is provided, which may be opened or closed at will. Suppose the plunger is moved upward with valve B closed, Fig. 11-5(a). Since the liquid is almost incompressible, it will force its way out through the path of least resistance. Check valve A, consequently, will open, and the liquid will be released through A. This condition represents the nozzle and pump during injection of fuel into the cylinder. Now suppose the plunger is again moved upward, but with valve B open, Fig. 11-5(b). In this case, the liquid will seek the easiest

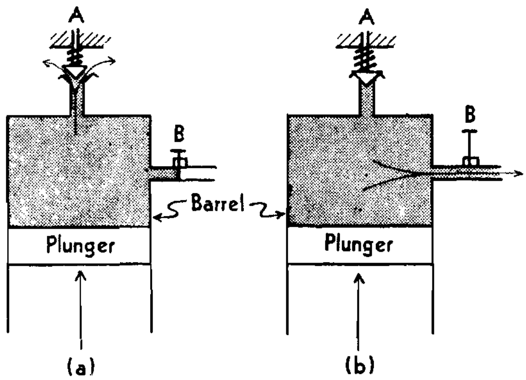


FIG. 11-5. Diagrams illustrating the basic principle of fuel metering in some injection systems.

escape route, which is that through valve B. This is the condition which exists in the pump system when fuel is not being injected, or has just completed injection.

Figure 11-6 is a schematic diagram which illustrates these principles in an actual injection system. Part (a) shows the physical configuration of a typical plunger, while the remaining parts are diagrams showing its operation in the barrel. The fuel is always available, under relatively low pressure, at the port A in the side of the barrel. The plunger is actuated axially along the barrel by mechanical means, and can be rotated about its axis by means of the rack D, Fig. 11-6(b), the movement of which is controlled by the fuel control lever. Port B is the nozzle orifice through which the fuel flows into the combustion chamber, after passing a check valve.

Figure 11-6(b) represents the plunger, rotated by the rack, to a position where no fuel will be injected through the port B into the cylinder. When the plunger is below port A, the barrel above the plunger will fill with fuel. As the plunger rises and closes off port A, the liquid

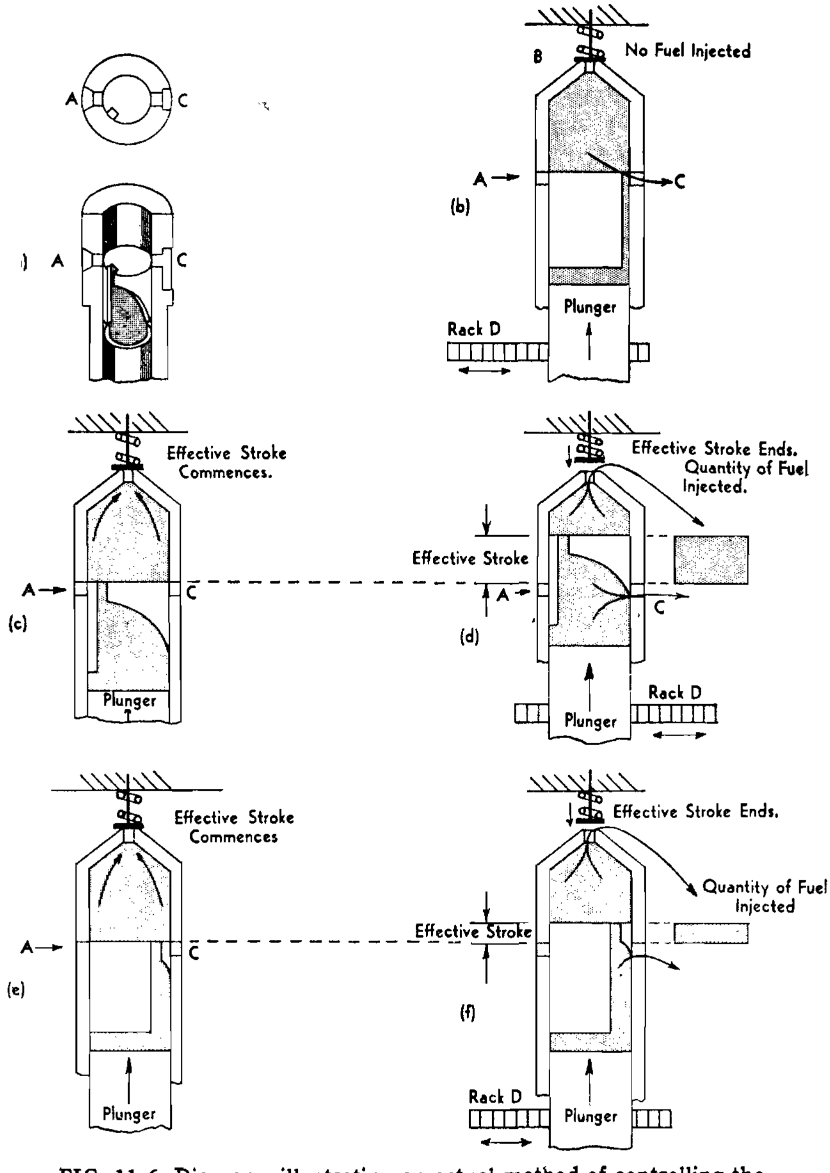


FIG. 11-6. Diagrams illustrating an actual method of controlling the quantity of fuel injected in a CI engine.

fuel has two possible escape paths, namely, through orifice B, or down along the linear channel on the plunger and out port C. Since it must overcome the force of the spring holding the check valve, in order to pass out through B, it takes the easier route through port C and is returned to the supply system. Therefore, no fuel is injected into the combustion chamber.

Now refer to Fig. 11-6(c) and (d). The plunger has been rotated by the fuel control lever, through the rack mechanism, until the plunger is in position to inject the maximum quantity of fuel per stroke. As the plunger moves upward past port *A*, Fig. 11-6(c), and traps the fuel in the barrel, it also closes off port *C*. This leaves only one escape route for the trapped fuel, past the check valve and through

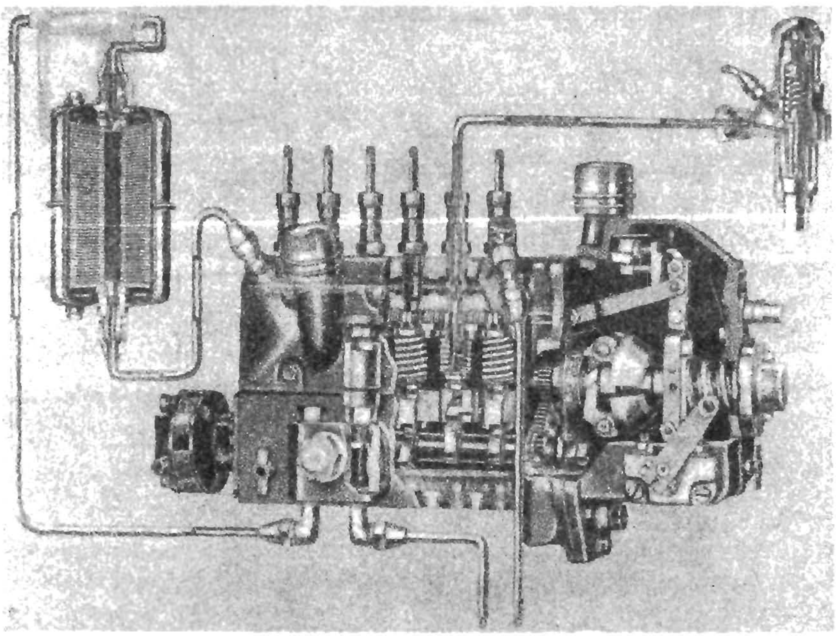


FIG. 11-7. Bosch fuel injection system (courtesy of American Bosch Corporation).

orifice *B* into the combustion chamber. The fuel will continue to be forced through orifice *B*, during the upward movement of the plunger, until the helical indentation on the plunger uncovers port *C*. At this point, the fuel will escape through the easier route out *C*, the check valve will close orifice *B*, and fuel injection into the combustion chamber will cease. The effective stroke of the plunger is the axial distance traversed between the time port *A* is closed off by the plunger, and the time port *C* is uncovered by the helix. Both the effective stroke and the relative quantity of fuel injected are indicated in Fig. 11-6.

If the plunger is now rotated to the position indicated in Fig. 11-6(e) and (f), the same sequence of events occurs, but since the port *C* is uncovered sooner, the effective stroke is shortened. Consequently, as indicated, a smaller quantity of fuel is injected into the combustion chamber.

It should be noted that the plunger continuously traverses the same

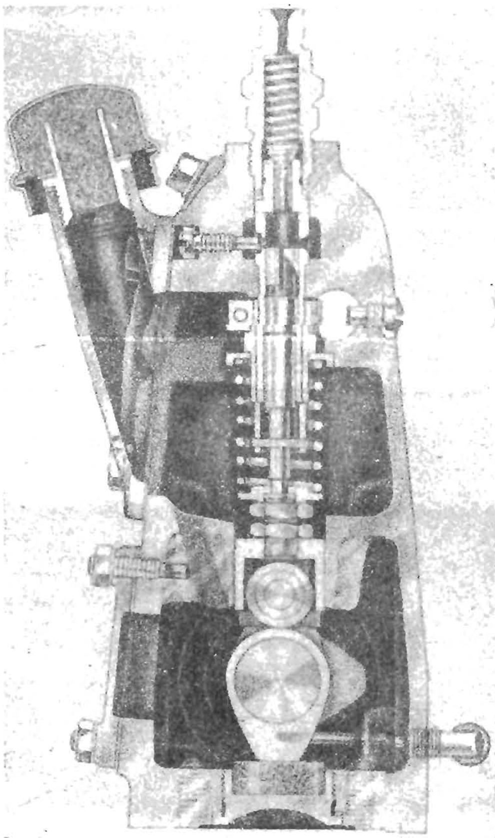


FIG. 11-8. *Cross sectional view of Bosch fuel pump (courtesy of American Bosch Corporation).*

axial distance on every stroke, as determined by the cam "lift." The rotation of the plunger by the rack, however, determines the length of the effective stroke, and thus the quantity of fuel injected.

While other systems are used to regulate the effective stroke of the plunger, the above discussed principle is the basis upon which the following specific injection systems operate.

(1) *The American Bosch Injection System*—The Bosch system of fuel injection consists essentially of two main parts, namely, a high pressure fuel pump and an injector. This system is of the type shown

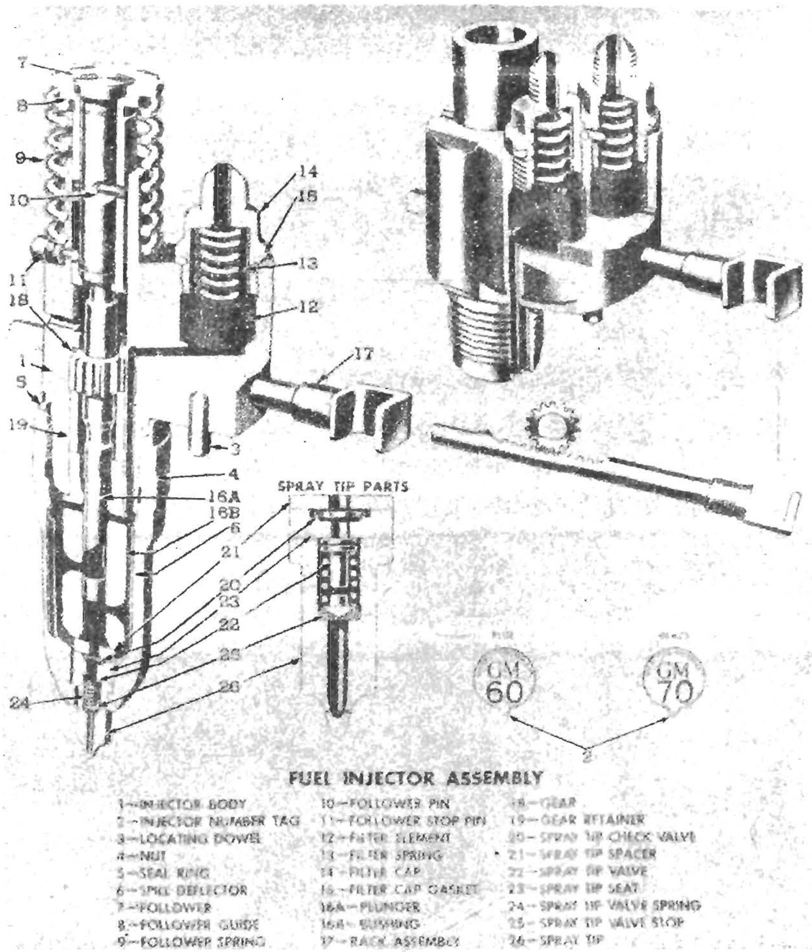


FIG. 11-9. Typical Unit Injector (courtesy of Detroit Diesel Engine Division, General Motors Corporation).

in Fig. 11-4(d), where all of the individual high pressure pumps are placed in one housing, and attached to the side of the engine. High pressure lines start at the pump and terminate at the injector, which is located in the cylinder head. The pump plungers are actuated by means of cams (one cam per pump), which are arranged "in-line" as shown in Fig. 11-7. The pump is of the positive displacement type, Fig. 11-8. At the proper moment the cam actuates the plunger, which forces the fuel, under high pressure, into the cylinder. The amount of fuel injected is controlled by a horizontal movement of the rack which, in turn, rotates the plunger and thus varies the effective stroke.

(2) *The Unit Injection System*—The Unit Injection system combines the high pressure pump with the injector nozzle in a single housing located at each cylinder, Fig. 11-9. This system is adapted to an engine in a manner shown in Fig. 11-4(b). As in the case of the Bosch system, the Unit Injector employs a positive displacement pump consisting of a plunger operating in a barrel. Again, the amount of fuel injected depends on the effective stroke of the plunger. The effective stroke and,

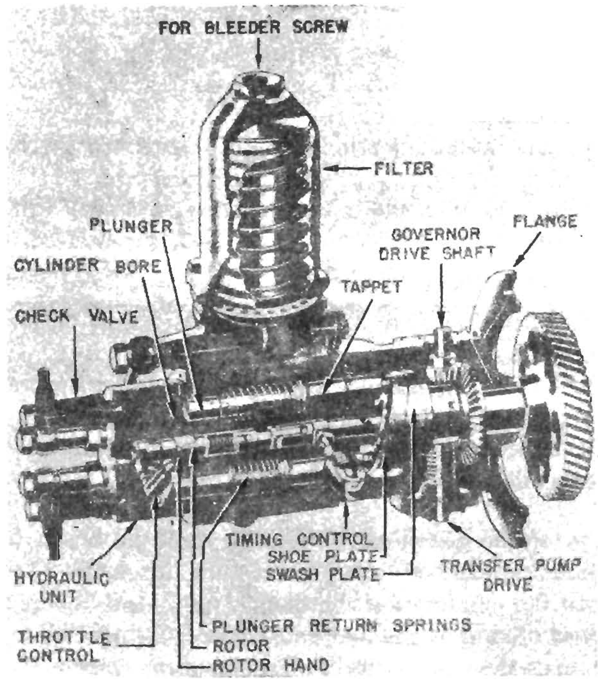


FIG. 11-10. Ex-Cell-O fuel injection pump (courtesy of Ex-Cell-O Corporation).

hence, the amount of fuel injected, may be varied by the horizontal movement of the rack which, in turn, rotates the plunger. In this type, the cam action is transmitted to the plunger through a push rod and a rocker arm. In the Bosch system, however, the cam directly actuates the plunger.

(3) *The Ex-Cell-O Injection System*—In the Ex-Cell-O system, a wobble plate actuates all of the pump plungers, which are circumferentially arranged in one housing. One plunger with its' barrel and one nozzle are provided for each cylinder. These are connected by individual lines which transmit the fuel, under the high pressure, from the pump to the nozzle. The system is adapted to the engine in a man-

ner shown in Fig. 11-4(d). Again, the pump is of the positive displacement type, consisting of a plunger operating in a barrel, Fig. 11-10. In this system, the effective stroke is regulated by a single common rotating element, the axial movement of which controls the opening of the by-pass ports, and thus controls the quantity of fuel injected.

11-6. The Injector Nozzle. Since there is no carburetor in the CI engine, the processes of atomization and distribution of the fuel, as well as the mixing of the fuel and air, must be accomplished within the combustion chamber. These processes affect both the power output and the fuel economy, and must therefore be done in the most efficient manner possible.

The liquid fuel in the injection system passes into the combustion chamber through the injector nozzle. This nozzle must be so designed that, in conjunction with the injection pressure, it will fulfill the following functions:

(1) Atomize the fuel
(2) Distribute the fuel to the required areas within the combustion chamber
(3) Prevent impingement of the fuel directly on the walls of the combustion chamber or on the piston, and
(4) In the non-turbulent type of combustion chamber, "mix" the fuel and air.

The design of the nozzle must be such that the liquid fuel forced through the nozzle will be broken up into fine droplets, or atomized, as it passes into the combustion chamber. This is the first phase in obtaining proper mixing of the fuel and air in the combustion chamber.

The fuel must then be properly distributed, or dispersed, in the desired areas of the chamber. In this phase, the injection pressure, the density of the air in the cylinder, and the physical qualities of the fuel in use, as well as the nozzle design, become important factors. Higher injection pressure results in better dispersion as well as greater penetration of the fuel into all locations in the chamber where its presence is desired. It also produces finer droplets which tend to mix more readily with the air. The greater the density of the compressed air in the combustion chamber, the greater the resistance offered to the travel of the fuel droplets across the chamber, with resultant better dispersion of the fuel. The physical qualities of the fuel itself, such as viscosity, surface tension, etc., also enter into the dispersion of the fuel.

The nozzle must spray the fuel into the chamber in such a manner as to minimize the quantity of fuel reaching the surrounding walls.

Any fuel striking the walls tends to decompose, producing carbon deposits, unpleasant odor, and a smoky exhaust, as well as an increase in fuel consumption.

The design of the nozzle is closely interrelated to the type of combustion chamber used. The subject of combustion chamber design will be discussed in Article 12-6. It is sufficient to state here that the "turbulent" type of combustion chamber depends upon chamber turbulence to produce the required mixing of the fuel and air. The "non-turbulent" type of combustion chamber, on the other hand, depends almost entirely on both the nozzle design and injection pressure to secure the

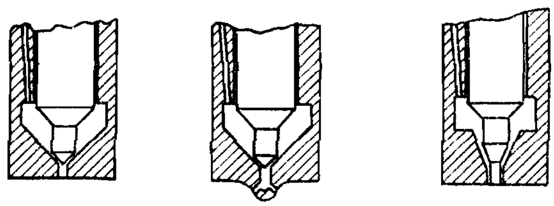


FIG. 11-11. Types of nozzles commonly used in CI engine fuel injection systems.

desired "mixing" in the combustion chamber; consequently, with this type of chamber, the nozzle must accomplish the additional function of "mixing" the fuel and air.

Various types of nozzles are used in CI engines, but the most common types are the *single orifice nozzle,* the *multi-orifice nozzle,* and the *pintle-nozzle.* These types are shown in Fig. 11-11.

The single orifice nozzle consists of a single hole in the end of the nozzle, through which the fuel passes into the combustion chamber. The multi-orifice type contains several such drilled passageways. The pintle nozzle consists essentially of a small plunger in the fuel passageway, the movement of which controls the flow of fuel into the chamber.

Each type of nozzle design has its own characteristic performance, with certain advantages and disadvantages. There is no single nozzle design which is ideal, or even suitable, for all applications. One type of nozzle may produce good results in one type of combustion chamber, but very poor results in another. Consequently, the nozzle must be designed and fitted to the type of combustion chamber used, and represents a compromise to secure the best over-all performance of the engine.

In general, the single and multi-orifice nozzles are used with the non-turbulent type of combustion chamber. The orifices of these nozzles are very small and are subject to clogging by carbon particles.

which may either interfere with the functioning of the nozzle stream, or may even completely stop the flow through some orifices. Consequently, this type of nozzle usually requires greater maintenance and higher operating expense.

The pintle type nozzle is generally employed with turbulent type combustion chambers, which utilize lower injection pressures. Due to the action of the pintle, this type of nozzle is less susceptible to clogging, and the maintenance expense is thereby usually reduced.

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EXERCISES

11-1. In general, which engine fuel, SI or CI, contains the greatest heating value per pound? Per gallon?

11-2. Why are high pressures required in CI engine injectors?

11-3. What are the compression ratio ranges used in SI and CI engines?

11-4. Why are higher thermal efficiencies obtained in the CI engine than in the SI engine?

11-5. For the same compression ratio, which has the higher thermal efficiency, the SI or the CI engine?

11-6. Does the SI or CI engine have higher specific weight (pounds per bhp)?

11-7. Is the four-stroke cycle SI engine similar in operation to the four-stroke cycle CI engine as far as cycle events are concerned? What may be said about the similarity in the scavenging process between SI and CI engines?

11-8. Draw p-V diagrams for four and two-stroke cycle CI engines indicating valve or port timing events.

11-9. Scavenging blowers use pressures of what range? What are the functions of the supercharger and the scavenging blower?

11-10. Describe single acting, double acting, and opposed piston type CI engines.

11-11. What are the two general types of injection systems? Describe briefly each of them.

11-12. There are three solid injection systems used with CI engines. What are they? Describe briefly each of them.

11-13. What are the five ideal requirements governing injection system design?

11-14. Describe briefly the major differences among Bosch, Unit Injector, and Ex-Cell-O injection systems.

11-15. What are the four functions of the nozzle?

11-16. What are the most common types of nozzles used?

11-17. Why is the pintle type nozzle generally employed with turbulent combustion chambers?

CHAPTER XII

COMBUSTION IN THE CI ENGINE

Chapter XI dealt with the method of supplying fuel and air to the combustion chamber in the CI engine. In this chapter, the combustion of this fuel and air, along with associated problems, will be discussed. Also, the types of combustion chambers, as well as their performance and their application, will be brought out and correlated with the types of nozzles discussed in Chapter XI.

12-1. Combustion in the CI Engine. It has been stated previously that certain fundamental differences exist between the SI and the CI engines. One of these major differences is the method by which the fuel is ignited. In the SI engine, a spark plug is used to initiate combustion. In a CI engine, the high temperature of the compressed air in the cylinder causes the fuel to ignite. Because of this different type of ignition, a number of new problems and operating characteristics, that are foreign to a SI engine, are involved. Subsequent discussion will bring out these peculiarities that are associated with the CI engine.

In order to understand combustion in a CI engine, it is necessary to review the mechanics of injection and the manner used to mix the compressed air and the injected fuel in the combustion chamber. By means of an injector, fuel is introduced into the chamber, and mixing of the air and the fuel then takes place. The degree of mixing will depend on the dispersion of the fuel and the turbulence of the air. The injector commences delivery of fuel before the piston reaches TDC. Since a finite time is required to deliver all of the fuel necessary for combustion during a given cycle, the injection of fuel continues for a number of degrees of crank angle (up to about 35). The exact duration depends on the speed and the size of the engine.

In a SI engine, a homogeneous air-fuel mixture is introduced into the combustion chamber and a single definite flame front progresses through this charge. In order to accomplish this flame front progression, the air-fuel mixture throughout the chamber must be within the combustible range (Art. 6-2). In a CI engine, however, a different situation exists. As the particles of fuel enter the chamber, they are dispersed and caused to mix with the air. This action results in a heterogeneous mixture, with A/F ratios varying widely in different areas within the chamber. In those areas where the mixture is very lean or very rich, a definite flame front can not be sustained. This situation is acceptable in a CI engine, however, since no definite *flame front* is required. In the CI engine, since the chamber is filled with compressed air at a temperature above the ignition temperature of the fuel, burning may be initiated at any point within the chamber where the *local* A/F ratio is conducive to combustion. In essence, this amounts to a multitude of "spark plugs" operating throughout the chamber. Consequently, any fuel, irrespective of how small the amount, will burn upon contact with the hot air. Theoretically, then, even though the *over-all* chamber A/F ratio is infinitely lean, burning can take place in any location where a fuel particle exists. Actually, however, the practical CI engine over-all A/F ratios vary from about chemically correct (usually on the lean side of chemically correct A/F) to about 100 to 1. In summary, it should be emphasized that *in a SI engine a homogeneous air-fuel mixture within the combustible range sustains the progress of a definite flame front across the combustion chamber. In a CI engine, on the other hand, the A/F ratios in the various parts of the chamber vary widely, no definite flame front is evident, and combustion occurs in many locations within the chamber.*

Air-fuel ratio plays an important role in CI as well as in SI engine operation, because it affects both power output and engine economy. The air supply in a CI engine is essentially constant for a given speed, and changing the quantity of fuel injected changes the A/F ratio. When decreasing amounts of fuel are used, the A/F ratios become leaner, and the indicated thermal efficiency approaches that of the air cycle. Due to the smaller amount of available energy, however, the mep and power output are lowered. If increasing amounts of fuel are injected, the A/F ratios become richer and the indicated thermal efficiencies are reduced. The mep and power output however, are increased with decreasing A/F ratios, up to a point in the vicinity of the chemically correct ratio.

As the amount of fuel is increased and the chemically correct A/F ratio is approached, the CI engine begins to produce a noticeably "black" smoke. This is due to the fact that in certain areas within the chamber, the A/F ratio will be so rich that some of the carbon and oxygen particles will be unable to combine in the time allotted for combustion. These uncombined carbon particles are the cause of the engine smoke. While higher power may be obtained by operating in the vicinity of the chemically correct A/F ratio, this objectionable smoke necessitates operation at slightly leaner A/F ratios, with consequent reduction in power output.

Figure 12-1 shows the power curve for a typical CI engine operating at constant speed. The approximate region of A/F ratios in which visible "black" smoke will occur is indicated by the shaded area.

The two branches of the power curve close to chemically correct A/F ratio were shown to indicate the fact that the power output of some of the CI engines reaches the peak on the rich side and in the immediate vicinity of the chemically correct A/F ratio, while other CI engines reach the power peak at A/F ratios considerably richer than the chemcally correct ratio.

At idling, light "whitish" smoke may appear. Under these operating conditions, lower temperatures in the combustion chamber, plus less turbulence, may result in only partial burning of some of the fuel particles with consequent smoky and unpleasant smelling exhaust.

Due to the practical limitations caused by smoke, therefore, CI engines are operated at A/F ratios leaner than the chemically correct

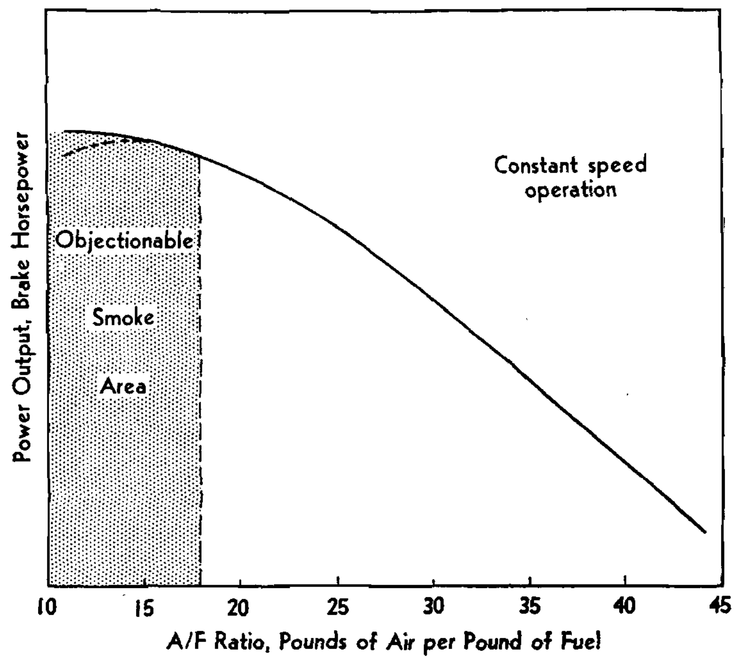


FIG. 12-1. Effect of A/F ratio on power output of a CI engine.

ratio. The additional amount of air (calculated on percentage-weight basis) above that required for the stoichiometric ratio is termed **excess air.** *Due to the shortcomings of distribution and limited intermixing* of fuel with air within the combustion chamber, the CI engine always operates with excess air. It should be pointed out, however, that the goal of the CI engine designer, when operating at full throttle, is to utilize, through combustion, as much of the trapped oxygen in the combustion chamber as possible, thereby reducing excess air to a minimum.

12-2. **Ignition Delay.** In Chapter VIII, the subject of ignition delay in a SI engine was discussed. It was stated that this delay occurs between the time the spark is produced and the time when the "actual burning" phase of combustion commences. Ignition delay of the fuel also occurs in the CI engine, and exerts considerable influence on both engine design and performance.

The fuel does not ignite immediately upon injection into the CI engine combustion chamber. There is a definite period of apparent inactivity between the time when the first droplet of fuel hits the hot air in the combustion chamber, and the time when it starts through the "actual burning" phase. This period is known as **ignition delay** or **ignition lag.** During this period there is no rise in pressure within the cylinder due to combustion. This delay is indicated on the *p-t* diagram, Fig. 12-2, as the distance between points *"a"* and *"b."* Point *"a"*

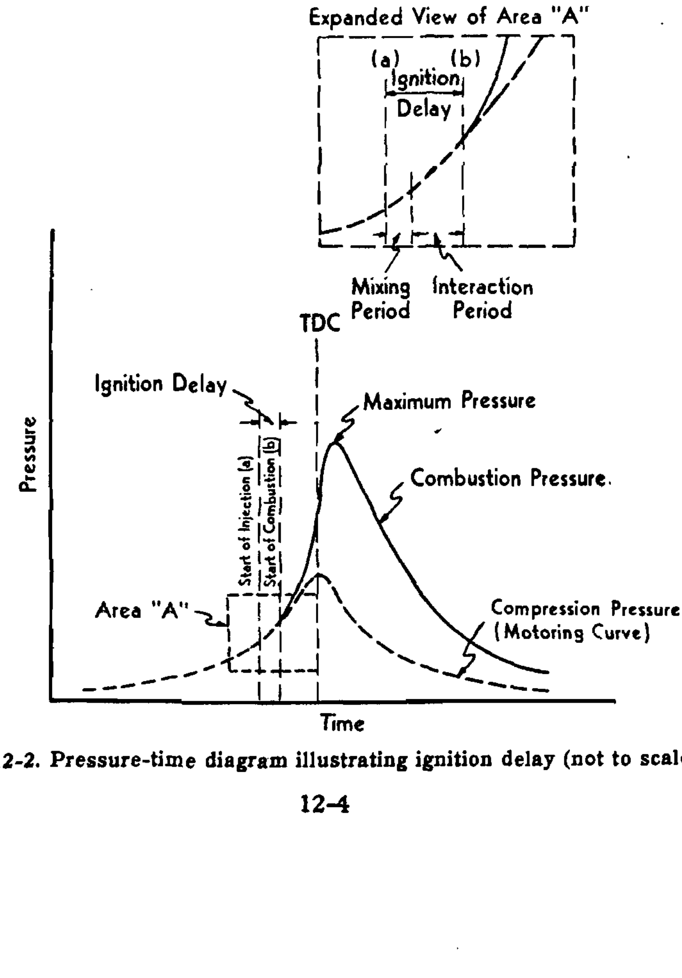


FIG. 12-2. Pressure-time diagram illustrating ignition delay (not to scale).

represents the time of injection, and point "b" represents the time at which the pressure curve (caused by combustion) first separates from the compression pressure (non-firing) curve.

The ignition delay period, as indicated in Fig 12-2, is divided into two parts:

(1) The *mixing period,* which is the time required for atomization and evaporation of the fuel, and physical mixing with the air, and
(2) The *interaction period,* in which the molecular interaction prepares the mixture for, and initiates, the "actual burning" phase of combustion. This is the longer of the two periods.

Ignition delay of the fuel is of extreme importance in CI engines primarily because of its effect on both the combustion rate and on detona-

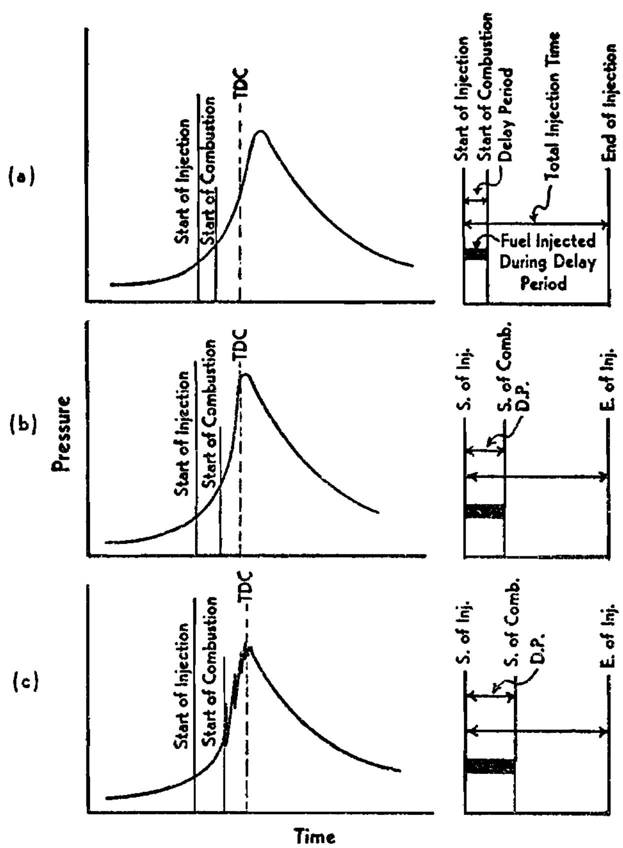


FIG. 12-3. Diagrams illustrating the effect of ignition delay on the rate of pressure rise in a CI engine (not to scale).

tion. It may also exert an influence on engine starting ability, and on the presence of smoke in the exhaust.

12-3. Combustion Knock in the CI Engine. The injection process takes place over a definite interval of time. Consequently, as the first droplets to be injected are passing through the ignition delay period, additional droplets are being injected into the chamber. If the ignition delay of the fuel being injected is short, the first droplets will commence the "actual burning" phase in a relatively short time after injection, and a relatively small amount of fuel will be accumulated in the chamber when "actual burning" commences. As a result, the mass rate of

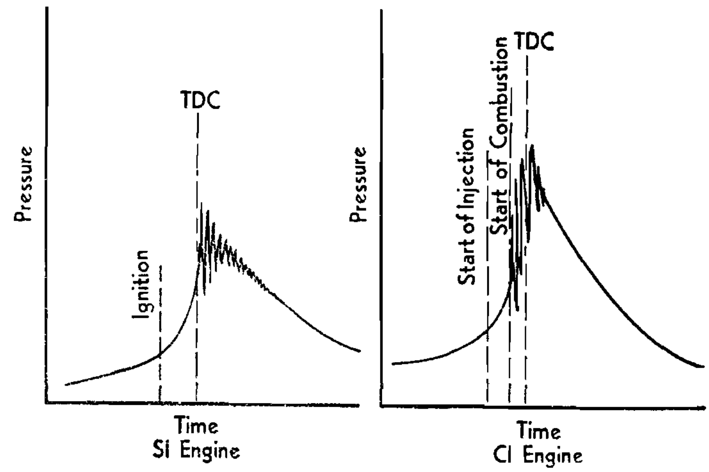


FIG. 12-4. **Diagrams illustrating knocking combustion in SI and CI engines (not to scale).**

mixture burned will be such as to produce a rate of pressure rise that will exert a smooth force on the piston, as shown in Fig. 12-3(a). If, on the other hand, the ignition delay is longer, the "actual burning" of the first droplets is delayed, and a greater quantity of fuel droplets will accumulate in the chamber. When the "actual burning" commences, the additional fuel can cause too rapid a rate of pressure rise, Fig. 12-3(b), resulting in a "jamming" of forces against the piston and rough engine operation. If the ignition delay is very long, so much fuel can accumulate that the rate of pressure rise is almost instantaneous, Fig. 12-3(c). Such a situation produces the extreme pressure differentials and violent gas vibrations known as detonation, and evidenced by audible knock. The phenomenon is similar to that in the SI engine. However, *in the SI engine, detonation occurs near the end of combustion,* *whereas, in the CI engine, detonation occurs near the beginning of combustion.* A schematic comparison of SI and CI knocking engines is shown on the *p-t* diagrams of Fig. 12-4.

In order to decrease the tendency to knock, therefore, it is necessary to start "actual burning" as early as possible after the injection takes place. In other words, it is necessary to decrease the ignition delay and thus decrease the amount of fuel present when the "actual burning" of the first droplets starts.

Through extensive studies in the laboratory, and from actual experience through operation of engines, it has been found that fuels made up of different hydrocarbons will have different ignition delays. For instance, cracked fuels produce longer ignition delay than the straight run fuels. This fact does not necessarily hold true if the straight run fuel is highly aromatic. As discussed in Article 5-7, ignition delay is generally measured in terms of cetane number. Fuels of higher cetane number have shorter ignition delay and, thus, have a lesser tendency to knock.

The ignition delay of CI engine fuels may be decreased by the addition of small amounts of certain compounds to the fuel such as ethyl nitrate, amyl thionitrite and others. These additives affect the combustion process by speeding the molecular interaction period, thereby decreasing the ignition delay and increasing the cetane number of the fuel.

12-4. Variables Affecting Ignition Delay. The preceding articles dealt with ignition delay, its effect on engine operation, and the possibility of decreasing ignition delay through medium of fuel quality (cetane number). There are, however, a number of engine variables that may also affect the ignition delay, and thereby affect engine performance. These engine variables are as follows:

(1) Compression ratio
(2) Inlet air temperature
(3) Coolant temperature
(4) Engine speed

(1) *Compression ratio effect*—For a given fuel, an increase in compression ratio will decrease the ignition lag. This fact is illustrated by Fig. 12-5. As the compression ratio increases, the minimum auto-ignition temperature decreases (curve "*a*"). This is apparently due to the increased density of the compressed air, resulting in closer contact of the molecules which thereby hastens the time of reaction when fuel is injected. Also, it can be seen from Fig. 12-5 that cylinder air tem-

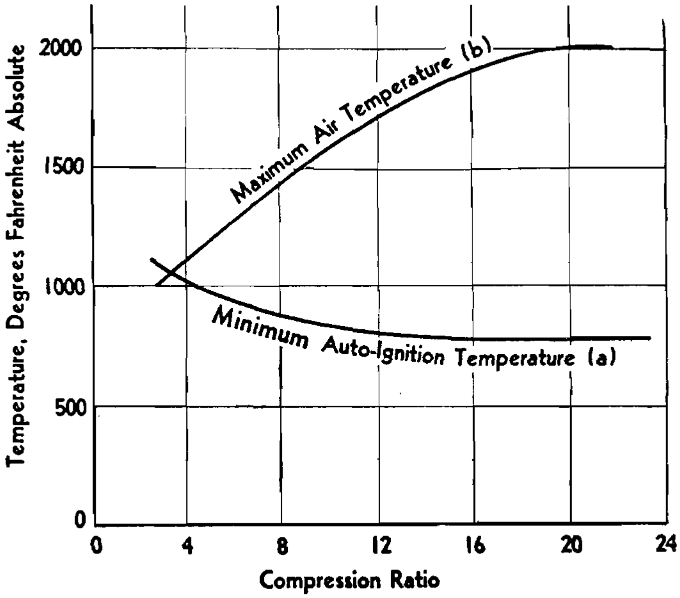


FIG. 12-5. Effect of compression ratio on minimum auto-ignition temperature and maximum cylinder air temperature (courtesy of NACA—Rothrock, A. M., Tech. Rept. 401, 1932).

perature increases with compression ratio (curve "*b*"). As the difference between cylinder air temperature and the minimum auto-ignition temperature increases, the ignition delay has been found to decrease. Fig. 12-5 shows that as the compression ratio increases, the difference between these two temperatures also increases. This results in a decrease in the ignition delay.

(2) *Inlet air temperature effect*—An increase in the intake air temperature will also decrease the ignition delay, because it raises cylinder air temperature. Increasing inlet air temperature raises curve "*b*" of Fig. 12-5, thereby increasing the temperature difference between curves "*a*" and "*b*," and thus decreasing ignition delay.

(3) *Coolant temperature effect*—The ignition delay decreases with an increase in coolant temperature, as shown in Fig. 12-6. Increase in the coolant temperature tends to decrease the heat transfer from the combustion chamber, thus producing higher cylinder air temperature. Higher cylinder air temperature, in turn, will increase the temperature difference shown in Fig. 12-5, resulting in decreased ignition delay.

(4) *Engine speed effect*—The degree of turbulence within the combustion chamber has a definite effect on the ignition delay. As the en-

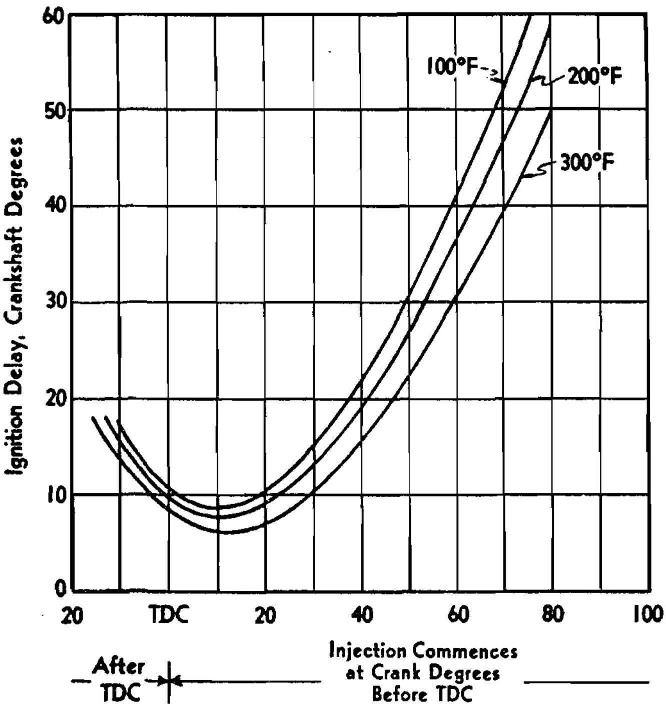


FIG. 12-6. Effect of coolant temperature on ignition delay (courtesy of NACA—Rothrock, A. M., and Waldron, C. D., Tech. Rept. 525, 1935).

gine speed increases the turbulence also increases. Unfortunately, however, the increase in turbulence also will increase the heat loss to the combustion chamber walls and, hence, the lower combustion chamber temperatures will result. On the other hand, if it would be possible to increase the engine turbulence as well as cylinder temperature and thus its pressure, then a decrease in ignition delay may be realized.

12-5. General Functions and Characteristics of the Combustion Chamber. Proper design of the combustion chamber is at least as important in the CI engine as in the SI engine. In the SI engine, a nearly homogeneous mixture enters the cylinder, is compressed, and then ignited by means of a spark plug. The fuel and air are mixed in the carburetor. In the CI engine, on the other hand, only the air is compressed in the cylinder, and the fuel is injected during a period of some 20 to 35 degrees of crank angle. In this short period of time, the fuel and air must be mixed. In essence, the mixing portion of the SI engine carburetor's duties are performed within the combustion chamber in a CI engine. Consequently, the combustion chamber in a CI engine must be designed to provide for this mixing of fuel and air.

COMBUSTION IN THE CI ENGINE

While there are many and varied CI engine combustion chamber designs in use today, they all belong essentially to one of two general types, or some combination of these types. These two general types are:

(1) The non-turbulent type, and
(2) The turbulent type

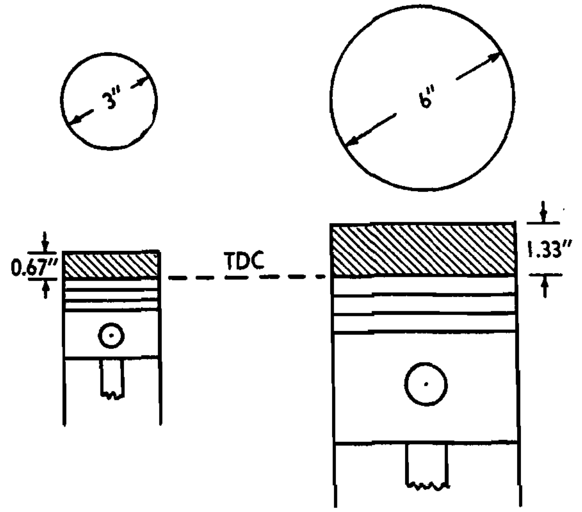
In the *non-turbulent* chamber, the mixing of the fuel and air depends only to a minor degree upon the turbulence of the air created by the chamber design. Primarily, the mixing is obtained through the use of orifice types of nozzles operating under high pressures. The fuel enters the cylinder in a highly atomized form at high velocities, and is sprayed throughout the chamber to produce the necessary contact between the fuel and air particles.

In the *turbulent* chamber, pintle type nozzles are generally used, and the fuel pressures are considerably lower. The combustion chamber is so designed, however, that it produces an exceedingly turbulent flow of air within the chamber, thereby physically mixing the fuel and air particles and causing them to come in contact with one another.

Each of these general types of chambers has certain advantages and disadvantages, which will be discussed in Article 12-6. The introduction of turbulence, however, produces a penalty which will be discussed at this point. Turbulent air in the chamber tends to sweep away any stagnant layers of gas near the walls of the combustion chamber. Removing this "insulator" causes higher heat transfer from the burning gases to the chamber walls, and tends to reduce the temperature within the chamber. Such a situation is particularly undesirable during starting when it is necessary to get the compression temperature as high as possible in order to ignite the fuel and air mixture.

Another factor entering into the transfer of heat to the cylinder walls is the *surface to volume ratio* (S/V) of the combustion chamber. As the surface area of the chamber increases, the gases are exposed to a greater cooling area. The quantity of heat added for a given load and a given A/F ratio, on the other hand, depends on the amount of fuel injected. For a given amount of heat added, the space or the volume of the combustion chamber in which this heat is confined is important, because part of this supplied heat will be lost through the combustion chamber walls, and the amount of this loss depends on S/V ratio. The larger the S/V ratio, the more heat will be transferred to the cylinder walls.[1] The S/V ratio, therefore, is an indication of the cooling that might be expected from a given combustion chamber.

[1] The piston crown area is not considered in computations of S/V ratio, because it does not provide a direct path for heat from combustion chamber to the water jacket.



$$\text{Surface} = \frac{\pi D^2}{4} + 2\pi r \times 0.67$$
$$= 13.42 \text{ square inches}$$
$$\text{Volume} = \frac{\pi D^2}{4} \times 0.67 = 4.75 \text{ cubic inches}$$
$$S/V = \frac{13.42}{4.75} = 2.83$$

$$\text{Surface} = \frac{\pi D^2}{4} + 2\pi r \times 1.33$$
$$= 53.4 \text{ square inches}$$
$$\text{Volume} = \frac{\pi D^2}{4} \times 1.33 = 37.7 \text{ cubic inches}$$
$$S/V = \frac{53.4}{37.7} = 1.415$$

FIG. 12-7. Diagrams illustrating surface to volume (S/V) ratios for two cylinders of different sizes.

Figure 12-7 indicates the S/V ratio of two simple combustion chambers of two engines having different size bore, but the same diameter to stroke ratio and the same compression ratio. It can be seen that the cylinder with the smaller bore tends to have the higher S/V ratio, indicating that it will tend to lose a greater proportion of its heat through the walls and consequently, run cooler. Because of this fact, engines of small bore, i.e., engines having a high S/V ratio, are generally difficult to start. In order to overcome this deficiency, special devices are often used with such engines to raise the temperature of the air in the combustion chamber until the engine is running. The *glow plug* is one type of device, and is a heating element installed in the combustion chamber and operated on current provided by the battery. It is placed in operation about 20 seconds before starting the engine, and turned off shortly after the engine starts and the combustion chamber is warmed up. Another method used to heat the air employs cutting off a part of the combustion chamber temporarily, thus increasing the compression ratio, and producing a higher temperature. One other interesting procedure entails delaying the intake valve opening, as well as reducing the valve lift, during starting. When the valve opens, the air rushes into the cylinder at high velocity. The kinetic energy is converted to heat as the gas comes to rest in the cylinder, thus raising the combustion chamber temperature.

12-6. Comparison of Some Basic Designs of CI Engine Combustion Chambers. There are many types of combustion chambers used in CI engines. Each of these has its own peculiarities, and desirable, as well as undesirable features. Any one of these combustion chambers may produce good results in one field of application, but less desirable, or even poor results in another. No one combustion chamber design has yet been developed which will produce the best results in all types of engines. The particular design chosen, then, must be that which accomplishes the best performance for the application desired. A presentation of each of the many variations of combustion chambers is beyond the scope of this book. Four specific designs which find wide use in the CI engine, however, with the general advantages and disadvantages of each, will be discussed. These are as follows:

(1) The non-turbulent type
  (a) Open combustion chamber
(2) The turbulent type
  (a) Turbulent chamber
  (b) Precombustion chamber
  (c) Energy cell

Figure 12-8 shows diagrammatically the differences in the physical shape of these four chamber designs.

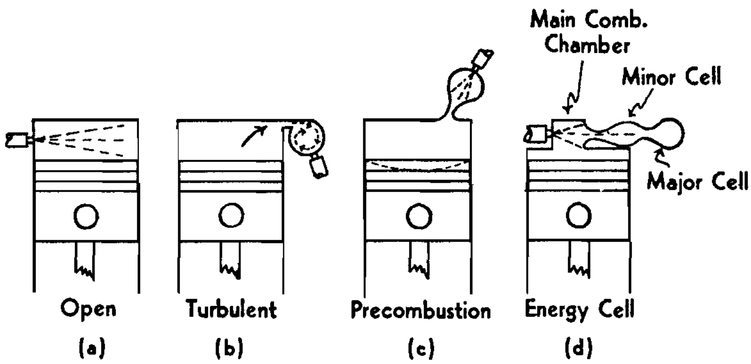


FIG. 12-8. Diagrammatic illustration of some commonly used CI engine combustion chambers.

(1) *The non-turbulent type*—Figure 12-8(a) illustrates the usual design of *open combustion chamber*, which is representative of the non-turbulent type. The fuel is injected directly into the upper portion of the cylinder, which acts as the combustion chamber. This type depends little on turbulence to perform the mixing. Consequently, the heat loss to the chamber walls is relatively low, and easier starting results. In order to obtain proper penetration and dispersal of the fuel necessary for mixing with the air, however, high injection pressures and multi-orifice nozzles are required. This necessitates small nozzle openings and results in more frequent clogging or diversion of the fuel spray by accumulated carbon particles, with consequent higher maintenance costs.

This type of chamber is ordinarily used on low speed engines, where injection is spread through a greater period of time and thus ignition delay is a relatively less important factor. Consequently, less costly fuels with longer ignition delay may be used. If attempts are made to use this type of chamber on high speed engines, the possibility of knock increases, since the allowable fuel injection time must be shortened. Ignition delay in this case becomes relatively more important, and greater quantities of fuel will accumulate during the delay period.

As stated above, the slow speed stationary engines are most likely to use the non-turbulent type of combustion chamber. Such engines, in general, for a given power output, require larger engine bore and stroke than higher speed engines, resulting in a greater weight to power output ratio.

Because of the small amount of turbulence found in a non-turbulent chamber, it is necessary to use considerable excess air. This greater excess air causes the engine to approach an air cycle performance, thus producing higher indicated thermal efficiencies as compared with engines requiring less excess air. This fact is accomplished, however, at the expense of power produced, i.e., lower mep will result.

In Great Britain, however, engines were designed and are being used at present time with the open type combustion chambers having predetermined air swirl. The swirl is produced by the designed cavities in the piston crown and introduction of directional air flow. The direction of the air flow is produced by either shrouding the intake valve or by designing an intake port in such a manner as to direct the air flow at right angles to fuel path, thus enabling a better distribution of fuel throughout the combustion chamber. The use of such combustion chambers resulted in improved engine performance, enabled variable speed engine operation and produced higher mep as compared with the usual type open combustion chamber.

(2) *The turbulent type*—The turbulent chamber, precombustion chamber, and energy cell are variations of the turbulent type of chamber, and are illustrated in Fig. 12-8(b), (c), and (d).

In the *turbulent chamber*, Fig. 12-8(b), the upward moving piston forces a flow of air into a small antechamber, thus imparting a rotary motion to the air passing the pintle type nozzle. As the fuel is injected into the rotating air, it is partially mixed with this air, and commences to burn. The pressure built up in the antechamber by the expanding burning gases force the burning and unburned fuel and air mixtures back into the main chamber, again imparting high turbulence and further assisting combustion.

In the *precombustion chamber*, Fig. 12-8(c), the upward moving piston forces part of the air into a side chamber, called the precombustion chamber. Fuel is injected into the air in the precombustion chamber by a pintle type nozzle. The combustion of the fuel and air produces high pressures in the precombustion chamber, which force the burning and burned mixture out into the main chamber, thus creating high turbulence and producing good mixing and combustion.

The *energy cell* is more complex than the precombustion chamber. It is illustrated in Fig. 12-8(d). As the piston moves up on the compression stroke, some of the air is forced into the major and minor chambers of the energy cell. When the fuel is injected through the pintle type nozzle, part of the fuel passes across the main combustion chamber and enters the minor cell, where it is mixed with the entering air. Combustion first commences in the main combustion chamber where the temperature is higher, but the rate of burning is slower in this location, due to insufficient mixing of the fuel and air. The burning in the minor cell is slower at the start, but due to better mixing, progresses at a more rapid rate. The pressures built up in the minor cell, therefore, force the burning gases out into the main combustion chamber, thereby creating added turbulence and producing better combustion in this chamber. In the meantime, pressure is built up in the major cell, which then prolongs the action of the jet stream entering the main chamber, thus continuing to induce turbulence in the main chamber.

All of the turbulent types of combustion chambers tend to exhibit the same general characteristics. This type of chamber depends primarily on turbulence to produce the required mixing of the fuel and air. Consequently, the heat loss to the chamber walls is relatively large, and starting is more difficult. The injection system, however, does not operate at such high pressures as in the non-turbulent type.

Furthermore, the pintle nozzle is less apt to clog, and maintenance difficulties are thereby reduced.

The turbulent type of chamber does not require as much excess air as the non-turbulent type, because it has the ability, through turbulence, to mix the air and fuel more thoroughly. A smaller amount of excess air tends to produce lower thermal efficiencies, although higher mep results.

The turbulent type combustion chambers are suitable for variable speed operation, because of their inherent high turbulence, which readily responds to an increase in the rate of fuel injection. Also, this type of chamber produces smoother operating engines. This is due to the fact that the antechambers absorb the initial shock of peak pressure, thus alleviating the piston from extreme pressure variations.

In summary, it may be said that a particular combustion chamber design must be chosen to perform a given job. No one combustion chamber can produce an ultimate of performance in all tasks. As in most engineering work, the design of the chamber must be based on a compromise, after full consideration of the following factors:

(1) Heat loss to combustion chamber walls
(2) Injection pressure
(3) Nozzle design—number, size, and arrangement of holes in the nozzle
(4) Maintenance
(5) Ease of starting
(6) Fuel requirement—ability to use less expensive fuels
(7) Utilization of air—ability to use maximum amount of air in the cylinder
(8) Weight relation of engine to power output
(9) Capacity for variable speed operation
(10) Smoothness with which forces created by expanding gases are transmitted to the piston.

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EXERCISES

12-1. Fuel that is used for combustion in the CI engine is injected through how many degrees of crank angle? What is the approximate maximum limit of injection in crank angle degrees?

12-2. How does the mixture of air and fuel in the combustion chamber of a CI engine differ from that of a SI engine?

12-3. What is the range of overall A/F ratios in a CI engine combustion chamber?

12-4. Does the flame front exist in a CI engine? Why?

12-5. What effect does the A/F ratio have on the indicated thermal efficiency in a CI engine?

12-6. What are the reasons for operating a CI engine at an A/F ratio higher than chemically correct?

12-7. What is meant by "excess air"?

12-8. What is meant by ignition delay or injection lag? It is usually divided into two parts. Name and describe them.

12-9. What causes the knock in a CI engine? In which part of the combustion process (beginning or the end) does it occur? How does this compare with the location of knock in a SI engine combustion process?

12-10. How do the injection timing and the fuel quality affect the engine knock?

COMBUSTION IN THE CI ENGINE

12-11. Name some engine variables that affect the ignition delay period. Explain briefly the effect of each of them.

12-12. In addition to confining the air within a given volume, what other important function does the CI engine combustion chamber perform?

12-13. What are the two general types of combustion chambers? Describe the process of mixing fuel and air in these chambers.

12-14. Why do the turbulent type chambers cause higher heat transfer than those of the non-turbulent type?

12-15. What is meant by S/V ratio? How does it affect the heat loss?

12-16. Name some of the methods used to increase combustion chamber temperatures when starting the CI engine in cold weather. Which of the two combustion chambers (turbulent or non-turbulent) is easier to start?

12-17. Name a design of combustion chamber representing non-turbulent type and three different designs representing turbulent type. Sketch each of them to show major variations.

12-18. What are the important points that should be borne in mind when considering a new combustion chamber design?

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CHAPTER XIII

COMPRESSION IGNITION ENGINE PERFORMANCE

Chapters XI and XII dealt with the characteristic component parts of the CI engine and their effect on engine operation. This chapter will cover the combined influence of these components on engine performance.

Many of the factors entering into the performance of four-stroke cycle CI engines are similar to those already analyzed for SI engines in Chapter X. These will not be again covered in detail in this chapter and, unless otherwise stated, this discussion will be concerned with the four-stroke cycle unsupercharged CI engine.

13-1. Heat Balance. A determination of the disposition of the energy provided to the engine by the fuel, or a heat balance, may also be made for a CI engine. As with a SI engine (Article 10-1) the results may be presented as a heat balance diagram. Such a diagram for a CI engine is illustrated in Fig. 13-1.[1] It should be noted that the portion of fuel energy going into the power output at full load is appreciably greater than that obtained in the SI engine. On the percentage input basis, therefore, heat losses will be smaller in the case of the CI engine as compared to the SI engine. The reason for this increase in energy at the drive shaft and decrease in losses, is attributed to the higher compression ratios used and the corresponding higher thermal efficiencies obtained in CI engines. The heat losses to the cooling water and to the exhaust increase with increase in load. Percentagewise, however, heat losses are little affected by increase in load, as indicated in Fig. 13-1. Also, it should be noted that friction losses on percentage basis for a given engine speed will decrease as the load is increased. This is due to the following reasons:

(1) Friction is affected considerably more by an engine speed rather than by a change in load.
(2) Heat loss remains about constant with the change in load.
(3) Power at the shaft increases with increase in load.

In general, all heat balances will resemble one another, but in detail they vary with both the particular engine and the operating conditions. The heat balance for a particular engine will change with variations in

[1] The reason why the heat balance diagram shown in Fig. 13-1 was plotted against load, rather than rpm, as in Fig. 10-1, is that most of the CI engines are operated at an approximately fixed rpm and a varying load. It should be understood, however, that for a variable speed CI engine a heat balance diagram of Fig. 10-1 type would be also plotted.

such factors as speed, load, fuel consumed, quantity of cooling water, cooling water inlet and outlet temperatures, and temperature and analysis of the exhaust gases. Figure 13-1, then, should be regarded as merely a representative diagram.

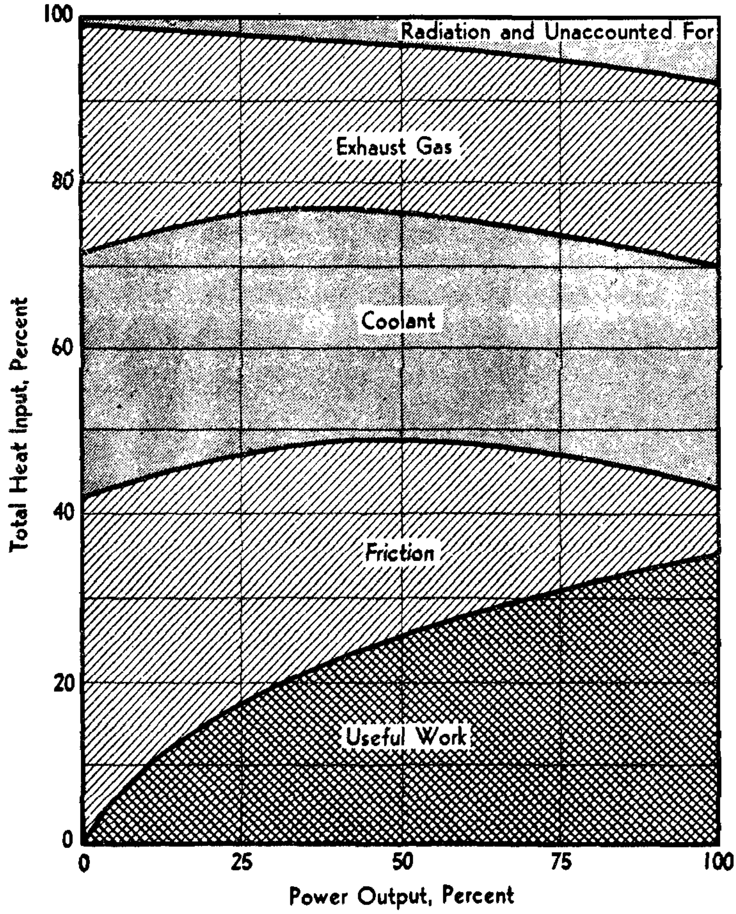

FIG. 13-1. Typical heat balance diagram of a four-stroke cycle unsupercharged CI engine operating at a given engine speed.

13-2. Variations Between the Air Cycle and the Actual Cycle of CI Engines. The variations between the air cycle and the actual cycle for a SI engine were discussed in Article 10-3. Some of these differences, such as variation of specific heat, heat losses, and the exhaust gas effect, apply similarly to the CI engine.

Following are the factors that influence CI engine cycle:

(1) Non-constant pressure combustion of the fuel
(2) Dissociation
(3) Blow-down losses
(4) Pumping loss

The number one item cited above is peculiar only to the CI engine cycle, while others are common to both the SI and CI engine cycles.

(1) *Non-constant pressure combustion of the fuel*—In the basic CI engine air cycle, it was assumed that the heat was added to the cycle at constant pressure. In the actual CI engine cycle, as shown by the indicator diagram, the heat is not added at constant pressure. In fact, in the case of high speed engines, it approaches a constant volume

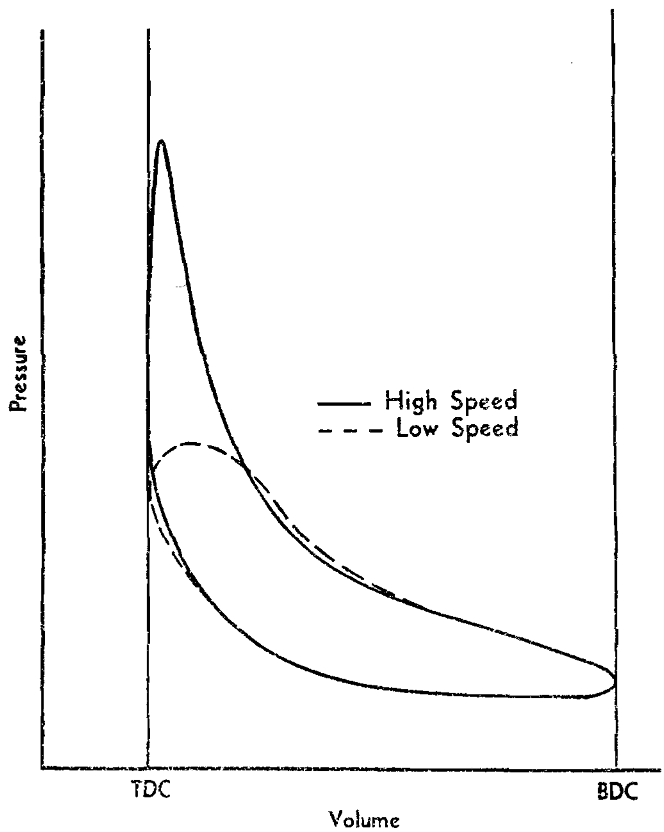

FIG. 13-2. Diagram illustrating the difference between a high speed and a low speed four-stroke cycle unsupercharged CI engine (not to scale).

addition of heat process. Only in some of the low speed engines does it manage to *approximate* the constant pressure process.

Figure 13-2 illustrates the relative tendencies of the low speed and high speed CI engine indicator diagrams.[2] The low speed engine tends toward a constant pressure combustion process, as indicated by the wider rounding of that portion of the diagram representing the combustion process. The high speed engine diagram, on the other hand, tends toward the SI engine diagram with its much more pointed peak.

(2) *Dissociation*—Dissociation apparently exists in the CI engine combustion process, but does not show as pronounced an effect as it does in the SI engine. This is due to the heterogeneous mixture existing in the combustion chamber, as well as the excess air present. Both of these factors, apparently, tend to reduce dissociation by lowering the overall temperature of the combustion gases.

(3) *Blow-down losses*—The four-stroke cycle CI engine has blow-down losses which are similar to those of the SI engine. The two-stroke cycle engine, however, exhibits an appreciably greater loss than the comparable four-stroke cycle engine. This is due to the shortened effective power stroke, which is necessary for the earlier release of the burned gas in order to obtain better scavenging. The blow-down loss, in this case, appears in the loss of area on a p-V diagram designated by a letter "*a*" in Fig. 11-2.

(4) *Pumping loss*—The pumping loss (as shown by the pumping loop) in a four-stroke cycle CI engine does exist, but it is rather small, because the air flow is not throttled for part load operation. In the two-stroke cycle CI engine, however, it is appreciable. The pumping loop, as such, does not appear on the p-V diagram since the cycle events are completed in only two strokes of the engine. The pumping loss in the two-stroke cycle CI engine is considered to consist of blow-down loss plus the power expended in driving the scavenging blower. The former loss appears on the indicator diagram, while the latter does not. It does reduce, however, the overall useful power output of the engine.

13-3. Air Consumption in the CI Engine. The CI engine is often referred to as a constant air supply engine, because the air supply is not throttled, and power output is varied by regulating the fuel supply alone. While this is true for a given engine speed, the air supply does vary with changing engine speeds. As the speed of the engine increases, the fluid friction of the entering air passing through the air manifold also increases. Also, the valve operation cannot produce optimum performance at both high and low speed operation. The combination of the

[2] For the sake of clarity, pumping loops of these indicator diagrams were omitted.

fluid friction and the limitations of valve performance for a wide speed range affect the amount of air flowing into the engine per stroke. As a result, this air quantity will vary with engine speed.

As in the SI engine, volumetric efficiency is an important factor entering into the air consumption of the CI engine. The greater the quantity of air consumed, the greater the *possibility* for the engine output to increase. In the CI engine, however, there is an additional criterion, namely, the *utilization* of the greatest possible portion of this inducted air in combination with the fuel during combustion. A good volumetric efficiency in a CI engine means little if the efficiently entrapped air is not utilized effectively. In the SI engine, the ihp developed is nearly proportional to the amount of air inducted, since the air and fuel are inducted as a mixture, and the proportion of each constituent is limited to a relatively narrow range. In the CI engine, however, the fuel and air are brought in separately. If the engine takes in a given amount of air, the greater the proportion of this air which is utilized in combining with the fuel (less excess air), the more fuel that can be added and the greater will be the power output. It is, therefore, desirable that the engine not only "swallow" the greatest possible amount of air through high volumetric efficiencies, but that it utilize, when operating at full power, the greatest possible amount of this captured air by decreasing the proportion of excess air to a minimum consistent with smoke limitations.

In two-stroke cycle engine operation the term *volumetric efficiency* is somewhat meaningless, because the engine requirements for air are different than those of the four-stroke cycle engine. In this case, in order to produce good scavenging, some of the air in the fresh charge is lost through the exhaust passageways. As a result, the definition of volumetric efficiency is usually not applied to this type of engine. In lieu thereof, engineers employ a wide variety of terms such as *scavenging efficiency*, *delivery ratio*, and others, which express generally the ability of the engine to utilize, through combustion, the maximum amount of air supplied by the blower.

Supercharging is used to produce a greater power output in CI engines as well as in SI engines (Articles 10-13 and 10-14), and is employed with both the two-stroke and four-stroke cycle types. Most installations utilize a positive displacement type of compressor coupled directly to the engine. The Roots blower (Fig. 10–18) is widely used, in either a two lobe (as shown), three lobe, or even a four lobe configuration. Turbine driven centrifugal superchargers are also used, and these utilize some of the kinetic energy remaining in the engine exhaust gas to produce the power necessary for compression.

13-4. Variables Affecting CI Engine Performance. Many of the variables which affect the performance of a SI engine exert a similar influence on the performance of a CI engine. The more important of these common to both engines are the engine speed, compression ratio, weight of inducted air, and frictional losses.

As in the SI engine, an increase in *engine speed* will increase the power output of the engine within its general operating limits. More power cycles per given time are thereby attained and a higher power output will result.

An increase in the engine *compression ratio* results in an increase in thermal efficiency, although this is limited by practical considerations. Too great an increase may result in excessive peak pressures in proportion to the gain in mep. The structure must be designed to withstand the excessive peak pressures, and results in a greater weight increase for the gain in mep with an attendant rise in the cost of production.

The higher the *weight of air inducted*, the greater the power output that may possibly be obtained in the CI engine, provided this air is

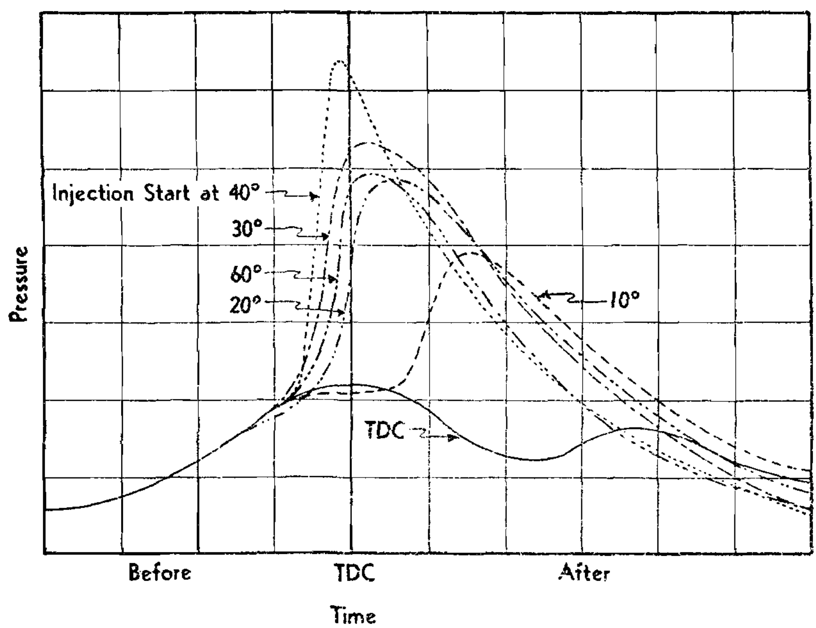

FIG. 13-3. Pressure-time diagram showing the effect of injection timing on maximum pressure and rate of pressure rise in a CI engine (courtesy of NACA—Rothrock, A. M., and Waldron, C. D., Tech. Rept. 525, 1935).

utilized efficiently, that is, that the excess air is kept to a minimum.

Again, as in the SI engine, proper engine "tuning" or adjusting of the operating variables to give the lowest poss'ble *heat losses* throughout the engine will tend to increase engine performance. Improper functioning of the fuel injection system, for instance, may cause greater heat losses and thereby increase specific fuel consumption.

In addition to the above, there are two major variables affecting engine performance that are characteristic only to the CI engine. These are the fuel injection timing and the timing of valves and ports in a two-stroke cycle engine.

(1) *Injection timing*—Injection timing should be adjusted to such a value as to produce maximum power output without engine roughness or undesirable smoke in the exhaust. Figure 13-3 shows the effect of the injection timing on the maximum pressure and the rate of pressure rise. It should be noted that when late injection occurs, say at TDC, maximum pressure decreases appreciably, because by the time combustion takes place the piston is on its way down on the power stroke. As a result, part of the pressure rise due to combustion is utilized in overcoming the expanding volume of the cylinder, thus decreasing the maximum pressure within the cylinder. Injection at 40 degrees before TDC, produces considerably higher maximum pressure as well as a greater rate of pressure rise, but at this early setting, engine knock is likely to result. If an early injection is used, say at 60 degrees crank angle before TDC, maximum attainable pressure again decreases. In this case, the burning becomes sluggish and seems to occur in two stages. During the first stage pressure rise it is considerably lower than during the second stage. Apparently, when too early an injection takes place, the fuel is not properly prepared for a rapid combustion. This fact may be attributed to the lower air temperature in the combustion chamber, at the time of fuel injection, and also to a lack of intermixing of fuel and air during the early part of the injection period.

(2) *Timing of valves and ports*—In general, the timing of the valves in a four-stroke CI engine is similar to that of a four-stroke SI engine. It is dictated by the desire to induce the greatest amount of air possible into the cylinder and to expel the burned gases with the smallest possible pressure differential between the cylinder and the atmosphere. In the two-stroke cycle CI engine, however, in addition to the above considerations, it is necessary to so time the operation of ports and valves as to minimize the loss of scavenging air through the exhaust port. Although fuel is not lost through this port along with the

scavenging air, as in a SI two-stroke cycle engine, the excessive loss of scavenging air is detrimental. Loss of scavenging air will demand a blower of greater capacity which, in turn, will require more power for operation, resulting in a reduction in overall useful engine power output.

13-5. Performance of Typical American CI Engines. This article will discuss a few typical American built CI engines which represent a cross-section of the many such successful models in present day use, and which may be encountered in automotive, industrial, and marine installations. Also, a presentation will be made of some of the characteristic engine performance curves. These curves "tell the story"

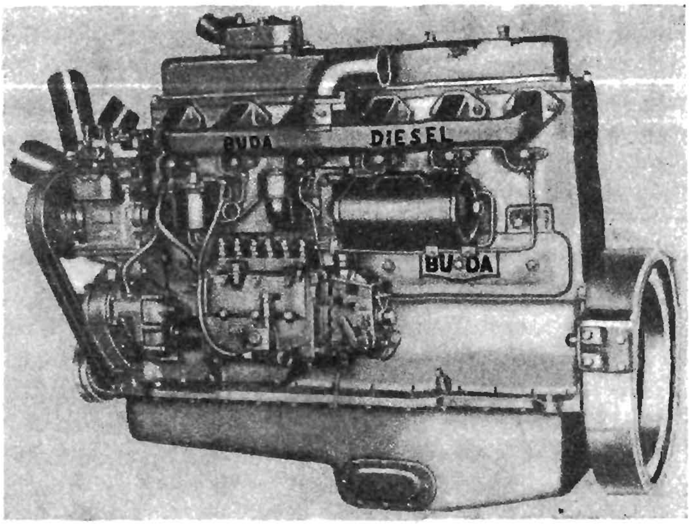

FIG. 13-4. Typical six cylinder, four-stroke cycle, automotive type CI engine of 5-1/4 inch bore and 6-1/2 inch stroke (courtesy of The Buda Company).

about the engine and its ability to perform a certain type of duty, as well as its cost of operation.

The engines discussed herein are divided into three fields of application, namely, (1) automotive, (2) industrial, and (3) marine. Some of these engines cited may be applied in more than one field and, therefore, the classification given should not be taken as the only one capable of their application.

(1) *Automotive type four-stroke cycle engines*—In spite of higher initial cost, the use of CI engines in the automotive field is growing in popu-

larity because of their ability to withstand more punishment in the field and because, at the present time, they are more economical to operate than SI engines. The reduction in fire hazard due to the type of fuel used in these engines is also a matter to be considered.

Figures 13-4 and 13-5 illustrate a typical six cylinder four-stroke cycle automotive type CI engine. The performance curves for this engine are shown in Figs. 13-6, 7 and 8. Typical data that is usually supplied the customer contains the torque, bhp, and bsfc curves, which are shown in Figure 13-6. Note that the engine is given three ratings for power produced with their maximum range of permissible speeds. Curves designated by the letter "A" represent maximum laboratory performance or the maximum power that the engine is capable of producing. For intermittent service, curves designated by "B," the

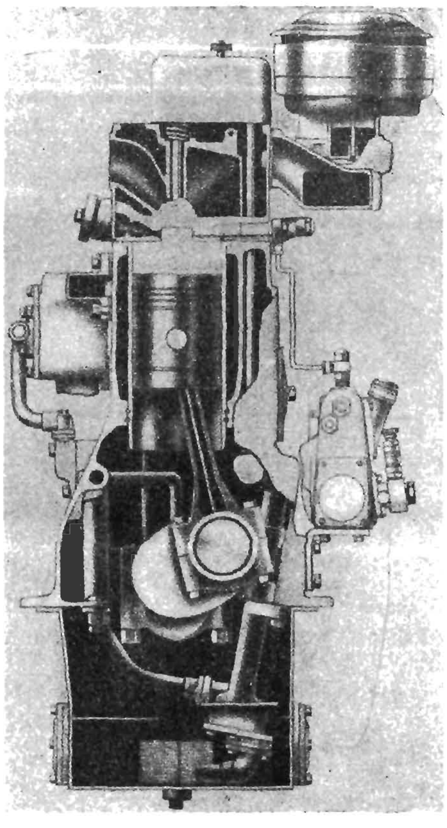

FIG. 13-5. Cross section of the typical six cylinder, four-stroke cycle, automotive type CI engine shown in Fig. 13-4 (courtesy of The Buda Company).

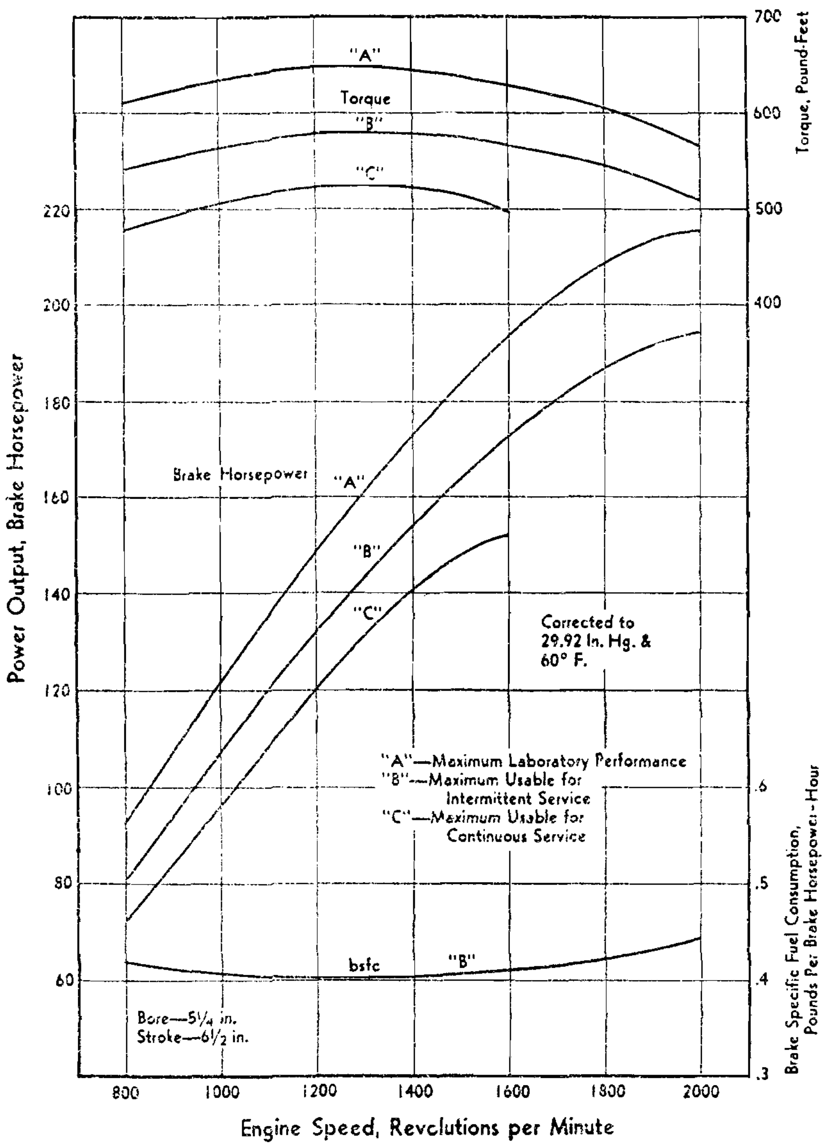

FIG. 13-6. Typical performance curves of a six cylinder, four-stroke cycle, automotive type CI engine at varying speed (data courtesy of The Buda Company).

engine rating is lowered, thus allowing a safety factor for the use of this engine. The maximum allowable operating speed, however, remains unchanged. Finally, curves designated by a letter "C" show the rating and the maximum allowable operating speed when the engine is employed for a continuous type of service. Also, note that the horsepower curves "A" and "B" do not reach the maximum possible output, because of the necessity of operating the engine at leaner A/F ratios than the chemically correct in order to avoid excessive smoke in the exhaust while the output of this engine as represented by curve "C" is limited by service considerations. Figure 13-7 shows the variation of bmep and mechanical efficiency with engine speed. It should be noted that there is a similarity in the shape of the torque curve shown in Fig. 13-6 and the bmep curve in Fig. 13-7. Both of these curves peak at about 1300 rpm. This similarity is due to the fact that mep and the torque

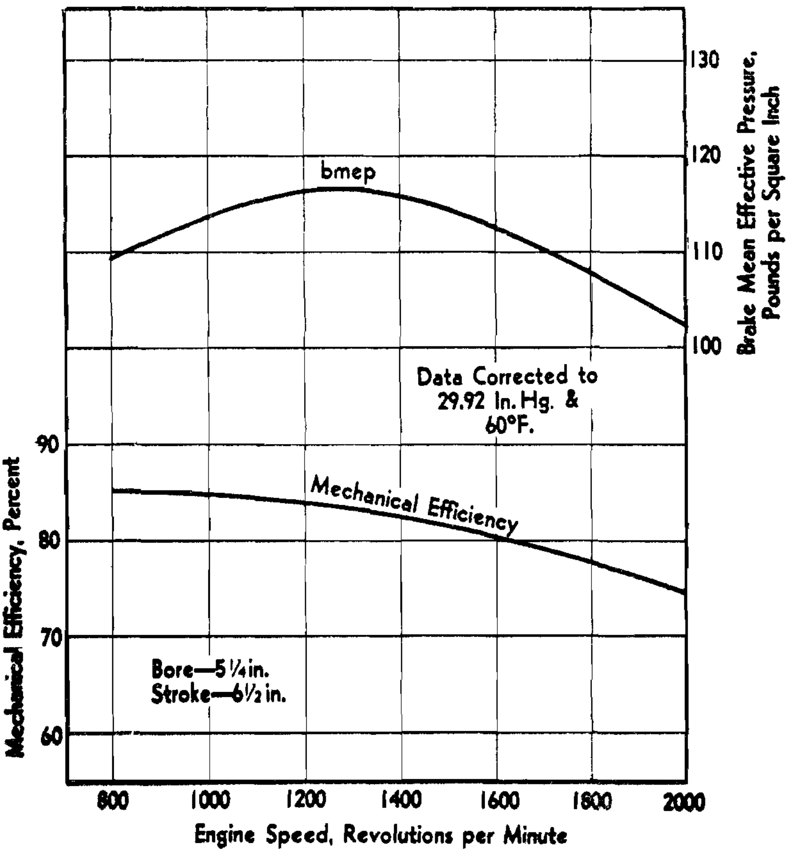

FIG. 13-7. Typical performance curves of a six cylinder, four-stroke cycle, automotive type CI engine at varying speed (data courtesy of The Buda Company).

are interrelated through combination of the two basic equations, 4-6 and 4-7. For instance, it is known that

$$\text{bhp} = \frac{p_b L A N C}{(33{,}000)(n)(12)},$$

and also that

$$\text{bhp} = \frac{2\pi R F N}{33{,}000}.$$

By equating these two formulae[3] and solving for p_b in terms of T ($T = RF$), it is possible to obtain the following expression:

$$\frac{p_b L A C}{n\ 12} = 2\pi R F$$

or

$$p_b = \left(\frac{2\pi\ n\ 12}{LAC}\right)T$$

then

$$p_b = KT$$

where

$$p_b = \text{bmep}$$
$$T = \text{torque}$$
$$K = \text{constant}$$

The above relationship indicates that bmep is equal to a constant times the torque. The value of torque, unlike that of bmep, may be easily obtained, at any given engine operating condition, through a dynamometer test. With this value of torque, the bmep may be readily calculated, then, by means of the relationship cited above.

It is of interest to note that the torque curves of a four-stroke cycle CI engine reach a peak at a certain speed as indicated in Fig. 13-6. The reason for this fact is that a naturally aspirated four-stroke CI engine will have one given speed at which the weight of air taken in per cycle will be maximum. Since the power produced will be proportional to the fuel added, it follows then that for a given A/F the maximum bmep and hence the torque will occur at the speed producing maximum air-charge per cycle. On the other hand, the torque curve for a two-stroke cycle CI engine will increase with the speed as shown in Fig. 13-21. In this case, a blower is used to supply scavenging air under pressure. As a result, increase in speed from 350 to 750 rpm changes the air-charge per cycle to a limited extent. The air consumed per unit

[3] Since p (mean effective pressure) is calculated on the basis of a single cylinder engine performance, the constant C (indicating number of cylinders in the engine) has been set to equal unity in the following calculations.

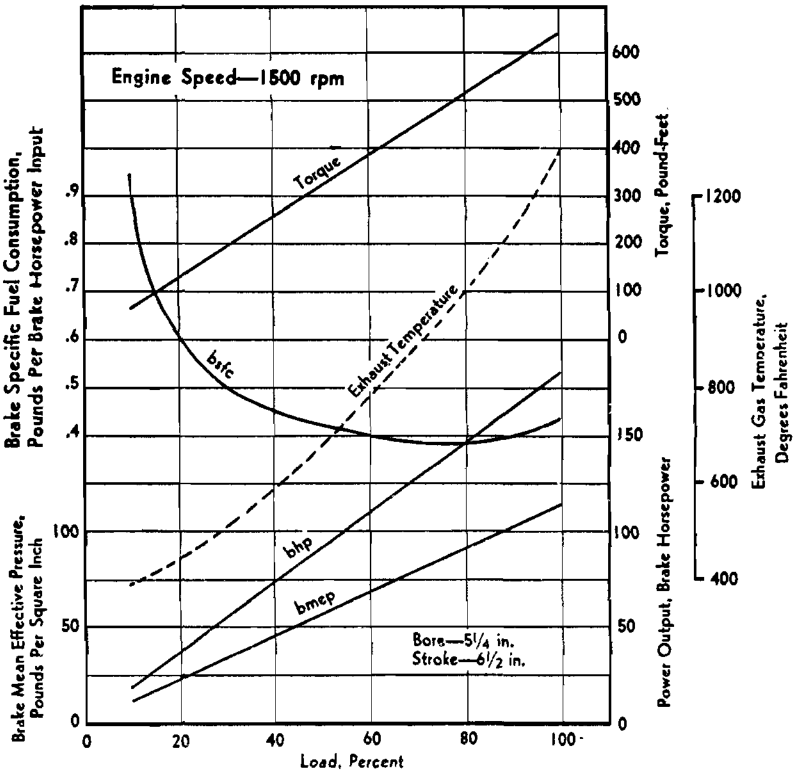

FIG. 13-8. Typical performance curves of a six cylinder, four-stroke cycle-automotive type CI engine at constant speed (data courtesy of The Buda Company).

time, therefore, increases with speed and hence, the hp. This explanation does not hold true for a high speed two-stroke cycle CI engine as indicated in Fig. 13-17. Apparently at higher speeds there is not enough time left for fuel to find available air within the combustion chamber, resulting in partial combustion of some of the fuel, thus in spite of the proper A/F ratio within the cylinder the torque will have a tendency to decrease with increasing engine speed.

Figure 13-8 represents the performance of the same engine at a constant speed of 1500 rpm and varying engine load. By inspection of the bsfc curve it may be observed that the most economical operating point, at this engine speed, is in the vicinity of 70 per cent load. By obtaining a family of such curves (one curve per given rpm), it is possible to determine the most economical load to be used at any speed throughout the engine speed range. The bsfc curve rises sharply with a

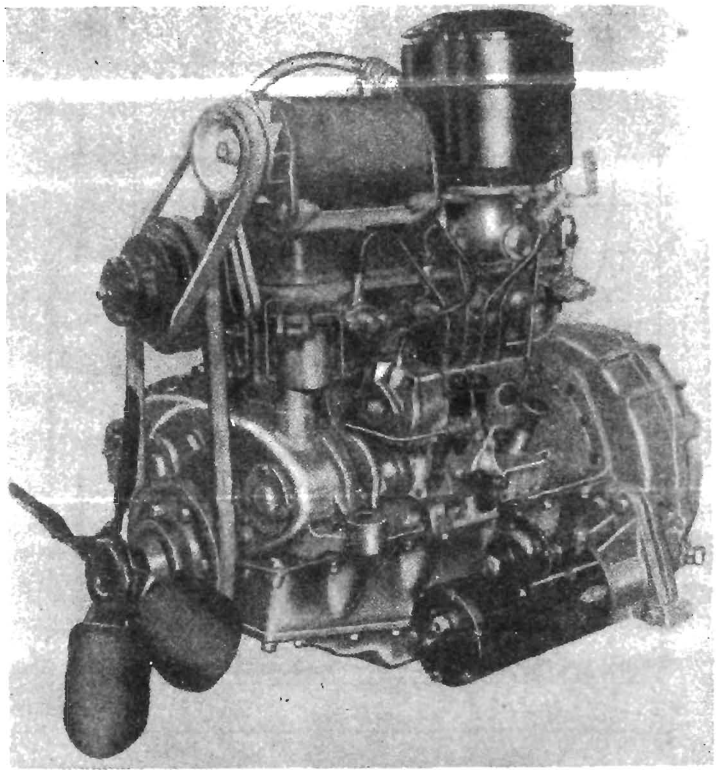

FIG. 13-9. Typical four cylinder, four-stroke cycle, automotive type CI engine (courtesy of Hercules Motors Corporation).

decrease in load since the per cent decrease in fuel required to operate the engine is less than the per cent decrease in bhp. The per cent heat loss at low loads and constant speed operation is greater than at the higher loads which also affects the shape of the bsfc curve and contributes toward its rise at low loads. At high load, the bsfc curve rises again. This is due to increase in friction (piston ring and bearing) brought about by increased cylinder pressures in the cylinder.

Exhaust gas temperature increases with the load, as shown in this figure, due to the increase in the amount of fuel used. Since the heat loss to the exhaust, on a percentage basis, is approximately constant throughout the load range, and since the mass of exhaust products is also approximately constant, the greater the quantity of fuel used, the more Btu's will be rejected to the exhaust, and hence, a higher exhaust temperature will result.

Another typical automotive type engine is shown in Fig. 13-9, and its performance curves in Fig. 13-10. These curves are of the type usually supplied to the customer upon request for the capabilities of the engine. Three bhp curves are shown for the three different displacements of the engines that are manufactured in this particular model.

The bsfc curve as shown in Fig. 13-10 rises at both ends of the speed range indicated. The rise at low speed is due to a greater per cent heat loss than at higher speeds and the upward trend of the bsfc curve at higher speeds is affected by the increase in fhp and the shortened time available for combustion. The decrease in time for combustion deprives some of the fuel particles of contacting the air in time to complete the combustion in a given period thus resulting in a waste of fuel.

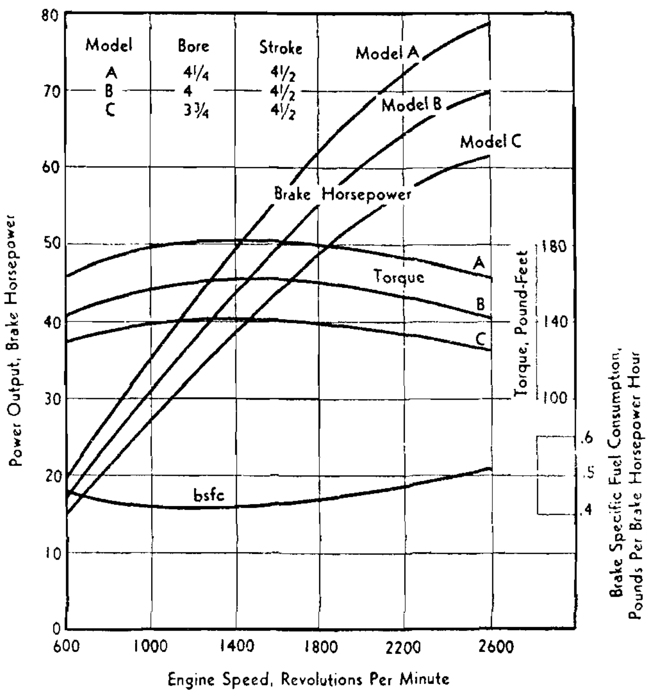

FIG. 13-10. Typical performance curves of a four cylinder, four-stroke cycle, automotive type CI engine at varying speed (data courtesy of Hercules Motors Company).

Figures 13-11 and 13-12 show typical p-t and p-V diagrams of an automotive four-stroke cycle CI engine operated at 1600 rpm. Figure

13-11 illustrates p-t diagrams obtained at full, 50 per cent, and no load engine operation, while Fig. 13-12 illustrates p-V diagrams for the same engine and under the same engine operating conditions. Note,

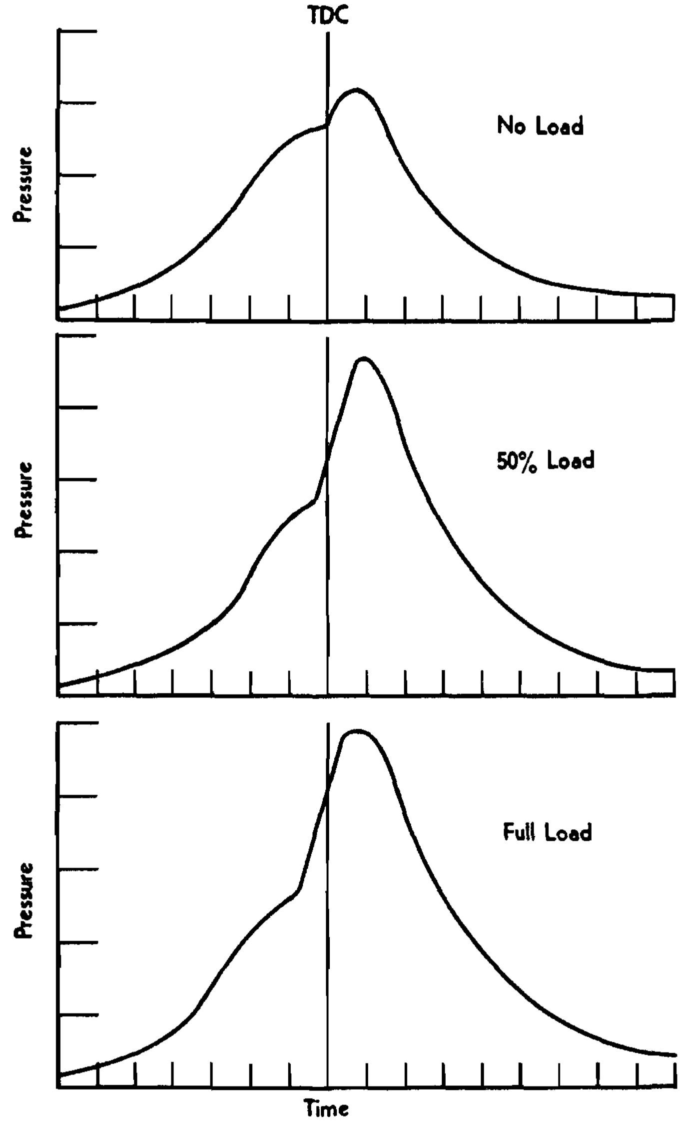

FIG. 13-11. Typical p-t diagrams for a four-stroke cycle CI engine showing full, half, and no load operation.

on the *p-t* diagrams, the rate of pressure rise as well as the maximum pressure developed as the load or the amount of fuel injected is varied from full load to no-load, or idling, conditions. Also, note the area of the *p-V* diagrams as the load is varied.

(2) *Industrial two-stroke type CI engine*—Two-stroke cycle CI engines, unlike their counterpart in the SI engine field, are of considerable importance in the CI engine domain. The power output of these en-

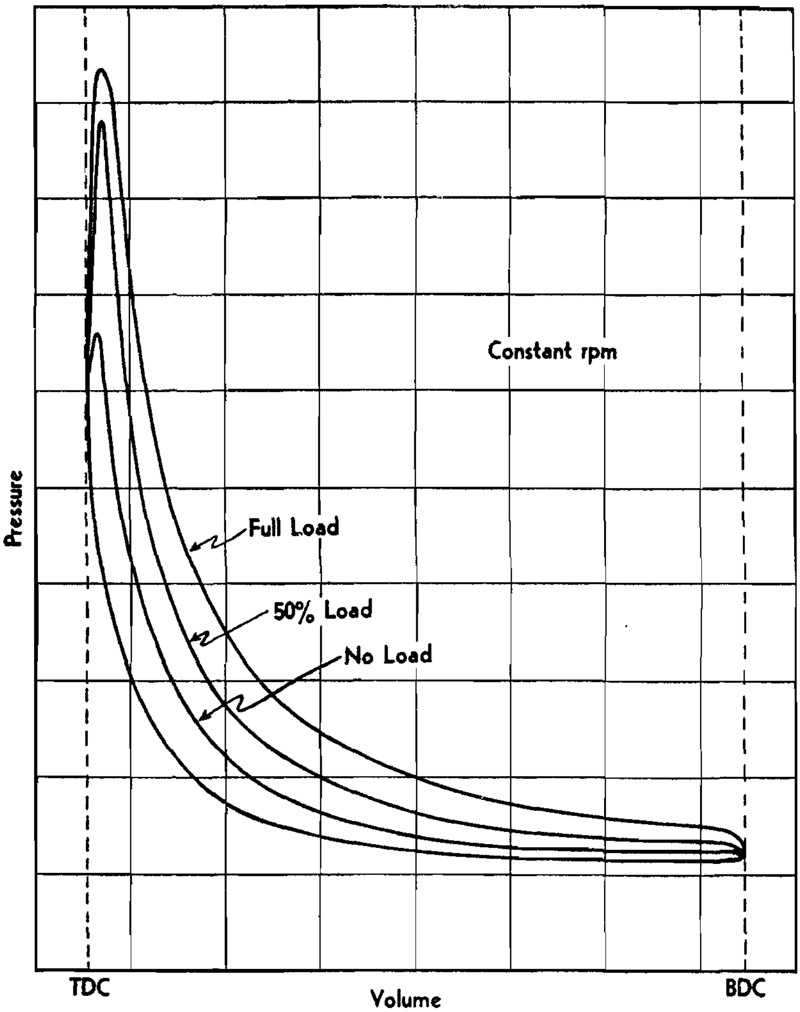

FIG. 13-12. Typical *p-V* diagrams for a four-stroke cycle CI engine showing full, half, and no load operation.

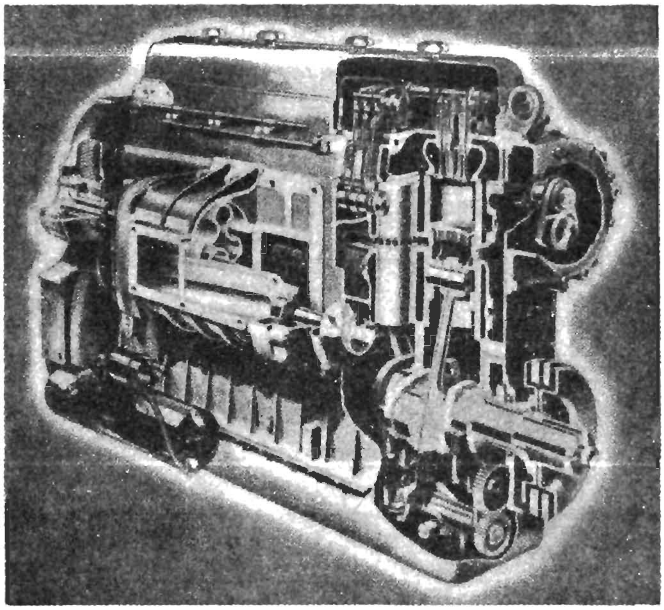

FIG. 13-13. Partial cutaway view of a typical six cylinder, two-stroke cycle CI engine (courtesy of Detroit Diesel Engine Division, General Motors Corporation).

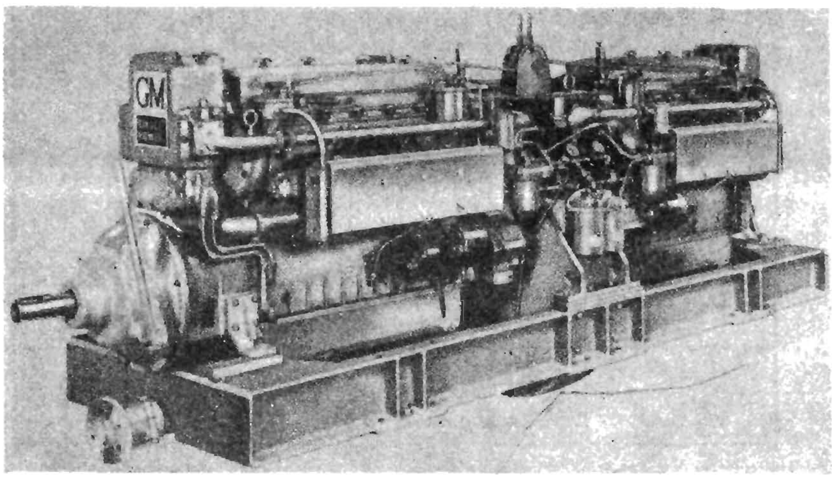

FIG. 13-14. Tandem arrangement of two basic engine units to form a greater capacity power plant—basic engine shown in Figure 13-13 (courtesy of Detroit Diesel Engine Division, General Motors Corporation)

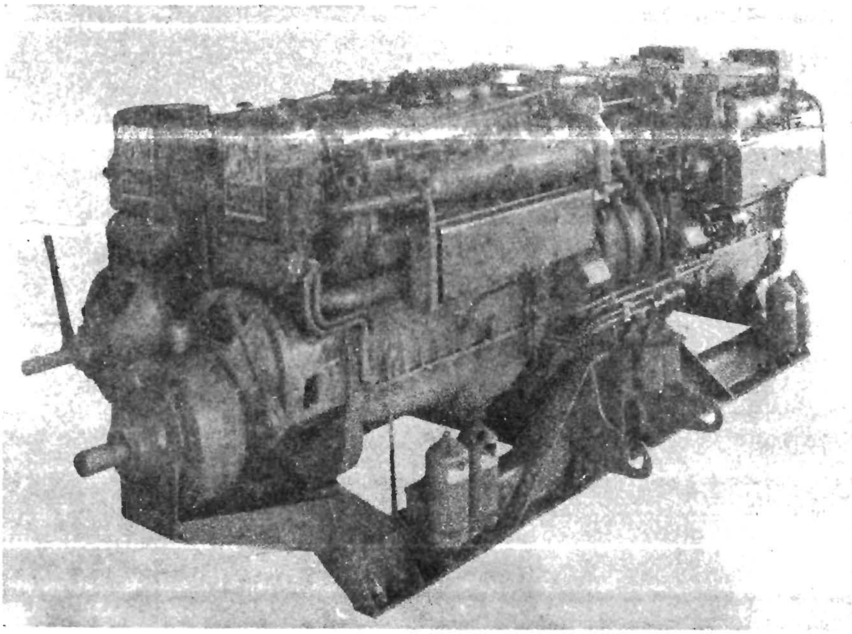

FIG. 13-15. Combination of four engine units to form a greater capacity power plant—basic engine shown in Figure 13-13 (courtesy of Detroit Diesel Engine Division, General Motors Corporation).

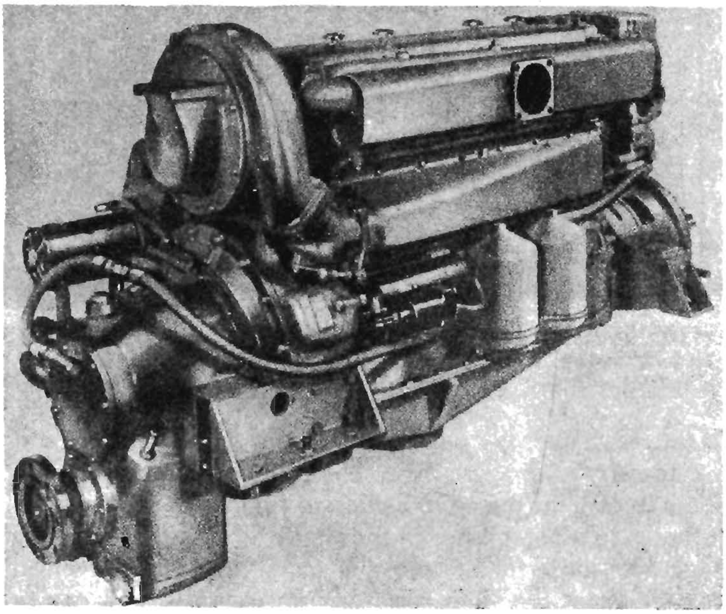

FIG. 13-16. Typical six cylinder, two-stroke cycle, marine type CI engine of 5 inch bore and 5.6 inch stroke (courtesy of Detroit Diesel Engine Division, General Motors Corporation).

gines ranges from about 10 bhp up to about 20,000 bhp per engine. At the present time, two-stroke cycle engines of power output above 10,000 bhp are usually made on special order and, hence, the cost of production of these engines is high. On the other hand, CI engines of

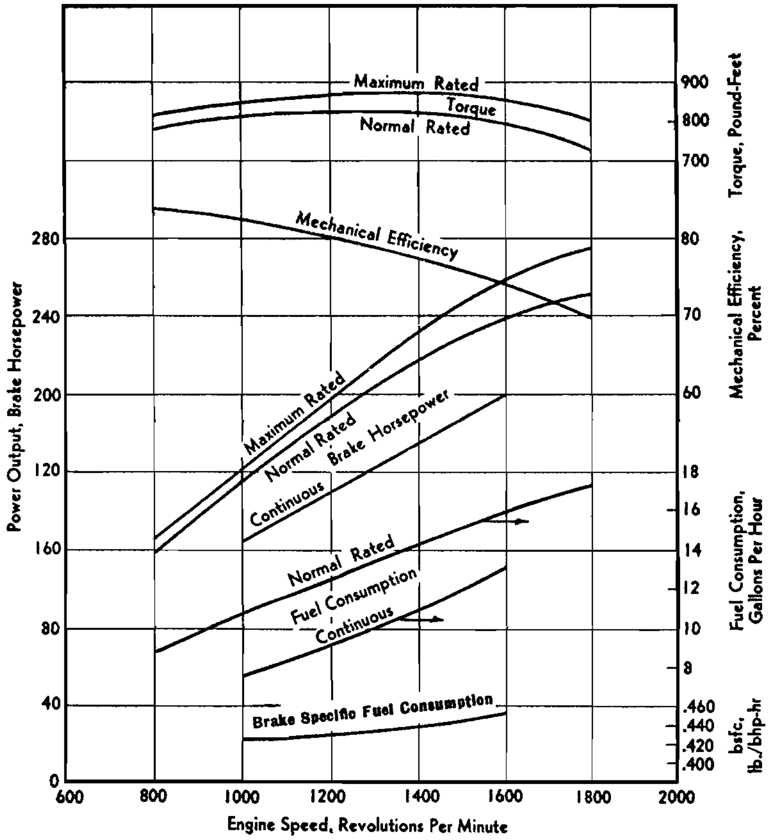

FIG. 13-17. Performance curves of the six cylinder, two-stroke cycle, marine type CI engine shown in Figure 13-16 (data courtesy of Detroit Diesel Engine Division, General Motors Corporation).

smaller capacity, up to about 3000 bhp per engine, are in a much greater demand.

Needless to say, the mass production of engines decreases their unit cost. In order to accomplish this end, however, it is necessary to use interchangeable parts for assembly of more than one size of engine There is a tendency, therefore, towards using a single standard size and shape of cylinder from which the engines may be built to deliver the re-

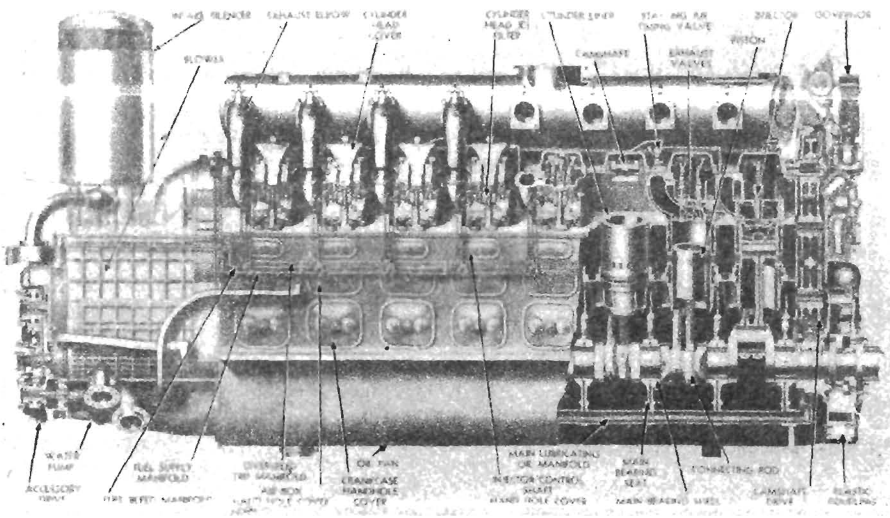

FIG. 13-18. Sixteen cylinder, two-stroke cycle, "V" type CI engine of 8-3/4 inch bore and 10-1/2 inch stroke (courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

quired power by the use of varying number of cylinders; or, from basic standard engine units, greater plant output may be secured by the use of engines in groups. An example of such a design is shown in Fig. 13-13. This partial cutaway view shows a six cylinder, two-stroke cycle engine. Engines of 2, 3 and 4 cylinders may be assembled having the same size cylinders and associated parts as that shown in Fig. 13-13. The arrangement of two engines comprising one power plant is shown in Fig. 13-14 while the arrangement of four engines is shown in Fig. 13-15.

Another example of a two-stroke cycle CI engine is shown in Fig. 13-16. This is a marine type engine with a 5 in. bore and a 5.6 in. stroke having an 18 to 1 compression ratio. Characteristic performance curves of this engine are shown in Fig. 13-17. These include three engine ratings, namely: continuous, normal rated, and maximum rated. The rating for continuous duty is such as to allow the engine 24 hours a day service. For normal intermittent duty, the engine rating specifies operation of not more than 2 hours at normal rated power followed by periods of the same duration with loads up to 75 per cent of the same rating. For maximum intermittent duty, the engine rating is defined as the operation of the engine at maximum rated power for periods not exceeding 30 minutes, followed by a period of operation of either one of the following two types before returning to maximum rated power:

(a) by loads up to the continuous rating for at least one hour's duration, or

(b) by loads not exceeding 50 per cent of maximum rated power for at least 30 minutes' duration.

In addition to the above, the curves showing bsfc, fuel consumption in gallons per hour, torque, and mechanical efficiency are also presented in Fig. 13-17.

(3) *Marine two-stroke cycle CI engines*—In the previous discussion

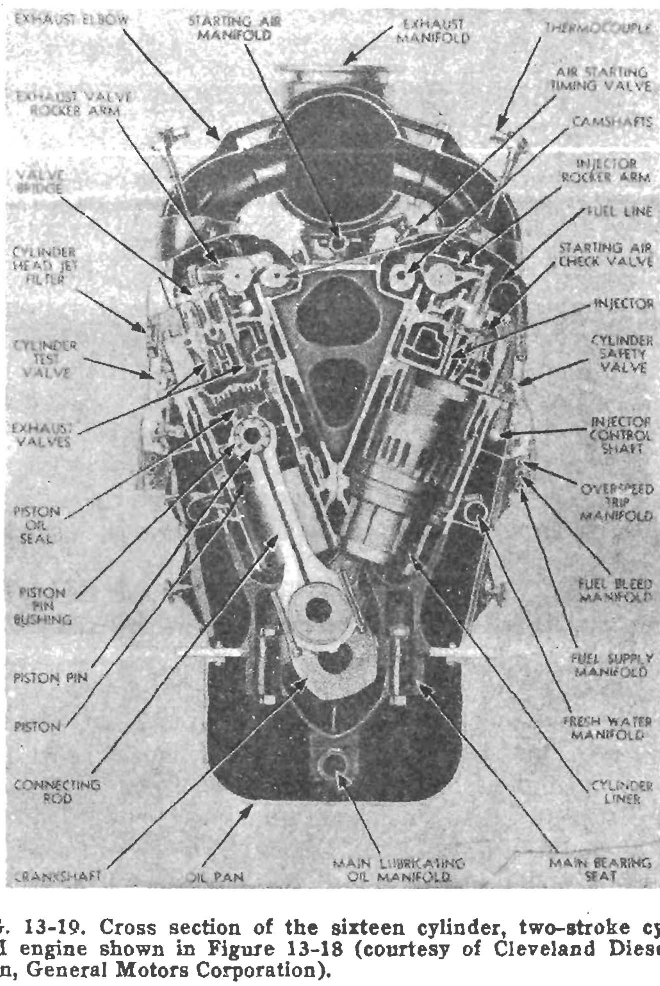

FIG. 13-19. Cross section of the sixteen cylinder, two-stroke cycle, "V" type CI engine shown in Figure 13-18 (courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

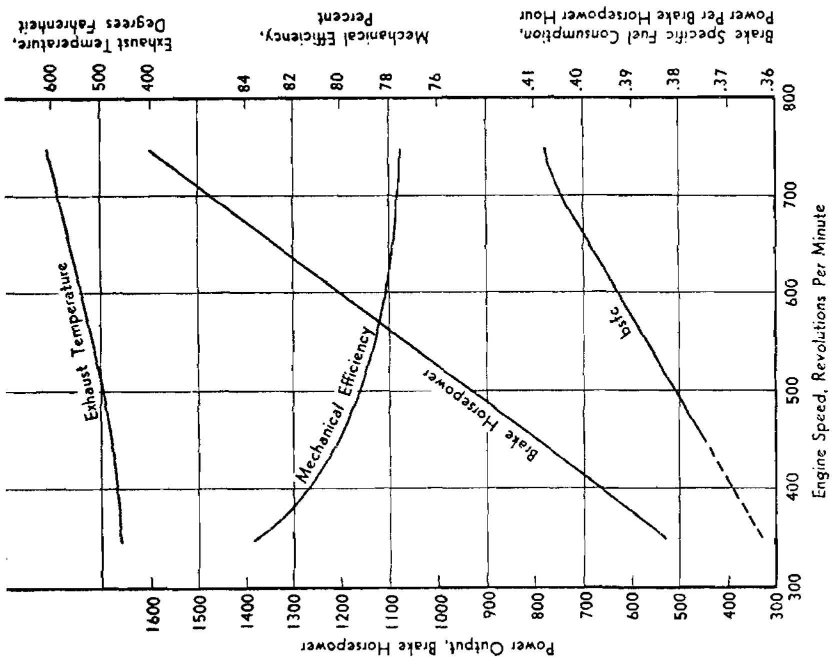

FIG. 13-20. Performance curves of the sixteen cylinder, two-stroke cycle, "V" type CI engine shown in Figures 13-18 and 13-19 (data courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

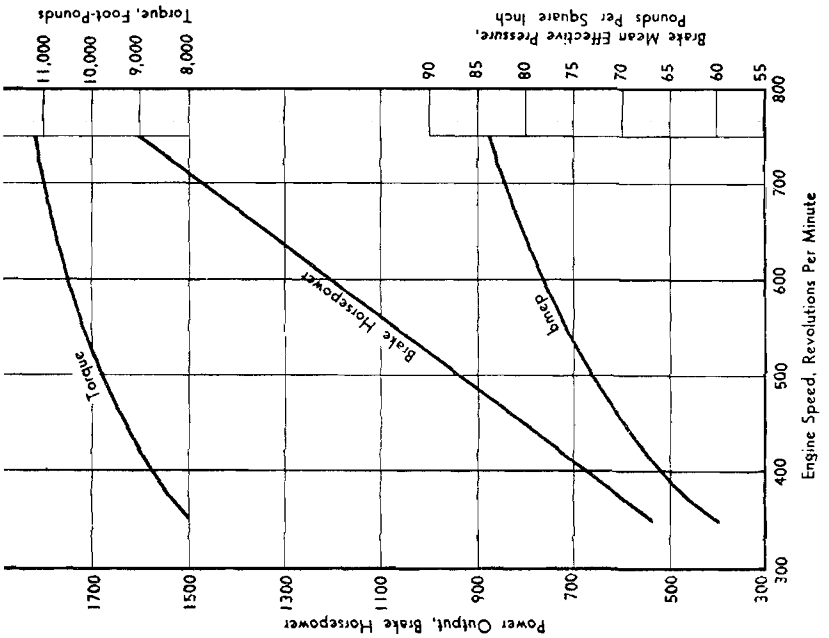

FIG. 13-21. Performance curves of the sixteen cylinder, two-stroke cycle, "V" type CI engine shown in Figures 13-18 and 13-19 (data courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

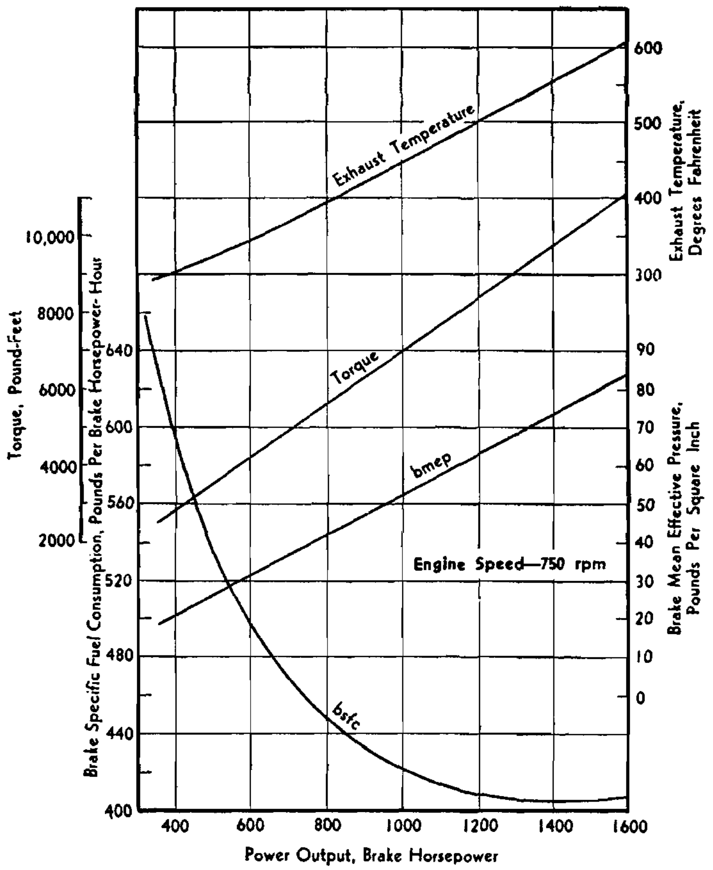

FIG. 13-22. Constant speed performance curves of the sixteen cylinder, two-stroke cycle, "V" type CI engine shown in Figures 13-18 and 13-19 (data courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

the performance of relatively small output CI engines was shown. Following will be the examination of performance of a relatively large sixteen cylinder engine depicted in Figs. 13-18 and 13-19. This engine is of the two-stroke cycle "V" type with $8\frac{3}{4}$ in. bore and $10\frac{1}{2}$ in. stroke and a 15.5 to 1 compression ratio. Performance curves of this engine are shown in Figs. 13-20, 21 and 22. Figure 13-20 presents variation of exhaust gas temperature, mechanical efficiency, bhp, and bsfc with speed. Note the effect of engine speed on exhaust gas temperature. As the speed increases the exhaust gas temperature also increases. This is due to the fact that bhp increases with speed, and increase in bhp will

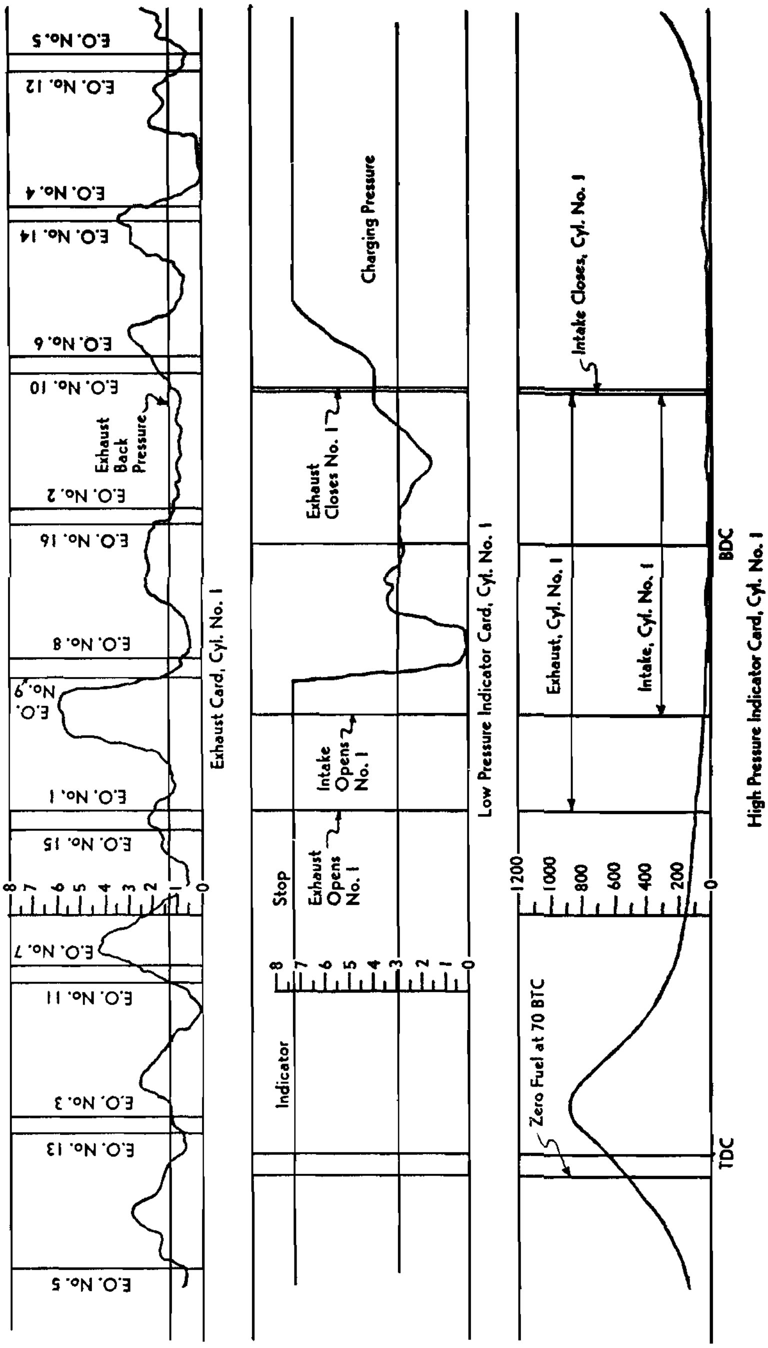

FIG. 13-23. Pressure-time diagrams of the sixteen cylinder, two-stroke cycle, "V" type CI engine shown in Figures 13-18 and 13-19 (data courtesy of Cleveland Diesel Engine Division, General Motors Corporation).

result in approximately proportionate heat loss to the exhaust. The brake horsepower curve never reaches its peak, because of the designed rating and the exhaust smoke limitations.

As shown in Fig. 13-20, bsfc increases with speed and wide open throttle engine operation. This fact is due to decreasing mechanical efficiency with increase in speed, thus indicating that the fhp increases at the greater rate than the bhp. As a result, more energy must be supplied to overcome the increased friction which does not show up in bhp. Consequently, bsfc increases as the speed increases. The bsfc curve for a four-stroke CI engine, Fig. 13-6, on the other hand, has a dip indicating a most economical speed of operation. The rise in the curve at the high speed end is due to decreased mechanical efficiency coupled with decreased time available for combustion, while the rise at the low speed end is attributed to the higher heat loss.

Figure 13-21 shows torque, bmep, and bhp. Again note the relationship between the torque and bmep curves at wide open throttle engine operation. Figure 13-22 illustrates the variation of bsfc, bmep, torque, and exhaust gas temperature under constant speed and varying load conditions.

Figure 13-23 shows *p-t* diagrams for the above engine. This figure depicts the high and low pressure indicator diagrams as well as the variation in the pressure existing in the exhaust pipe. Note the pressure variation within the cylinder during the scavenging process and compare it with the charging pressure. Also, note that while there is variation in the exhaust pressure of the individual cylinders, the overall back pressure remains constant.

Since the low pressure indicator operates at pressures considerably lower than those recorded by the high pressure indicator, it is imperative that the low pressure indicator is connected to the pressure source only when the pressure is within its range of operation, otherwise the instrument might be damaged. In this particular case, part of the low pressure indicator record appears as a straight line, Fig. 13-23, during the events that involve higher pressures than the indicator is designed for. This straight line is designated as "indicator stop"—meaning that it is not connected to the pressure source at that time. However, when the pressure in the cylinder is at the point where the intake and exhaust events are taking place and are within the range of the instrument, then the low pressure indicator is automatically connected and the record of pressures existing in the cylinder, at the time, are recorded.

COMPRESSION IGNITION ENGINE PERFORMANCE

References

Detailed information on the subject matter contained in this chapter may be located in the following references:

13-1. Spanogle, J. A., and Whitney, E. G., *A Description and Test Results of a Spark Ignition and a Compression Ignition Two-Stroke Cycle Engine*, NACA, Rept. 495, 1934.

13-2. Rothrock, A. M., and Waldron, C. D., *Some Effects of Injection Advance Angle, Engine-Jacket Temperature, and Speed on Combustion in a CI Engine*, NACA, Rept. 525, 1935.

13-3. Moore, Charles S., and Collins, John H., Jr., *Friction of Compression-Ignition Engines*, NACA, TN 577, August 1936.

13-4. Rogowski, A. R., Bouchard, C. L., and Taylor, C. Fayette, *The Effect of Piston-Head Shape, Cylinder-Head Shape, and Exhaust Restriction, on the Performance of a Piston-Ported Two-Stroke Cylinder*, NACA, TN 756, March 1940.

13-5. Lester C. Lichty, *Internal Combustion Engines*, McGraw-Hill Book Company, Inc.

EXERCISES

13-1. In general, how does the heat balance of the CI engine differ from that of the SI engine?

13-2. When discussing the performance of several engines, can the heat balance data obtained from one of the engines be applied to any other? Why?

13-3. In what manner does the actual CI engine cycle differ from the theoretical air cycle?

13-4. Is the shape of the *p-V* diagram of a slow speed CI engine similar to that of a high speed engine? Why?

13-5. Why does the two-stroke cycle CI engine have greater blow-down loss than that of the four-stroke cycle engine?

13-6. How serious is the pumping loss, at part throttle, of the four-stroke CI engine as compared to that of the SI engine? Is there a pumping loop for a two-stroke cycle engine? Why?

13-7. How important is the air consumption in a CI engine? Does the air consumption vary with the engine speed? Load?

13-8. In general, is the excess air in the CI engine desirable? Why? What effect does it have on indicated thermal efficiency? Mep?

13-9. Is the term "volumetric efficiency," as such, used in discussing two-stroke CI engines? Why?

13-10. Can supercharging be used profitably in connection with two-stroke cycle CI engines? Why?

13-11. Why is injection timing important in CI engine operation?

13-12. Is there any fuel lost during the scavenging process of a two-stroke cycle CI engine? Why? Why is the port timing so important in a two-stroke cycle CI engine?

13-13. What are the usual performance curves supplied by the manufacturer in his advertising literature?

13-14. Do some of the CI engines have more than one power rating? If so, what are they?

13-15. Why do torque and bmep curves have similar shapes? How can one calculate bmep without the aid of an indicator card?

13-16. The characteristic performance curves are usually based on two types of independent variables (plotted as abscissa). What are they?

13-17. How would one go about determining the most economical operating load for various engine speeds?

13-18. How does the mechanical efficiency vary with engine speed? Why?

CHAPTER XIV

COMPARISON OF SI AND CI ENGINES

In determining whether the SI or the CI engine is best suited for a particular application, one must know both the requirements of the job to be performed, and the relative capabilities of the engine types. Considerable discussion has been presented, in the previous chapters, on the operation of each of these types and on their performances. The subject matter of this chapter will be concerned with the general relative capabilities and limitations of the SI and the CI engine.

This comparison must be considered as being on a *general* basis, as there are many specific exceptions to the broad observations to be presented.

14-1. Differences in Operating Variables. The SI engine, as the name implies, embodies an electrical ignition system which ignites a combustible charge in the cylinder. This nearly homogeneous air-fuel mixture is usually proportioned by means of a carburetor, and its quantity is regulated by a throttle which provides the necessary speed and load control of the engine.

The CI engine does not utilize a spark for ignition of the charge. A nearly constant supply of air is drawn into the combustion chamber and is compressed by the piston. The fuel is injected directly into the chamber, and the compression temperature is utilized to ignite a rather heterogeneous air-fuel mixture. The speed and load control of this engine is secured through the regulation of the quantity of fuel injected.

Because of the basically different types of operation, there are certain diversities between the operating variables of the two types, as follows:

(1) *Compression ratio*—Because of the limitations imposed by detonation, the compression ratio in the standard SI engine is restricted by the anti-knock quality of the fuel. Present day SI engines operate in a range of compression ratios from about 5 to 10.5. Detonation in the CI engine, on the other hand, does not restrict this type of engine to low compression ratios. In fact, increasing the compression ratio tends to reduce the possibility of detonation. Compression ignition engine compression ratios presently run in the range of about 12 to 20. The higher ratios employed in the CI engine result in higher thermal efficiencies, hence a greater power output for the quantity of fuel consumed.

(2) *Operating pressures*—Higher pressures are found in the CI engine as a result of the higher compression ratios utilized. In fact, they are necessary to provide a compression pressure sufficient to heat the air in the cylinder to a temperature greater than the ignition temperature of the fuel. Compression pressures in the CI engine run in the range of 400 to 700 pounds per square inch, in comparison to a value of 100 to 200 for the SI engine. These higher pressures require a stronger engine structure, and CI engines are, therefore, sturdier and heavier.

(3) *Operating speed*—The brake horsepower of a given engine is a function of engine speed. The maximum brake horsepower is ordinarily identified with the speed at which this output occurs. This speed will vary with different engines. As a general rule, SI engine speeds will be higher than those of comparable CI engines. The SI engine is usually, therefore, considered as a high speed engine. For example, present day passenger car engine speeds for maximum output occur in the range of 3000 to 5000 rpm. Many CI engines operate at speeds considerably below those of SI engines, in the 400 to 1200 rpm range. In recent years, however, the high speed diesel (1200 to 3000 rpm) has seen rapid development and is becoming increasingly popular.

(4) *Distribution of fuel to the cylinders*—In the multi-cylinder SI engine, considerable difficulty is encountered in distributing the mixture to the various cylinders in such a manner that each cylinder will receive the same desirable air-fuel ratio. The combined effect of the liquid droplets adhering to the intake manifold walls and in "suspension" in the mixture passing through the manifold, causes the A/F ratio reaching the various cylinders to differ. The mixture actually reaching some of the cylinders is, therefore, not that conducive to optimum performance. The injection system of the CI engine, on the other hand, provides generally excellent distribution of fuel to the cylinders.

(5) *Supercharging*—Both the SI and the CI engine may be supercharged. The amount of supercharging that can be used with the SI engine is limited by the detonation of the fuel. In the CI engine, on the other hand, supercharging tends to prevent detonation, and is limited by the amount of power required to drive the supercharger. In either type, it is used to increase the power output of the engine, and may be used (as in aircraft SI engines) to maintain a higher power output at altitudes above sea level.

(6) *Exhaust gas temperature*—The thermal efficiency of the CI engine runs higher than that of the SI engine, because of the higher compression ratios used. The exhaust gases in the CI engine are, therefore, returned to the atmosphere at lower temperatures, since a greater portion

of the available energy goes into useful work and the losses to the exhaust are lower. Also, the excess air in the CI engine has a pronounced effect on lowering the exhaust temperatures, particularly in the 2-stroke cycle engine.

(7) *Starting*—The CI engine is inherently more difficult to start than the SI engine, primarily due to the greater cranking effort required to overcome the higher compression ratios used. This condition is accentuated in cold weather, since the lower initial temperature of the air and the greater heat transfer through the chamber walls make it more difficult to raise the compression temperatures to values sufficient to ignite the injected fuel.

14-2. Comparison of Performance Characteristics. On the basis of the following performance characteristics, each of the engine types has certain advantages and disadvantages:

(1) *Power output per unit weight*—Because of the higher compression ratios and the higher pressures involved, CI engines require stronger engine parts and are inherently heavier. They generally weigh about 5 to 20 pounds or more per horsepower delivered, whereas the SI engine usually weighs about 1 to 7 pounds per horsepower.

(2) *Power output per unit piston displacement*—This factor presents a rough comparison of the space requirements of the engine types. Most high speed CI engines deliver approximately 0.3 hp for each cubic inch of piston displacement, whereas this ratio for SI automobile engines is around 0.5, and some SI aircraft engines reach 0.9 or more. This indicates the tendency of the SI engine to occupy less space for the same power output.

(3) *Acceleration*—The CI engine inherently produces the best acceleration, since the injection system offers direct control of the quantity of fuel injected, and a rapid and positive means of changing this quantity. A change in the quantity of mixture supplied to the SI engine, on the other hand, is accomplished through a relatively indirect control depending upon throttle opening and the resultant velocity of the air through the carburetor venturi. The use of accelerating jets and pumps in the SI engine, however, effectively overcomes most of this deficiency.

(4) *Reliability*—Both SI and CI engines have been developed to the extent that they now are reliable power plants. The CI engine has greater stamina and can generally stand rougher duty. This is not only because it is built stronger, but also because smoke limitations cause it to be rated considerably below its maximum possible power output. This, in effect, further increases the extra strength factor. Operational difficulties of the CI engine are usually concerned with such components

as the costly injection system or the complicated speed governor, whereas, in the SI engine, the ignition system or the carburetor are inclined to create many of the difficulties.

(5) *Fuel economy*—Probably the greatest advantages accruing to the CI engine are better fuel economy ***at both full and part load***, as well as the ***safety of the fuel that can be utilized***. Because of their importance, these factors will be discussed in the following article.

14-3. Comparison of Fuel Economy and Fuels Used. The CI engine is more economical than the SI engine from the standpoint of fuel consumption. Since the CI engine operates at higher compression ratios, the expansion ratios and the thermal efficiencies are higher, resulting in a lower specific fuel consumption, or a greater power output per pound of fuel per hour. At full throttle, the gain in specific fuel consumption for the CI engine over the SI engine ranges around 10 to 25 per cent. ***At part throttle and at idling, the CI engine presents an even more favorable relative fuel consumption than the SI engine.*** Under these operating conditions, SI engine specific fuel consumption rises rapidly, due primarily to the throttling of the mixture. In the CI engine, the specific fuel consumption rises less rapidly, since the fuel is injected directly, and no throttling exists. This factor becomes particularly important in engines driving vehicles, fishing craft, and Navy small boats, since they operate to a considerable extent at part load.

In addition to gaining through the lower fuel consumption, the CI engine utilizes fuels which presently cost less than SI engine fuels, thereby further reducing the relative operating fuel expense. Although CI engine fuels usually contain fewer Btu's per pound than SI engine fuels, they are denser and contain more Btu's per gallon, the usual basis upon which they are sold. From a fuel economy point of view, then, the CI engine is less costly to run than the SI engine.

Another, and very important, consideration advantageous to the use of the CI engine is the ***reduced fuel fire hazard***. Fuels that are used today in SI engines are, of necessity, relatively volatile hydrocarbons and are a great fire hazard when not carefully handled. CI engine fuels on the other hand, need not be as volatile and the fire hazard is greatly reduced since they are less likely to form an explosive mixture when spilled. This factor is of extreme importance with marine or other confined installations.

14-4. Comparison of Other Operational Costs. In addition to fuel economy considerations, which are definitely in favor of the CI engine, there are other operational costs which should be considered in the selection of a particular engine type. Some of the more important of these are as follows:

(1) *Initial cost of the engine*—Due to the fact that the SI engine is not as sturdily built as the CI engine, and since it has a less costly fuel supply system, the SI engine costs less initially than a comparable CI engine. This factor is accentuated by the relatively larger production of the SI engine at the present time, particularly in the automotive field. CI engines are rising in popularity, and increased production should narrow this gap in initial cost, although the CI engine will probably always cost more to purchase than the SI engine.

Some of this difference should be made up by the CI engine through longer life, since the CI engine is limited by smoke from operating near its peak output, and is rated considerably below its possible maximum power. Consequently, this added safety factor should result in less wear and tear on engine parts and a generally longer life.

(2) *Maintenance costs*—The maintenance costs of the two engine types are generally about the same, with the CI engine costs possibly slightly higher. Replacement parts for the CI injection system are costly due to the close tolerances involved. Furthermore, at the present time, skilled technicians for the repair of CI engines are generally in shorter supply than are SI technicians, and this tends to raise the repair cost of the CI engine.

14-5. Comparison of SI and CI Engine Applications. As seen from the above discussion, both the SI and CI engines have certain advantages and disadvantages. The determination of the type selected to perform a given job must be based upon these relative advantages and disadvantages in relation to the requirements of the task to be performed.

SI engines offer the following advantages:

(1) Low original cost
(2) Low weight for power output
(3) Smaller size for power output
(4) Low starting, or cranking, effort required
(5) Less objectionable exhaust gas odor.

Consequently, they are widely used for mobile power, particularly in automobiles and airplanes. They are also used to a considerable extent in trucks, although the CI engine is rapidly gaining favor in this field. They are also widely used as the power plant in pleasure boats, because of the lower initial cost and in spite of the added fire hazard.

In this field, the two stroke cycle SI engine is used for outboard motors. Some SI engines are used for stationary work such as in motor-generators sets, air compressors, and water pumps, where the operation is conducted primarily at constant speed.

The CI engine, which has developed rapidly in recent years, presents the following advantages:

(1) Low specific fuel consumption at both full load and *part load conditions.*
(2) Reduced fire hazard
(3) Utilizes less expensive fuels
(4) Better suited to two-stroke cycle operation
(5) Longer operating life.

In general, the CI engine finds wide use in those fields where it is operated continuously, or nearly so. Through such utilization, the gain through lower fuel consumption assists in counteracting the disadvantages of higher initial cost. Compression ignition engines power most heavy duty equipment such as bulldozers, tractors, and earth moving machinery. They are widely used in large trucks, and are gaining in popularity as improvements are made through increased recent development. Also, they are employed to a considerable extent in stationary plants, and because of the reduced fire hazard, they are particularly suitable for confined installations and marine use.

14-6. Comparison of Two and Four Stroke Cycle CI Engines. Since the 2-stroke cycle CI engine has one power stroke for every revolution and is theoretically capable of twice the power output of a comparable 4-stroke cycle engine, the question often arises as to why 4-stroke cycle CI engines are built. The answer lies in many practical considerations and limitations, some of the more important of which will be discussed briefly.

Particularly at high speeds, the exhaust valve of the 2-stroke cycle engine must open relatively earlier than that of the 4-stroke cycle engine if reasonable scavenging of the cylinder is to be obtained. This reduces the expansion ratio of the 2-stroke cycle and results in lower utilization of the energy in the burned gases, resulting in poorer fuel economy. Also, the 2-stroke cycle engine must utilize a blower if reasonable scavenging is to be realized. The power required to drive this blower must be charged to the engine and reduces the net output accordingly.

The separate exhaust and intake strokes of the 4-stroke cycle provide greater opportunity for the dissipation of heat from critical parts such as the piston, and essentially permit the 4-stroke cycle engine to run at higher speeds than the 2-stroke cycle engine.

The result of these, and other less influencing practical considerations, is that the 4-stroke cycle CI engine is widely used despite the preponderance of theoretical advantages for the 2-stroke cycle engine. In general, the large capacity, slow speed diesels are predominantly of the 2-stroke cycle type, while the high speed, lower output CI engines tend toward the 4-stroke cycle.

EXERCISES

14-1. What limits the maximum allowable compression ratio in SI engines? CI engines?

14-2. In general, how do the compression pressures in CI and SI engines compare?

14-3. What are the general operating speed ranges for SI engines? For low speed and high speed CI engines?

14-4. Is proper distribution of fuel to the cylinders generally more difficult to achieve in a SI or CI engine? Why?

14-5. Under the same loading conditions, would you expect the exhaust gas temperature to be lower in a SI or a CI engine? Why?

14-6. Would you expect the CI or the SI engine to be generally more difficult to start? Why?

14-7. Which type of engine, CI or SI, gives the better fuel economy at full load? At part load? Why?

14-8. Why are CI engines considered to be less of a fire hazard?

14-9. In general, would a CI or a SI engine be more advantageous from the standpoint of:

(a) Original cost?
(b) Size and weight for power output?
(c) Starting effort required?
(d) Exhaust gas odor?
(e) Fuel economy?
(f) Fire hazard?
(g) Suitability for 2-stroke cycle operation?

14-10. What are some of the practical considerations that cause the 4-stroke cycle CI engine to be widely used despite the theoretical advantages enjoyed by the 2-stroke cycle CI engine?

CHAPTER XV

LUBRICATION

The maintenance of proper lubrication of all moving parts is an important problem in the operation of an internal combustion engine. The functions of lubrication are to decrease the power required to overcome friction and to reduce wear between the rubbing and bearing surfaces, thereby increasing the power output and the engine service life. If the lubrication system should fail to operate properly, resulting in the breakdown of the lubricating films, the engine will be subject to seizure and serious damage.

A secondary function of the lubricant is to act as a coolant, carrying heat away from the bearings, cylinders, and pistons. Also, the lubricating film on the cylinder wall must act as a seal to prevent the gases of combustion from blowing by the piston rings and entering the crankcase. Thus, the effectiveness of engine lubrication plays an important role in determining the service life and the performance characteristics of an engine.

The basic problems associated with the proper lubrication of the various types of bearings encountered in an internal combustion engine will be discussed in general in this chapter, along with the properties of lubricating oils and the effect of engine operation on these properties. In addition, the various types of lubricating systems and their components will be discussed.

15-1. Mechanism of Lubrication. The reciprocating internal combustion engine has a vast number of moving parts. Without an adequate film of oil between the surfaces of the reciprocating, oscillating, and rotating metal parts, the power required to overcome the frictional resistance and the wear on the parts would be prohibitively high.

The lubrication process is illustrated in Fig. 15-1 for a slipper bearing that has inclined surfaces. If one surface is moving and is inclined to the other, the viscous drag of the oil tends to draw lubricant into the space between the surfaces and build up a wedge. This develops an oil-film pressure that can sustain a load. If the two surfaces were parallel or if they didn't have relative motion, the oil-film pressure could not be developed and a load could not be supported by the lubricant.

A parameter utilized as a comparison of the effectiveness of a lubricating film is called the friction coefficient. The *friction coefficient* may be defined as the dimensionless ratio of the resistance due to the friction in the direction of the motion to the supported load normal to the line of motion. The coefficient varies depending on the type of lubricant and the type of film, i.e., thick or thin.

If the sliding surfaces of Fig. 15-1 are completely separated by a film of oil, there is no metal contact and the wear on the surfaces is a minimum. This is termed *thick-film* (perfect or complete-film) lubrication. The phenomenon involved with this type of film is a viscous flow process where the frictional resistance is due principally to the shearing of the lubricant. The friction coefficient for a thick-film will generally range from 0.002 to 0.012 and the load carrying capacity of the bearing may be as high as 18,000 psi.

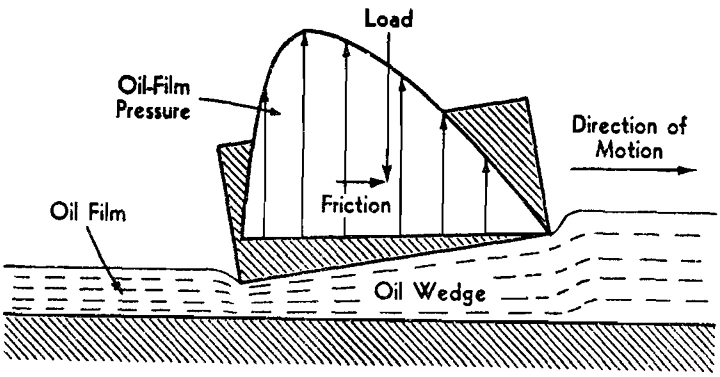

FIG. 15-1. Slipper bearing with inclined surfaces showing oilwedge and oil-film pressure.

An increase in the load or a decrease in the relative surface velocity or the viscosity of the lubricant decreases the film thickness. When the film becomes so thin that surface irregularities break through and come in contact with one another, there will be metal contact, wear, and possible seizure. This type of lubrication is called *thin-film* (imperfect or partial-film) lubrication. With thin-film lubrication the coefficient of friction is greater than that of the thick-film and generally ranges from 0.012 to 0.10, while the load capacity is decreased to around 300 psi. The frictional resistance depends on the properties of the lubricant, the roughness of the surfaces, the materials of the bearing surfaces, and the quantity of oil supplied. In the thin-film region, the *oiliness* characteristic of the lubricant is a major factor in determining the friction coefficient or the frictional characteristics of the oil. Among oils that have the same viscosity at the same test conditions, the greater the oiliness the lower the friction coefficient. This characteristic is most pronounced in organic oils, i.e., animal and vegetable oils.

Dry or unlubricated surfaces have a very high coefficient of friction, 0.10 and up. The load bearing capacity of this type is in the order of a few pounds per sq in.

TABLE 15-1

Motion	Surface
a. Sliding contact	a.
1. Rotating	1. Journal bearings—crankpins, crankshafts, camshafts, valve mechanisms, etc.
2. Oscillating	2. Journal bearings—piston pins, knuckle pins, rocker arm bearings, etc.
3. Reciprocating	3. Slipper bearings—pistons, piston rings, valve stems, cross heads, etc.
b. Meshing contact	b. Worm, bevel, spur, and helical gears.
c. Rolling contact	c. Ball, roller, and needle bearings.

In reciprocating internal combustion engines, oil films must be established and maintained under extreme operating conditions (temperature, speed, load, and pressure) and in a variety of different types of bearings and motion. Table 15-1 lists the various bearings and motions normally encountered in internal combustion engines. There is a wide variation in the lubrication requirements for the different bearings. Therefore, the lubricant selected for a particular engine must provide the greatest number of lubricating properties that are required.

15-2. Journal Bearings. One of the most widely used types of bearings is the journal bearing. A journal bearing with exaggerated clearances is shown at rest in Fig. 15-2(a). Since there is no relative motion

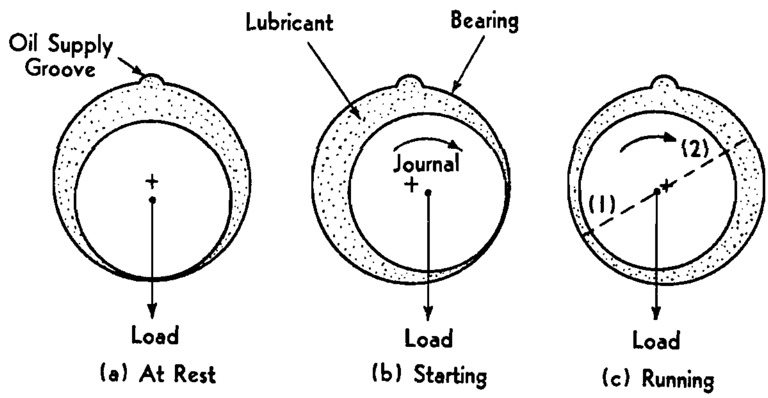

FIG. 15-2. Journal bearing with exaggerated clearances.

between the bearing and the journal when the journal is at rest, there is no film-pressure and the oil cannot support the load. This results in a metal to metal contact.

As the journal begins to rotate, in a clockwise direction as shown in Fig. 15-2(b), the journal crawls up the bearing to the right. Since the lubricating oil adheres to the surfaces, the journal is operating in the region of thin-film or partial film lubrication. As the speed is increased, a viscosity pump action is developed and the journal center moves to the left, Fig. 15-2(c). Oil is drawn by the pump action from the unload portion (2) of the bearing and a wedge of oil is formed under the jour-

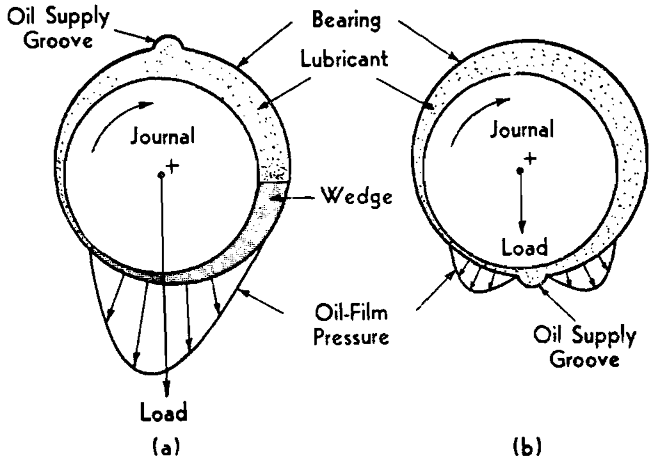

FIG. 15-3. Oil-film pressure in journal bearing (a) with oil supply in unloaded sector and (b) with oil supply in loaded sector.

nal. An oil-film pressure is developed as shown in Fig. 15-3(a). The journal is then operating in the thick-film region and the load is carried by the wedge of oil, shown as the shaded area in Fig. 15-3(a).

The oil supplied to the bearing should enter in the unload region or diverging section. If the oil supply groove is cut in the load region, Fig. 15-3(b), the oil-film pressure is reduced, resulting in a large reduction in the load carrying capacity of the bearing.

When operating normally, journal bearings run with a thick-film lubrication so that friction and wear are minimized. However, they operate in a thin-film region when the engine is started, Fig. 15-2(b). Thus nearly all the wear on a journal bearing occurs during the starting process. To keep the wear down, an engine should be started under a no load or light load and at slow speed. A fast start with a cold engine will increase the wear on the bearings of an automotive engine.

Wear during starting is also increased by abrasive particles present in the oil that bridge the thin-film. Wear is decreased by eliminating these abrasive particles and by minimizing the duration of thin-film operation at starting.

Oil is usually supplied to the bearings under pressure by a pump system (Fig. 9-6). The supply pressure does not aid in the load carrying capacity, but it does give a positive rate of oil flow to carry away the heat developed by friction in the journal. If the heat flow from the bearing is restricted, the temperature may rise rapidly causing the bearing to melt and seize.

An oscillating journal bearing does not have the tendency to form a wedge and a viscous pump action as in a rotating journal. If the load on the bearing reverses, such as with piston pins, a very high load may still be supported. The viscous oil between the approaching surfaces develops an oil-film pressure as it is squeezed out between the surfaces. The oil then flows to the unloaded side and is ready to support the load when it is reversed.

When the load does not reverse itself, it is difficult to maintain the proper film and sustain a load in a plain journal bearing. Thrust bearings utilizing ball or roller bearings and the Kingsbury thrust bearings are used since they can maintain proper lubrication where a journal bearing would fail.

15-3. Reciprocating Bearings. Pistons, piston rings, and valve stems operate with reciprocating motion. Throughout the engine cycle, these parts are subjected to high flame temperatures caused by the combustion of the fuel and to varying pressures. Thus, the oil film that is supplied to the cylinder walls must provide proper lubrication under extreme conditions. In addition, the lubricant on the cylinder walls must function as a seal to minimize the amount of the gases of combustion that pass the piston rings and enter the crankcase.

Tests indicate that the piston rings and cylinders operate in the thin-film lubrication region a great deal of the time. To provide a thick-film lubrication would result in excessive oil consumption or excessive friction due to the required high viscosity of the lubricating oil. In the thin-film region the surface finish, as well as the oiliness of the lubricant, plays an important role in reducing wear and scuffing. Tests indicate that a ground and lightly honed finish produces the best results. With this type of finish the surface has flat spots with many small indentations interspersed among them. These irregularities help the oil to maintain a film on the surface and act as individual oil reservoirs. Surface damage may be reduced further by utilizing sulphur and chlorine compounds in the lubricant. These anti-fusing compounds prevent *the welding of stress points* by converting the material into oxides, sulfides, and chlorides.

During the breaking in process, erosion or scuffing is caused by the high spots that break through the film surface. Additional damage to the cylinder is caused by abrasive materials in the oil, cylinder or ring distortion, and corrosion. Corrosion occurs not only when the engine stands idle for prolonged periods, but by the condensation of water and other vapors that combine with other compounds to form corrosive acids.

15-4. Gear Teeth. Most gears used in an internal combustion engine operate in the thin-film region. Lubrication of gears may be provided by an oil jet directed at the teeth as they disengage or by the gears dipping into an oil bath. The amount of oil must be regulated. If too much is provided, oil may become trapped between the teeth as they approach. Forcing the trapped oil out creates an oil-film pressure that tends to push the gears apart. This results in vibration, high bearing loads, and power loss.

15-5. Needle, Ball, and Roller Bearings. Needle bearings have rollers that are small in diameter compared to the shaft diameter. With this type of bearing there is some slippage between the roller surfaces and the shaft. In operation, this type is generally flooded with oil to reduce friction, slippage, and wear.

Ball and roller bearings have very little sliding contact and a thin mist or spray of oil is sufficient for adequate lubrication. An excess of oil causes violent churning which generates heat and results in a rise in temperature and a loss of power.

The friction coefficient of ball and roller bearings is in the range of 0.001 to 0.007. The frictional coefficient for a properly designed thick-film lubricated journal bearing will approach the coefficients of ball and roller bearings.

15-6. Properties of Lubricating Oils. Lubricating oils are obtained by refining crude petroleum. The properties of the lubricants are dependent upon the crude oil from which they were derived, the type and extent of refining, and the additive treatment. There are many different types of crude oils, refining processes, and additives, resulting in a very wide variation in the physical properties and performance characteristics of lubricating oils.

In order to evaluate and compare the many different lubricants, extensive laboratory tests and standards have been set up over the years.

The major consumers, such as the military services and automotive manufacturers, have set up standard specifications for lubricating oils. The physical and performance properties of the lubricants as determined by strict laboratory tests must stay within the limits set forth by the specifications. The properties listed in the normal specifications, as shown in abbreviated form in Tables 15-2 and 15-3, are discussed briefly in the following paragraphs.[1]

TABLE 15-2

ABBREVIATED MILITARY SPECIFICATIONS FOR FORCED-FEED OIL—LOW V.I. FROM BUREAU OF SHIPS PAMPHLET NBS 431

Military Symbol No.	2110	2135	2190
SAE Viscosity No.	10	20	30
Viscosity Saybolt seconds at 210° F	—	—	—
Viscosity Saybolt seconds at 130° F	90–120	120–145	185–205
Pour Point, Degrees F (Max.)	0	0	35
Flash Point, Degrees F (Min.)	325	340	350
Color[1]	Report	Report	Report
Carbon Residue, per cent (Max.)	0.20	0.30	0.40
Total Sulfur, per cent (Max.)	0.50	0.50	0.50
Ash, per cent (Max.)	0.003	0.003	0.003

[1] No requirements test made as matter of record.
[2] See NBS 431 for other requirements, such as viscosity-index, reaction, corrosion, neutralization number, etc.

TABLE 15-3

MILITARY SPECIFICATIONS FROM "LUBRICATING OIL; AIRCRAFT-ENGINE" MIL-L-6082A, OCTOBER 9, 1951

Military No.	1065	1100
Viscosity, Saybolt seconds at 210° F	62–68	93–103
Viscosity Index	100	95
Flash Point, Deg. F (Min.)	420	470
Pour Point, Deg. F (Max.)		
Undiluted	0	10
Diluted	−65	−65
Carbon Residue, per cent (Max.)	0.6	1.2

[1] See MIL-L-6082A for other requirements, such as corrosion, sulfur, ash, etc.

Viscosity. The viscosity of an oil is a measure of its fluid resistance to flow and is regarded as its internal or fluid friction. Although the

[1] See reference 15-3 for details on refining, specifications, and tests.

absolute unit of viscosity is the poise (dyne seconds per square centimeter), the standard practice is to express viscosity in terms of Saybolt seconds or seconds Saybolt Universal.

The Saybolt viscosimeter test is the method most widely employed for determining the viscosity of lubricating oils. The oil sample is placed in the Saybolt tube, which is a 60 ml tube with a fixed orifice, and heated in a bath to the test temperature. A cork is withdrawn from the orifice and the time in seconds for the 60 ml of oil to flow through the orifice is taken as the Saybolt viscosity. Thus, the Saybolt seconds is purely empirical and based on arbitrarily chosen conditions.

The Society of Automotive Engineers (SAE) has adopted a system using SAE viscosity numbers. Each number represents a viscosity range expressed in minimum and maximum Saybolt seconds at a particular test temperature. An example is an oil with a SAE number of 30 must have a Saybolt viscosity in the range between 185 and 255 Saybolt seconds at a test temperature of 130° F. This is shown in Table 15-4.

TABLE 15-4

SAE VISCOSITY NUMBER FOR CRANKCASE LUBRICANTS

SAE Viscosity Number	Viscosity Range, Saybolt Seconds			
	At 130° F		At 210° F	
	Min.	Max.	Min.	Max.
10	90	119		
20	120	184		
30	185	254		
40	255			80
50			80	104
60			105	124
70			125	150

Normally a lubricant is selected for an engine on the basis of its SAE rating. The selection of the proper oil viscosity for any given application depends on the type of engine, and its operating load, speed, and temperature. In internal combustion engines, temperature is the most critical of the operating conditions since the temperatures encountered in normal operations vary over a wide range. At low temperatures, such as encountered in starting, the viscosity of an oil is high resulting in an increase in friction and difficulty in starting. As the speed and temperature increase, the viscosity decreases resulting in a lowering of the power required to overcome the frictional resistance. Thus, the temperatures over the entire operating range are important and the viscosity at one set test temperature has little general significance.

The viscosities at various temperatures may be determined from the ASTM Saybolt viscosity-temperature charts. However, the common measure of viscosity-temperature relationship is viscosity index. *Viscosity index* is a measure of the change in viscosity of an oil with temperature as compared to two reference oils having the same viscosity at 210° F. It is an empirical system wherein a typical Pennsylvania (paraffinic-base) oil was assigned an index of 100 and a Gulf Coast (naphthenic-base) oil was assigned an index of zero. In general, the viscosity index number indicates the relative resistance of a given oil to change viscosity with a radical change in temperature. A low index number indicates a low resistance for a given oil so that it would have a relatively high viscosity when cold and a relatively low viscosity at elevated temperatures. A high viscosity index oil is preferred for good engine lubrication.

Pour Point. The temperature at which an oil will not flow when the test container is tilted under set test conditions is termed the pour point. It indicates the temperature below which the oil loses its fluidity and will not flow or circulate in a lubricating system. At sufficiently low temperatures, the partial separation of the paraffin waxes or the congealing of the hydrocarbons composing the oil causes the oil to become stiff. The oils from paraffinic-bases tend to have a higher pour point than the naphthenic-base oils.

Flash Point. The temperature to which an oil must be heated in order to give off sufficient vapors to form a combustible mixture with air is termed the flash point. The flash point is determined by heating a sample of oil in a metal cup at a given rate. A lighted taper is passed over the surface of the heated oil and the temperature is recorded when a flash appears. The minimum flash point for internal combustion engine lubricants varies from 345° F to 500° F. The flash point, in conjunction with the fire point, is a rough indication of the volatility of an oil. The *fire point* is usually determined at the time the flash point test is made and is defined as the temperature at which an oil will continue to burn after the flammable vapor-air mixture is ignited.

Carbon Residue. The carbon residue is determined by heating, igniting, and burning a 10 gram sample of oil under strict control conditions in a Conradson apparatus until only a residue remains in the crucible. The weight of the residue is carefully determined and its percentage based on the original sample weight is calculated. The carbon residue test has little significance in regard to the amount of carbon that will be formed in an engine during operation. The value of the test, in conjunction with other specifications, is to indicate the type of crude oil base from which the oil was refined and the degree or control of the refining.

Color. The color of an oil is an indication of the degree of refining and is an aid in identifying an oil.

Reaction and Neutralization. All lubricating oils should be neutral in reaction, i.e., they should not retain any mineral acids or alkalies that are employed in the refining process. The neutralization number indicates the amount of constituents contained in the lubricant that will react with potassium hydroxide or with sulphuric acid.

Classification of Military Symbol Oils from Bureau of Ships Pamphlet NBS-431 of March 1, 1950.

(1) A four-digit identifying symbol is used in all cases, the first digit indicating the series to which the oil belongs and the last three digits indicating the viscosity in Saybolt seconds. Viscosities are measured at 130° F for all series 2, 8, and 9 oils. All other series are measured at 210° F.

(2) "The series of lubricating oils, as indicated by the first digit, are as follows:

Series 1 Lubricating-oil, Aircraft Engine.
2 Lubricating-oil, General Purpose.
3 Lubricating-oil, General Purpose.
4 Lubricating oil, Compounded.
5 Lubricating-oil, Cylinder Material.
6 Lubricating-oil, Compounded.
7 Lubricating-oil, Compounded.
8 Lubricating-oil, Compounded.
9 Lubricating-oil, Diesel-engine."

(3) "The following illustrates the symbol system:

Example: 2135 indicates a series 2 low V.I. (viscosity-index) forced-feed oil having a viscosity of approximately 135 seconds at 130° F.

Example: 4065 indicates a series 4 compounded marine engine oil having a viscosity of approximately 65 seconds at 210° F."

15-7. Additives. In addition to the properties listed in the specifications, lubricants used in internal combustion engines must have good stability, detergent, oiliness, and film strength characteristics, plus having anti-corrosive, anti-oxidizing, and anti-foaming qualities. In an endeavor to improve these properties, certain oil-soluble organic compounds, containing inorganic elements such as phosphorus, sulfur, amine derivatives, and metals, are added to the mineral based lubricating oils. These oil-soluble organic compounds are called additives and one or more of the following types are usually added to present day lubricants:

1. Oxidation and corrosive inhibitors
2. Detergent agent
3. Anti-foam agent
4. Oiliness and film-strength agents
5. Pour point improvers
6. Viscosity-index improvers

Oxidation and Corrosive Inhibitors. Normal engine operation subjects the lubricating oil to high temperatures and to oxygen, particularly in the cylinders, resulting in the possible oxidation and decomposition of the oil. The ability of the oil to resist this oxidation and decomposition, which yield oil-soluble acids, lacquers, and sludge, is termed stability.

Oxygen is present in the air in the crankcase and in the gases of combustion that blow-by the piston rings. Some of this oxygen combines with the hydrocarbons of the oil under high temperature to produce oil-soluble acids and insoluble particles which tend to form a hard shiny deposit called lacquer on the pistons and piston rings. Water vapor, which is also present in the gases of combustion, tends to condense at a low operating temperatures and mix with the products of oxidation. The water and oxidation products, along with carbon and other foreign particles, such as dust and dirt, tend to form a soft, sticky sludge. The sludge causes clogged oil passages and sticking valves, which may result in eventual damage to the engine. The water and oil-soluble acids also tend to attack and corrode the metal parts.

Stability additives are used to decrease the oxidation and corrosive characteristics of lubricating oils. The inhibitors are generally in the form of sulfur and phosphorus compounds or amine and phenol derivatives. These compounds have a greater affinity for oxygen than the hydrocarbons of the lubricants. Different additives will have varying degrees of effectiveness depending on the oil-base, the operating conditions, and length of time the oil is to be used. A further aid in the prevention of oxidation and corrosion is an effective crankcase ventilation system. Usually breathers are installed to remove some of the gases of combustion that have blown-by the piston rings which aids in preventing these condensibles from forming sludge and lacquer.

Detergent Additives. The detergent additives improve the properties of an oil by keeping the oxidation products, carbon, water, dirt, and other insoluble matter in suspension or dispersion in the oil. By keeping the insoluble compounds dispersed, the tendency for the compounds to settle out and stick to the metal surfaces forming sludge and lacquer is decreased. Detergents are not intended to cleanse and purge a dirty engine, but will keep a clean engine free of sludge and other deposits. Since the main function of the detergent additives is to disperse the insoluble particles and reduce the tendency to accumulate in the form of sludge, they have often been termed *dispersants.*

The chemical structure and physical action of the detergents, or dispersants, are quite complicated and involved. Some of the chemical compounds used are: aluminum naphthenate, calcium phenyl stearates, calcium alkyl salicylates, metal salts of cetyl phenol, and alkaline earth metal petroleum sulfates.[2] The physical action is analogous to the action of soap in keeping dust and oil suspended in water.

Anti-foam Agents. All oils foam to some extent due to the violent agitation and aeration that occurs in a running engine. The minute particles of air in a foaming oil increase oxidation and reduce the mass flow of oil to the bearings. In addition, foaming may cause abnormal loss of oil through the breather.

Anti-foam agents are used to reduce the foaming tendencies of the lubricant. Although many chemical compounds may be used, the most effective are the silicone polymers.

Pour Point Improvers. In order to obtain fluidity or flow of oil at low temperatures, pour depressants are added to the lubricating oils to lower the pour point. These additives tend to prevent the formation of wax at the low temperatures encountered in starting. Two of the pour depressants on the commercial market are "Santopour" and "Acryloid."

Viscosity Index Improvers. High molecule polymers are added to the lubricating oils to increase their viscosity index. An increase in the viscosity index increases the resistance of an oil to change viscosity with a change in temperature. A high viscosity index oil will have good starting characteristics plus satisfactory operation at high-speed, heavy-load conditions.

Oiliness and Film-strength Agents. Oiliness and high film-strength are important in partial or thin-film lubrication. Many organic sulfur, chlorine, and phosphorus compounds are used as additives to improve the film-strength of a lubricant.

15-8. Lubricating Systems. The function of a lubricating system is to provide a sufficient quantity of cool, filtered oil to give positive and adequate lubrication to all the moving parts of an engine. The two basic types of lubricating systems in use to meet the requirements for proper lubrication of the various types of internal combustion engines are the *wet sump* and the *dry sump* systems. The wet sump system is employed

[2] Reference 15.

in relatively small engines, such as automobile engines, while the dry sump system is used in large stationary, marine, and aeronautical engines.

Wet Sump. In the wet sump system, the bottom of the crankcase contains an oil pan or sump that serves as the oil supply or reservoir tank and in most cases it also serves as the oil cooler. Oil dripping from the cylinders and bearings flows by gravity back into the wet sump where it is picked up by a pump and recirculated through the engine lubricating system. The types of wet sump systems in general use are (1) the *splash and circulating pump* system, (2) the *splash and pressure* system, (3) the *force-feed system*, and (4) the *full force-feed* system. Most of the relatively small four-stroke cycle gasoline and diesel engines now being manufactured utilize either the force-feed or the full force-feed lubricating systems.

The splash and circulating pump method of lubricating an engine is illustrated in the schematic diagram of Fig. 15-4 (a). The oil supply is carried in an engine crankcase at a predetermined level. A circulating pump, generally of the gear type, delivers oil to the troughs located under the ends of the connecting rods. Dippers on the ends of the connecting rods strike the oil in the troughs and splash it over the various parts of the engine. Some of the oil collects in cups or pockets on the main and crankshaft bearings and feeds these bearings. The crankpin bearings receive oil from the dippers through slots cut in the lower ends of the connecting rods. Part of the oil is splashed by the dippers onto the cylinder walls and lubricates the piston skirt and piston rings. Oil splashed into the underside of the hollow pistons collects under the piston heads and passes through grooves to lubricate the piston pins. The oil dripping from the cylinders and gears, and the excess overflow oil from the troughs, collects in the sump tank where it is cooled by the air flowing around the outside of the sump. The cooled oil is then recirculated.

The splash and pressure lubricating system is shown in Fig. 15-4(b). The oil pump supplies oil under pressure to the main and camshaft bearings. The oil pump also supplies oil under pressure to pipes which direct a stream of oil against the dippers on the connecting rod bearing cups. The crankpin bearings receive oil from the dipper through slots cut in the lower ends of the connecting rods. The other parts of the engine are lubricated by the splash or spray of oil thrown up by the dipper.

In the force-feed lubricating system, the oil pump forces oil under pressure to the main, connecting rod, and camshaft bearings, and to the timing gears. Drilled passages in the crankshaft carry oil from the main

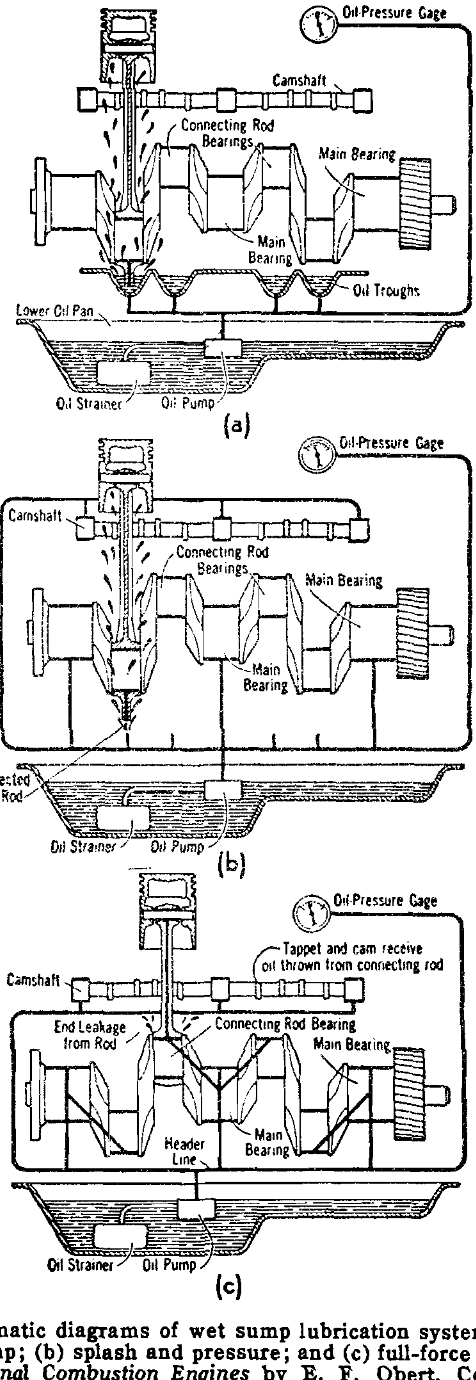

FIG. 15-4. Schematic diagrams of wet sump lubrication systems: (a) splash and circulating pump; (b) splash and pressure; and (c) full-force feed. (By permission from *Internal Combustion Engines* by E. F. Obert. Copyright 1950. International Textbook Company.)

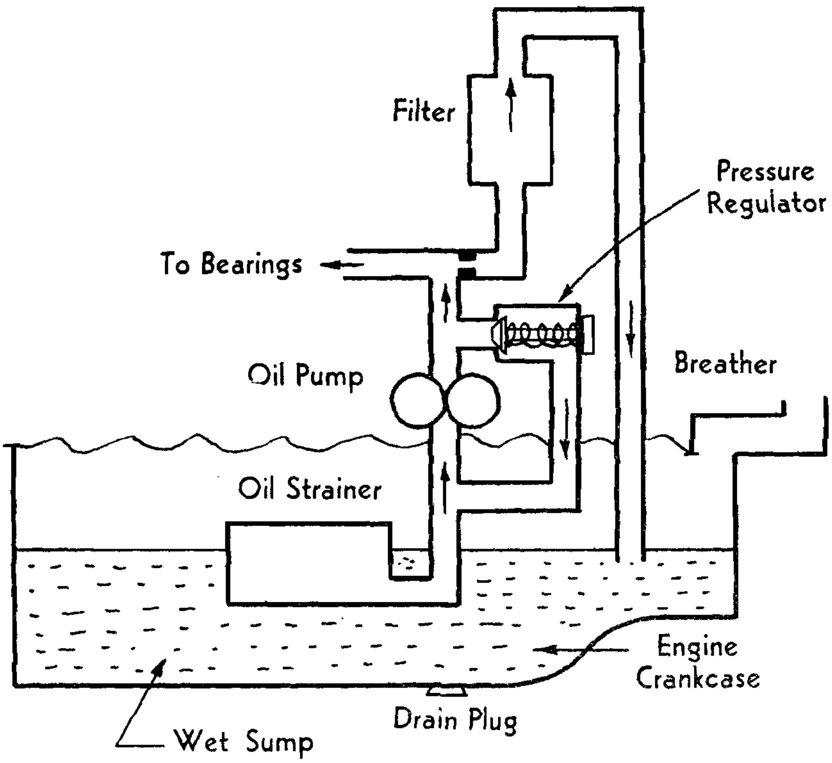

FIG. 15-5. Schematic diagram of wet sump lubricating system.

bearings to the connecting rod bearings. The cylinder walls, piston, and piston pins rely on the oil spray thrown off from the connecting rods and crankshaft for lubrication.

The full force-feed type of lubricating system is illustrated in Fig. 15-4(c). In this system, oil under pressure from the pump is forced through drilled passages to all the bearings. The drilled holes in the connecting rods permit oil to flow from the connecting rod bearings to the piston pins. The cylinder walls, piston, and piston rings are lubricated by oil spray from around the piston pins and the main and connecting rod bearings. In some engines, holes are drilled in the upper part of the connecting rod bearings so that oil under pressure is sprayed on the cylinder walls and underside of the pistons.

The basic components of the wet sump lubricating systems are the (1) pump, (2) strainer, (3) pressure regulator, (4) filter, and (5) breather. The arrangement of the components in a typical wet sump system is shown in Fig. 15-5. Oil is drawn from the sump through a strainer by

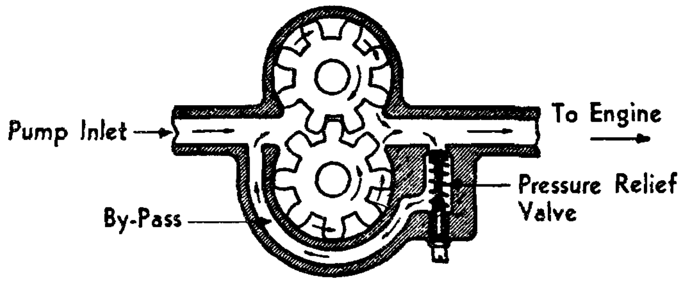

FIG. 15-6. Oil pump, pressure relief valve, and by-pass. (U. S. Navy Photograph.)

a gear or rotor type of oil pump. The strainer is usually a fine mesh screen which prevents foreign matter from entering the oil circulating system. An oil pressure relief valve is provided to prevent the build up of excessive oil pressures. Figure 15-6 shows a typical gear pump, pressure relief valve, and by-pass. Most of the oil from the pump goes directly to the engine and a portion of the oil passes through a cartridge filter which removes the solid particles from the oil. This reduces the amount of contamination from carbon, dust and other impurities present in the oil. Since all of the oil coming from the pump does not pass directly through the filter, the filtering system shown in Fig. 15-5

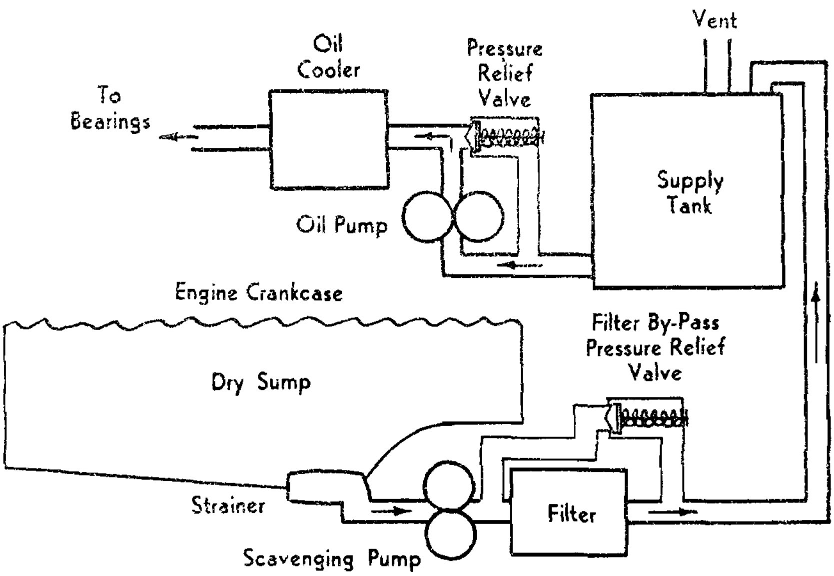

FIG. 15-7. Schematic diagram of a dry sump lubricating system.

is called a by-pass filtering system. Over a period of operation all the oil will eventually pass through the filter. In this filtering system, a clogged filter will not restrict the flow of oil to the engine.

Dry Sump. In a dry sump system, the supply of oil is carried in an external tank. This is illustrated in Fig. 15-7. Oil dripping from the cylinders and bearings into the sump is removed by a scavenging or sump pump, passed through a filter, and returned to the supply tank. Since the capacity of the scavenging pump is greater than that of the oil pump, oil is prevented from accumulating in the base of the engine. All the oil passing from the engine to the supply tank passes through the filter. If the filter becomes clogged in the full-flow filter system shown in Fig. 15-7, the oil filter pressure relief valve opens permitting oil to by-pass the filter. The oil pump draws oil from the supply tank and circulates it under pressure to the engine bearings. The majority of engines using the dry sump have a full-force feed lubricating system as exemplified in Fig. 15-4(c). A separate oil cooler, either water or air, is usually provided in the dry sump system to remove heat from the oil.

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EXERCISES

15-1. What are the functions of a lubricating system?

15-2. Will proper lubrication aid in increasing the service life of an engine? Why?

15-3. How is an oil-film pressure developed? Will an oil-film pressure develop if the two metal surfaces are parallel?

15-4. Define friction coefficient.

15-5. How does thick-film lubrication differ from thin-film lubrication?

15-6. Will an increase in the temperature increase or decrease the viscosity of a lubricant?

15-7. How is a wedge of oil developed in a journal bearing?

15-8. When does most of the wear occur in a journal bearing? Why?

15-9. How may the wear be minimized in a journal bearing?

15-10. What causes wear or scuffing in a cylinder wall?

15-11. How is the viscosity of an oil measured? What are the units commonly used?

15-12. How are SAE viscosity numbers determined?

15-13. What governs the selection of the proper lubricant?

15-14. What is viscosity-index?

15-15. Which oil provides the best lubrication over a large range of temperature, a high or a low viscosity-index oil? Why?

15-16. What is the significance of the pour-point? Carbon residue? Color?

15-17. What properties of lubricating oils do the additives improve?

15-18. Why are oxidation and corrosive inhibitors necessary?

15-19. What do the detergent additives accomplish?

15-20. Why are anti-foam additives used?

15-21. What are the two main types of lubricating systems? Describe the differences between the two systems.

15-22. What are the four types of wet sump systems?

15-23. Describe how the engine parts are lubricated in each of the wet sump systems.

15-24. What are the basic components of a lubricating system?

15-25. Draw a schematic diagram, labelling all components, of a typical wet sump system.

15-26. Draw a schematic diagram, labelling all the components, of a dry sump system.

CHAPTER XVI

THE THEORY AND FUNDAMENTALS OF GAS TURBINES

16-1. Brief History of Gas Turbines. The fundamentals and development of the gas turbine have held the imagination and energy of eminent scientists for many centuries. The basic concepts date back to the first known gas turbine built by Hero of Alexandria in 130 B.C. However, up until 1935 a practical gas turbine had not been produced that could compete with any degree of success with the reciprocating internal combustion engines and the steam plant in the fields of propulsion and stationary power plants.

Even though the gas turbine has a long history (16-1, 2),[1] the first important design was not made until 1791 in England by John Barber. The next important development was accomplished in the early part of the nineteenth century by Stirling and Ericsson who developed hot-air engines that led to the establishment of the Stirling constant volume cycle and Ericsson constant pressure cycle, both using isothermal compression and expansion. In the middle of the nineteenth century, Joule proposed the constant pressure cycle using isentropic compression and expansion. This cycle is now used as the fundamental cycle for gas turbines.

The first approach to the modern gas turbine was made in 1872 by Dr. F. Stolze. The Stolze gas turbine consisted of a multistage axial flow compressor coupled directly to a reaction turbine. Heat was added to the system by an externally fired combustion chamber. Tests of this gas turbine were made between 1900 and 1904. The unit was not successful due to low compressor and turbine efficiencies caused by the lack of knowledge of aerodynamics and due to the low turbine inlet temperatures caused by the inability, at that time, to supply metals for turbine blading that could withstand high temperatures and stresses.

During this period, Mr. Charles Curtis and Dr. S. A. Moss started the gas turbine research in the United States. Dr. Moss built the first gas turbine to operate in this country at Cornell University in 1902. However, the compressed air was furnished by a steam driven compressor which required greater power than that supplied by the turbine giving the plant a negative net power output.

Sir Charles Parson, inventor of the reaction steam turbine, undertook the development of an efficient axial flow compressor. Due to lack

[1] Numbers in parenthesis denote the references listed under bibliography.

of information about aerodynamics, his compressor was of little success and the development was abandoned in 1908.

Shortly before this time, a more efficient centrifugal compressor was developed by Rateau. The Societé des Turbomoteurs made several experimental gas turbines in Paris between 1903 and 1906 that operated on a cycle similar to the modern gas turbines using a multi-stage centrifugal compressor. The operating temperature of the turbine was 1030° F abs. The thermal efficiency obtained was between two and three per cent. This unit was the first gas turbine to produce a positive net useful work.

The lack of efficient compressors caused designers to investigate cycles that avoided the use of air compressors. The only cycle to offer a practical solution was the explosion or constant volume cycle developed by Dr. Holzworth in 1906. However, the disadvantages caused by the complicated valve gear and cooling system outweighed the advantages of this system. Thus, today, the simpler type of gas turbine cycle, which is the constant-pressure combustion gas turbine cycle, is in general use and will be studied in this text. Henceforth, in order to shorten the nomenclature, this cycle will be termed the gas turbine cycle.

Between 1905 and 1930 much work was accomplished by many scientists in the field of gas turbine research. The development and knowledge gained in two fields of research parallel to that of the gas turbine have contributed heavily toward the present day gas turbines. The development of the turbo-superchargers for reciprocating engines under the research of Dr. Moss, Dr. Buchi, and Dr. Lorenzen produced important advancements in the science of aerodynamics which aided in the design of more efficient compressors and turbines. The turbo-supercharger work also aided in the advancement of the science of metallurgy. The other parallel field of research which contributed to the advancement of the gas turbine was the development of the Velox steam generator under the supervision of the Brown-Boveri Company of Switzerland. This steam generator is a boiler fired under pressure. The pressure is produced by a compressor driven by a gas turbine which is actuated by the flue gases from the boiler. Before this system could operate, a compressor of high efficiency was essential; otherwise, the exhaust gas turbine would be unable to develop the power required to drive the compressor and the deficiency would have to be supplied by another source.

The improvement in both materials and compressor efficiency resulted in a renewed world wide effort to produce a constant-pressure gas turbine. One of the significant results of this renewed interest was the development of a turbojet engine by Air Commodore Frank Whittle of the British Royal Air Force. Although Guillaume of France and others had as early as 1922 operated types of turbojet engines, the engines were too inefficient for production.

In 1930, Whittle applied for a patent on a turbojet propulsion system incorporating all the essential elements of a simple gas turbine. Due to the prevailing international political tension in 1936, the Royal Aircraft Establishment and the newly organized Power Jets, Ltd., undertook the serious development of a turbojet engine. The Royal Aircraft Establishment group worked on an engine using an axial flow compressor while the Power Jets, Ltd., went ahead with Whittle's first design which used a centrifugal compressor. In 1937, Whittle's engine with some modifications proved to be successful. This started a decade in which many other possibilities of the gas turbine were investigated with unprecedented vigor.

During the period of 1935 and 1938, definite gas turbine programs were undertaken in Switzerland by Brown-Boveri, Sulzer Brothers, and Escher Wyss. Their principal aim was the development of gas turbines for locomotives and shore power plants.

In 1938, the Navy Department began serious investigations into the feasibility of utilizing the gas turbines for aviation and marine propulsion power plants. World War II accelerated the research and development of the gas turbine in all fields of propulsion. During the war the major progress and production was in the field of jet propulsion engines for aircraft and guided missiles. Since the end of World War II, rapid advancements have been accomplished in the various fields of application of the gas turbine engines.

16-2. Principles of the Gas Turbine. The basic, simple open cycle gas-turbine consists of a compressor, combustion chamber, and a turbine (See Fig. 16-1). The compressor takes in ambient air and raises its pressure. In the combustion chamber, heat is added to the air, raising its temperature and hence its heat energy. The heat added to the air, or working media, may be accomplished by (a) injecting fuel (liquid, gaseous or perhaps solid) into the air, and burning it in a combustion chamber, or (b) it may be added by the use of an air heater. This working fluid or media is then available, at a high temperature and pressure (3 to 10 atmospheres), to be expanded through a turbine to develop mechanical energy, in a manner similar to the familiar steam turbine. Since ambient air enters at the compressor and the gases of combustion are rejected to the atmosphere, the working medium must be continuously replaced. This cycle is termed an open cycle, and is a continuous flow process. However, part of the power developed by the turbine must be utilized to drive engine accessories, as well as the compressor with only the remainder available as useful work.

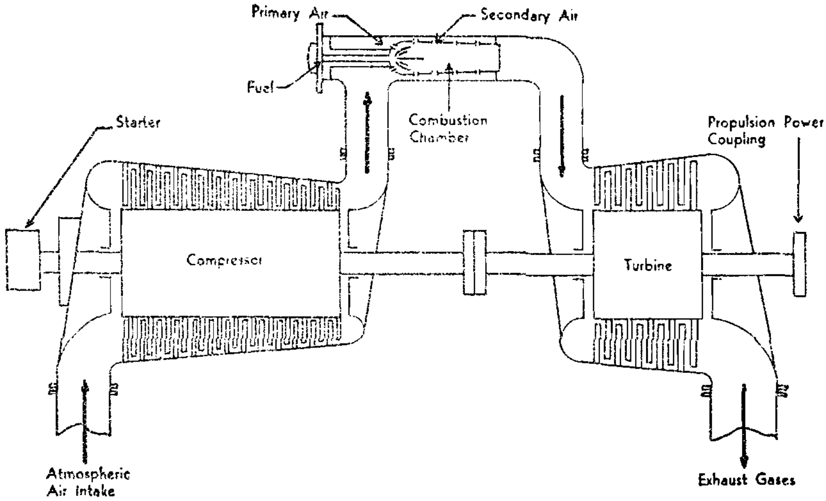


FIG. 16-1. Component parts of a simple open cycle (constant-pressure combustion) gas turbine.

16-3. Gas Turbine Performance Parameters. A clear understanding of the various steps in the gas turbine cycle is necessary in order to appreciate the operation of the gas turbine. The only known way to make a heated working fluid produce mechanical energy in a continuous flow process, is to allow it to expand from a higher to a lower temperature and pressure. To achieve this end in a gas turbine, fuel must be added not only to increase the energy of the working media to produce the necessary work but also to make up for losses which are radiation, leakage, pressure drops and efficiencies of component parts. Thus the problem becomes one of largest yield for least fuel consumed.

An initial step in this direction is to design the turbines and compressors in such a manner as to obtain the highest efficiency. The efficiencies obtained in present day designed compressors and turbines are in the order of 85 to 90%, and further advances in this direction probably will be slow. Figure 16-2 shows the improvement in turbine and compressor efficiencies since 1900. The upper limit of the compressor efficiency curve gives the efficiency of an axial flow compressor while the lower limit gives the efficiency of a centrifugal compressor. Figure 16-3 illustrates the very critical effect of compressor and turbine efficiencies on the thermal efficiency of the simple open gas turbine.

Therefore, other methods must be found to increase over-all efficiencies. Perhaps the most promising way is to increase the allowable turbine inlet temperature. Figure 16-4 shows the improvement in metallurgy that has been made to allow higher permissible inlet temperatures to the turbine. Figure 16-5 illustrates the critical effect of turbine inlet temperature on the overall thermal efficiency. A practical limitation to increasing the turbine-in temperature, however, is the ability of materials available for the turbine blading to withstand the high rotative and heat stresses.

For a turbine-in temperature range of 1200 to 1350° F, a simple gas turbine may realize an efficiency of between 18 to 26%, depending upon design. Considerable effort is being expended to find new materials, coatings and techniques, to increase the permissible turbine-in temperatures in gas turbines. Designers have also turned to other ways of raising the overall efficiency of the gas turbine engines. As has been the case in steam plants, higher efficiencies may be obtained by more complex plants utilizing various efficiency-improving devices.

One method of increasing the thermal efficiency, η_t, is to reduce the amount of fuel required to bring the working media to the allowable turbine inlet temperature. This can be accomplished by preheating the working fluid after it leaves the compressor and before it enters the combustion chamber by passing the working fluid through a heat exchanger called a *regenerator*. The heat available for this purpose, in the heat exchanger, is derived from the turbine exhaust gases that pass through it, and thus the net heat rejected to the atmosphere is also reduced.

The compressor work required may be reduced by dividing the compression into two or more stages, and to cool the air or working fluid between them. Most of the heat of compression may then be removed by *inter-cooling*. The effect of *inter-cooling*, when carried to the theoretical limit, is to have the work of compression approach an isothermal process (i.e., compression at a constant temperature). From a practical standpoint, however, the effect of inter-cooling is to reduce the work of compression required to achieve a given pressure rise. The reduction of compressor work achieved in this manner results in an increase in the overall gas turbine output and usually improves the overall plant efficiency.

In a reverse manner, the output of the turbine may be increased by dividing the expansion of the working media into a number of steps

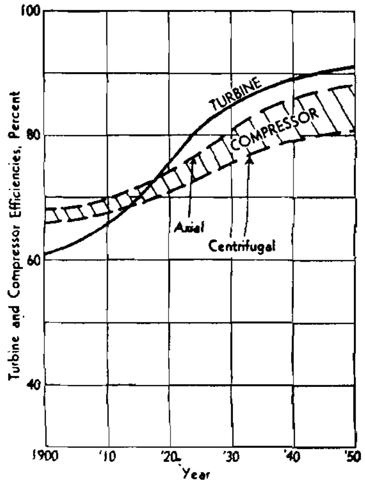


Fig. 16-2

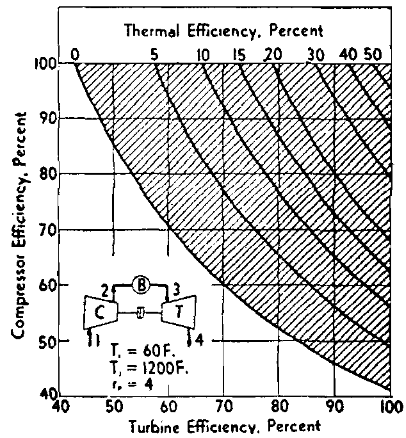


Fig. 16-3

FIG. 16-2. Advancement in turbine and compressor efficiencies since 1900 (courtesy of *Engineering* and Institute of Mechanical Engineers, England, ref. 16-19).

FIG. 16-3. Effect of compressor and turbine efficiencies on the overall thermal efficiency of a simple open cycle gas turbine (courtesy of *Engineering* and Institute of Mechanical Engineering, England, ref. 16-19).

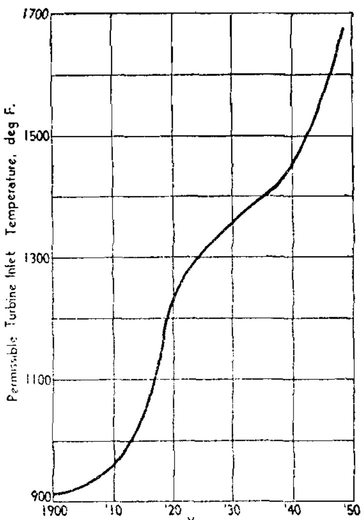


Fig. 16-4

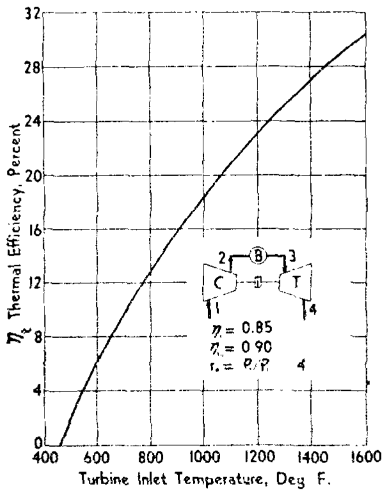


Fig. 16-5

FIG. 16-4. Increase in permissible turbine inlet temperature since 1900 (courtesy of *Engineering* and Institute of Mechanical Engineers, ref. 16-19).

FIG. 16-5. Effect of turbine inlet temperature on the thermal efficiency of a simple open cycle gas turbine (courtesy of *Engineering* and Institute of Mechanical Engineers, England, ref. 16-19).

and the gas *reheated* between them. The *reheating* of the gas or working media back to the limiting turbine-inlet temperature allows a greater portion of the expansion to take place at higher temperatures. The theoretical limit of reheating would, of course, be an isothermal expansion at the turbine inlet temperature. Again, from a practical standpoint, the result of reheating is an increase in the output of the turbine through the same expansion pressure range although it has a negligible effect upon the overall efficiency. However, when *reheating* is properly utilized in conjunction with *regeneration*, the increase in overall efficiency is appreciable.

It becomes apparent therefore, that there are a number of ways in which to increase the efficiency of the gas-turbine engine. In addition, modifications of the basic gas-turbine engine have been developed, namely, closed, semi-closed systems, and free piston gas generators, which further increase the overall plant efficiency.

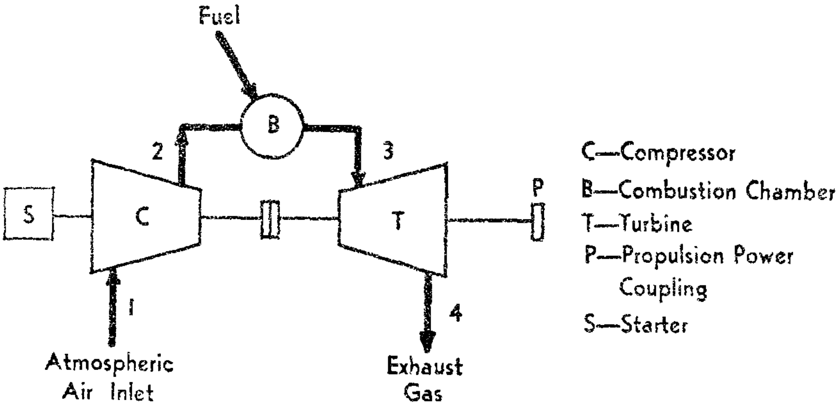


FIG. 16-6. Simple open cycle gas turbine.

16-4. Simple Open Gas Turbine Cycle and Assumptions for Simplifying Computations. A diagrammatical sketch of a simple open cycle gas turbine is shown in Fig. 16-6. The analysis of the gas turbine process is based on the Joule or Brayton cycle shown in Fig. 16-7. The atmospheric air entering the compressor is compressed to several times its atmospheric pressure. The pressure ratio is defined as the pressure of the compressed air upon discharge from the compressor, p_2, to the pressure of the air entering the compressor, p_1, and is denoted as r_p, i.e.,

$$r_p = \frac{p_2}{p_1}. \tag{16-1}$$

The high pressure air discharged from the compressor enters the combustion chamber, *B*, in Fig. 16-6 and state 2 in Fig. 16-7. In the simple open gas turbine the primary air is normally mixed with the injected fuel (liquid, gaseous or powdered solids) and burned continuously once the ignition is started. When using a hydrocarbon fuel, the mixture of primary air and fuel is approximately a stoichiometric air-fuel mixture. Since a normal hydrocarbon fuel burns with a temperature around 3500° to 4000° F, the major part of the air, called secondary air, is utilized to cool the products of combustion down to the permissible turbine inlet temperature. This gives an overall A/F ratio for the gas turbine engine utilizing a hydrocarbon fuel of between 50:1 and 250:1. Ideally the combustion process, 2–3, would take place at constant pressure, but in the operating gas turbine engine there is a small pressure drop due to the friction of the working fluid and to a change in momentum of the fluid caused by a change in density during combustion. The mixture of the products of combustion and the secondary air at the exit to the combustion chamber, state 3, are expanded through the turbine and exhausted to the atmosphere, state 4. Approximately two thirds of the power developed by the turbine is utilized to drive the compressor and the auxiliaries and about one third of the power of the turbine goes for useful work through the propulsive coupling.

In the Joule cycle, Fig. 16-7, the heat rejection process line, 4-1, is shown as a dotted line to indicate that the cycle is exhausting the working substance to the atmosphere at state 4 and taking in atmospheric air at state 1. Whether the cycle is completed from 4-1 by a rejection of heat as in the idealized cycles, i.e., closed cycle heat engines, or the cycle is not completed by the rejection of the working substance to the atmosphere as in many actual cycles, the effect insofar as net work and thermal efficiency of the cycle are concerned is exactly the same. It should be noted that the gas turbine and all its elements are steady flow devices and the **steady flow** energy equation applies to the cycle (16-3).

The general equations for the analysis of an actual operating gas turbine engine are very complex and the computations are long and difficult. However, by making certain simplifying assumptions it is possible to obtain equations that can be easily and readily used to compute the performance of a gas turbine. These assumptions are selected in such a manner that the precision of the resulting data or the validity of comparison of the general characteristics of the different gas turbines over the range of their operating variables is not sacrificed to any important degree. The errors created by these assumptions are very small and are of the order of a few per cent. This method of analysis can be used for a fairly close approximation; but in detailed design work, the simplifications are not used.

The following simplifying assumptions will be used throughout this chapter for all equations and computations:

(1) *Working fluid.* The working substance is dry air having the properties as stated in reference 14, i.e., dry air with variable specific heats. The working fluid is considered to have a constant weight and composition throughout the cycle.
(2) *Heating process.* Heat is added from an external source in place of a combustion process. The weight of fuel burned in an operating engine or real cycle is small compared to the weight of the air consumed due to the very high A/F ratios.
(3) *Parasitic losses.* It is assumed that no pressure drops occur between the components of the system or in the combustion chamber, reheater, intercooler, or regenerator. It is also assumed that the system has no radiation or leakage losses.
(4) *Potential and kinetic energy.* It is assumed that the potential and kinetic energy changes in the system as applied to the general energy equation are negligible. The kinetic energy change is small, only a few per cent, compared to the enthalpy change; therefore, the actual velocities of the working fluid will not be taken into account.

16-5. Ideal and Actual Theoretical Cycles for Simple Open Cycle Gas Turbine. The cycle analyses of a gas turbine, i.e., the approximations from an idealized cycle using a perfect gas as the working fluid to the actual engine cycle, are different from those encountered in cycle analyses (three approximations) in the field of the reciprocating engines. A very close approximation can be made to the actual gas turbine cycle performance by using the simplifying assumptions in the preceding Article. Thus, a different terminology will be used to define the gas turbine approximations than those used for reciprocating engine analysis.

The basic cycle for the analysis of a simple open cycle (constant-pressure combustion) gas turbine is the **Joule** (Brayton) cycle as shown in Fig. 16-7. The following idealized processes take place:

(1) Process 1-2. Isentropic or reversible adiabatic compression through the compressor.

(2) Process 2-3. Constant pressure addition of heat (combustion chamber).
(3) Process 3-4. Isentropic or reversible adiabatic expansion through the turbine.
(4) Process 4-1. Constant pressure rejection of heat to the atmosphere.

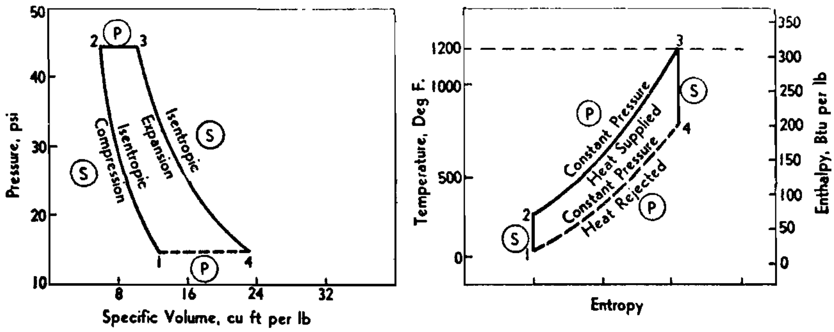


FIG. 16-7. Joule (Brayton) cycle.

This cycle will be termed the **ideal** cycle. The ideal cycle will have dry air with variable specific heats as the working fluid and will take into account the simplifying assumptions of Article 16-4.

An analysis of the ideal gas turbine cycle must be based on the steady flow equation. Since the changes in the kinetic energy and the potential energy of the gas turbine are assumed to be negligible, the steady flow equation for the compressor, process 1-2, becomes

$$h_1 + q_{12} = h_2 + wk_{12} \qquad \text{(Btu/lb)}$$

where each term has units in Btu per lb of air. Since the compression is assumed to be isentropic in the ideal case, q_{12} is equal to zero and the ideal work of the compressor, iwk_C, is

$$iwk_C = wk_{12} = h_2 - h_1 \qquad \text{(Btu/lb).} \qquad \textbf{(16-2)}$$

The heat added in combustion chamber is expressed as

$$q_S = q_{23} = h_3 - h_2 \qquad \text{(Btu/lb).} \qquad \text{(16-3)}$$

The ideal work of the turbine, iwk_{T_u}, with an isentropic expansion, is given as

$$iwk_{T_u} = wk_{34} = h_3 - h_4 \qquad \text{(Btu/lb).} \qquad \text{(16-4)}$$

The net work of the cycle is therefore

$$iwk_{(net)} = iwk_{T_u} - iwk_C \tag{16-5}$$

$$iwk_{(net)} = (h_3 - h_4) - (h_2 - h_1) \quad \text{(Btu/lb)}. \tag{16-6}$$

The thermal efficiency, η_t, of the ideal cycle is calculated from

$$\eta_t = \frac{iwk_{(net)}}{q_S} = \frac{iwk_{T_u} - iwk_C}{q_S}$$

$$\eta_t = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)} \tag{16-7}$$

An equation can be developed expressing the thermal efficiency of the Joule cycle in terms of the pressure ratio and the ratio of the specific heats.

$$\eta_t = 1 - \left(\frac{1}{r_p}\right)^{(k-1)/k} \tag{16-8}$$

The steps in the development of equation 16-8 are shown in Appendix "C." The working fluid in the ideal case, as defined on page 16-9, has a variable specific heat. As a result, "k," in the above equation, must be considered as variable and will be dependent upon the operating temperatures.

All computations in the analysis of the gas turbine cycles will be worked using the values of enthalpy obtained from the Air Charts in Appendix "B." This nomograph type of Air Charts was developed by the Bureau of Ships from the properties of dry air obtained from "*Thermodynamic Properties of Air*" by Keenan and Kaye. An example problem for the Joule or ideal cycle for the simple open cycle gas turbine engine is given in Appendix "A."

In the ideal cycle, it was assumed that the processes were reversible, i.e., isentropic or reversible adiabatic compression (compressor) and expansion (turbine). This implied that the compressor and turbine efficiencies were 100 per cent. In the actual gas turbine, the processes are irreversible adiabatic, i.e., the component elements (compressor and turbine) have efficiencies less than 100 per cent. The irreversibility is caused by the fluid friction losses in the compressor and in the turbine. The cycle taking into account the actual compressor and turbine efficiencies will be termed the actual theoretical cycle. This cycle will use dry air as the working fluid and will utilize the assumptions made in Article 16-4 for simplification of calculations.

The T-s diagram for the actual simple open cycle gas turbine as defined above is shown in Fig. 16-8. Note particularly the manner in

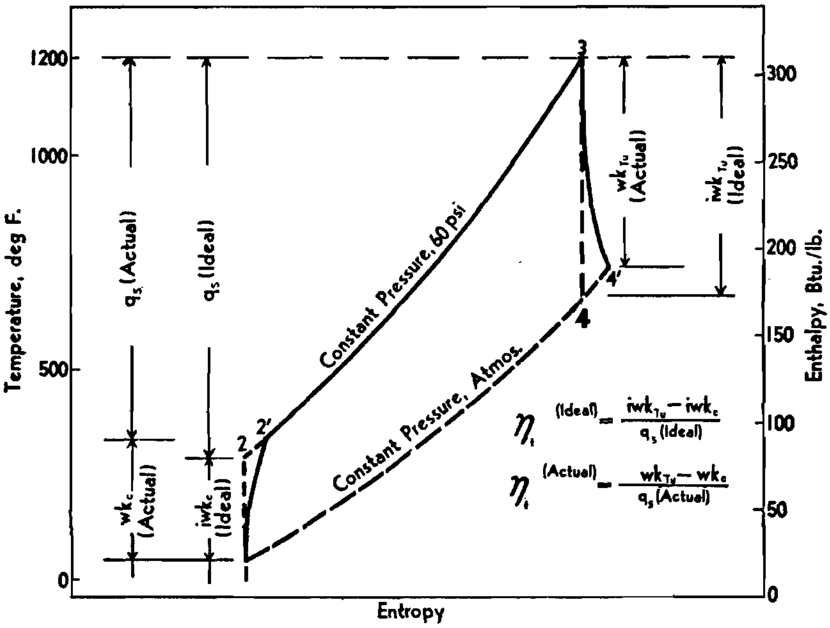


FIG. 16-8. Temperature-entropy diagram of simple open cycle gas turbine.

which the actual compression and expansion processes depart from the vertical, showing that they are no longer isentropic.

The actual work of the compressor, wk_C, is

$$wk_C = iwk_C/\eta_C = (h_2 - h_1)/\eta_C$$

$$wk_C = h_2' - h_1 \qquad \text{(Btu/lb).} \qquad (16\text{-}9)$$

The efficiency of the compressor, η_c, is defined as: the ratio of work required for ideal adiabatic or isentropic air compression through a given pressure range, to work actually needed by the compressor to accomplish the pressure rise.

$$\eta_C = \frac{\textbf{isentropic compressor work}}{\textbf{actual compressor work}} = \frac{h_2 - h_1}{h_2' - h_1}. \qquad (16\text{-}10)$$

The heat supplied in the actual cycle becomes

$$q_S = h_3 - h_2' \qquad \text{(Btu/lb).} \qquad (16\text{-}11)$$

The actual work of the turbine, wk_{Tu}, is expressed as

$$wk_{Tu} = iwk_{Tu}\eta_{Tu} = (h_3 - h_4)\eta_{Tu}.$$

Therefore, $$wk_{Tu} = h_3 - h_4' \qquad \text{(Btu/lb)}. \tag{16-12}$$

The efficiency of the turbine, η_{Tu}, can be defined as: the ratio of work actually available from the turbine to the work available from the turbine in an ideal adiabatic or isentropic expansion through a given pressure drop.

$$\eta_{Tu} = \frac{\text{actual turbine work}}{\text{isentropic turbine work}} = \frac{h_3 - h_4'}{h_3 - h_4}. \tag{16-13}$$

The thermal efficiency, η_t, of the actual simple open cycle gas turbine is

$$\eta_t = \frac{\text{net work}}{q_S} = \frac{iwk_{Tu}\eta_{Tu} - (iwk_C/\eta_C)}{q_S}$$

$$= \frac{(h_3 - h_4)\eta_{Tu} - [(h_2 - h_1)/\eta_C]}{h_3 - h_2'}$$

$$\eta_t = \frac{wk_{Tu} - wk_C}{q_S} = \frac{(h_3 - h_4') - (h_2' - h_1)}{h_3 - h_2'}. \tag{16-14}$$

An example problem of an actual simple open cycle gas turbine is given in Appendix "A."

16-6. Effect of Operating Variables on Thermal Efficiency. The thermal efficiency of an actual simple open cycle gas turbine is heavily dependent upon the following operating variables:

(1) Pressure ratio, $r_p = p_2/p_1$.
(2) Turbine efficiency, η_{Tu}.
(3) Turbine inlet temperature, T_3.
(4) Compressor efficiency, η_C.
(5) Atmospheric or compressor inlet air temperature, T_1.

Since there are five variables, it is not possible to represent the variation in thermal efficiency by a single curve.

In a simple open cycle gas turbine as shown in the schematic diagram in the upper left hand corner of Fig. 16-9, the cycle output is reduced from a full-load condition by decreasing the quantity of fuel being injected which lowers the quantity of heat added. A reduction in the heat supplied will cause a reduction in the speed of the rotor, i.e., turbine and compressor. A change in the speed of the compressor will make a corresponding change in the pressure ratio and the mass rate

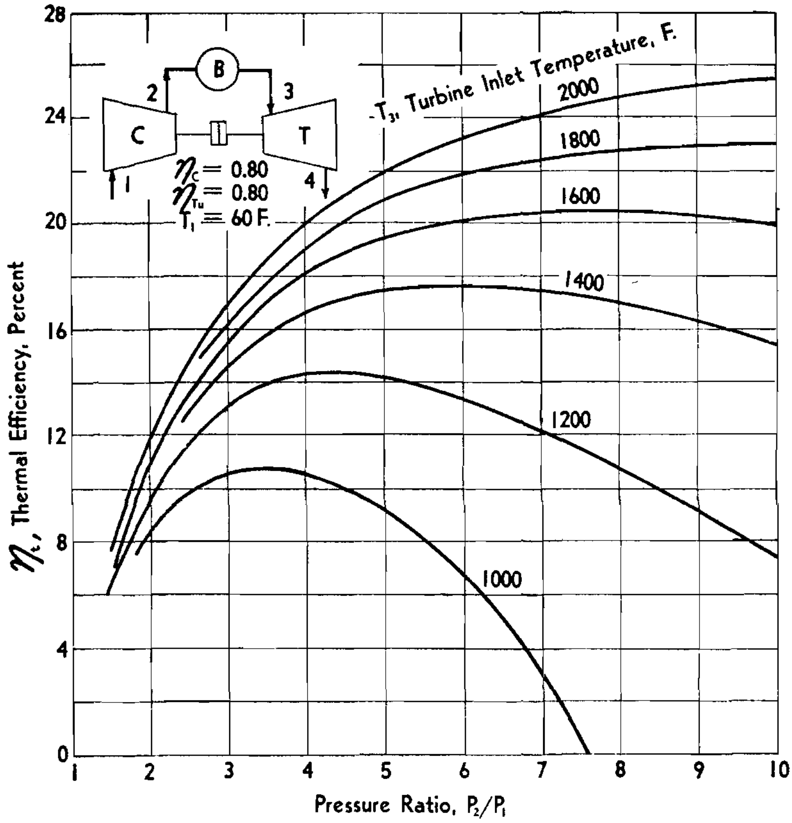


FIG. 16-9. Effect of turbine inlet temperature on the thermal efficiency of a simple open cycle gas turbine (reproduced from *Combustion Engines*, by A. P. Fraas, 1948. Courtesy of McGraw-Hill Book Co., ref. 16-14).

of air flow through the engine. Also, a variation in the speed from the design or full-load condition of the compressor and turbine will cause a change in the efficiencies of the components. The determination of the part-load thermal efficiencies of the gas turbine cycle becomes complex and the performance characteristics curves of the components must be known. Therefore, in this section the effect of the variation in the pressure ratio will be considered only from the full-load performance of the components at each pressure ratio.

1. *Effects of variation of turbine inlet temperature and pressure ratio.* Figure 16-5 indicates the great improvement in the thermal efficiency that is possible from an increase in the permissible turbine inlet temperature with the other variables being held constant. The constant values of the variables used in the calculation of the curves are listed

below the schematic sketches in the figures. Figure 16-9 indicates that there is an optimum pressure ratio for each given turbine inlet temperature.

The thermal efficiency, η_t, of the actual simple open cycle gas turbine is:

$$\eta_t = \frac{wk_{tu} - wk_c}{q_s} = \frac{(h_3 - h_{4'}) - (h_{2'} - h_1)}{h_3 - h_{2'}} \qquad (16\text{-}14)$$

where

$$q_s = h_3 - h_{2'} \qquad (16\text{-}11)$$

and the heat rejected to the atmosphere may be expressed:

$$q_R = h_{4'} - h_1. \qquad (16\text{-}15)$$

The thermal efficiency, η_t, of the actual simple open cycle gas turbine expressed in equation (16-14) then becomes:

$$\eta_t = \frac{(h_3 - h_{4'}) - (h_{2'} - h_1)}{h_3 - h_{2'}} = \frac{q_S - q_R}{q_S}.$$

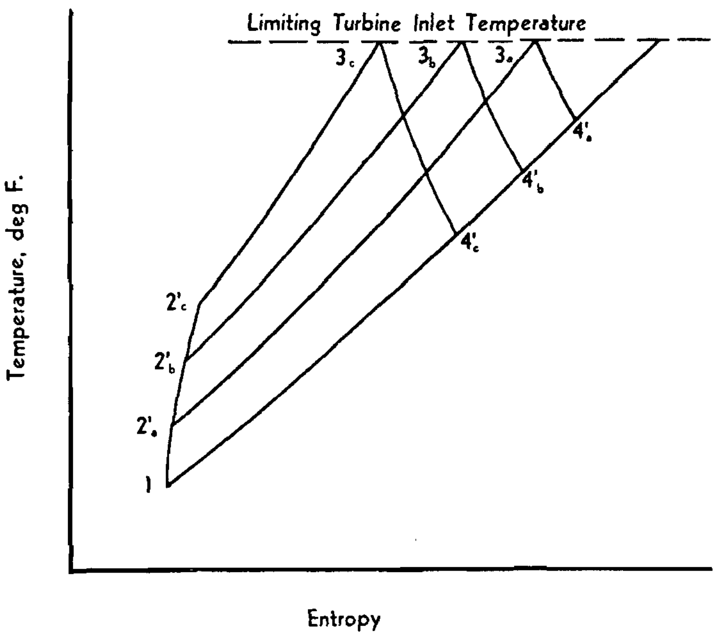


FIG. 16-10. Effect of increase in pressure ratio on the heat supplied and heat rejected of a simple open cycle gas turbine.

Figure 16-10 shows the T-S diagrams of a simple open cycle gas turbine with a limiting permissible turbine inlet temperature and increasing pressure ratios. The figure 16-10 indicates that as the pressure ratio increases the heat supplied, q_S, decreases as well as the heat rejected, q_R. However, the rate of change of the heat supplied (q_S) with changing pressure ratio is not the same as the rate of change of heat

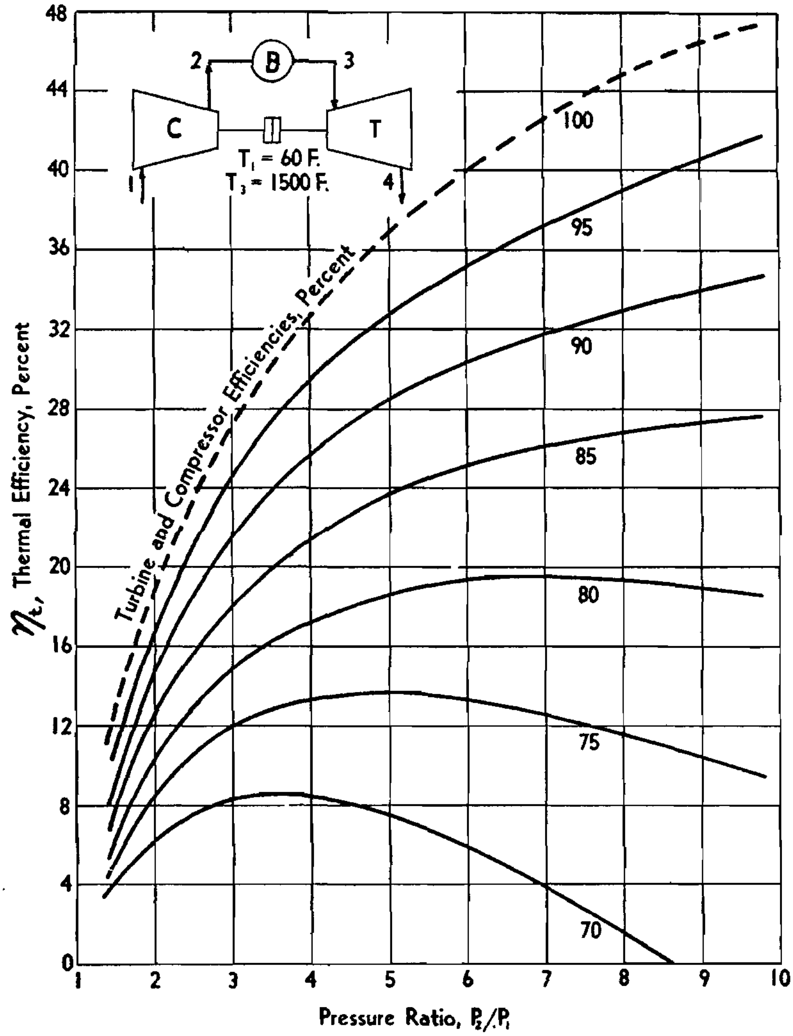


FIG. 16-11. Effect of turbine and compressor efficiencies on the thermal efficiency of simple open cycle gas turbine (reproduced from A. P. Fraas, *loc. cit.*).

rejected (q_R). As a result, there exists an optimum pressure ratio producing maximum thermal efficiency for each turbine inlet temperature. The thermal efficiency increases to a maximum as the pressure ratio increases and then drops off with a further increase in the pressure ratio, see Fig. 16-9. As the turbine inlet temperature increases, the peaks of the curves flatten out giving a greater range of pressure ratios of optimum efficiency. The importance of the research in metallurgy for the improvement of the turbine blading material to give higher permissible turbine inlet temperatures is born out in Fig. 16-9.

2. *Effect of variation of component efficiencies.* Figures 16-3 and 16-11 indicate how very sensitive the thermal efficiency of the gas turbine cycle is to variations in the efficiencies of the compressor and turbine. The dotted line in Fig. 16-11 is the curve for an ideal simple open cycle gas turbine. The curves of component efficiencies of less than 100 per cent show that there is an optimum pressure ratio for each set of component efficiencies. As the component efficiencies are reduced, the peak of the thermal efficiency occurs at a lower pressure ratio.

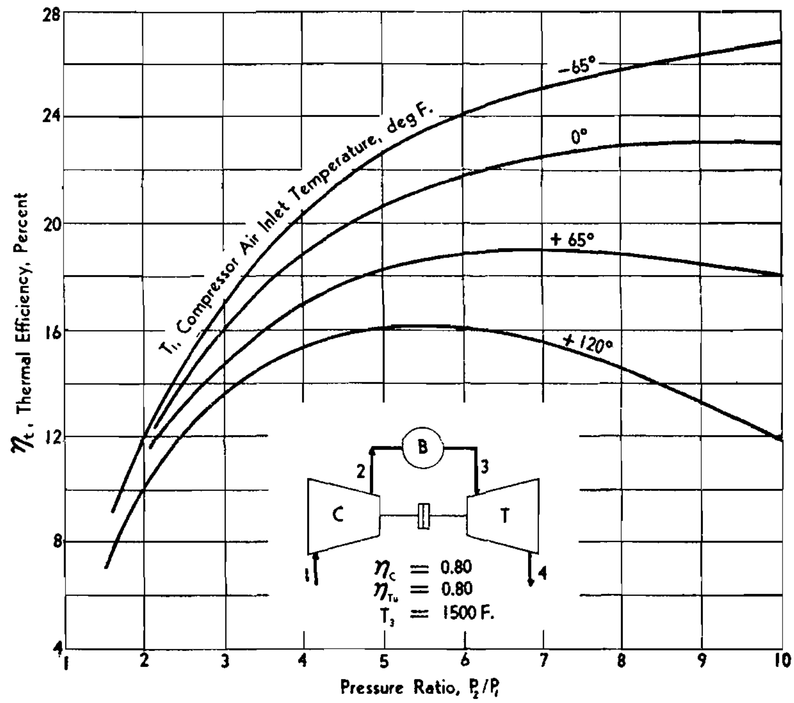


FIG. 16-12. Effect of compressor inlet air on the thermal efficiency of a simple open cycle gas turbine (reproduced from A. P. Fraas, *loc. cit.*).

3. ***Effect of variation of atmospheric or compressor air inlet temperature.*** The effect of the air inlet temperature on the thermal efficiency of a simple open cycle gas turbine is demonstrated in Fig. 16-12. As the air temperature is lowered, the thermal efficiency increases, peaks at a higher pressure ratio, and has a flatter curve giving optimum efficiency over a greater pressure ratio range. The curve reveals that there will be large variations in the thermal efficiency when there are extreme climatic conditions and that the best thermal efficiency will be obtained under winter conditions. This shows also why the efficiency of the turbojet aircraft engine increases with an increase in altitude, for as the altitude increases the atmospheric air becomes colder.

16-7. Regenerator. The thermal efficiency of a simple open cycle gas turbine may be improved by the utilization of the energy present

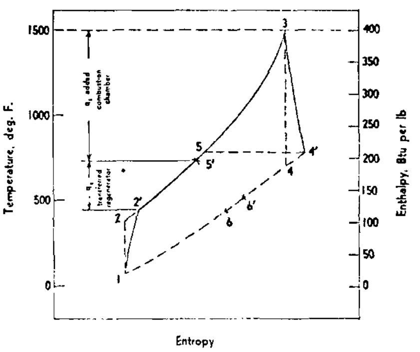


FIG. 16-13. Temperature-entropy diagram of an open cycle gas turbine with regenerator.

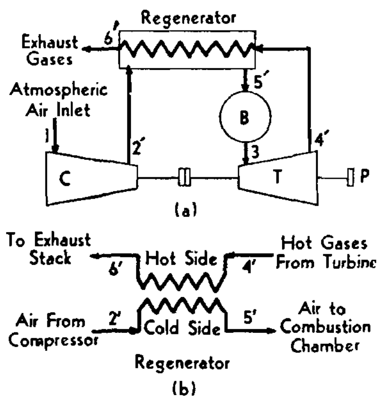


FIG. 16-14. (a) Open cycle gas turbine with regenerator. (b) Heat balance for regenerator.

in the exhaust gases in a regeneration process. Inspection of Fig. 16-13 reveals that the temperature of the exhaust gases leaving the turbine at state 4′ is higher than the temperature of the compressed air at state 2′. This difference in temperatures makes the regeneration possible. The recovery of part of the thermal energy of the exhaust gases is accomplished by installing a heat exchanger called a *regenerator* in the flow system as shown in Fig. 16-14(a). The exhaust gases at a high temperature enter the hot side of the regenerator and are circulated around tubes or plates containing the cold compressed air in the cold side of the regenerator. In this system, the temperature of the compressed air is increased before it reaches the combustion chamber and therefore, less fuel is required to raise the air to the specified turbine inlet temperature.

The compressor, turbine, and net work are not affected by the addition of the regenerator to the cycle. However, the heat required to be supplied to the cycle in the combustion chamber is decreased. Thus, the thermal efficiency of the cycle will be increased. The regenerator must be carefully designed; for if there is an excessive pressure drop across the regenerator, it will cancel out any gain in the thermal efficiency. Regeneration is only possible when the temperature of the turbine exhaust gases is higher than the temperature of the air discharged from the compressor. This is usually the case for the present range in gas turbine engine pressure ratios, i.e., up to pressure ratios of seven.

Ideal open cycle with regeneration. This cycle is shown in Fig. 16-13 as 1-2-5-3-4-6-1. The heat added to the compressed air in an ideal regenerator, q_{iX}, of an ideal open cycle gas turbine is (Fig. 16-13),

$$q_{iX} = h_5 - h_2 = h_4 - h_2 \qquad \text{(Btu/lb)},$$

where $h_4 = h_5$ for ideal regeneration. Thus, the heat required to be supplied in the combustion chamber of this cycle in order to raise the air to the permissible turbine inlet temperature is reduced by the amount q_{iX} and becomes

$$q_S = h_3 - h_5 \qquad \text{(Btu/lb)}.$$

The thermal efficiency of the ideal open cycle gas turbine with an ideal regenerator then becomes

$$\eta_t = \frac{\text{ideal net work}}{q_S} = \frac{iwk_{T_u} - iwk_C}{q_S}$$
$$= \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_5}. \qquad (16\text{-}16)$$

Also, the thermal efficiency of this ideal cycle may be expressed in terms of temperatures and pressure ratio,

$$\eta_t = 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}.$$

The steps for the development of this equation are given in Appendix "C."

Actual open cycle with regeneration. This cycle is shown in Fig. 16-13 as 1-2′-5′-3-4 -6′-1. Depending on the type of design and the magnitude of the heat-transfer surface, any desired degree of regeneration is theoretically possible. In order to obtain an ideal regenerator, i.e., one having a heat exchanger effectiveness or regenerator efficiency of 100 per cent, the temperature of the compressed air, $T_{2'}$, must be raised to the temperature of the exhaust gases, $T_{4'}$, entering the regenerator. This could only be accomplished by having a heat-transfer surface of infinite area.

Since the regenerators are restricted in size due to weight and space limitations, the regenerators have a maximum effectiveness of about 75 per cent. From Fig. 16-14, the regenerator effectiveness, η_x, is the ratio of the actual heat transfer, $h_{5'}-h_{2'}$, to the maximum possible or ideal heat transfer, $h_{4'}-h_{2'}$, and therefore is

$$\eta_X = \frac{\textbf{actual heat transfer}}{\textbf{ideal heat transfer}} = \frac{h_{5'} - h_{2'}}{h_{4'} - h_{2'}}. \qquad (16\text{-}17)$$

The heat added to the compressed air by the regenerator in an actual open cycle gas turbine (Fig. 16-13 and 16-14) is

$$q_X = (h_{4'} - h_{2'})\eta_X = h_{5'} - h_{2'} \qquad (\text{Btu/lb}). \qquad (16\text{-}18)$$

Therefore, the heat required to be supplied to the air in the combustion chamber in order to raise it to the permissible turbine inlet temperature becomes

$$q_S = (h_3 - h_{2'}) - (h_{4'} - h_{2'})\eta_X$$
$$q_S = (h_3 - h_{2'}) - (h_{5'} - h_{2'})$$
$$q_S = h_3 - h_{5'} \qquad (\text{Btu/lb}). \qquad (16\text{-}19)$$

Since the compressor work, the turbine work, and the net work of an open cycle gas turbine are not effected by the addition of a regenerator to the cycle, the thermal efficiency from Fig. 16-13 is then

$$\eta_t = \frac{wk_{T_u} - wk_C}{q_S}$$
$$\eta_t = \frac{(h_3 - h_{4'}) - (h_{2'} - h_1)}{h_3 - h_{5'}}. \qquad (16\text{-}20)$$

An example problem for an open cycle gas turbine with a regenerator is given in Appendix "A."

The enthalpy of the exhaust gases leaving the hot side of the regenerator, $h_{6'}$, may be found from

$$h_{4'} - h_{6'} = (h_{4'} - h_{2'})\eta_x \quad \text{or}$$
$$h_{4'} - h_{6'} = h_{5'} - h_{2'} \qquad (16\text{-}21)$$

The factors determining regenerator design for specific applications are: (1) space limitations, (2) maintenance and cleaning restrictions, (3) manufacturing limitations, (4) desired effectiveness, and (5) pressure losses. It is important to keep the pressure losses through the regenerator as low as possible. A large pressure drop on the cold air side increases the compression work while a large pressure rise on the hot exhaust side increases the back pressure on the turbine and reduces the turbine output. The limiting regenerator effectiveness compatible with reasonable space, weight, and cost is between 70 and 75 per cent with heat transfer materials available. This means from 2.5 to 3.5 square feet of heating surface per horsepower. The heating surface required for an increase in heat transfer above 75 per cent increases very rapidly for each per cent gained in effectiveness; therefore, the accepted economic limit is around 75 per cent.

16-8. Intercooler. The work of compression can be reduced if the isentropic process is replaced by an isothermal process. Figure 16-15 shows

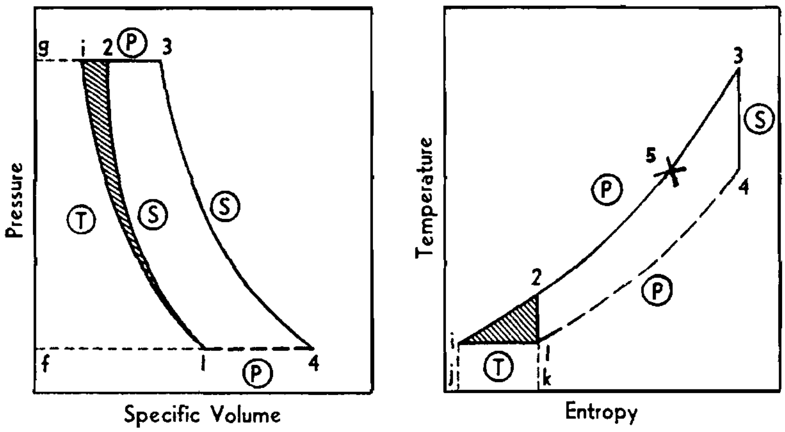


FIG. 16-15. *P-v* **and** *T-s* **diagrams of two ideal open cycle gas turbines, one with an isentropic compression process and one with an isothermal compression process.**

the effect upon the gas turbine cycle of the substitution of an isothermal compression process for the isentropic compression process. The work of compression is reduced by the amount of the shaded area (1*i*21) in the *p-v* diagram. Since the turbine expansion work, process 3 to 4, remains the same in each system, the net work of the cycle with isothermal compression (area 1*i*341) is increased over that of the cycle having isentropic compression (area 12341) by the amount of the shaded area. Therefore, the work ratio (Article 16-11), i.e., the ratio of the net work to the turbine expansion work, is increased.

Although the net work is increased, the replacement of the isentropic compression with an isothermal compression will normally produce less than one per cent change in the thermal efficiency of the open cycle gas turbine when operating at pressure ratios below the range of 16 to 20. The increase in net work in the isothermal case is offset by the increase in the amount of heat, q_S, required to be supplied to the combustion chamber to raise the compressed air to the turbine inlet temperature, state 3. The isothermal compressed air must be raised from state i to state 3, while the isentropic compressed air is raised from state 2 to state 3. However, when the open cycle gas turbine has a *regenerator* in the system, the thermal efficiency of the cycle with the isothermal compression process will be greater than the cycle with the isentropic compression process due to the net work being larger in the isothermal case while the heat added, q_S, to the cycles in the combustion chambers is the same in each case, state 5 to state 3.

Since isothermal compression is not practical, a two-stage compressor with an intercooler is normally used in the actual open cycle gas turbines. However, the greater the number of stages of compression with intercooling between each stage the closer will be the approximation to an isothermal process. The effect of intercooling on the compression work and the equations for a multistage compressor with intercooling may be found in any standard thermodynamics textbook. It has been determined (16-3) that the maximum reduction in power required for a two-stage compressor with intercooling occurs when the total pressure rise is divided into two equal pressure ratios, i.e., $r_{p(1a)} = r_{p(b2)}$ (Fig. 16-16). The pressure of the air entering the intercooler, p_{int}, is

$$p_{int} = \sqrt{(p_1)(p_2)} \qquad (16\text{-}22)$$

The pressure ratio across the two compressors, as shown in Fig. 16-16, is expressed as

$$r_p = r_{p(21)} = \left(\frac{p_{int}}{p_1}\right)\left(\frac{p_2}{p_{int}}\right) = r_{p(1a)} r_{p(b2)}.$$

By substituting $p_1 = \dfrac{(p_{int})^2}{p_2}$ from equation (16-22) in the above equation, it is found that the pressure ratio across the second stage of compression is

$$r_{p(b2)} = \frac{p_2}{p_{int}} = \sqrt{r_p}$$

and therefore, from the above equation

$$r_{p(b2)} = r_{p(1a)} = \sqrt{r_p}. \tag{16-22a}$$

Actual open cycle with intercooling and regeneration. A schematic diagram of an open cycle gas turbine with a regenerator and an intercooler in the system is shown with the T-s diagram of the cycle in

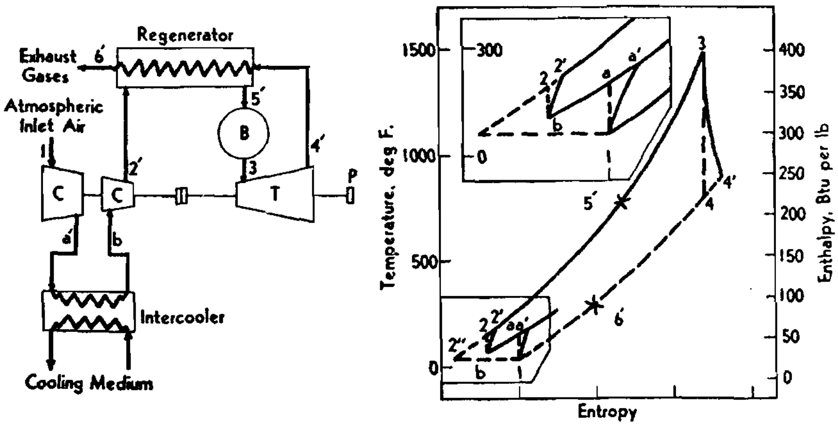


FIG. 16-16. Open cycle gas turbine with regenerator and intercooler.

Fig. 16-16. The compressor work is the sum of the work for the first and second stage of compression, that is

$$wk_C = wk_{C(1a')} + wk_{C(b2')}$$

$$wk_C = \frac{(h_a - h_1) + (h_2 - h_b)}{\eta_C}$$

$$wk_C = (h_{a'} - h_1) + (h_{2'} - h_b) \qquad \text{(Btu/lb)}. \tag{16-23}$$

The enthalpy of the air at the exit from the intercooler, state b, is dependent upon the intercooler effectiveness, η_{int}, and the enthalpies at states 1 and a'. The intercooler effectiveness is the ratio of the actual heat transfer, $h_{a'} - h_b$, to a theoretical heat transfer, $h_{a'} - h_1$, based on T_b being equal to T_1. Therefore,

$$\eta_{int} = \frac{h_{a'} - h_b}{h_{a'} - h_1}. \tag{16-24}$$

The equations for the turbine work and the heat added in the combustion chamber are the same as in the open cycle gas turbine with regeneration, that is

$$wk_{T_u} = h_3 - h_{4'}$$

and

$$q_S = h_3 - h_{b'}.$$

Therefore, the thermal efficiency of the open cycle gas turbine with intercooling and regeneration becomes

$$\eta_t = \frac{wk_{T_u} - [wk_{C(1a)} + wk_{C(b2)}]}{q_S}$$

$$\eta_t = \frac{(h_3 - h_4)\eta_{T_u} - \left[\dfrac{(h_a - h_1) + (h_2 - h_b)}{\eta_C}\right]}{(h_3 - h_2') - (h_{4'} - h_{2'})\eta_X}$$

$$\eta_t = \frac{(h_3 - h_{4'}) - [(h_{a'} - h_1) + (h_{2'} - h_b)]}{h_3 - h_{b'}} \tag{16-25}$$

Intercooling will reduce the work of the compressors and will increase the net work, the power output of the cycle, and the work ratio. It has little effect on the thermal efficiency of an open cycle gas turbine below pressure ratios of 16 to 20. However, a large increase in thermal efficiency over a larger pressure ratio range will be gained by using a combination of intercooling and regeneration in the system. This is shown in Fig. 16-26.

16-9. Reheater. The work of the turbine may be increased if the expansion of the working fluid is made to approach an isothermal process. This is accomplished in an actual operating gas turbine by expanding the working fluid in two or more stages and reheating the gases between stages at constant pressure back up to the maximum permissible turbine inlet temperature. Since there is a great excess of air in the working fluid that can be utilized for the combustion of fuel, reheating is done by inserting a combustion chamber between the two turbine stages as shown in the diagrammatic sketch of Fig. 16-18.

Figure 16-17 shows *p-v* and *T-s* diagrams of an open cycle gas turbine with reheating. The working fluid is expanded in the first stage of the turbine, process $3c'$, heated back up to the permissible turbine inlet temperature by the combustion of fuel in the reheater at constant pressure, process $c'd$, and expanded through the second stage of the turbine, process $d4'$. The work of the turbine and the net work of the cycle is increased by the amount of the shaded area ($c'd4'xc'$). The reheater has little effect on the thermal efficiency since the increase in the net work of the cycle is offset by the increase in the amount of the heat added, q_S, due to the addition of the reheat process, $c'd$.

An examination of the *T-s* diagram of Fig. 16-17 shows that the

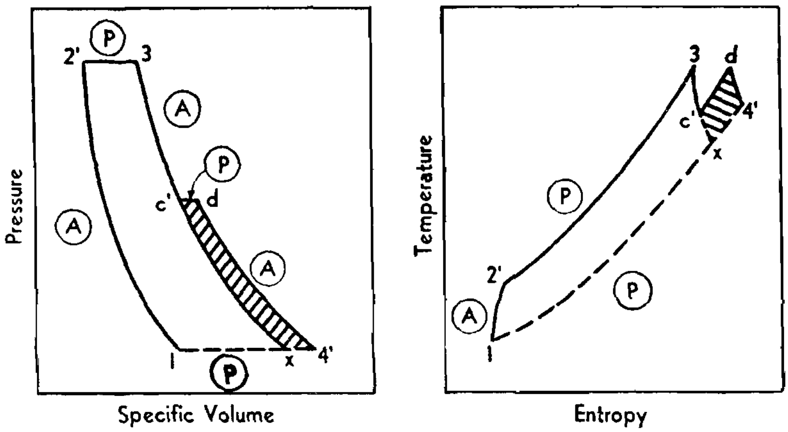


FIG. 16-17. ***P-v*** **and** ***T-s*** **diagrams of an actual open cycle gas turbine with reheating.**

temperature of the exhaust gases leaving the second stage of the turbine, state $4'$, is higher than the temperature would have been if there were no reheater in the cycle, state x. Therefore, when a *reheater* is inserted in an open cycle gas turbine with a *regenerator*, the thermal efficiency of the cycle is increased over that possible with only a regenerator in the cycle.

Actual open cycle with reheating. From the *T-s* diagram of Fig. 16-17, the turbine work is the sum of the work of the first and second stages,

$$
\begin{aligned}
wk_{Tu} &= wk_{Tu(3c')} + wk_{Tu(d4')} \\
&= [(h_3 - h_c) + (h_d - h_4)]\eta_{Tu} \\
&= (h_3 - h_{c'}) + (h_d - h_{4'}). \qquad (16\text{-}26)
\end{aligned}
$$

The total heat added in the cycle is the sum of the heat added in the first and the second combustion chambers (the second combustion chamber is the reheater),

$$q_S = (h_3 - h_{2'}) + (h_d - h_{c'}). \qquad (16\text{-}27)$$

The thermal efficiency of the open cycle gas turbine with reheating then becomes,

$$\eta_t = \frac{[(h_3 - h_{c'}) + (h_d - h_{4'})] - (h_{2'} - h_1)}{(h_3 - h_{2'}) + (h_d - h_{c'})}. \qquad (16\text{-}28)$$

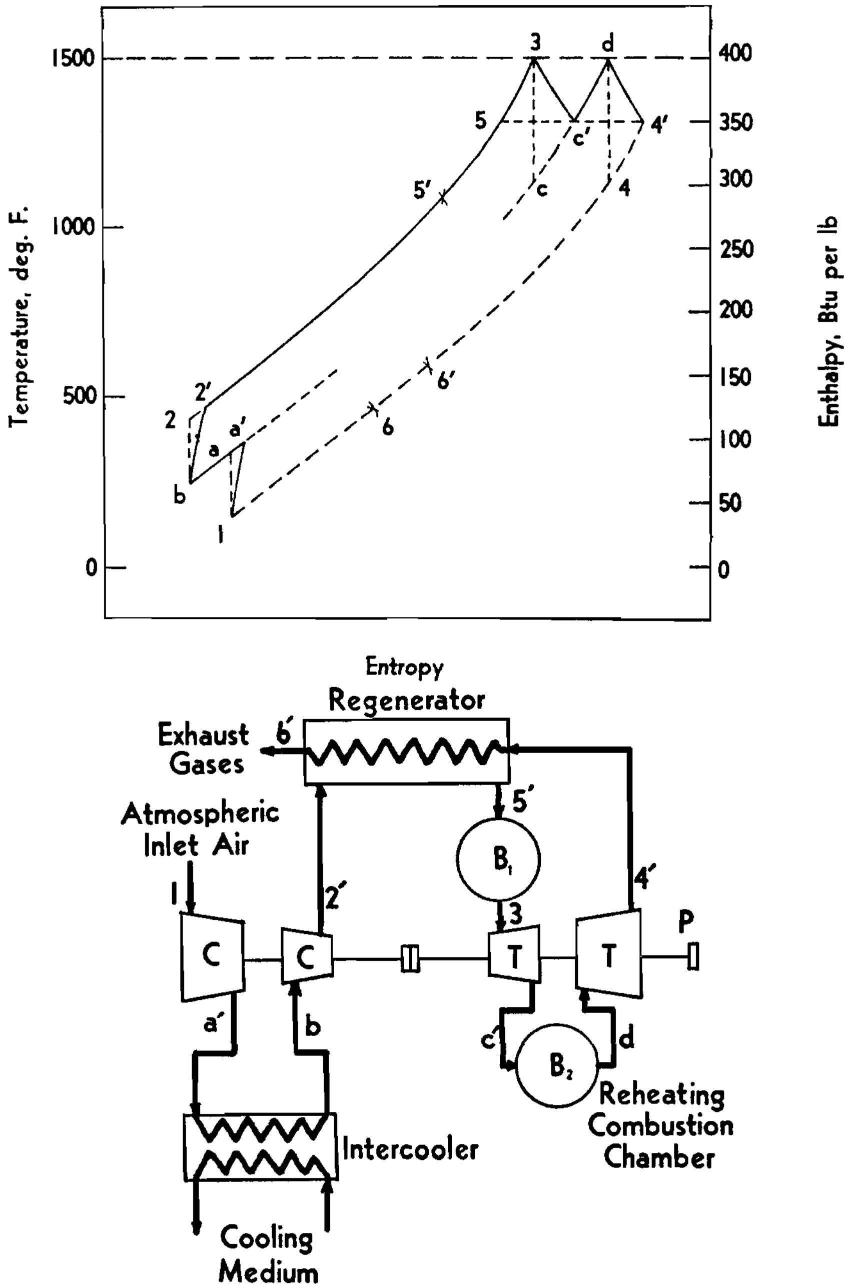


FIG. 16-18. Open cycle gas turbine with regenerator, intercooler, and reheater.

Actual open cycle with reheating and regeneration. Taking Fig. 16-18 and removing the intercooler, the thermal efficiency of an open cycle gas turbine with reheating and regeneration becomes

$$\eta_t = \frac{[(h_3 - h_{c'}) + (h_d - h_{4'})] - (h_{2'} - h_1)}{(h_3 - h_{5'}) + (h_d - h_{c'})} \tag{16-29}$$

Actual open cycle with reheating, intercooling, and regeneration. From Fig. 16-18, the thermal efficiency of the cycle is expressed as

$$\eta_t = \frac{[wk_{Tu(3c')} + wk_{Tu(d4')}] - [wk_{C(1a')} + wk_{C(b2')}]}{q_{S(5'3)} + q_{S(c'd)}}$$

$$\eta_t = \frac{[(h_3 - h_{c'}) + (h_d - h_{4'})] - [(h_{a'} - h_1) + (h_{2'} - h_b)]}{(h_3 - h_{5'}) + (h_d - h_{c'})} \tag{16-30}$$

16-10. Air Rate. In addition to the thermal efficiency which is a measure of the fuel economy, the size of the plant is equally important in many applications, particularly in the field of aviation. For a given duty, the size of a plant is dependent on the rate flow of air in relationship to the useful horsepower output. This is expressed as pounds of air per horsepower-hour and is called the air rate. Air rate, AR, is defined as

$$\text{AR} = \frac{w_a}{\text{hp output}} = \frac{w_a}{w_a(wk_{\text{net}}/2545)} = \frac{2545}{wk_{\text{net}}} \quad \left(\frac{\text{Btu/hp-hr}}{\text{Btu/lb air}}\right)$$

$$= \frac{2545}{wk_{Tu} - wk_C} \quad \left(\frac{\text{lb air}}{\text{hp-hr}}\right) \tag{16-31}$$

The reciprocal of air rate, termed specific power (hp-hr per lb air), is sometimes used; however, air rate will be used in this text.

Air rate is a criterion of the size of the plant, i.e., the lwer the air rate the smaller the plant. From the mechanical and metallurgical standpoint, the lowering of the air rate results in turbines of smaller physical dimensions with a more nearly uniform temperature distribution. Any means by which the physical dimensions can be reduced and the inherent distortions minimized are steps toward greater reliability of the gas turbine.

1. *Effect of turbine inlet temperature and pressure ratio on the air rate.* Figure 16-19 shows that as the turbine inlet temperature is increased, the air rate is decreased (16-4). As the pressure ratio is increased, the air rate will decrease to a minimum value, and any further increase in the pressure ratio will increase the air rate.

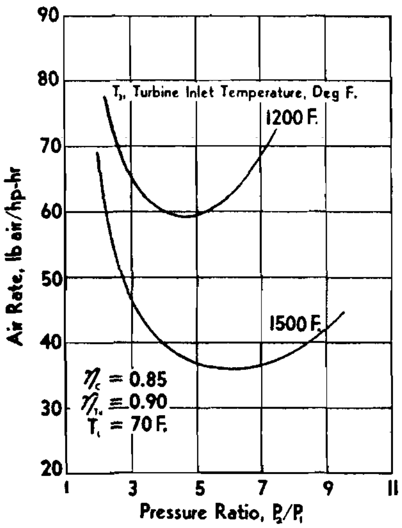


Fig. 16-19

FIG. 16-19. Effect of turbine inlet temperature and pressure ratio on the air rate of simple open cycle gas turbine (reference 16-20).

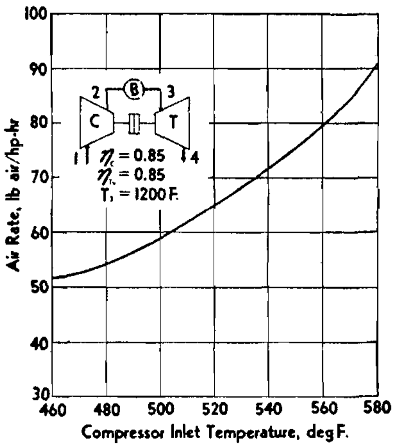


Fig. 16-20

FIG. 16-20. Effect of the compressor inlet temperature on the air rate of a simple open cycle.

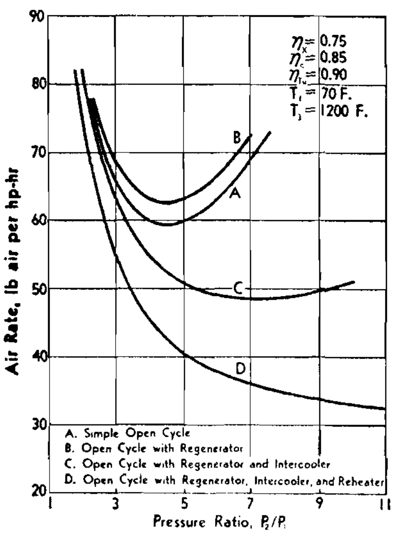


Fig. 16-21

FIG. 16-21. Effect of the additional elements on the air rate of an open cycle (reference 16-20).

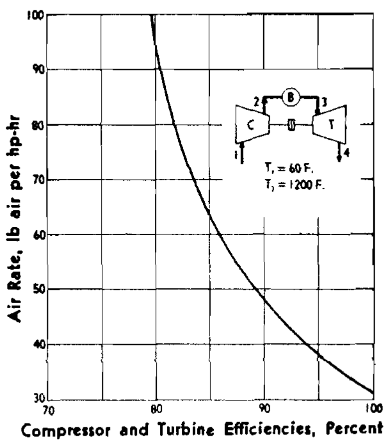


Fig. 16-22

FIG. 16-22. Effect of the compressor and turbine efficiencies on the air rate of a simple open cycle.

2. *Effect of compressor inlet temperature on the air rate.* An increase in the compressor inlet air temperature increases the compressor work. Since the turbine output is not affected by variation in the compressor inlet temperature, the net work of the gas turbine is decreased. An increase in compressor inlet temperature will decrease the net work and hence will increase the air rate as shown in Fig. 16-20.

3. *Effect of regeneration, intercooling, and reheating on the air rate.* The effect of the addition of elements to a simple open cycle on the air rate is shown in Fig. 16-21. A comparison of the curves shows that one of the main advantages of intercooling and reheating is to reduce the air rate, particularly at higher pressure ratios, which results in a reduction in the size of the gas turbine and the elements therein. Regeneration alone, however, will increase the air rate a small amount.

4. *Effect of compressor and turbine efficiencies on the air rate.* An increase in the compressor and turbine efficiencies will decrease the air rate. Figure 16-22 shows the reason why the present day gas turbines with their relatively high component efficiencies are small in size compared to the early gas turbines.

16-11. Work Ratio. A good gas turbine design not only requires that the gas turbine have a high thermal efficiency and low air rate, but that it also have a high work ratio. The work ratio is defined as the ratio of the net or useful work to the turbine expansion work. Hence the work ratio, α, is

$$\alpha = \frac{wk_{net}}{wk_{T_u}} = \frac{wk_{T_u} - wk_C}{wk_{T_u}}$$

$$\alpha = \frac{[(iwk_{T_u})(\eta_{T_u})] - [iwk_C/\eta_C]}{(iwk_{T_u})(\eta_{T_u})}. \tag{16-32}$$

The work ratio may be also used as a guide in the determination of the size of the gas turbine.

In addition to being an aid in measurement of the effective utilization of the component parts, the work ratio also serves as an indicator of the sensitivity of the plant to the decrease or deterioration of the component efficiencies, i.e., the compressor and turbine efficiencies. The component efficiencies will decrease during part load performance from their optimum values and they will also be lowered from their design values during continuous operation. If the work ratio is high the variations in the compressor and turbine efficiencies will have less effect on the thermal efficiency of the cycle than if the work ratio is low. Furthermore, a plant with a high work ratio, i.e., not as sensitive

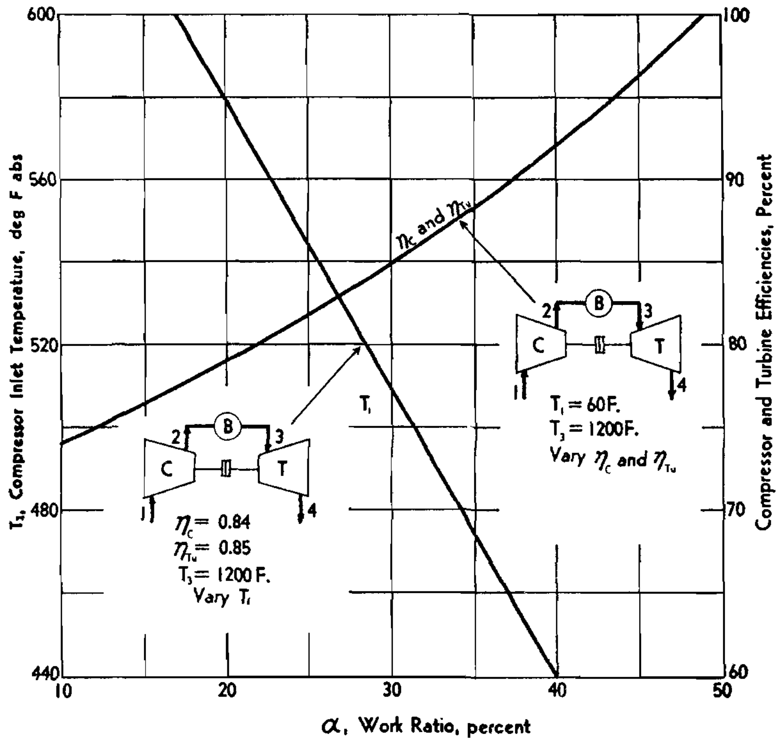


FIG. 16-23. Effect of the compressor inlet temperature and compressor and turbine efficiencies on the work ratio of a simple open cycle gas turbine.

to component efficiency changes, will have a higher part load performance efficiency than a plant with a low work ratio.

A gas turbine plant with a work ratio of 40 per cent will develop 400 horsepower useful work for every 1000 horsepower output of the turbine. The remaining 600 horsepower would be required for the compressor and the auxiliaries. A gas turbine plant with a work ratio of 50 per cent would develop 500 horsepower useful output for every 1000 horsepower output of the turbine.

1. *Effect of compressor inlet temperature on the work ratio.* The work ratio is increased as compressor inlet temperature is decreased, Fig. 16-23, due to the decrease in the power required by the compressor while the turbine output remains the same.

2. *Effect of compressor and turbine efficiencies on the work ratio.* Figure 16-23 indicates the great effect the compressor and turbine efficiencies have on the work ratio. A slight change in the element efficien-

cies has a large effect on the work ratio indicating the sensitivity of the plant to the compressor and turbine efficiencies.

3. *Effect of turbine inlet temperature and pressure ratio on the work ratio.* Figure 16-24 shows that the work ratio is decreased as the pressure ratio is raised and that the work ratio is increased as the turbine inlet temperature is raised.

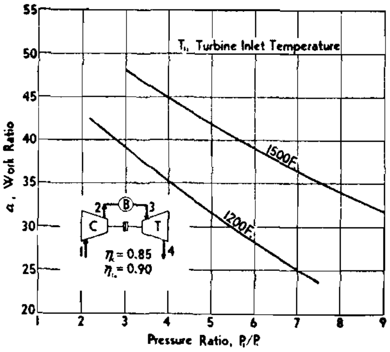


FIG. 16-24. Effect of the turbine inlet temperature on the work ratio of a simple open cycle gas turbine (reference 16-20).

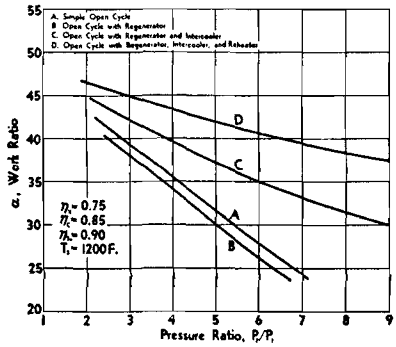


FIG. 16-25. Effect of the additional elements on the work ratio of an open cycle.

4. *Effect of regeneration, intercooling, and reheating on the work ratio.* Figure 16-25 shows the effect of the addition of elements to a simple open cycle on the work ratio. A comparison of the curves shows that the intercooler and reheater not only improve the work ratio but also decrease the slope of the curve to give greater improvement as the pressure ratio is raised. This shows that the intercooler and reheater decrease the sensitivity of the gas turbine to variations in pressure ratio and, therefore, produce better gas turbine characteristics at part-loads.

16-12. Effect of Regeneration, Intercooling, and Reheating on Performance. The additional cost, weight, and complexity of the gas turbine engine caused by the insertion of a regenerator, an intercooler, and a reheater into the system is more than offset, in many applications, by the improvement in the performance. The effect of these elements on the thermal efficiency, power output, air rate, and work ratio are interrelated.

The regeneration by the exhaust gases is one method of increasing the thermal efficiency of an open cycle gas turbine. A regenerator which is a comparatively inexpensive piece of equipment with a rela-

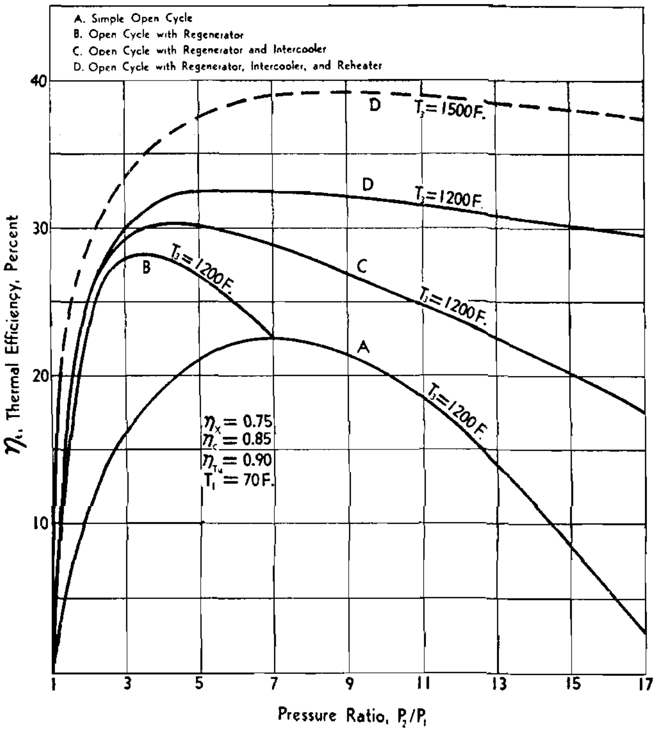


FIG. 16-26. Effect of the additional elements on the thermal efficiency of an open cycle gas turbine.

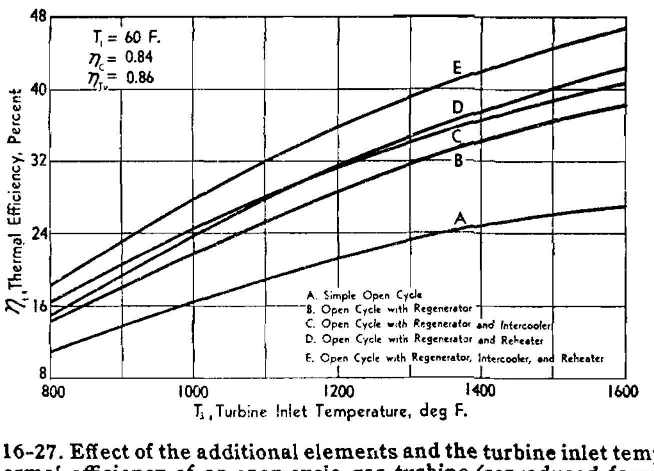


FIG. 16-27. Effect of the additional elements and the turbine inlet temperature on the thermal efficiency of an open cycle gas turbine (reproduced from *Journal of American Society of Naval Engineers*, Inc., August 1947, ref. 16-21).

tively low maintenance will reduce the fuel consumption, but the addition of this element will increase the weight and overall dimensions of the plant. While the regenerator alone will increase the thermal efficiency, it will do so only at relatively small pressure ratios as shown by curve *B* of Fig. 16-26. The maximum efficiency of the open cycle gas turbine with regeneration occurs at a much lower pressure ratio than the maximum efficiency of the simple open cycle, curve *A*. Figure 16-28, which shows the *T-s* diagrams of two gas turbines with different

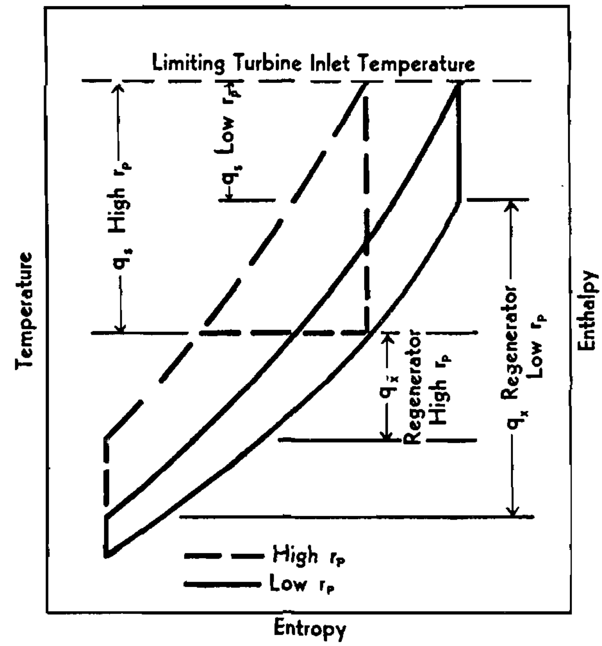


FIG. 16-28. Effect of increase in pressure ratio on the addition of heat to an open cycle gas turbine with regenerator.

pressure ratios but with the same permissible turbine inlet temperature, demonstrates the reason why regeneration alone in the cycle is the most effective at low pressure ratios. As the pressure ratio is increased, the turbine exhaust temperature decreases while the compressed air temperature increases, and, therefore, the heat available for regeneration is reduced. This means that as the pressure ratio is increased the heat added in the form of fuel in the combustion chamber must be increased. When the pressure ratio of curve *B*, Fig. 16-26, is increased beyond the point where curve *B*, crosses curve *A*, the turbine exhaust temperature is lower than the compressed air temperature and regeneration then will become negative, i.e., the compressed air will then be heating up the exhaust gases.

The disadvantage of restricting an open cycle gas turbine with regeneration to low pressure ratios can be eliminated by the addition of an intercooler or a reheater or a combination of both to the gas turbine cycle. Intercooling and reheating alone in a system will have little effect on the thermal efficiency at relatively low pressure ratios; but in combination with a regenerator, the thermal efficiency is increased and the range of pressure ratios with improved thermal efficiency is

TABLE 16-1

EFFECT OF INTERCOOLING AND REHEATING ON GAS TURBINE CYCLE PERFORMANCE

Cycle	η_t Per cent	Air Rate AR lb/hp-hr	Work Ratio α Per cent	optimum $r_p = p_2/p_1$
Simple open cycle	14.6	89	24.4	5
Open cycle with regenerator	25.8	91	30.1	2.6
Open cycle with regenerator and intercooler	26	71	33.8	3.5
Open cycle with regenerator, intercooler, and reheater	28.9	51	35.8	6

also increased as shown by curves *C* and *D* of Fig. 16-26. The flattening out of the curves *C* and *D* illustrates that the part load performance of a gas turbine will be improved by intercooling and reheating. These curves demonstrate that the maximum efficiency and best part load performance of an open cycle gas turbine occurs when all three elements, regenerator, reheater, and intercooler are added to the system.

Figure 16-27 shows the effect of turbine inlet temperature as well as the effect of the addition of elements to the system on the thermal efficiency of the cycle. The increase in the thermal efficiency of each system with an increase in the turbine inlet temperature is readily apparent. The curves reveal that the rate of improvement of the thermal efficiency is the highest with the most complex cycle.

The addition of an intercooler or a reheater or a combination of both will increase the useful power output of a cycle (Articles 16-8 and 16-9). The influence of these two elements on the air rate and the work ratio of a cycle are shown in Figs. 16-21 and 16-25 respectively. Intercooling and reheating will lower the air rate which will reduce the size of the plant. They will also increase the work ratio which will make the plant less sensitive to the changes in compressor and turbine efficiencies.

Table 16-1 gives a comparison of the optimum performance of four different systems. The table was computed using compressor, turbine, and regenerator efficiencies of 0.80, turbine inlet temperature of 1200° F, and compressor inlet temperature 60° F.

Water injection. Another method used to improve the performance of the gas turbine is to inject water into the working air at the entrance to the compressor. The compressed air is cooled by the latent heat of vaporization of the water. The total mass flow of the working medium is increased by the mass of the injected water. The increase in the flow rate through the turbine increases the power output. In this manner the net work or useful power output of the cycle is increased, the work ratio is increased, and the air rate is lowered. The water injection will cause an increase in the range of optimum thermal efficiencies similar to that accomplished by intercooling.

Water injection is most commonly used as a power boost for take-off and emergencies with jet propelled aircraft (Article 17-11). However, the power boost is of short duration in aircraft due to the weight and storage requirements of the water. The marine or land based gas turbines may have long duration or continuous water injection for power boost. However, water, particularly if it has impurities, may cause corrosion or deposits on the blades which will have a detrimental effect on the performance and maintenance. Other liquids, such as alcohol, may be injected to obtain a power boost, but only at the expense of an increase in operating costs.

16-13. Basic Gas Turbine Components.

Compressors. The design theory and rigorous analysis of the flow through the various types of compressors are beyond the scope of this text. The following limited discussion, therefore, is an endeavor to impart only a basic concept of compressor operation.

The primary requirements of a gas turbine compressor are that it handle a relatively large volume of air, or working media, at the highest possible efficiency. The principal compressors now being used in gas turbine engines comprise the following types:

(1) positive-displacement
(2) centrifugal flow
(3) axial flow

The Lysholm, an example of a *positive-displacement* compressor, probably will not be used to a great extent in the future unless improvement is made in the design, Fig. 16-29. In general, it has the greatest stability range and its efficiency is comparable with that of the centrifugal compressor. However, the possibilities of its use in the Navy are small because of the very high noise level and limited rotative speed which requires either additional turbine stages or a reduction gear. It has the further disadvantages of complexity of production with

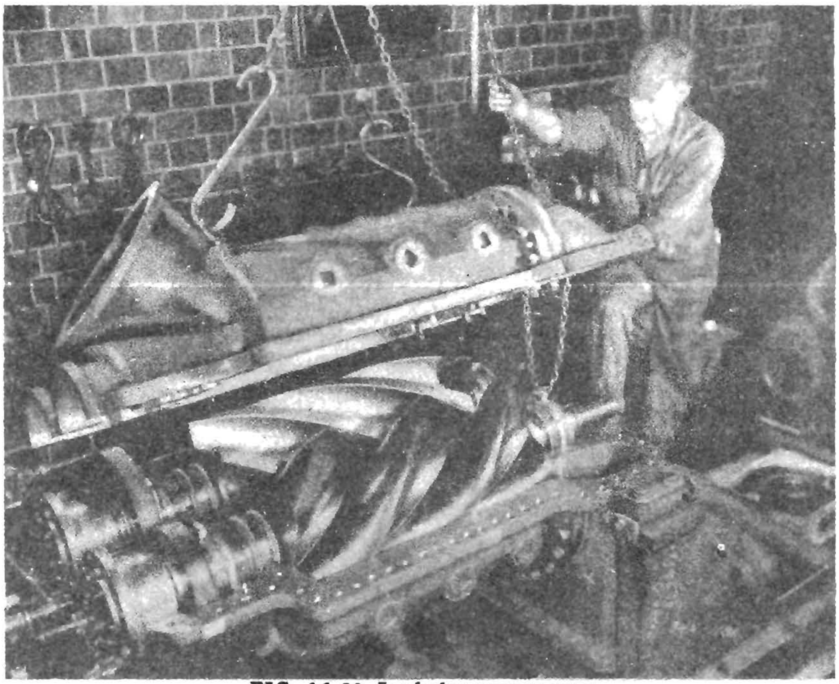


FIG. 16-29. Lysholm compressor.

higher cost, limitation on maximum obtainable pressure ratio, and high specific weight.

The free piston gas generator, which is a combination of compressor and combustion chamber, will be discussed in Art. 16-16.

The *centrifugal* compressor is comprised of two major parts, the impeller, or rotating component and the diffuser, Fig. 16-30. The air, or working media, enters the compressor at the hub of the centrifugal impeller and it then moves radially outward through the impeller and into the diffuser. The impeller converts the mechanical energy, available to the compressor, into kinetic energy, plus heat due to friction, in the working media. The diffuser then transforms the kinetic energy in the working media, or air, into pressure energy in accordance with Bernoulli's principle. The flow through the diffuser is subject to frictional losses as well. Also, because the working media leaves the impeller radially and must normally be turned 90° to enter the combustion chambers or regenerator, additional friction losses are encountered. In some designs the working media is turned 90° upon leaving the impeller and only then enters the diffuser section of the compressor.

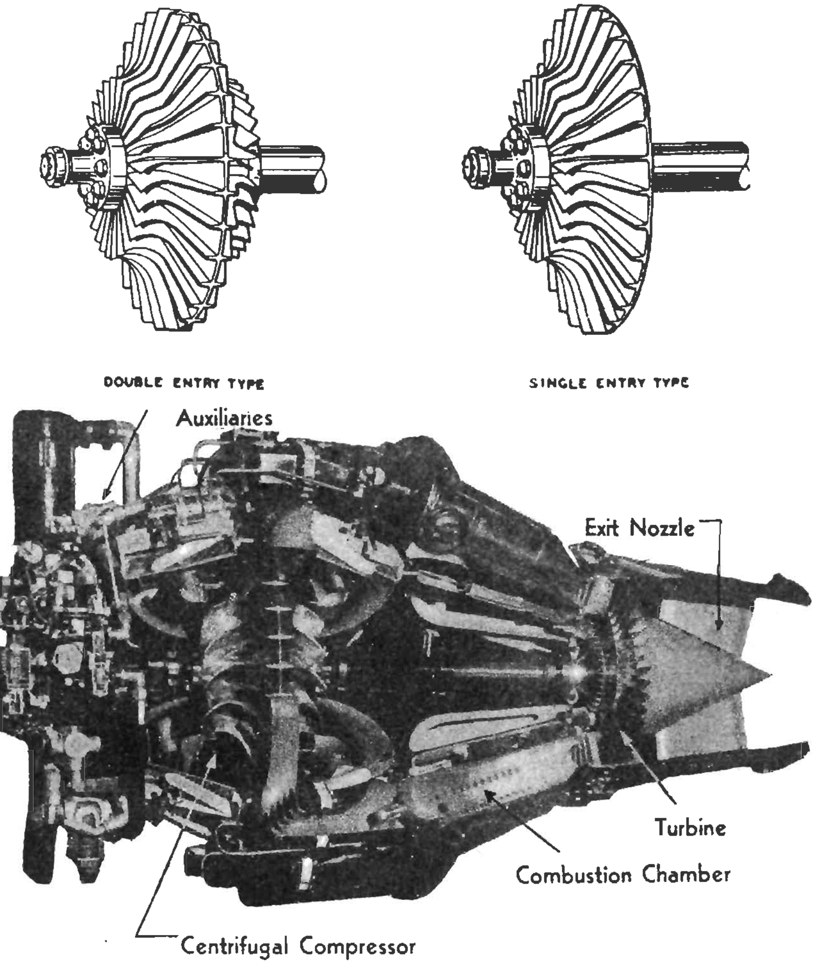


FIG. 16-30. Centrifugal flow compressor.

From a practical design standpoint, it is not feasible to have supersonic inlet velocities in the diffuser. This limitation on impeller exit velocities is reflected in the choice of impeller blade shapes (i.e., bent backward, forward or straight radial), and compressor rpm. A further consideration in the choice of impeller blade shapes is one of manufacturing costs and rotative stress limits. The result of these considerations is that the usual choice of impeller blading in the centrifugal compressor is a compromise resulting in a straight radial blade shape, Fig. 16-30.

In general, the *centrifugal compressor*, as compared to the axial flow, is more rugged, simpler, relatively insensitive to surface deposits, has a wider stability range, less expensive, and attains a higher pressure ratios per stage. However, the efficiency is lower, the diameter greater, and it is not as readily adaptable to multi-staging. The single stage compressors for use in industry may obtain efficiencies from 80 to 84% at pressure ratios between 2.5 and 3, while for aircraft use, pressure ratios are between 4 and 4.5, with efficiencies in the range of 76 to 81%.

The important characteristics of the *axial flow* compressor are its high peak efficiencies, adaptability to multi-staging to obtain higher overall pressure ratios, high flow rate capabilities, and relatively small diameter. However, the axial flow compressor is sensitive to changes in air flow and rpm, which result in a rapid drop off in efficiencies, i.e., the stability range of speeds for good efficiencies is small. These latter characteristics limit the part load capabilities of this type of compressor and are considered undesirable in some installations.

The axial flow compressors consist of a series of rotor-stator stages. The rotor, or moving component, comprises a series of blades that move relative to the series of stationary blades called the stator. See Fig. 16-31. The rotor blades transmit the mechanical energy into kinetic energy in the working media or air.

Compression is accomplished in both the rotor and stator blade passages by converting the kinetic energy relative to the blades into pressure energy (i.e., continually diffusing it from a high velocity to a lower velocity with a corresponding rise in static pressure). It should be noted that the diffusion process through the rotor is dependent upon the velocity of the air, or working media, relative to the rotor, and that through the stator is effected by the movement of air, or working media, relative to the stator blades.

Thus, as Fig. 16-32 shows, the velocity of the air or working media entering the rotor V_{r_1}, is the velocity relative to the rotor, and is a vectorial resultant of the actual velocity of the working media, V_1, and the rotational velocity of the rotor, V_b. The rotor blades turn the air or working media through an angle, and thus impart kinetic energy to the working media such that the actual velocity of the working media is increased. The velocity of the working media relative to the rotor blades, however, has decreased, because of the increase in area through

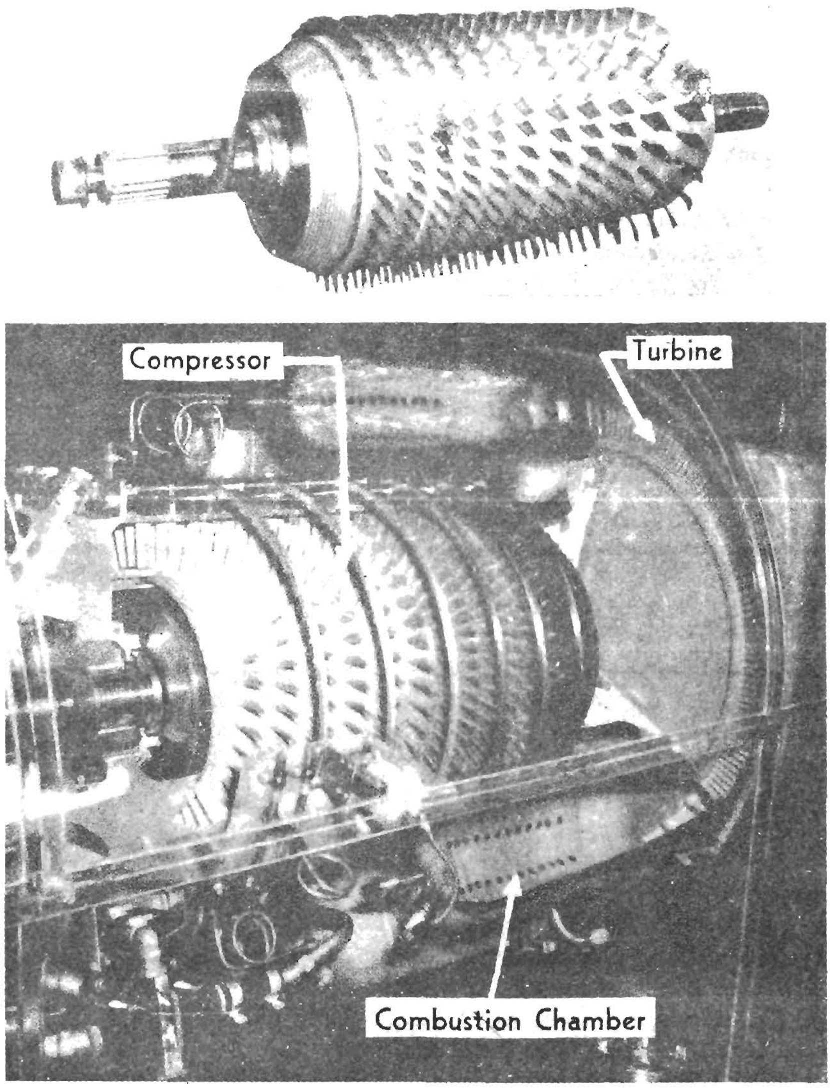


FIG. 16-31. Axial flow compressor.

the rotor blades, and hence, increases the static pressure in the normal diffusion process.

The velocity of the air, or working media, entering the stator is a vectorial resultant of the velocity of the working media relative to the rotor blades, V_{r_2}, and the rotational velocity of the rotor, V_b, as shown

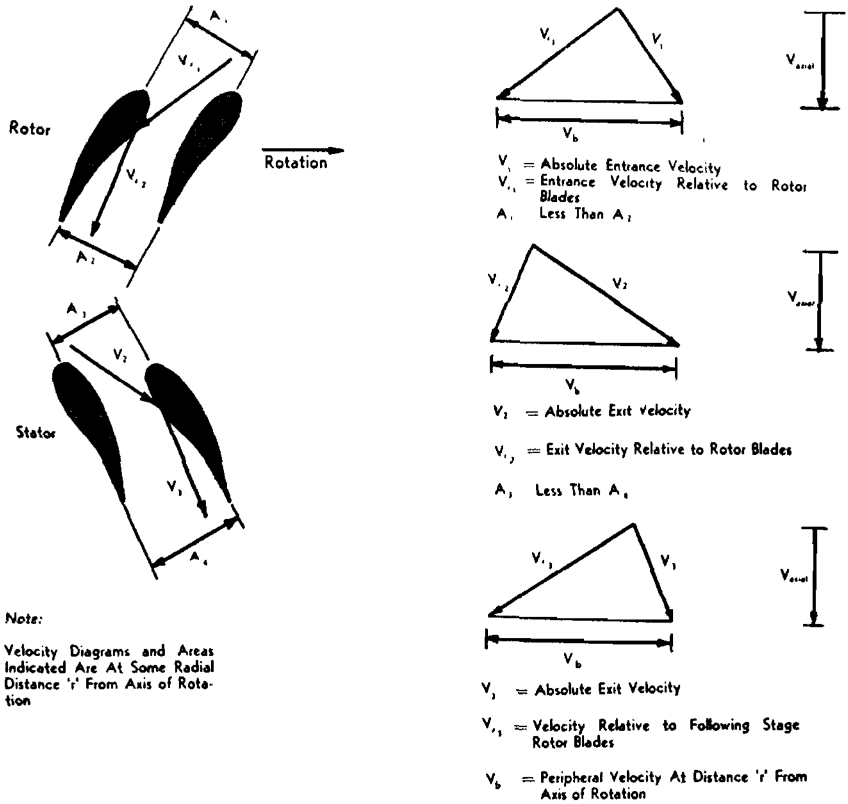


FIG. 16-32. Axial flow compressor. Rotor-stator blade sections and associated velocity vector diagrams.

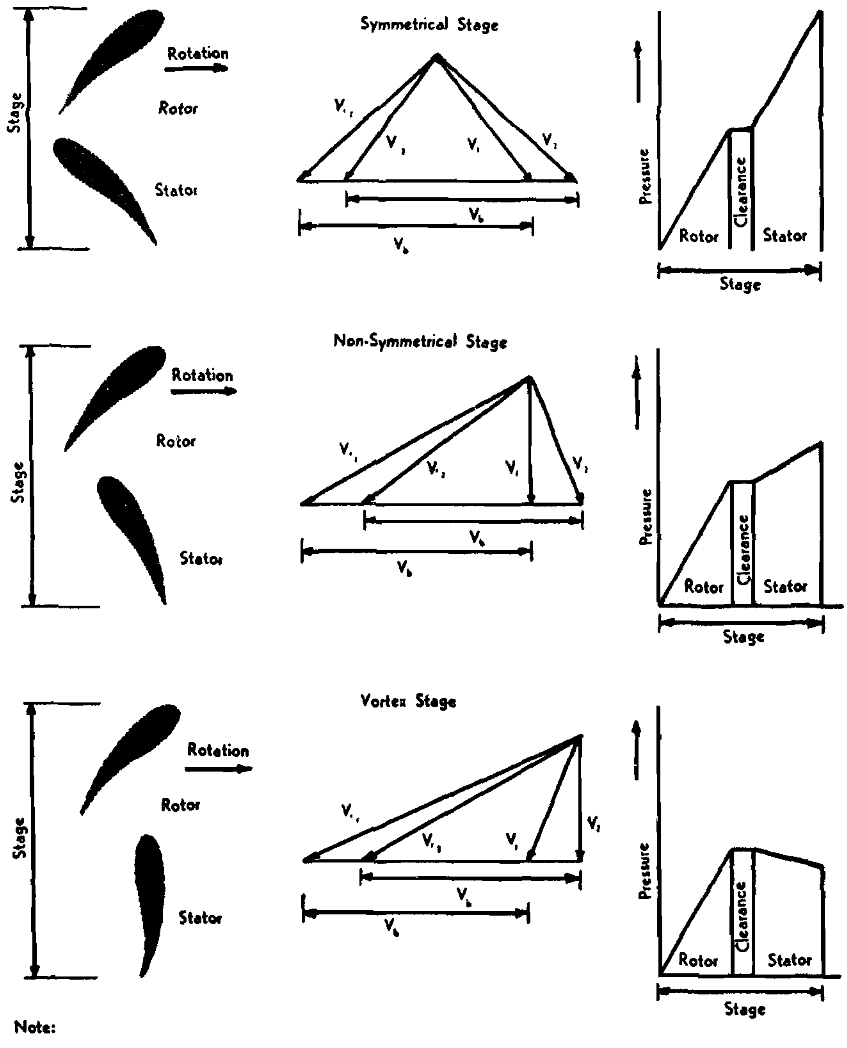
in Fig. 16-32. The stator blades change the direction of flow of the working media and, because of the increasing area through the stator blades, the static pressure is again increased in the normal diffusion process. This type of compressor is normally designed so as to provide a constant axial flow of the working fluid, i.e., the V_{axial} throughout the compressor is essentially constant (See Fig. 16-32).

The present day compressors are designed on the basis of airfoil aerodynamic theory that has been modified and substantiated by NACA cascade test information (i.e., interaction effect of adjacent blades).

The axial flow compressors consist of three basic design types:

(1) *symmetrical flow*
(2) unsymmetrical flow
(3) Vortex flow

The three basic types are classified according to the relative pressure rises through the rotor and stator blades and depicted by the air, or working media flow velocity vector diagram through each stage, Fig. 16-33.



Note:

V = Absolute Velocities

V_r = Relative Velocities

V_b = Rotational Velocity

Diagrams depict velocities and pressures at some distance 'r' from the axis of rotation.

FIG. 16-33. Three types of *Axial Flow* Compressor stages (not to scale).

The symmetrical stage is defined as one in which the pressure rise through the moving blades is equal to the pressure rise through the stationary blades at any given radial distance from the compressor axis. The velocity triangle for the stationary blades is thus the mirror image of the velocity triangle for the moving blades at any radius as shown in Fig. 16-33.

The unsymmetrical stage is defined as one in which the ratio of the pressure rise through the rotor blades and stator blades is constant along the blade length. However, the pressure rise through the rotor is not equal to the pressure rise in the stator at a given radius. These conditions result in the unsymmetrical velocity diagram as shown in Fig. 16-33.

The vortex stage is one in which the angular momentum acting on a particle of fluid, in the radial direction, is constant. This satisfies the flow requirement that $V_{b_r} \times r = \text{constant}$ at all points along the blade length, where V_b is the tangential velocity at radius r. Theoretically, the particle is in radial equilibrium, because the centrifugal force acting on the fluid particle is balanced by a radial pressure differential acting toward the axis of rotation. This type of flow, theoretically, minimizes tip losses and approaches two dimensional flow which contributes to very high efficiencies. However, as shown in Fig. 16-33, the absolute velocity increases through the stator blading in the vortex stage, and results in a pressure drop rather than a pressure rise through the stator. This condition results in a comparatively small pressure rise per stage, and for a given pressure ratio, requires a greater number of stages. The vortex stage is a particular type of unsymmetrical stage in which the absolute velocity leaving the moving blades is axial.

A comparison of the approximate number of stages required by the aforementioned types, to obtain a given pressure ratio, say of six, are 14, 24, and 31 for the symmetrical, non-symmetrical and vortex stages, respectively.

It should be noted that it is possible to combine the high efficiency of the vortex stage with the high pressures obtained with the symmetrical or non-symmetrical type stage design in order to obtain the most efficiently performing compressors. This type of combination is present day design practice.

Supersonic axial-flow compressor. A supersonic axial-flow compressor is a compressor in which the air flow is supersonic at some station or stations in the compressor. The control of the location of the supersonic flow and hence the shock wave, and its affect on the flow pattern through the compressor is the problem in the design and construction of the compressor. The potentialities of the super-sonic axial flow compressor lie in the sizeable pressure increase that is available across the shock waves as well as in the weight and size reduction.

The development of the super-sonic axial flow compressor is being pursued extensively at the present time by the N.A.C.A. and industry in an effort to realize the aforementioned potentialities.

The axial flow supersonic compressors are classified in accordance with the type and location of the shock wave. These are:

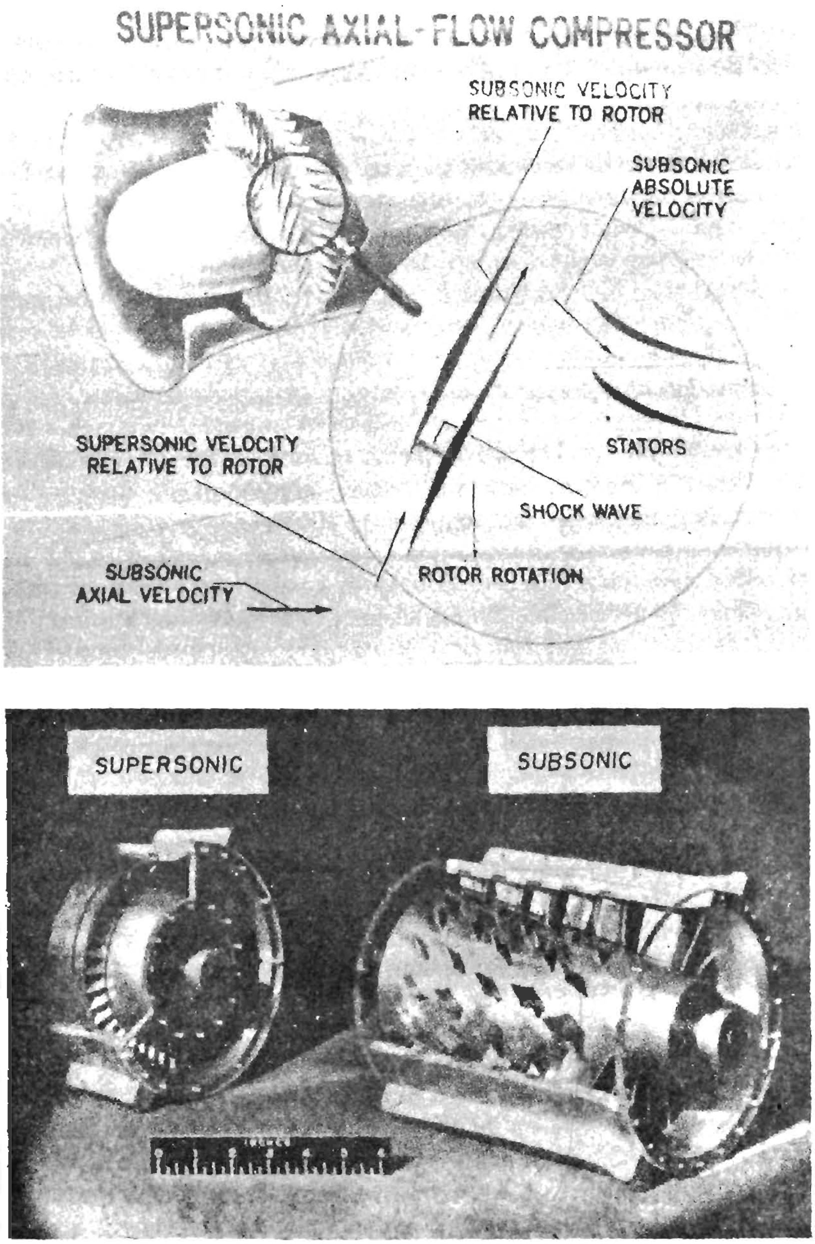
(1) Normal shock in the stator
(2) Normal shock in the rotor
(3) Combination of above
(4) Weak oblique shocks in place of the normal shocks

The compression or pressure rise is primarily obtained across the shock wave produced and is due to the pressure gradient phenomenon of the shock wave itself. The problem, as defined previously, is that of the control of this shock wave phenomenon. The correct velocities and hence, shock wave positions, are obtained, in each case, by suitable blade angles, shapes, rotor rotational speeds and stage inlet velocities.

The supersonic axial flow compressor in which the normal shock wave is produced in the stator has subsonic entrance velocities into the rotor. The velocity in the stator entrance is supersonic, and is obtained by having large turning angles through the rotor blading, which is a source of possible flow separation and its associated problems of flow pattern disturbances. To avoid these problems associated with the large turning angle through the rotor blading the entrance velocity to the rotor must be low. Therefore, this low velocity relative to the rotor imposes a very large restriction on the possible compressor design range.

The shock-in-stator compressor is superior to the shock-in-rotor compressor in pressure ratio, efficiency, lower rotor rpm, ease of construction, and control of the shock phenomenon. The range of theoretical pressure rise per stage at the present time for this type of compressor is 1.4 to 2.1.

The supersonic axial flow compressor in which the normal shock is produced in the rotor requires a high rotative velocity of the rotor. See Fig. 16-34. However, smaller turning angles in the rotor blading are required, which minimizes the flow separation problems encountered in the shock-in-stator supersonic compressor. As indicated in Fig. 16-34, the entrance velocity to the stator is subsonic. Theoretical analysis made concerning the shock-in-rotor supersonic compressor indicates a lower efficiency range than for that of the shock-in-stator



(b)

FIG. 16-34. (a) Schematic diagram of supersonic axial flow compressor. (b) Relative proportions of subsonic and supersonic compressors of equal performance. (Courtesy of *Journal of Aero. Sciences*, A. Silverstein, April, 1949.)

type. Pressure ratios from 1.0 to 2.0, however, can be expected in the shock-in-rotor supersonic compressor. Also, in the shock-in-rotor compressor the cascade design is not so critical, nor does it require a subsonic diffuser between stages.

There is very little known, at the time of this writing, concerning the combination of the shock-in-stator and shock-in-rotor and weak oblique shock types of compressors, except that they are more complicated than the first two types discussed in this article.

It should be noted that to date only experimental models of supersonic compressors have been built, and that their development is continuing.

Combustion chambers. Much research and experimentation is still being conducted to improve the design of combustion chambers for gas turbines. The problems involved are quite different from those encountered in the combustion chambers of other prime movers due to the high air velocities required (150 to 400 ft/sec) to keep the combustion chamber size within reasonable limits and due to the high air-fuel ratios necessary (50:1 to 250:1) to keep turbine inlet temperatures down to permissible limits. These two factors tend to make it difficult to start and maintain the combustion in open cycle gas turbine engines.

In open cycle gas turbines fuel is introduced into the combustion chamber under pressure and is mixed with the primary air so that the air-fuel ratio is close enough to the stoichiometric ratio to support combustion (Fig. 17-9). The gases of combustion are cooled by mixing them with the balance of the air flow, called secondary air, which flows between the inner and outer shells of the combustion chamber and aids in the cooling of the chamber. Therefore, in addition to the combustion problem there is the problem of thoroughly mixing the cool air and the hot products of combustion. Since the maximum allowable temperature of the final mixture is governed by the permissible turbine inlet temperature, it is important that the temperature of the final mixture be uniform across the gas front leading to the turbine blades in order not to produce a localized hot spot on the blades.

The performance criteria of the combustion chamber are (1) low pressure loss, (2) high combustion efficiency, and (3) good flame stability. Low pressure losses are extremely desirable from the point of view of the overall thermal efficiency. High combustion efficiency is necessary for high over-all thermal efficiency. Low combustion efficiency may indicate that all the fuel is not being burned in the combustion chamber but continues burning in the turbine and exhaust. This late burning cannot be tolerated because of its effect on turbine blade life and heat loss due to unburned constituents being discharged. Flame stability implies steady and continuous flame. This problem has been serious due to the phenomena of resonant or pulsating combustion, and due to blowouts or flameout, where the flame is blown out the exit of the combustion chamber and is thereby extinguished (Fig. 17-8). This phenomenon is particularly serious in aircraft applications.

In addition to the three performance criteria, the main requirements for a good combustion chamber are: (1) low carbon deposit in the combustion chamber, turbine, and regenerator; (2) low weight and frontal area; (3) reliability and serviceability with reasonable life; and (4) thorough mixing of the cold air with the hot products of combustion to give uniform temperature distribution so that one section of the turbine blades will not have too high a temperature compared to the other sections. Since some of these requirements call for conflicting design features, the final design is a compromise to obtain the most satisfactory results. Furthermore, since there are so many variables and phenomena involved, the design of combustion chambers has to be based on experimental data.

The types of construction of the combustion chambers in use (Fig. 17-9) are in general: (1) tubular or "can" counterflow (Fig. 16-31); (2) tubular or "can" straight-through flow (Fig. 16-30); and (3) annular parallel flow. Although theoretically the annular chamber possesses advantages over the "can" type, this has not been realized in practice. Also, for test and for replacement of burned out chambers, the "can" type is cheaper and more practical. For these two reasons, the "can" type predominates in current practice.

The combustion chamber in the open cycle gas turbine engine is the most efficient component of the gas turbine. Efficiencies of between 95 and 98 per cent are obtained over a fairly large operating range.

The combustion chamber of a closed cycle gas turbine engine (Art. 16-15), is actually a heat exchanger. The heat added to the working media, air or gases of higher density, must be supplied through a heat exchanger from an external source. The working media is thus not contaminated with the products of combustion. The problem has been to design an efficient heat exchanger of a practical size, capable of supplying the heat addition required. However, cheaper fuels such as soft coal may be used in the heat exchanger of a closed cycle gas turbine.

Figure 16-40 shows the semi-closed cycle gas turbine used by the Sulzer Brothers in which the combustion chamber combines the features of the open cycle combustion chamber and the closed cycle heat exchanger. The working media, air in this case, must be replenished because part of it has been used in the process of combustion. However, the air heater is smaller than the heat exchanger for the closed cycle gas turbine.

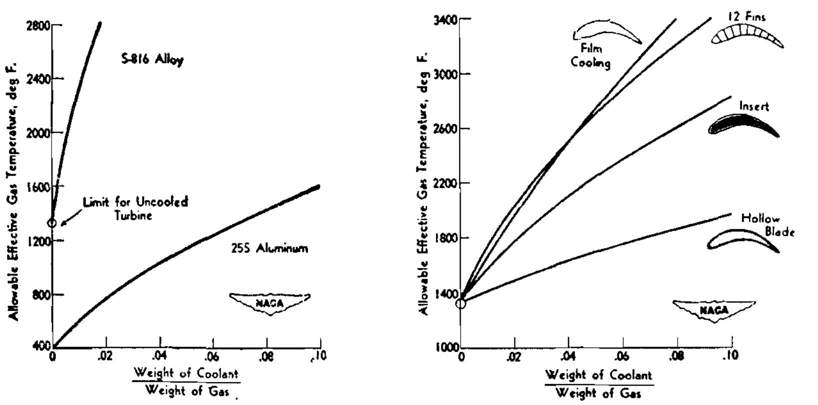
Turbine. The basic theory of the design and evaluation of the operating characteristics of the turbines used in the gas turbine cycles is similar to that of the turbines used in the steam plant. The turbine using gas differs from the turbine using steam in the blading material, the means for cooling the bearings and highly stressed parts, the thermal distortion due to higher temperatures, and the high ratio of blade length to wheel diameter to accommodate large gas flows. The axial flow types of turbines in use are the reaction, the impulse, and a combination of the two. The radial flow type (Fig. 16-48) will probably be used only for auxiliary machinery such as auxiliary fire pumps requiring powers of less than 300 hp. The radial turbine is similar in construction to the centrifugal flow compressor and the gases of combustion are expanded radially inward toward the shaft.

The basic requirements for the turbines are light weight, high efficiency, ability to operate at high temperatures for long periods, reliability, and serviceability. The determination of blading material depends on the stress-rupture and creep characteristics of the various blading materials, in combination with mechanical and thermal stresses, resistance to mechanical and thermal shock, and resistance to corrosion and vibration. The stresses are only approximately known at the present time and much research and experimentation is being conducted along this line. In general, the operating life of the turbine blading is accepted as the governing factor in the determination of the gas turbine plant life, i.e., the number of hours of operating service life will depend on the turbine inlet temperature, the higher the temperature the lower the life.

The great effect of the permissible turbine inlet temperature on the thermal efficiency of the gas turbine is shown in Fig. 16-5. The effect of the turbine inlet temperature on performance will be discussed later in the chapter. Since the maximum temperature obtainable in gas turbines using hydrocarbon fuels is of the order of 3500° to 4000° F and since the materials now being used limit the temperature to between 1200° and 1700° F, the current gas turbine plant requires approximately two-thirds of the air passing through the plant be used to cool the hot gases of combustion. One of the major problems in the gas turbine field is, therefore, the development of materials or cooling methods that will permit higher turbine inlet temperatures in order to improve the efficiency and power output of the cycle.

Considerable research is being conducted in the field of air and liquid cooling of the turbine blades. Air and liquid cooling includes hollow blade cooling, rim cooling (16-6, 7), and the injection of a fluid through porous material (16-8). Investigations that have been conducted by various research organizations indicate that significant gains may be expected along this line. Figure 16-35a shows the results of research conducted by the NACA Flight Propulsion Laboratory, Cleveland, in raising the allowable turbine inlet temperature of two different metals by water cooling. In addition to the possibility of raising the turbine inlet temperature, it also indicates that with water cooling there is a possibility of using non-strategic and non-critical materials for the turbine blades. Figure 16-35(b) illustrates the results of research conducted by the NACA Flight Propulsion Laboratory, Cleveland, in raising the permissible turbine inlet temperature of S-816 alloy by air cooling with various blade configurations. Blade cooling, however, complicates the turbine construction and unless applied properly may have a detrimental effect on the efficiency.

Another phase of research is the study and development of materials of very high melting points, such as refractory materials, ceramics, cermals (ceramic-metal combinations) (9), and the use of powder metallurgy. Investigations indicate that cermals may be able to operate at temperatures about 800° F above that of the present metal alloys. Cermals have an additional advantage in that raw materials are not critical, and are low in cost as compared to the strategically critical, high cost metal alloys now used for turbine blading.



(a) (b)

FIG. 16-35(a) Cooling effectiveness of water cooled blades using two different materials (courtesy of NACA, ref. 10); (b) Cooling effectiveness of various blade configurations for air cooling of S-816 high-temperature alloy blades (courtesy of NACA, ref. 16-6).

The gas turbine engine will not replace reciprocating engines and steam plants, but it has become a competitor in many fields. It is the best power plant available in some fields, such as engines for high speed aircraft. The reciprocating plant and steam plant have undergone development since the turn of the century in order to reach their position as reliable prime movers. The gas turbine has undergone serious development only since about 1935. There is much research, experimentation, and development necessary to improve the performance of the gas turbine engine and its components. One of these major fields of research is in the development of turbine blades that will permit higher inlet temperatures and still give a reliable plant with a long service life.

16-14. Open Cycle Gas Turbine. The simple open cycle gas turbine, i.e., one without an intercooler, reheater and regenerator in the cycle, have wide fields of application. In the field of Aeronautics (Chapter XVII), the gas turbine engines now play a major role as a propulsion power plant. In the marine field (Article 16–17), they are being developed as a main propulsion engine, as an emergency high power boost engine, and as an engine to drive auxiliaries and emergency devices. On land (Article 16-18), they are being used more and more for stationary power plants for a great variety of applications and as a propulsion engine for locomotive drive. In each field the advantages and disadvantages will vary depending on the designed duty of the gas turbine engine. However, the advantages and disadvantages in general will be discussed.

Advantages. (1) *Warmup time.* After the engine has been brought up to speed by the starting motor and the fuel ignited, the gas turbine engine will accelerate from a cold start to a full load without a warmup period. This is particularly important in aviation, marine, locomotive, and stand-by emergency power plants.

(2) *Simplicity.* The rotor, which consists of the turbine and compressor wheels connected by a shaft, and the gear trains that drive the auxiliaries are the only moving or rotating parts in the system. There are then no unbalanced forces so that the engine is vibrationless, and the lubrication of the engine is easy and inexpensive. The ignition system is comparatively simple, for a spark is only required for a very short period to start the burning, after which the combustion is self-sustained. The combustion chamber is inexpensive, lightweight, and small with a high rate of heat release.

(3) *Flexibility.* Since different processes within the cycle take part in separate components (compressor, combustion chamber, turbine), a great variety in the arrangement of the system is possible. The ar-

rangement will depend upon the duty, performance desired, and the space allocation of the plant. A few of the great number of different arrangements possible are shown in Fig. 16-36. A regenerator, intercooler, and reheater may be added to any one of these systems. Since the components are separate, the replacement of components and the maintenance is simplified compared to reciprocating engines.

(4) *Low weight and size.* The simple open cycle gas turbine has a

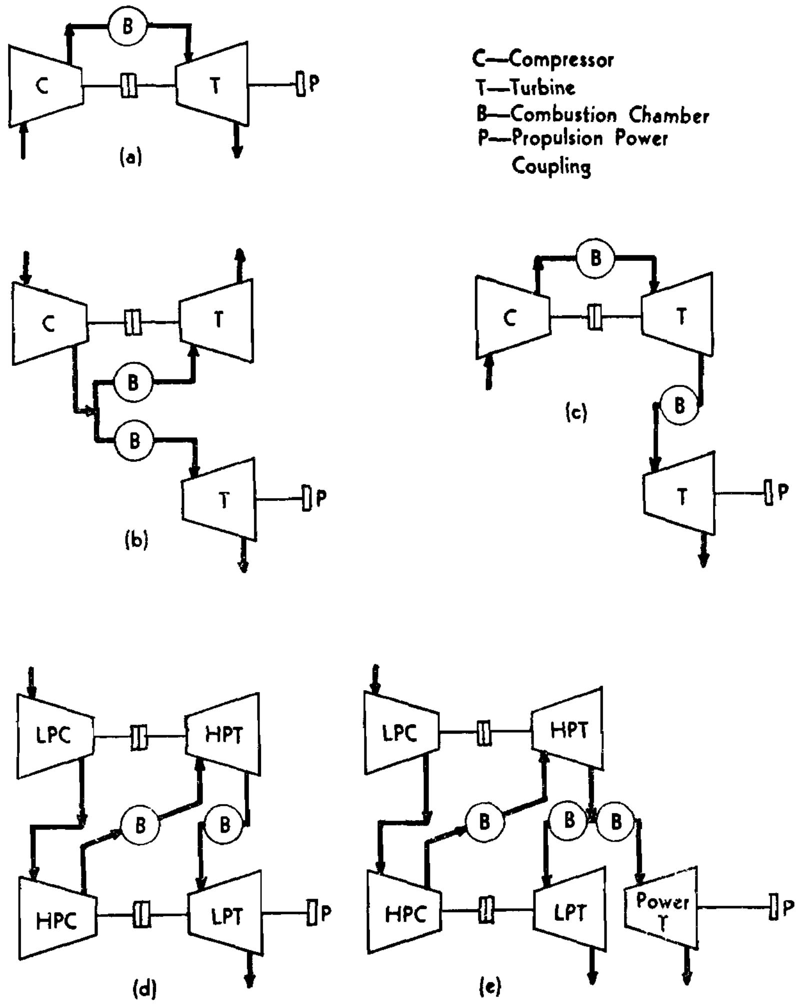


FIG. 16-36. Some of the possible arrangements of the basic components the gas turbine.

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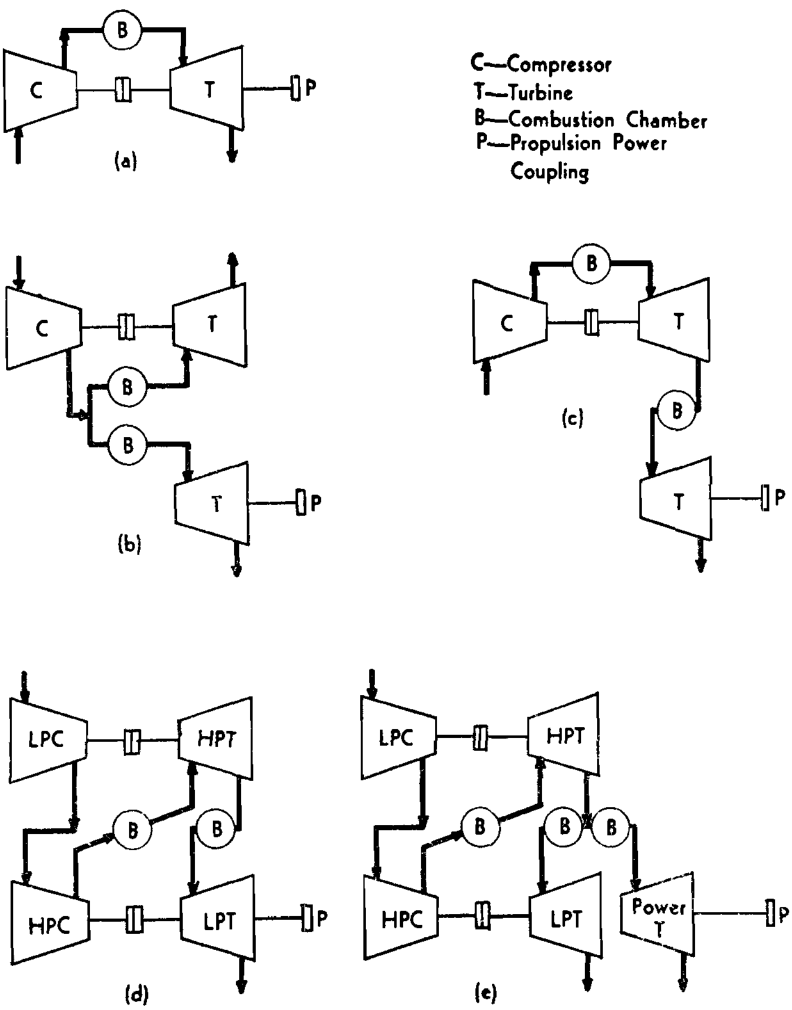

FIG. 16-36. Some of the possible arrangements of the basic components the gas turbine.

lower specific weight (lb weight of engine per hp output) and a lower volume space per hp output than any of the reciprocating and steam prime movers. The high power output in a small space and with a low weight is particularly advantageous in aviation engines and in some marine installations which may be used for high emergency speed and power without having to resort to a large, heavy, complex steam or Diesel plant. Other advantages of the jet propulsion engines over that of the aircraft reciprocating engines will be discussed in Chapter XVII.

(5) *Independent system.* Open cycle gas turbines, except those having an intercooler in the system, require no cooling water. Also, if a powder charge or other mechanical device is used in place of an electric starter, no electric power or battery is required. The engine is then self-contained and independent of outside power or cooling medium, which is advantageous in many power applications.

(6) *Fuels.* The combustion chambers may be designed to burn almost any of the hydrocarbon fuels, from high octane gasoline down to heavy Diesel oil including solid fuels. In spite of careful design, some fuels produce too high a carbon or other foreign deposit in the combustion chamber, turbine, and regenerator which lowers the component efficiencies and in turn the thermal efficiency of the plant. A periodic cleaning of these units may be required. An efficient and thorough method of removing the ash and other foreign matter from the combustion of solid fuels, such as coal, in the open cycle gas turbine has not been developed. However, much research is being devoted to the development of such a system.

To make a general statement as to whether the gas turbine engine is more economical than a reciprocating or steam engine for a particular application is impossible for each engine has its advantages and disadvantages for every type or field of application. From the economic point of view the gas turbine engine cannot be placed strictly in either the advantage or disadvantage column for the placement will depend entirely on whether or not the engine fulfills the requirements of a particular application most economically. From the thermal efficiency point of view, the simple open cycle gas turbine has a maximum of between 23 to 26 per cent with the turbine materials and component efficiencies available at the present time. This maximum is lower than that of the reciprocating and steam engines (Fig. 16-39). However, the thermal efficiency of the simple open cycle gas turbine engine at designed load can be increased so that it will compete with other prime movers by making it a complex cycle, i.e., by adding an intercooler, reheater, and regenerator. These additional elements detract somewhat from a few of the advantages, such as simplicity, low weight, small space, and possible low capital cost. However, it is impossible to make a complete comparison without having fuel, lubrication, maintenance, capital, and mileage costs on a balance sheet.

Disadvantages. (1) *Part load performance.* The calculation of the part load characteristics of a gas turbine engine is very complicated and complex. The system is sensitive to changes in component efficiencies, particularly the axial and centrifugal flow compressors which have a small speed range of optimum efficiencies. A large portion of the turbine output must go to drive the compressor. This, coupled with the fact that the air flow rate or load through the compressor cannot be varied without changing the speed which varies the pressure ratio, produces a system that has poor efficiency at part load conditions. Figure 16-26 shows the effect of a change in pressure ratio on the efficiency of a simple open cycle gas turbine. This indicates that gas turbine engines should be run as close to the designed optimum load, usually around 90 to 95 per cent of full load, as possible.

The part load performance may be improved by utilizing an intercooler and reheater in the cycle. This is borne out in Figs. 16-26 and 16-37. The part load efficiency may also be improved by the use of a twin shaft arrangement such as systems (*b*) and (*c*) of Fig. 16-36. With the twin shaft arrangement, the turbine driving the compressor can be operated at a constant optimum speed for the compressor while the inlet temperature is maintained at its designed value. Part or variable loads at the power turbine are obtained by varying the amount of fuel and therefore the temperature of the working fluid, which varies the speed of the turbine. In this manner, part of the cycle is operated at its maximum efficiency while the inefficiency caused by variable speed is confined to only a portion of the system. The effect on the efficiency of a twin shaft arrangement is illustrated in the curves of Fig. 16-37.

(2) *Sensitivity.* The simple open cycle gas turbine is sensitive to changes in the component efficiencies, i.e., a reduction in the component efficiencies will rapidly lower the thermal efficiency of the cycle (Fig. 16-3). The component efficiencies may be lowered by dust or dirt being deposited on the compressor blades, by carbon or other foreign deposits from combustion in the combustion chamber, turbine, and regenerator, by reduced speeds such as at part load, and by failure to maintain designed efficiencies with continuous operation. The work ratio, which is a measure of the sensitivity, may be improved by the addition of intercooling and reheating to the cycle (Fig. 16-25).

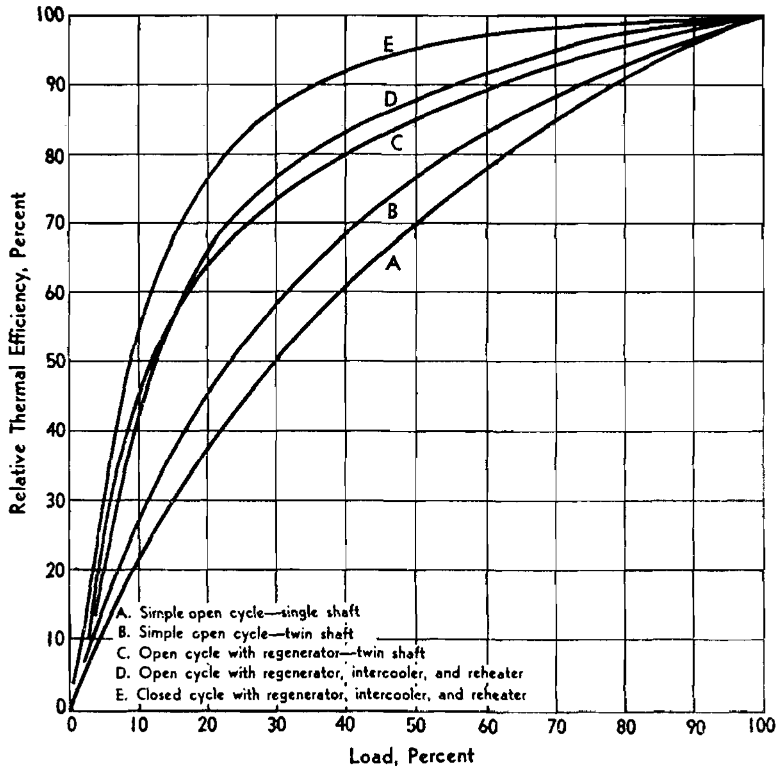

FIG. 16-37. Relative thermal efficiency at part loads for various cycles with the full load thermal efficiency of each cycle being considered as 100 percent (reproduced from *Journal of American Society of Naval Engineers*, Inc., August 1947, ref. 16-21).

The open cycle gas turbine is sensitive to changes in the atmospheric temperature as shown in Fig. 16-12, i.e., an increase in atmospheric temperature will lower the thermal efficiency of the engine. This is a disadvantage where atmospheric conditions are extreme, but is advantageous for the operation of the gas turbine engine in cold climates, such as high altitude operations with jet propulsion aircraft.

(3) *High air rate.* The simple open cycle gas turbine has a very high air rate compared to other prime movers. The large quantity of air required to be inducted and exhausted is not a disadvantage in most applications, but it is a prime factor in naval design where large air intakes and exhausts do not lend themselves to watertight integrity of the ship. The air rate may be lowered by intercooling and reheating (Fig. 16-21). For marine main propulsion, the complex open cycle gas

turbine will require higher pressure ratios and higher turbine inlet temperatures before the size of the ducts and piping are decreased sufficiently to permit their use in combatant naval vessels. However, this will not prevent the engine from being used for small emergency high speed power plants.

16-15. Closed Cycle Gas Turbine. The engines discussed so far have been the *open* cycle gas turbines, i.e., a system in which the working medium must be continuously replaced from the atmosphere since air is drawn in at the compressor and the gases of combustion are rejected to the atmosphere. In order to overcome some of the inherent disadvantages and difficulties encountered with an open cycle, the *closed* cycle gas turbine, i.e., a system in which the same working medium is continually circulated, was developed. *The performance characteristics, the effect of the various variables and component elements on the performance, and the theory and equations developed for the open cycle apply equally as well to the closed cycle.*

Since the closed cycle continuously circulates the same working fluid, air or gases of a higher density than air, the heat added must be supplied through a heat exchanger from an external source and the heat rejected from the system must be through a heat exchanger and a cooling medium. A schematic sketch of a simple closed cycle gas turbine is shown in Fig. 16-38a. Combustion of the fuel takes place in the *air heater* and is external to the working medium of the system. The working fluid leaving the turbine is cooled down by the cooling water in the *precooler* and is recirculated to the compressor.

The **advantages** of this system over that of the open cycle are:

(1) *Reduced size.* The density of the working fluid is increased in the closed cycle by placing the system under an initial overall high pressure. Also since the working medium is not required to support combustion it is not mandatory that it be air. It is possible to use a gas of heavier density and higher specific heat than air, such as the monatomic gases: krypton, argon, xenon, and mercury vapor. This increase in the density reduces the physical size of all components and ducts of the system for the same power output and permits the use of higher temperatures for a given stress limit. An example developed in reference 20 shows that a closed cycle gas turbine under an initial pressure of 30 atmospheres having an output of 100,000 kw will have the same dimensions as an open cycle gas turbine of 12,000 kw output.

(2) *Improved part-load efficiency.* The control of a closed cycle system is different from the open cycle. The output is varied by either withdrawing part of the working medium to the low pressure accumu-

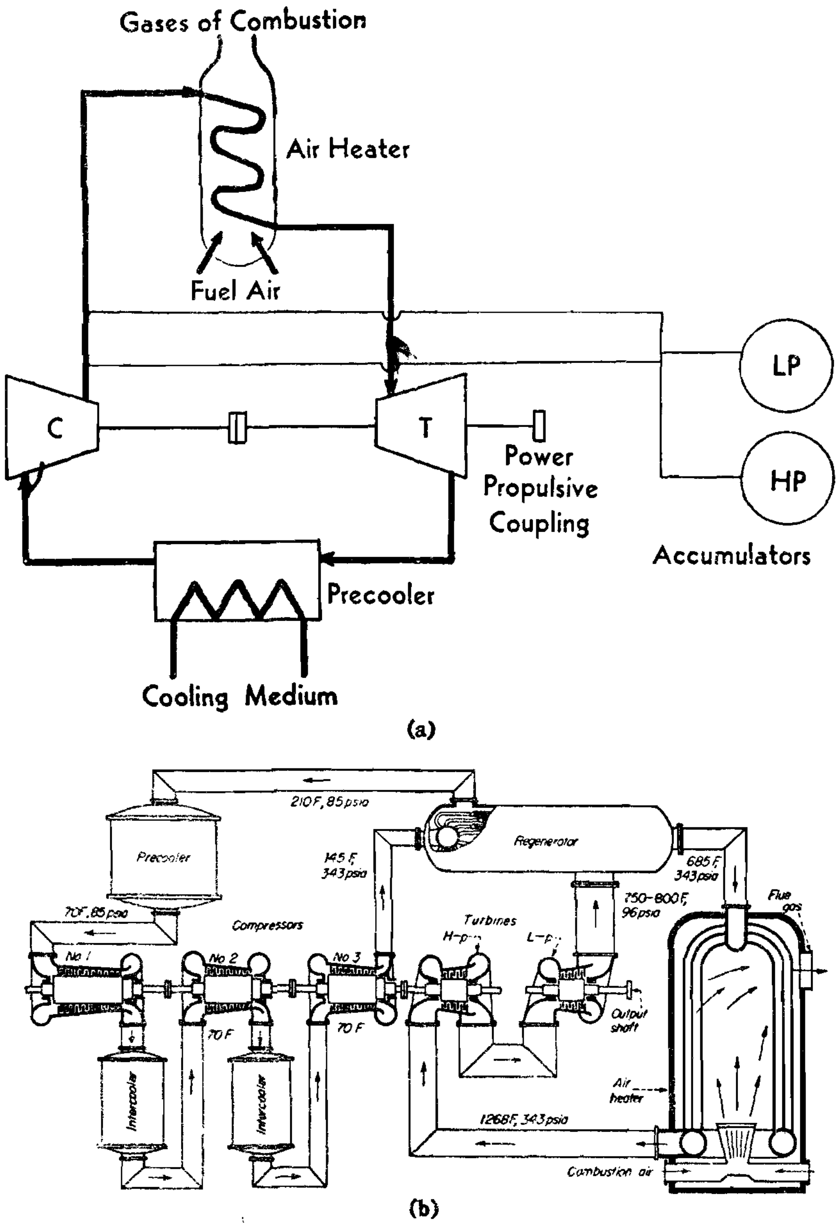

FIG. 16-38. (a) Simple closed cycle gas turbine; (b) Escher Wyss 2000 kw closed cycle gas turbine (reproduced from "How A Gas Turbine Works," by H. L. Rowley and G. B. Skrotzki, *Power*, October, 1946).

lators for a reduction or by admitting more working medium from the high pressure accumulator for an increase in the output. The temperatures and pressure ratios of the system remain constant at their optimum operating values and efficiencies during load change. Theoretically, then, the efficiency of the plant should remain constant over the part-load range; however, it will vary due to the changes in frictional losses. The improvement in the part load efficiency of the closed cycle over the open cycle is illustrated in Fig. 16-37.

(3) *No contamination.* Since the working medium does not contain the gases of combustion, the turbine and regenerator are not subjected to carbon deposits and should remain relatively clean. The compressor should remain free of dust and other foreign deposits since the working medium can be cleaned before being put into the system. This means that the periodic cleaning of the components is not necessary and the component efficiencies should not change appreciably with continued operation. Thus, continued operation should not reduce the thermal efficiency.

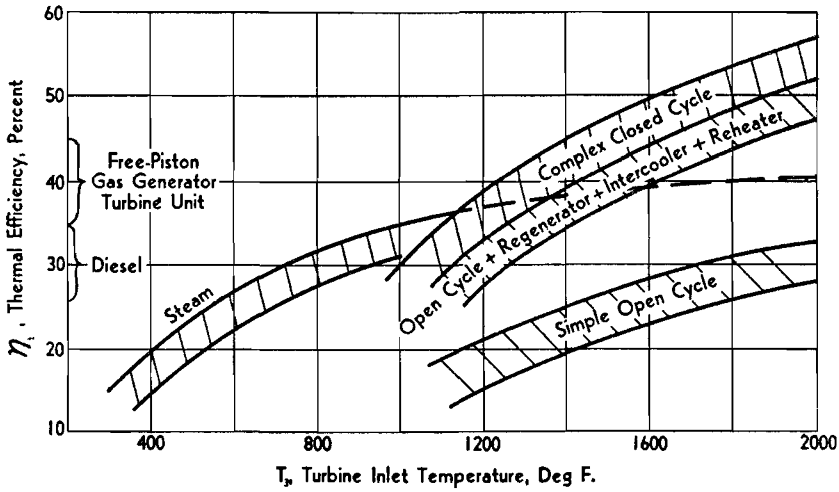

FIG. 16-39. Relative comparison of the thermal efficiencies of diesel, steam, and gas turbine plants.

A comparison of a complex closed cycle gas turbine with a complex open cycle in Fig. 16-39 indicates the closed cycle to have a higher efficiency than the open cycle. This is mainly a result of the reduction in component efficiencies due to the deposit of dust and carbon in an open cycle gas turbine system.

(4) *Fuel.* The closed cycle utilizing external heating can use an inexpensive solid fuel, such as coal.

The **disadvantages** of the closed cycle as compared to the open cycle gas turbine engine are:

(1) *Dependent system.* Cooling water must be provided for the precooler. This eliminates the use of the system as an aeronautical engine. The provision of cooling water is not a problem in marine propulsion and many land based applications.

(2) *Complexity.* The complexity and cost of the system, particularly in the load control, is increased. Since the system is under an initial high pressure with a working medium other than air, it is necessary that the system be gas tight. This adds to the cost and increases the engineering problems.

(3) *Air heater.* A heavy, large air heater is required. The air heater is relatively inefficient compared to the internal combustion chambers of the open cycle gas turbine engine.

In spite of the disadvantages and complexities of the closed cycle, it is higher in efficiency, smaller in weight and space, and is easier to adapt to marine propulsion than the open cycle gas turbine. It has a comparable or better efficiency than steam plants of the same power output with a great saving in weight and space. Figure 16-39 shows a comparison of the closed cycle gas turbine with other prime movers. The closed cycle offers the best possibilities of all the prime movers for use in conjunction with nuclear energy for a propulsion engine (Chapter XX).

Escher Wyss 2000 kw closed cycle gas turbine. The firm of Escher Wyss in Switzerland has been working with the closed cycle gas turbine for many years and has been foremost in the development of the closed system. The 2000 kw closed cycle gas turbine shown in Fig. 16-38(b) was designed and built between 1936 and 1939. Since this time it has successfully passed operating tests and many hundreds of hours of operation. The plant is a closed cycle gas turbine with intercooling and regeneration with maximum operating turbine inlet temperature of 1260° F. The over-all thermal efficiency of the plant from official load tests in 1945 is 31.6 per cent, which is good considering that the component efficiencies of a plant built in 1936 are relatively low compared to those obtainable at the present time.

Semi-closed cycle gas turbines. A semi-closed gas turbine system has been developed that may be placed between the open and the closed cycle gas turbines as far as the advantages and disadvantages of each system are concerned. It is basically a high pressure system so that component parts are smaller than an open cycle for the same power output. Better part-load performance is obtained with this system than with an open cycle. The air heater is smaller than the air

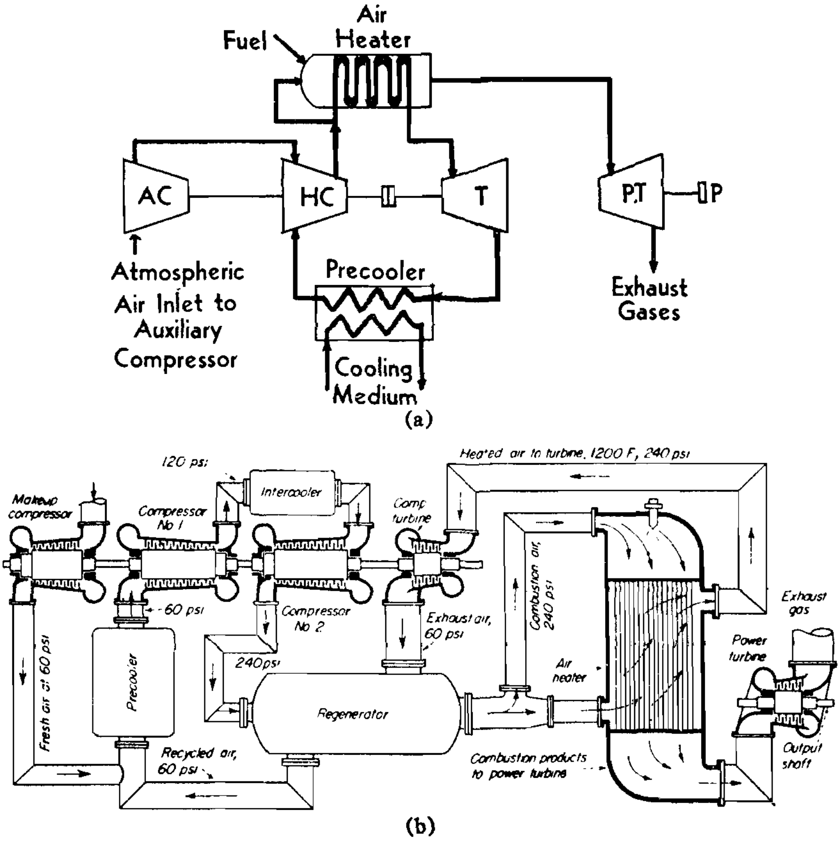

FIG. 16-40. (a) Schematic diagram of the basic Sulzer Brothers semi-closed cycle gas turbine; (b) Sulzer Brothers, 7000 hp marine semi-closed cycle gas turbine (reproduced from *Power, loc. cit.*).

heaters of the closed cycle. The semi-closed cycle was developed primarily by Sulzer Brothers of Switzerland. Some investigation and research is being conducted in the United States, mainly by Westinghouse Electric Corporation.

The basic elements of the semi-closed cycle used by Sulzer Brothers are shown in Fig. 16-40(a). The working medium, which is air, has to be replenished since part of the air is used for the combustion of fuel in the air heater. The air of combustion is replaced by the auxiliary compressor which keeps the main system under an initial pressure of several atmospheres. That part of the air which is not contaminated by the products of combustion passes through the turbine driving the compressor and is cooled down in the precooler before being recirculated.

A schematic diagram of the Sulzer Brothers 7000 hp marine semi-closed gas turbine is shown in Fig. 16-40(b). This plant operated at a turbine inlet temperature of 1200° F and obtained a thermal efficiency well over 30 per cent for a wide range of loading.

16-16. Free-Piston Gas Generator-Turbine System. The utilization of the hot high pressure exhaust gases from a modified Diesel cylinder to drive a turbine has been under study and experimentation both abroad and in this country. Such a system for the generation of hot

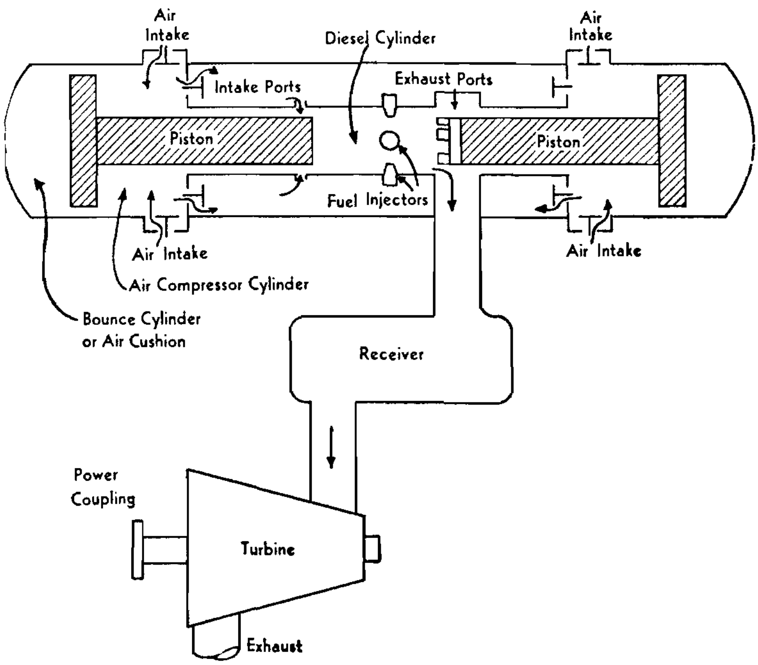

FIG. 16-41. Schematic diagram of the Pescara designed free-piston gas generator-turbine system.

gases for a turbine is termed a *free-piston gas generator.* It combines the high thermal efficiency of the Diesel cycle with the simplicity of the turbine for the expansion of the hot gases down to atmospheric pressure.

The free-piston gas generator is an outgrowth of the free-piston air compressor developed after World War I by R. P. de Pescara in France, and Junkers[2] in Germany. In 1939, Pescara applied a modified

[2] A sectionalized Junkers free-piston high-pressure air compressor is on display in the model room of the Internal Combustion Engines Laboratory of the Department of Marine Engineering, U. S. Naval Academy.

free-piston air compressor to the generation of hot gases to be expanded in a turbine for useful power output. This development was taken up by Sulzer Brothers in Switzerland, and by the Lima-Hamilton Company in the United States under a contract from the Bureau of Ships of the Navy Department.

Figure 16-41 shows the schematic diagram of a free-piston gas generator-turbine system developed by Pescara. The plant may be considered as a highly supercharged two-cycle opposed piston Diesel engine with the hot gases of combustion expanding down to atmospheric pressure in the turbine. The gas generator phase consists of the Diesel cylinder and combustion chamber in the center. Fuel is injected in the center of the Diesel cylinder and burned. The expansion of the gases of combustion forces the opposed pistons apart or outward. Each of the pistons is made integral with another piston which acts as a single stage air compressor on its inner face, and as an air bounce cushion on its outer face. The energy stored in the bounce or air cushion cylinder by the outward movement of the pistons is utilized to drive the pistons inward compressing the air in both the air cylinder and in the Diesel cylinder. Air at 75 to 100 psi passes through valves from the compressor cylinder into a central air space from which it enters the Diesel cylinder through intake ports on the left. This air scavenges and supercharges the Diesel cylinder. The mixture of the scavenging air and gases of combustion at a temperature around 1000° F pass through the exhaust ports and are expanded down to atmospheric pressure in the turbine. All the useful shaft power of the plant is produced by the turbine.

The pistons are connected together by a light mechanical linkage which acts as a synchronization gear to aid in keeping the pistons in step and to time the injection of the fuel. The control of the motion of the pistons, both as to the rate of oscillation and the distance travelled, is accomplished by the variation in the pressure in the air cushion by a stabilizer located and interconnected between the central space and the air cushion. Any number of gas generators may be used in parallel to supply a single turbine (Fig. 16-42). The power output of the turbine may then be varied by cutting the gas generators in or out of the system.

The free-piston gas generator functions as the compressor and combustion chamber does in a conventional open cycle gas turbine plant. In this manner, the high thermal efficiency of the Diesel engine is obtained. Thermal efficiencies higher than those of Diesel engines and in excess of 40 per cent are obtained by this system. Since there are no unbalanced forces, and no side forces on the cylinder wall, the engine itself is vibrationless and requires no extensive hold-down bolts and plates as required by a standard Diesel engine. It is smaller and lighter than a Diesel engine of the same power output. The turbine is about one third the size of the turbine of a conventional open cycle gas turbine plant which has to provide power for the compressor in addition to useful work. Also, the air rate is much lower in the free-piston generator than in the conventional gas turbine, which means that smaller intake ducts can be used. Due to its smaller size and lower temperatures, the turbine

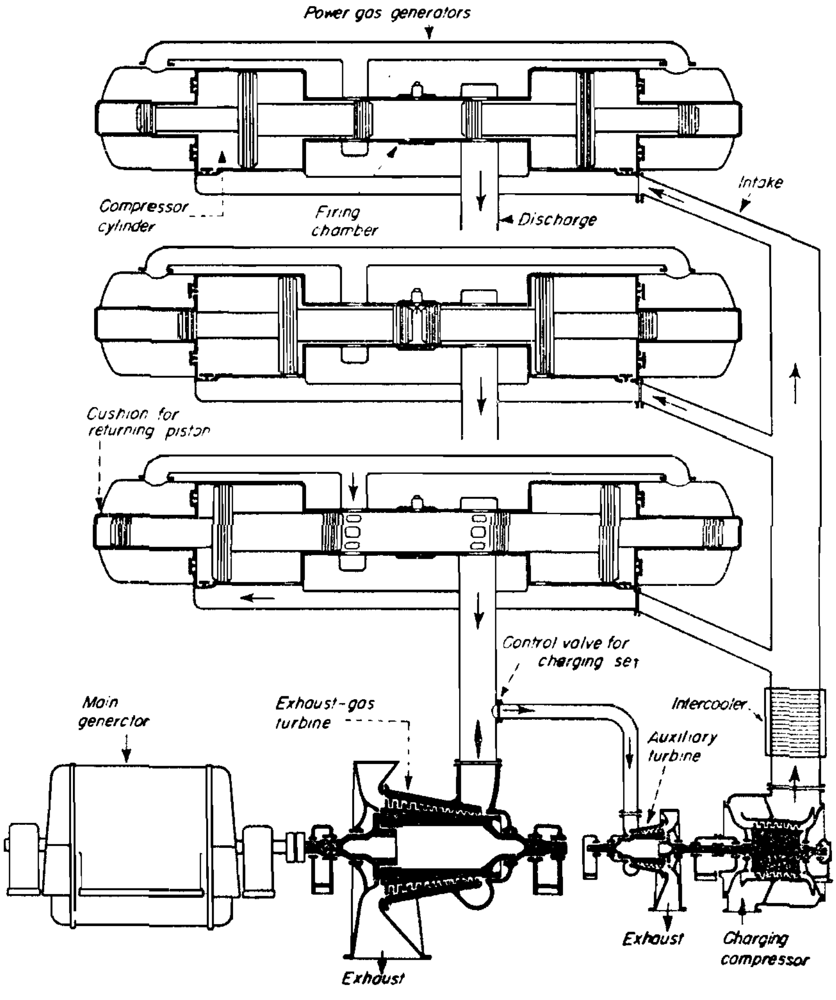

FIG. 16-42. Sulzer Brothers 7000 hp free-piston gas generator-turbine system (reproduced from *Power*, *loc. cit.*).

should require fewer critical materials and have a longer reliable service life.

In spite of all the theoretical advantages of this system, it is still in the laboratory stage of development and is not ready to compete in the power propulsion field due mainly to mechanical faults. There are a number of mechanical problems of starting and control that require research and development. The problem of synchronization of the pistons in a single unit and in a multi-unit have not been solved with any degree of success. As with any new engine, it requires a decade or two to solve all the problems to obtain an engine that is reliable, simple, controllable, and efficient.

In 1943, a contract was given to Lima-Hamilton Corporation for the development of a free piston gas generator-turbine system for ship propulsion. The first unit was delivered to the U. S. Naval Engineering Experiment Station for development tests in the fall of 1950. A schematic diagram of a 7000 hp unit built by Sulzer Brothers, Switzerland, in 1945, for ship propulsion is shown in Fig. 16-42. This unit could not compete successfully with the Diesel and semi-closed cycle gas turbine plant in cost, starting characteristics, and ease of control.

16-17. Marine Gas Turbine Engine History. Important advancements were made in the development of the gas turbine plants, mainly

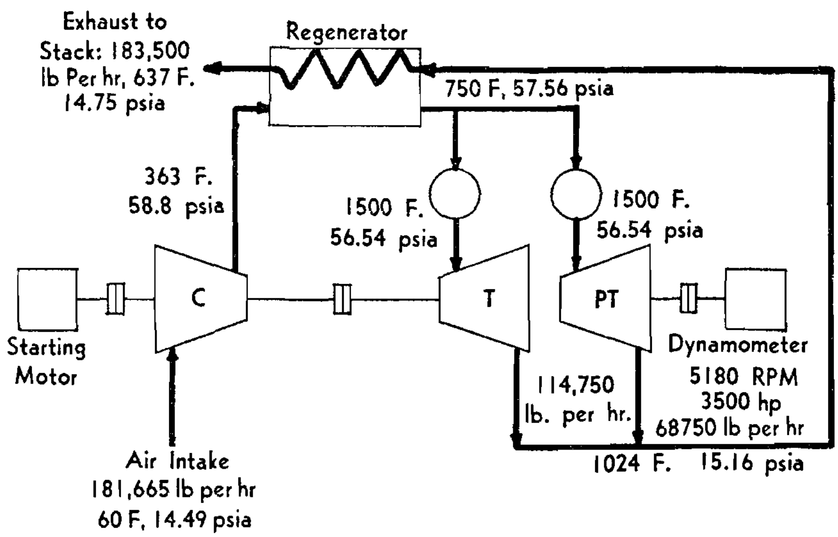

FIG. 16-43. Schematic diagram of the Allis-Chalmers gas turbine showing temperatures, pressures, and flow at the various stations (reproduced from *Power*, May 1946).

in the fields of shore power stations and locomotive power plants, by both England and Switzerland during the decade of 1930 to 1940. The reports on their progress were so enthusiastic that the Department of the Navy began an investigation of the gas turbine engine field in 1938 and established a research and development program to utilize the gas turbine engine for naval propulsion in 1940.

The Bureau of Ships issued a contract in 1940 to the Allis-Chalmers Manufacturing Company for the design and construction of a 3500 hp open cycle gas turbine with regeneration. The contract called for an axial flow type of compressor. This gas turbine engine was built for research and test purposes with the definite objective of furnishing design and operating data for future use. The unit was delivered to the U. S. Naval Engineering Experiment Station at Annapolis in 1944 for test. The tests were completed in the summer of 1949.

The Allis-Chalmers gas turbine is shown diagrammatically in Fig. 16-43 with typical weight flow, temperature, and pressure test data given for various points in the cycle. A 20 stage axial flow compressor with an efficiency of 84.2 per cent and a pressure ratio of 4 delivers air

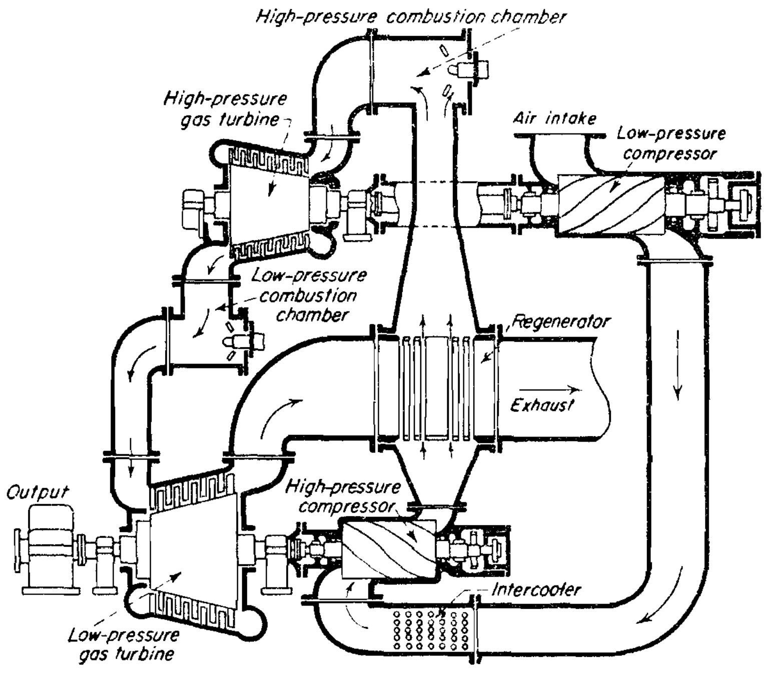

FIG. 16-44. Elliot 2500 hp naval gas turbine (reproduced from *Power*, *loc. cit.*).

to a 57 per cent efficient regenerator. From the regenerator the air is divided so that approximately two thirds of the flow passes through the compressor turbine and one third through the power turbine. The turbine efficiencies of this unit are 81.2 per cent. It should be noted that this gas turbine engine and its component elements were designed and constructed in the early 1940's and that rapid progress has been made in this field in the intervening years; consequently, this particular engine is now outmoded and outdated. However, much valuable scientific and technical data has been accumulated from the tests that will aid in future design and construction.

In order to investigate other types of compressors and gas turbine cycles, the Bureau of Ships awarded a contract in 1942 to the Elliott Company for the construction of an open cycle gas turbine with intercooling, reheating, and regeneration. The contract required the compressor to be of the Lysholm positive displacement type. Figure 16-44 shows a schematic diagram of the 2500 hp Elliott gas turbine.[3] Due to the necessity that the Lysholm compressor be operated at a relatively low speed of around 3400 rpm at full load, the choice of the turbine was limited to either a simple short-path type, i.e., one with a relatively low number of stages, in combination with a reduction gear, or a direct connected turbine spindle with 12 to 14 stages. The latter type was selected for the design. In 1944, the development was completed and the plant was tested with good results. The test results are shown in Fig. 16-45.

As a result of these tests, the Elliott Company submitted in 1945 a proposal for two 3000 hp plants incorporating several improvements and having a turbine inlet operating temperature of 1400° F. These new plants, which were originally aimed at actual shipboard installation, were authorized by the Bureau of Ships. After many setbacks and development difficulties, the components were hot-tested early in 1948. On testing the low pressure turbine, it was discovered that incipient cracks were prevalent in the welds between the discs. The multi-stage wheels necessitated by the low rpm of the Lysholm compressors and the earlier decision to avoid the complication of reduction gears proved to be a misguided choice since the inspection of the earlier 2500 hp plant disclosed the same cracks. The tests were completed and the units were delivered to the U. S. Navy in the spring of 1951.

In 1943, the Department of the Navy began an investigation into

[3] A plastic scale model of the Elliott gas turbine is on display in the model room of the Internal Combustion Engine Laboratory, Department of Marine Engineering, U. S. Naval Academy.

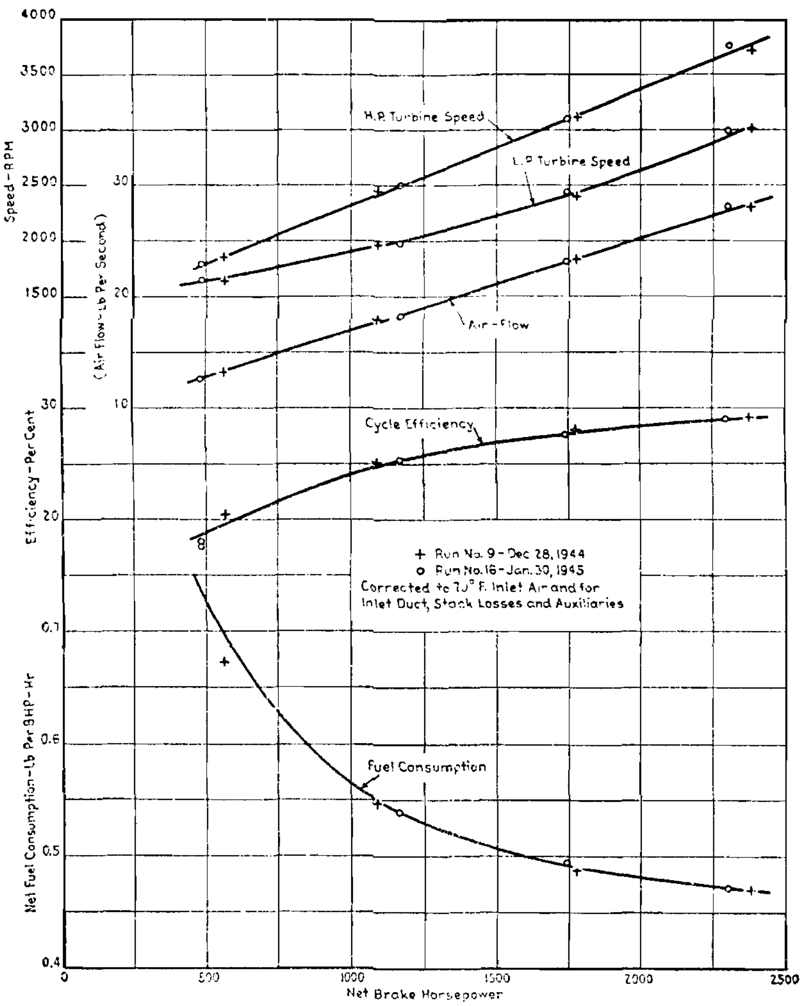

FIG. 16-45. Performance data obtained in an actual test run of the Elliot 2500 hp gas turbine (Soderberg, Smith, Scott, "A Marine Gas Turbine Plant," ref. 16-22).

the possibility of utilizing the free-piston gas generator-turbine system for marine propulsion by awarding a contract to the Lima-Hamilton Corporation for the design and construction of two units. The first tests on these units began at the U. S. Naval Engineering Experiment Station in 1950.

Great Britain has had a prominent lead in the gas turbine field and has been most active in the application of the gas turbine engines to marine propulsion. During 1947, the Metropolitan-Vickers Electric Co. succeeded in operating the first gas turbine engine installed in a ship for marine propulsion. This plant was a 2500 shp unit installed in an experimental naval ship designated the MGB-2009. The MGB-2009 has an overall length of 117 feet, a beam of 19 feet, and was originally equipped with three 1250 bhp Packard engines, with each engine driving its own propeller through a reduction gear. The central Packard engine was removed and the gas turbine engine was installed to drive the central propeller through a reduction gear. This engine was used for maximum or emergency ahead power while the two outboard reciprocating engines were used for cruising and maneuvering. In this manner, good cruising efficiencies and fuel consumption were obtained and emergency power could be rapidly developed from a cold start by the gas turbine engine of low overall specific weight and size (16-10).

The engine was a modified aircraft turbojet engine consisting of a nine stage axial flow compressor with a pressure ratio of 3.47, a combustion chamber, a two-stage turbine driving the compressor, and a four-stage turbine on a separate shaft to drive the propeller through the reduction gear. It was operated for 50 hours at sea before being removed and thoroughly inspected. Data from tests show the unit to obtain 2550 bhp at 7220 rpm, with a thermal efficiency of 16 per cent and a specific fuel consumption of 1.06 lb per bhp-hr. A weight comparison indicated that the reduction gear weighed 2600 pounds, while the gas turbine weighed 3710 pounds. A modified gas turbine engine of this type is now being tested in the MGB-2009 (16-10).

The British Thomson-Houston Co., Ltd., has designed and constructed a 1200 shp open cycle gas turbine with regenerator for installation in the merchant ship *Auris*, an oil tanker of 12,600 tons. The ship is under construction and was designed to be propelled by four Diesel-engine driven alternators. One of the Diesel engines was replaced with the gas turbine engine to drive the alternator in order to obtain extensive tests on the unit at sea. The engine consists of an axial flow compressor with a pressure ratio of 4:1, two turbines with the low pressure turbine driving the alternator and the high pressure turbine driving the compressor, and a regenerator of 55 per cent efficiency. The plant has a thermal efficiency of 20 per cent with a turbine inlet temperature of 1200° F, and a specific fuel consumption of 0.7 lb per shp-hr. Trial runs of this plant were made in 1951.

The English Electric Co., Ltd., is designing and constructing the largest marine gas turbine plant yet ordered by the British Admiralty. It is a new design to suit the main propulsion requirements of one of the *Captain* class frigates. A ship of the same class is propelled by steam-driven alternators of 6600 hp running at 5600 rpm. By designing a gas turbine plant of the same output and speed, a direct comparison of a steam and a gas turbine plant will be made.

Rolls-Royce, Ltd., has a contract from the British Admiralty to design and construct a gas turbine engine for the main propulsion of H.M.S. *Gray Goose.* The *Gray Goose* is a gunboat of some 250 tons. The gunboat is now propelled by a steam plant with an output of about 8000 shp with a speed of 30 knots.

Pametrada (Parsons and Marine Engineering Turbine Research and Development Association) has designed and is constructing a 3500 shp gas turbine engine complete with maneuvering and speed reduction gear for a shaft speed of 85 rpm. This engine has a turbine inlet temperature of 1200° F and consists of two compressors with pressure ratios of 2:1 and 2.8:1 respectively, two turbines, and a regenerator of 75 per cent efficiency. Pametrada expects to design and construct a 15,000 shp gas turbine engine in the near future (16-11).

Sulzer Brothers, Switzerland, has completed the preliminary tests on a 7000 shp semi-closed gas turbine engine for marine propulsion, Fig. 16-40. Thermal efficiencies of over 30 per cent are expected from this plant with a marked reduction in size, weight, and air rate from that of the open cycle.

Escher-Wyss, Switzerland, has designed an 8000 hp closed cycle gas turbine using a controllable pitch propeller for main propulsion. Much research, both in Europe and the United States, is being conducted on the design and study of the closed cycle both as a straight gas turbine and as a system to be used with nuclear energy.

The preceding paragraphs give a brief resumé of the gas turbine developments in the field of marine propulsion. Other research and developments in this field can not be discussed in this section of the text for security reasons. However, it can be stated that much research is going into the development of satisfactory power transmission and starting systems, fuels, and the component elements. Also, the research and tests conducted on the engines and their components in the fields of aviation, power stations, and locomotives are being utilized in the development of the gas turbine engines for marine propulsion.

Two of the research and development problems in connection with gas turbines for marine propulsion are suitable methods for the transmission of power and a method for reversing. Basically, a successful gas turbine for marine application requires high speed for optimum efficiency while the characteristics of marine propellers require low rotative speed for their optimum efficiency; and a means of reversing must be provided for maneuverability. There have been several methods of transmission and reversing proposed: (1) electric drive, direct current, (2) electric drive, alternating current, (3) mechanical-geared drive with controllable pitch propeller,[4] (4) mechanical-geared drive containing reversing features such as pinions at the reduction gear connected through suitable hydraulic or magnetic couplings, and (5) mechanical geared drive with a reversing gas turbine (16-11). The electric drive, direct current, is not well suited for gas turbine application due to high first cost and maintenance, low efficiency, and large weight and space. The electric drive, alternating current, is currently being used for gas turbines in England and appears to be suitable; however, it requires large space, its efficiency is lower than a mechanical drive, and it does not provide rapid reversal. The mechanical drive with a controllable pitch propeller is comparable and superior to other transmission systems in all respects for horsepowers up to about 25,000. The Bureau of Ships has been conducting tests with controllable pitch propellers since 1940. Over 2400 were installed and used successfully on patrol craft, tugs, and landing craft during the war. Tests on the destroyer USS *Dahlgren* in 1941 with a controllable pitch propeller indicated that losses would not exceed 1 to 2 per cent with proper design, that ships with controllable pitch propeller could stop within two-thirds the distance required by reversing turbines with fixed propeller, and that fuel savings can be obtained at cruising speeds by using lower pitch and higher shaft speed (16-12). For engines of largè horsepower output, the best type of drive appears to be a mechanical-geared drive containing reversing features; however, a power turbine on a different shaft than the compressors will be required. There is little prospect that a mechanical-geared drive coupled with a reversing turbine, separate astern turbine, will be employed due to the extra weight, space, cost, and mechanical troubles, plus the fact that there are other acceptable methods.

16-18. Marine Applications. The development of the gas turbine engine for ship propulsion has been slow as compared to the rapid advancements made by the gas turbine engine in the field of aeronautics (Chapter XVII). In aviation, it was mandatory that the engine be devel-

[4] Sectionalized reversible pitch propeller is on display in the model room of the Internal Combustion Engine Laboratory, Department of Marine Engineering, U. S. Naval Academy.

oped rapidly, since it was the only type of engine that could fulfill the military requirements for high speed combat aircraft with a relatively reasonable economy. In the field of naval and marine propulsion, the existing types of power plants available—steam and Diesel—have been suitable, satisfactory, and economical. Thus, there has been a tendency to hold back until the time arrived when the gas turbine plant could show positive advantages over the other types of propulsion engines.

Another principal reason for the slow development has been the low permissible turbine inlet temperatures which is a direct criterion of the service life of the engine and also of the efficiency of the plant. If the turbine inlet temperature is raised, the efficiency increases but the number of hours of operating life of the engine is radically decreased. A turbine inlet temperature above 1550° F will give a predicted service life below 1000 hours with the available turbine materials. In order to obtain a predicted service life of 100,000 hours, a relatively low turbine inlet temperature of 1250° to 1350° F must be used. The service life and restriction on the permissible turbine inlet temperature has been more of a restriction on naval and marine propulsion than on the aeronautical power plant. In the field of aviation, packaged power plants are used so that a new complete engine can easily be installed in place of an old engine. These aviation packaged engines have a low expected useful life of 500 to 2000 hours. The expected power plant life for a standard ship propulsion engine in the past has been 100,000 hours, which corresponds to twenty years service with relatively few opportunities for engine checks and with a great difficulty of replacing one engine with another. Due to the restriction on the turbine inlet temperature by the requirement for a 100,000 hour service life, many naval constructors in the United States and in England are advocating 10,000 hour service life for cruising engines and 500 to 1000 hour service life for separate high speed engines which will be packaged gas turbine plants having the same principles used in aviation.

Statistical studies have been made both in the United States and in England as to the operating power and speeds requirements of combat naval vessels over a period of years. These studies build up a strong case for low-life, packaged engines to be used for high speed or emergency operation and, also, show an application where the gas turbines have definite advantages over the other types of naval propulsion engines. The statistical studies over both peace and wartime conditions show that naval ships over 70 cent per of their total operating time use only a very small proportion of their installed power.

The steam-propulsion plant on an average destroyer has a specific weight of around 16 lb per hp. For a 60,000 hp twin-screw ship, statistical figures indicate that for 99 per cent of the operating life of the vessel during which 97 per cent of the fuel is consumed the ship carries over 300 tons of propulsion machinery as reserve against the demand for high speed which would require more than 30 per cent of the installed power. It has been proposed that the top 70 per cent, 42,000 hp, of the existing propulsion machinery, which is required to operate during only one per cent of the operating life, be replaced by a relatively short-life, packaged gas turbine plant that has a specific weight between 0.75 and 2 lbs per shp. This would give a saving of approximately 260 tons in ships propulsion plant and would reduce the length of the overall machinery plant by more than ten feet. The remaining 18,000 hp of steam propulsion plant would be used for economical cruising and for backing requirements (16-13). In addition to the weight and space saving, the gas turbine has a definite advantage

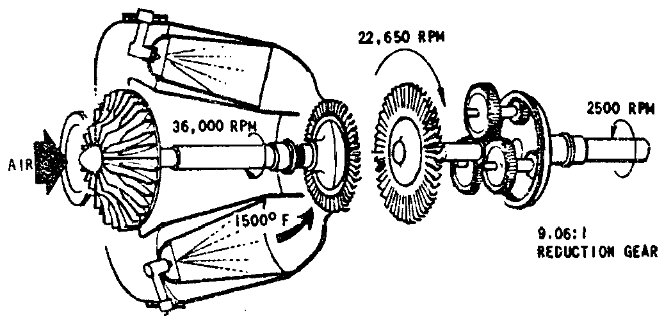

FIG. 16-46. Boeing Model 502, 175 hp, gas turbine (permission Boeing Aircraft Co.).

over the steam for emergency high power requirements in that it requires only a very small warm-up period in going from a cold start to a full load condition. The low service life requirement would permit higher turbine temperatures and, consequently, a gas turbine plant that would have an efficiency as high or higher than the remaining steam installation.

Early in 1947, the Boeing Aircraft Company disclosed details of their privately sponsored turbojet aircraft engine development. This engine with the addition of a propeller and shaft appeared to offer some superiority as a small boat drive. After further study and negotiation, the development of an advanced version of a 175 hp and a 400 hp engine for installation in landing craft in combination with the reciprocating engines for cruising was subsidized by the Bureau of Ships. This was the first admission into the Bureau of Ships' program of a plant having the basic characteristics of an aircraft engine, such as a packaged engine of extreme lightness, relatively short life, high operating temperatures, and high rotating speeds. The engine consists of a centrifugal compressor, two tubular can type combustion chambers, and two axial flow single stage turbine wheels, Fig. 16-46.

The Boeing gas turbine model 502, the 175 hp unit, was tested in an LCVP at the U. S. Naval Engineering Experiment Station during the Spring of 1951. The Boeing test data on this model is shown in Fig. 16-47. An interesting comparison made by the Boeing Aircraft Company between a Diesel engine of the equivalent hp and the Boeing gas turbine is as follows:

	Gas Turbine Engine	*Diesel*
Length	38 in.	53.2 in.
Width	23 in.	29 in.
Height	18.5 in.	45.2 in.
Volume	9.4 cu ft	40.5 cu ft
Weight	185 lbs	2850 lbs

This type of engine is being installed in twenty minesweep boats.

Besides the possibilities and potentialities of gas turbines for the main propulsion plants for ships in the future and for high peak load or high speed plants in combination with Diesel or steam plants, the gas turbine is being used in several applications for auxiliary purposes and it has the future possibility of many additional applications. Gas turbines are being used in the Velox steam plants as an auxiliary to raise the pressure of the incoming air. It has wide application as an auxiliary for both spark ignition and compression ignition engines for supercharging purposes. Gas turbine driven generators for the main power supply and for emergency or standby power sources are finding wide application from the 25 kw generators of aircraft up to power supplies of 25,000 kw and above.

Early in 1947, the Bureau of Ships began an investigation into the possibilities of the use of aircraft type gas turbines for other shipboard applications as auxiliaries where 1000 hours or less at rated power would satisfy durability requirements giving high permissible inlet temperatures with good efficiencies. Several designs were considered for emergency generator prime movers. Two types of open cycle gas turbines using axial flow compressors have been built and tested as emergency generator drives. A 400 hp engine for emergency generator drive weighs 600 lbs as compared to a weight of 8300 lbs for a Diesel drive.

In 1948, the Bureau of Ships began an investigation into the possibilities of using an open cycle gas turbine of an output below 100 hp for a 500 gallons per minute portable fire pump which would not only reduce the unit weight by about 60 per cent but would provide a pump unaccompanied by the usual quantity of spare parts and the temperament of the two cycle internal combustion engine now in widespread use for fire fighting purposes throughout the Navy. Two types of pumps of this nature utilizing the gas turbine were constructed and delivered in 1949. The first type of pump built by the Solar Aircraft Company has an integral centrifugal compressor-turbine wheel similar to a double faced impeller. Compression is on one side radially outward

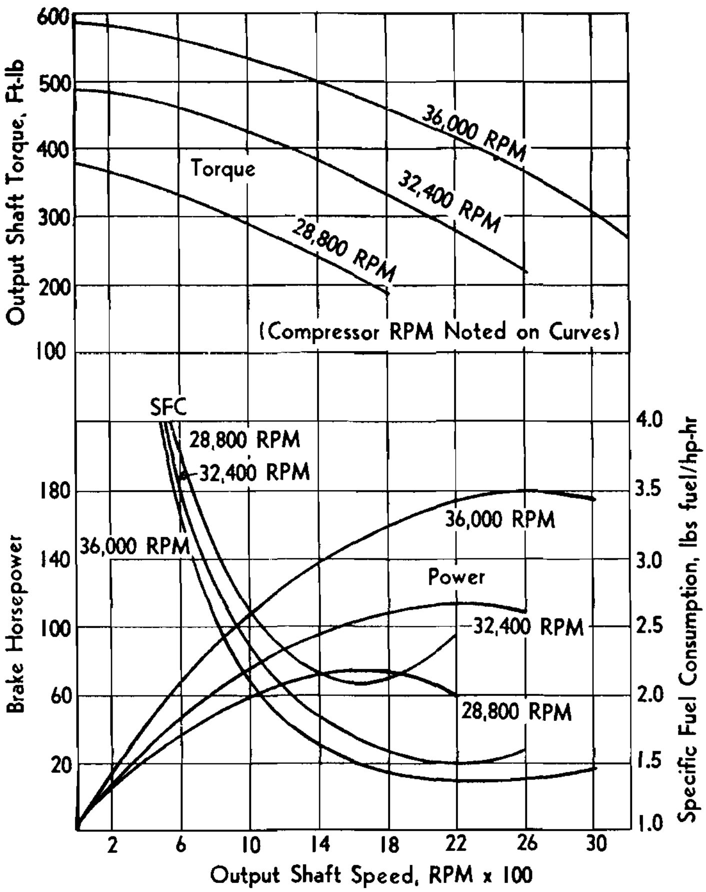

FIG. 16-47. Performance Curves on Boeing Model 502, Gas Turbine (reproduced from "Potent Power Packages—The Boeing Gas Turbines" by H. M Jacklin, at SAE Meeting).

while expansion is on the other side radially inward. This type lends itself to economical mass production and also provides an easy method of conduction cooling. The second type, which was designed by the Continental Aviation and Engineering Company uses a centrifugal compressor and a single-stage axial flow turbine. Both types are intended to be capable of manual starting. A 50 hp portable gas turbine fire pump weighs about 60 lbs, compared to a weight of about 185 lbs for the gasoline piston-engine. The gas turbine engine has another distinct advantage over the gasoline engine in that a non-volatile, high flash point fuel may be used (16-13). The Solar gas turbine engine driven fire pump is shown in Fig. 16-48.

At the present time, it appears that the gas turbine can best be utilized for marine propulsion in the range of 2500 to 25,000 horsepower, i.e., above that of the large Diesel engines and below the best range of the steam plants, 25,000 horsepower and above. However, gas

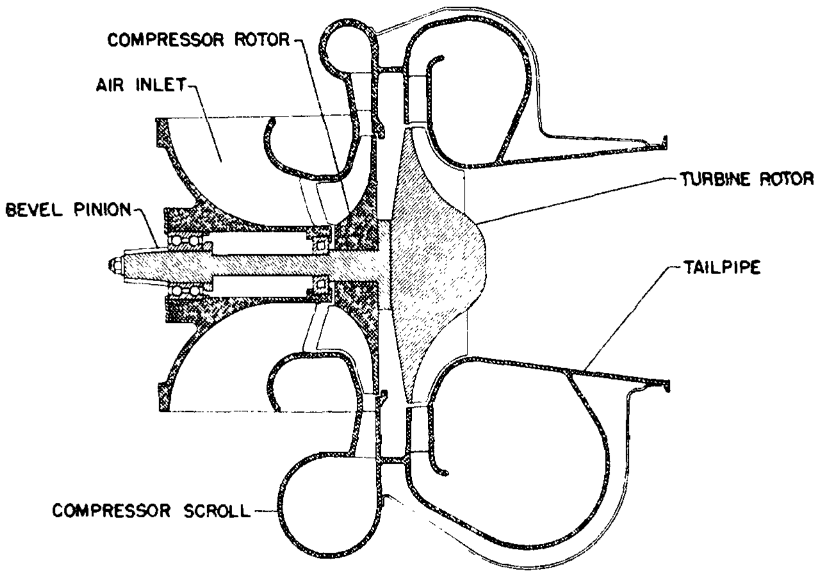

FIG. 16-48. Sectionalized view of the rotor (turbine and compressor) of the gas turbine drive fire pump. The turbine is a radial flow type.

turbines below 2500 horsepower will be utilized as auxiliaries and perhaps as peak load or high speed engines in combination with Diesel or spark ignition engines in landing craft, motor torpedo boats, etc. No thorough and extensive analysis of the comparison of the various

Diesel, steam, and gas turbine plants has been published. However, a very rough general comparison of the three types can be made for a plant between 10,000 and 15,000 horsepower.

In this operating range, the thermal efficiencies of the gas turbine will be comparable to and above that of the steam plant and below that of the Diesel plant, but will approach and be comparable with the Diesel as the permissible turbine inlet temperature is raised. The specific fuel consumption will be between that of the steam plant, 0.60 to 0.78 lb per shp-hr, and that of the Diesel plant, 0.36–0.40 lb per shp-hr; and it will approach that of the Diesel as temperature and pressure ratios are increased. The cost per gallon of the fuel in a gas turbine will be equal to that of the steam plant and lower than that of the Diesel. Also, cost of the lubricating oil in a marine Diesel engine may amount to as much as 15 per cent of the total cost of the fuel oil, while that of the steam plant and the gas turbine will only be about 0.5 per cent. The specific weight of the gas turbine will be between 40 and 70 per cent of that of the Diesel and between 20 and 30 per cent of that of the steam plant. This indicates that a ship with a gas turbine plant will be able to carry a correspondingly greater amount of fuel, and therefore obtain a greater range. The gas turbine will give a great saving in space over both that of the steam plant and the Diesel engines. There will be a great simplification of the plant over that of the steam plant, due to the absence of the boilers with their feed-water evaporating and condenser systems. The gas turbine will have lower maintenance, operating and upkeep costs than that of the Diesel plant.

However, there is a great amount of research and development to be accomplished in the coming years before the gas turbine will take its place along with the Diesel and steam plants in the field of marine propulsion. The previous articles in this chapter have indicated the amount of work that remains to be done on the gas turbine and its component parts. It must be emphasized that the gas turbine will not replace the Diesel and steam plants but will supplement them; for there will be applications where each one will surpass and be superior to the others. In the future, naval officers will not only come in contact with the various Diesel engines, the steam plants, the straight gas turbine plants, and the gas turbine plants in combination with either the Diesel or steam plant, but there are also strong possibilities that they will be operating with liquid oxygen and hydrogen peroxide installations and nuclear energy plants. These last two types will be discussed in Chapters XIX and XX of this book.

16-19. Gas Turbines for Commercial Land Application. The various types of gas turbines have been applied to commercial land use far more extensively than to marine propulsion and progress in this field has been rapid. Due to space limitation, only a general summary of the commercial land applications will be given. A more detailed description of these gas turbines may be found in periodicals and in references 16-10, 11, 14, 15, 16, and 17.

In general, some of the widespread applications of the gas turbine for commercial land use employing the open cycle, semi-closed cycle, closed cycle, and free piston gas-generator, are as follows:

(1) Locomotive propulsion.
(2) Central Power Stations.
  (a) Fully automatic booster stations at end of transmission lines.
  (b) Standby plants for hydro installations.
  (c) Standby and peak load plants for small systems.
  (d) In locations where water is not available.
  (e) Bombproof power plants.
(3) Industrial.
  (a) By-products power such as the Houdry process for oil refining.
  (b) Standby plants for steam or purchased power.
  (c) Peak load plants supplementing power plants.
  (d) Main power sources for laboratories.
  (e) Pumping stations.
(4) Automotive—Some research, design, and construction in this field has been done both in England and in the United States.

Engines ranging from 100 to 30,000 hp are being designed and constructed, and are in current use for the above applications (16-18).

Both in the United States and in Europe, the gas turbine is being used more and more for locomotive propulsion with several in use now and more being constructed each year. The great majority have been simple open cycle or open cycle gas turbines with regeneration using liquid fuel. They are being built in this country by Westinghouse Electric Corp., General Electric Co., Allis-Chalmers Co., and the Elliott Co. An example of this type is the 4000 hp unit consisting of two 2000 hp gas turbines built by the Westinghouse Electric Corp., which has been under successful tests for several years. The length of the locomotive is 78 feet as compared with a Diesel-electric locomotive of the same output employing two units of about 150 feet total length. The over-all fuel rate at full load is 0.78 lb per bhp-hr, with a corresponding thermal efficiency of 16.7 per cent. Indications are that the maintenance, operational, and fuel costs will be below that of the Diesel-electric locomotives. The control has been reduced to pushbutton simplicity, and the unit can be brought from a full cold stand-still condition to idling speed ready to take full load in one and a half minutes.

In addition to the open cycle gas turbine, the Lima-Hamilton Corp. has a unit consisting of six free-piston gas generators in combination with a turbine unit and conventional electric drive under construction for the Pennsylvania Railroad. The Cooper-Bessemer Corporation also has a free piston gas generator turbine unit under development for

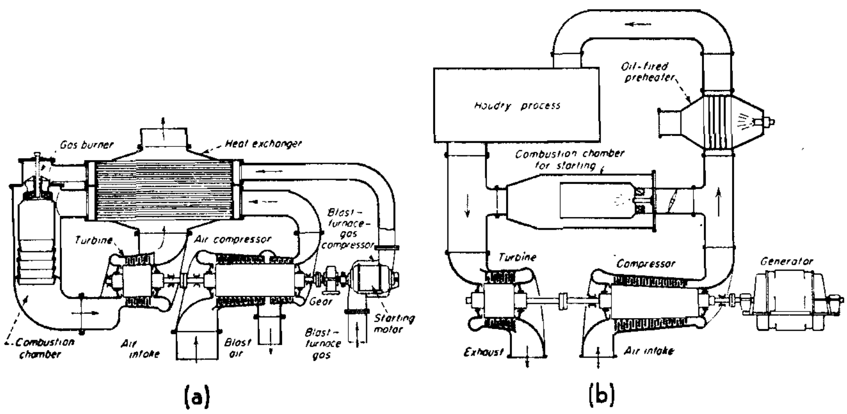

FIG. 16-49. (a) Blast-furnace gas-fired turbine plant supplying blast air for furnaces. (b) Gas turbine for Houdry Process (reproduced from *Power*, *loc. cit.*).

utilization in ships, electric power generation and pipe line pumping stations. A further development is the 4200 hp gas turbine locomotive being designed to use powdered coal and being built by Allis-Chalmers Corp., in conjunction with the Bituminous Coal Research, Inc., for the American Locomotive Company. It is a single shaft open cycle gas turbine with regenerator having an expected thermal efficiency of 24 per cent with a turbine inlet temperature of 1300° F when using oil as fuel.

The utilization of the gas turbine generating plants for the various types of central power stations is far more widespread in Europe than in the United States. In Europe, a great number of central power stations using gas turbines from 500 kw up to 27,000 kw are in use and more are being designed and constructed each year. These plants include the open cycle, semi-closed cycle, and closed cycle. Liquid fuel is being used in the majority of the plants, but coal is being used in some of the closed cycle plants, and being experimented with in the open and semi-closed cycles. In general, the turbine inlet temperatures are 50 to 100° F below that used in the United States. However, thermal efficiencies in the range of 25 to 35 per cent are being obtained with turbine inlet temperatures in the range of 1200° to 1350° F. The leading nations in Europe in this field are Switzerland and England. In the United States, several open cycle and semi-closed cycle plants are being designed and constructed to use natural gas as the fuel.

The most extensive use of the gas turbine in industry in the United States has been in the Houdry oil-refining, cracking process. The first plant was installed in 1939, and since then over 30 such plants have been constructed and installed. A schematic diagram of a gas turbine for the Houdry process is shown in Fig. 16-49(b). Figure 16-49(a) shows the gas turbine employed to furnish blast air for blast-furnaces.

The gas turbine has made rapid progress during the past decade due mainly to the large amount of research, talent, and money available during the war. It is no longer a theoretical and impractical engine, but one that can well compete in many fields of application with the steam plants and Diesel engines. The employment and application of the gas turbine as well as the free piston gas generator and its turbine will be more widespread and more common in the future in the field of locomotive propulsion, land power stations, and other industrial uses.

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EXERCISES

16-1. Name the three major components of a gas turbine engine.

16-2. What is the function of the secondary air? Why is it necessary?

16-3. List the three types of compressors in current use.

16-4. What are the advantages of an axial flow compressor? A centrifugal flow compressor?

16-5. List the requirements for a good combustion chamber.

16-6. Why is blade cooling required?

16-7. Sketch a schematic diagram of a simple open cycle gas turbine engine, label all components, and show flow of working media.

16-8. List the simplifying assumptions.

16-9. The following data was obtained by test and calculation for a simple open cycle gas turbine engine (See Fig. 16-8 for state points):

State point	T °F abs	h Btu/lb	State point	T °F abs	h Btu/lb
1	520	28.8	3	1500	274
2	672	65.3	4	1184	192
2′	691	70	4′	1216	208

(a) Draw T-s diagram for the cycle, label process lines.

Calculate:

(b) Actual turbine work.
(c) Compressor efficiency
(d) Net work
(e) Thermal efficiency

Ans:
(a) See Figure 16-8
(b) 66 Btu/lb
(c) 88.6%
(d) 24.8 Btu/lb
(e) 12.15%

16-10. List the five operating variables that strongly effect the thermal efficiency of an open cycle gas turbine. Show by diagrams how the thermal efficiency varies with a change in these operating variables.

16-11. Sketch a schematic diagram and a T-s diagram of an open cycle gas turbine engine with a regenerator. Label all components and process lines.

16-12. What limits the regenerator effectiveness?

16-13. Given the following data for an open cycle gas turbine engine with regenerator (See Figs. 16-14 and 16–13 for state points):

(a)

State point	h Btu/lb	State point	h Btu/lb
1	42	4	202
2	92.7	4′	213
2′	98.0	5′	182

(b) Heat supplied in combustion chamber = 126 Btu/lb

Calculate:

(a) Regenerator effectiveness.
(b) Enthalpy at exit from combustion chamber, state 3.
(c) Turbine efficiency.
(d) Thermal efficiency.
(e) Enthalpy at exit from regenerator, state 6′.

Ans:
(a) 73%
(b) 308 Btu/lb
(c) 89.6%
(d) 30.9%
(e) 129 Btu/lb

16-14. Given the following data for an open cycle gas turbine engine with intercooler and regenerator.

State point	h Btu/lb	State point	h Btu/lb
1	28.8	2′	82.3
a	61.1	3	398.4
a′	67.2	4	220.8
b	40.3	4′	243.9
2	75.6	5′	187.3

(a) Draw T-s diagram of cycle.

Calculate:

(b) Regenerator effectiveness
(c) Intercooler effectiveness
(d) Compressor work
(e) Turbine efficiency

Ans: (b) 65%
(c) 70%
(d) 80.4 Btu/lb
(e) 87%

16-15. Given the following data for an open cycle gas turbine engine with a regenerator and a reheater (See Figure 16-17 for T-s diagram):

State point	h Btu/lb	State point	h Btu/lb
1	29	c′	337
2		d	437
2′	129	4	339
3	437	4′	356
c	326	5′	269

$w_a = 90$ lbs air/min
$r_p = 5$
Compressor efficiency = 85%

Calculate:

(a) Regenerator effectiveness
(b) First stage turbine efficiency
(c) Enthalpy at state 2
(d) Thermal efficiency

Ans: (a) 61.7%
(b) 90%
(c) 114 Btu/lb
(d) 30.2%

16-16. Given the following data for an open cycle gas turbine engine with intercooler, reheater, and regenerator:

State point	T °F abs	h Btu/lb	State point	T °F abs	h Btu/lb
1	540	33.6	c	1238	206
a	738	81.3	c′	1282	217
a′	764	87.7	d	1650	314
b	585	44.4	4	1238	206
2	799	96.1	4′	1282	217
2′	828	103.2	5′	1149	183
3	1650	314			

(a) **Draw a schematic diagram and a T-s diagram of the above cycle, labelling all components, process lines, and state points. Show flow of working media through the engine.**

Calculate:

(b) Intercooler effectiveness
(c) Total of heat supplied in combustion chamber and reheater.

Write equations for:

(d) Net work
(e) Thermal efficiency
(f) Regenerator effectiveness
(g) Enthalpy of exhaust gas leaving regenerator.

Ans:

(a) See Figure 16-18
(b) 80%
(c) 228 Btu/lb
(d) See numerator equation (16-30)
(e) See equation (16-30)
(f) See equation (16-17)
(g) See equation (16-21)

16-17. Define air rate. Does a low air rate increase or decrease the size of an engine?

16-18. Define work ratio.

16-19. Which is better, a high or a low work ratio? Why?

16-20. What is the effect of regeneration on the performance of an open cycle gas turbine?

16-21. What are three major effects on the performance of an open cycle gas turbine with regeneration caused by the addition of an intercooler and a reheater?

16-22. Show by graph the effect on the thermal efficiency by the successive addition of a regenerator, an intercooler, and a reheater or a gas turbine engine at various pressure ratios.

16-23. How is the performance of a gas turbine engine increased by water injection?

16-24. What are six advantages of an open cycle gas turbine engine as compared to other types of internal combustion engines? What are the disadvantages?

16-25. What are three methods of improving the part load performance of a gas turbine engine?

16-26. Draw a schematic diagram of a closed cycle gas turbine, label all components, and show flow of working media.

16-27. Give four advantages and three disadvantages of a closed cycle gas turbine engine as compared to an open cycle gas turbine engine.

16-28. Why may the size of a closed cycle gas turbine engine be smaller than the same power output open cycle gas turbine engine?

16-29. Draw a schematic diagram of a semi-closed cycle gas turbine engine, label all components, and show flow of working media.

16-30. Describe how a free piston gas generator works.

16-31. List three proposed methods of transmission and reversing.

16-32. Why was the development of a marine gas turbine engine slow compared to the jet propulsion engines?

16-33. What would be the advantages of using a gas turbine engine for emergency or high speed power?

16-34. What auxiliaries will be driven by gas turbine engines?

CHAPTER XVII

JET PROPULSION ENGINES

17-1. Theory of Jet Propulsion. Jet propulsion, like all means of propulsion, is based on Newton's Second and Third Laws of Motion. Newton's Second Law states that *the rate of change of momentum in any direction is proportional to the force acting in that direction.* Newton's Third Law states that *for every action there is an equal and opposite reaction.*

With regard to vehicles operating entirely in a fluid, the reaction principle is based on imparting momentum to a mass of fluid in such a manner that the reaction of the imparted momentum furnishes a propulsive force. Peculiar to jet propulsion, however, this mass of fluid, whose velocity has been increased, is ejected from the vehicle in a jet stream. The jet aircraft draws in air and expels it to the rear at a markedly increased velocity; the rocket greatly changes the velocity of its fuel which it ejects rearward in the form of products of combustion. In each case the *action* of accelerating the mass of fluid in a given direction created a *reaction* in the opposite direction in the form of a propulsive force. The magnitude of this propulsive force is defined as *thrust.*

17-2. Types of Jet Engines. Within the category of jet engines, there are many individual types and combinations of types. Foremost of these, at least from an historical standpoint, is the *engine-jet,* for it was this engine that powered the first flyable jet aircraft. In 1932 an Italian, Campini, successfully designed and operated such an engine. It consisted of a reciprocating engine, which drove a compressor, a combustion chamber and an air exit nozzle. Since this type of system combined the inherent disadvantages of the reciprocating engine and the relatively low efficiency of the turbojet (discussed later), it is not now being produced. In contrast to the engine-jet, the *hydro-jet* engine that produces thrust directly in the water has been developed and currently shows promise for use in torpedoes and similar underwater weapons.

As a matter of interest, there is at least one ferryboat in the United States that utilizes hydro-jet propulsion. At White's Ferry, Maryland, one ferryboat, which was to have been placed in service shortly after the time of this writing, September 1954, is designed to take water in at the bow and, by means of an eight cylinder Ford industrial engine, expel it from tubes in the stern.

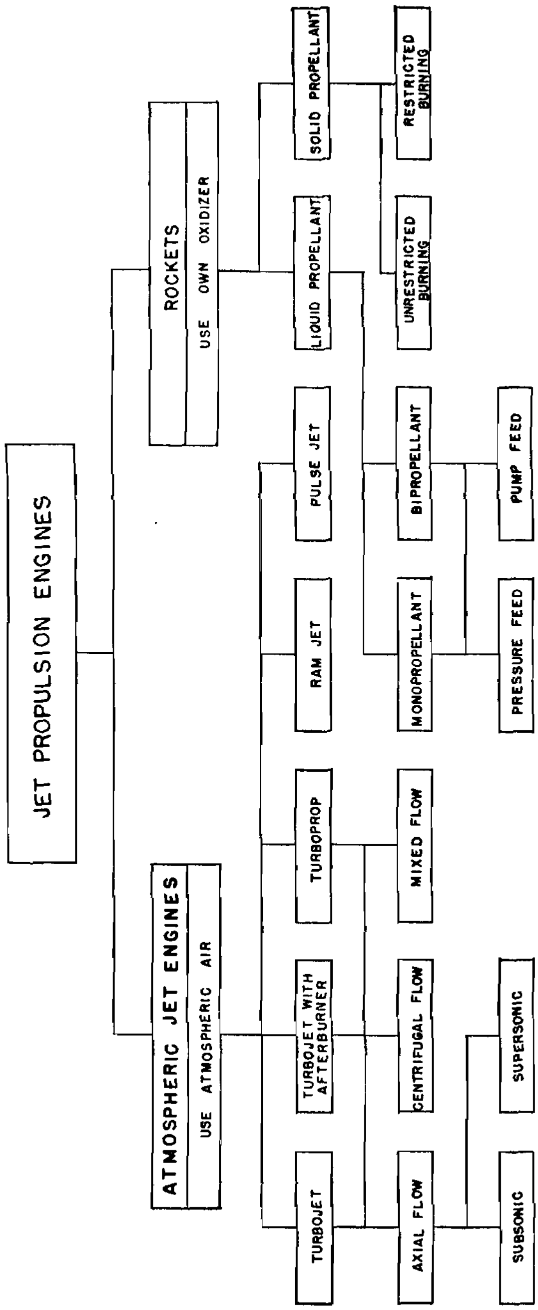


FIG. 17-1. Jet propulsion engines in current use as power plants for airplanes and guided missiles.

JET PROPULSION ENGINES

Although interesting, the study of the complete spectrum of jet engines is not within the purview of this course. This and the succeeding chapter will treat only the *thermal jet type propulsion engine.* In this engine, the heat energy is released directly into the working medium as it passes through the power plant. The heat energy is converted into kinetic energy in the form a relatively high increase in the velocity of the working medium. The acceleration of the working fluid reacts against the components of the engine creating the thrust as the working medium is ejected from the exhaust nozzle.

The jet propulsion engines are classified basically as to their method of operation. The two main categories of jet propulsion engines are the atmospheric jet engines and the rockets. The atmospheric jet engines require oxygen from the atmospheric air for the combustion of fuel. As a result, their performance depends to a great degree on the forward speed of the engine and upon the atmospheric pressure and temperature.

The rocket engine differs from the atmospheric jet engines in that the entire mass of the jet is generated from the propelants carried within the engine, i.e., the rocket engine carries its own oxidizer for the combustion of fuel and is, therefore, independent of the atmospheric air. The performance of this type of power plant is independent of the forward speed and affected to a maximum of about ten per cent by changes in altitude. Chapter XVIII of this text is devoted to the discussion of rockets. Detailed study of rocket theory, design and associated problems is therefore omitted here.

The various types of atmospheric jet engines in current use and which will be discussed in this chapter are as follows: (See Figs. 17-1 and 17-2)

(1) Steady combustion systems; continuous air flow:
- (a) Turbojet
- (b) Turbojet with afterburner (also known as turboram jet, turbojet with tail-pipe burning, and turbojet with reheater).
- (c) Turboprop (also known as propjet).
- (d) Ram jet (also known as athodyds and Lorin tube).

(2) Intermittent combustion system; intermittent air flow:
- (a) Pulse jet (also known as aeropulse, resojet, Schmidt tube, and intermittent jet).

The turbojet, turbojet with afterburner, and turboprop are modified simple open cycle gas turbine engines. The turbojet engine, Fig. 17-2 consists of an open cycle gas turbine engine (compressor, combustion

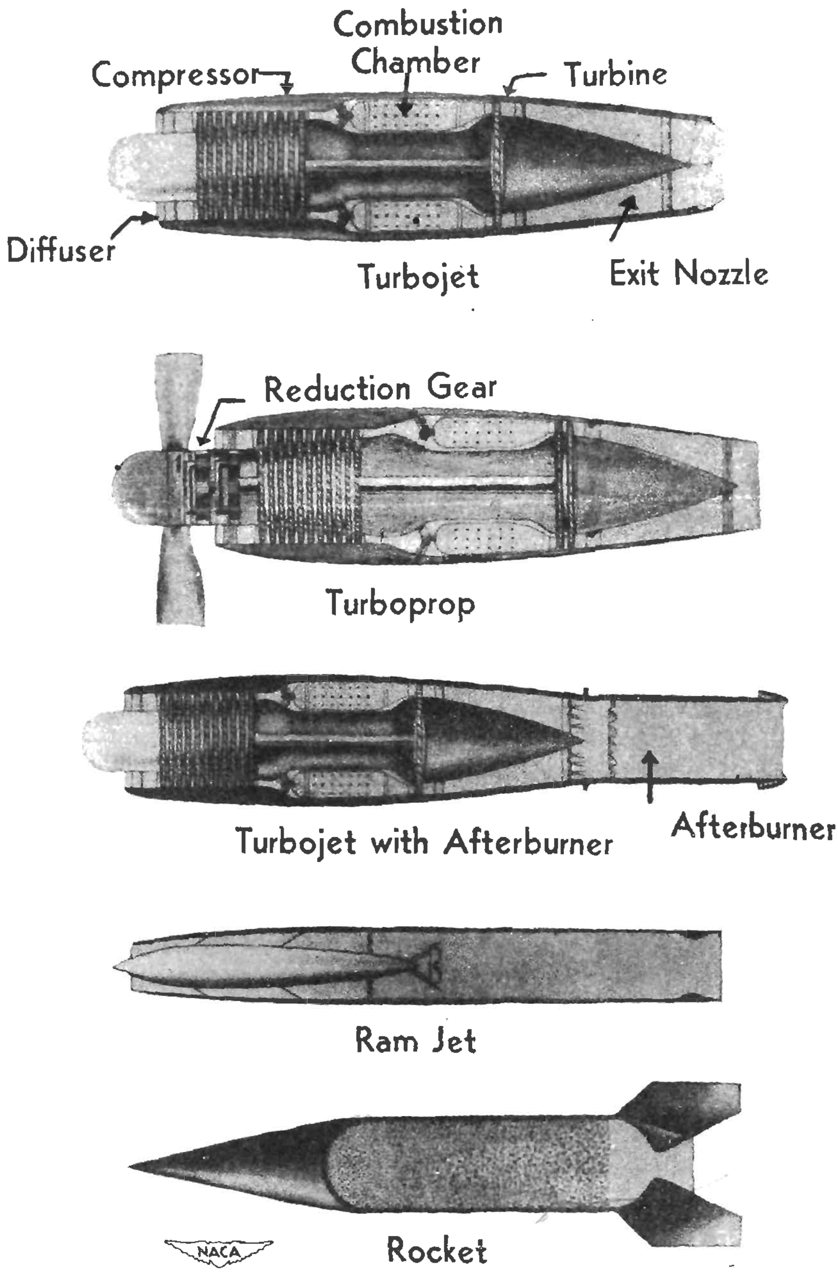


FIG. 17-2. Schematic diagrams of jet propulsion engines.

chamber, and turbine) with an entrance air diffuser added in front of the compressor and an exit nozzle added aft of the turbine. The turbojet with afterburner, Fig. 17-2 is a turbojet engine with a reheater added to the engine so that the extended tail pipe acts as a combustion chamber. The turboprop, Fig. 17-2 is a turbojet engine with extra turbine stages, a reduction gear train, and a propeller added to the engine. Approximately 80 to 90 per cent of the thrust of the turboprop engine is produced by acceleration of the air outside the engine by the propeller and about 10 to 20 per cent of the thrust is produced by the jet exit of the exhaust gases. The ram jet and the pulse jet are athodyds (aero-thermo-dynamic-ducts), i.e., a straight duct type of jet engine without compressor and turbine wheels.

The theoretical performance of these engines vary with factors which are discussed in this chapter. In addition to these factors, however, the installation configuration of a particular engine in a given vehicle introduces other factors which influence the engine's performance. For example, variations in the shape of the air inlet or exhaust nozzle may cause up to a 10% to 20% decrease in engine performance from the test stand rating. From this it can be seen that maximum coordination must be maintained between airframe and engine designers throughout the development of a new jet aircraft. Without such coordination the potential power of the engine may never be achieved.

The numbering system used by the Air Force and Navy to identify jet engines is given in Appendix D.

17-3. Energy Flow Through a Jet Engine. The flow of energy through a jet engine has certain similarities to that of a reciprocating engine. However, the manifestation of the energy differs, and also the terminology employed.

In article 17-2 the energy flow was briefly outlined. It was stated that the heat energy supplied to the engine was converted into kinetic energy. That portion of the heat supplied to the combustion chamber which was utilized to effect the change of kinetic energy of the air is termed the propulsive power. The difference between the heat supplied and the propulsive power comprises the heat loss, and the thermal efficiency of the engine is expressed as the ratio of the propulsive power to the heat supplied.

That power which is the rate of the actual useful work developed by the engine, termed thrust power, is equal to the propulsive power less the kinetic energy losses. From this it can be deduced that the propulsive efficiency of the engine is the ratio of the thrust power to propulsive power, and the overall efficiency becomes the ratio of the thrust power to the heat supplied. The propulsive and overall efficiencies of the jet engine are then comparable to the mechanical and brake thermal efficiencies, respectively, of the reciprocating engine.

17-4. Thrust, Thrust Power, and Propulsive Efficiency. The basis for comparison of jet engines is thrust, T, a term previously defined as the magnitude of propulsive force created by a jet engine. It was determined that it was dependent upon the rate of change of momentum of the working medium, air, as it passes through the engine. Since the weight rate of flow of fuel through the engine is normally in the vicinity of one per cent of the weight rate of flow of air, it will not introduce any appreciable error if it is assumed that the working medium is comprised entirely of air. Based upon this assumption, the thrust of the atmospheric jet engine can be expressed as

$$T = \frac{w_a}{g}(V_j - V_0) \qquad \text{(lbs)} \qquad \textbf{(17-1)}$$

where

w_a = air flow, lb per sec
g = 32.2 ft per sec^2
V_j = exit velocity of gases relative to the exit nozzle, ft per sec
V_0 = vehicle velocity through the air, ft per sec

Since the atmospheric air is assumed to be at rest, the velocity of the air entering the engine, relative to the engine, is the velocity of the vehicle V_0.

Thrust power, TP, the time rate of development of the useful work achieved by the engine, is the product of the thrust times the flight velocity of the vehicle, or

$$TP = TV_0 = \frac{w_a(V_j - V_0)V_0}{g} \qquad \text{(ft-lbs/sec).} \qquad \textbf{(17-2)}$$

Propulsive power, PP, representing the energy required to change the momentum of the mass flow of air, may be expressed as the difference between the rate of kinetic energies of the entering air and the exit gases, or

$$PP = \frac{w_a(V_j^2 - V_0^2)}{2g} \qquad \text{(ft-lbs/sec).} \qquad \textbf{(17-3)}$$

Therefore, the propulsive efficiency, η_p, may be expressed as

$$\eta_p = \frac{TP}{PP} = \frac{2(V_j - V_0)V_0}{V_j^2 - V_0^2} = \frac{2(V_j - V_0)V_0}{(V_j + V_0)(V_j - V_0)}$$

$$\eta_p = \frac{2V_0}{V_j + V_0} = \frac{2}{1 + V_j/V_0} \,. \qquad (17\text{-}4)$$

This is also called the Froude efficiency. From this expression it may be seen that the propulsion system approaches maximum efficiency as the velocity of the vehicle approaches the velocity of the exhaust gases. But as this occurs, the thrust and the thrust power approach zero. Thus, the ratio of velocities for maximum propulsive efficiency and for maximum power are not the same.

An alternate way to express propulsive efficiency is to express propulsive power in terms of thrust power and kinetic energy losses, i.e.,

$$PP = TP + \text{K.E. losses.}$$

The propulsive efficiency then becomes

$$\eta_p = \frac{TP}{PP} = \frac{TP}{TP + \text{K.E. losses}} \,.$$

This is the ideal propulsive efficiency equation and is equally valid for engine driven propellers and hydraulic jet propulsion as well as for the thermal jet engine.

17-5. The Turbojet Engine. The turbojet engine, while unique in regard to its *detailed* mechanisms, embodies *all of the basic principles* employed by each of the atmospheric engines mentioned in Article 17-2. A firm grasp of these principles will facilitate the understanding of *all* of the variations of the atmospheric jet engine. For this reason the turbojet should, and does, enjoy a fuller treatment in this text than do the other engines discussed.

The turbojet engine consists of a diffuser which slows down the entrance air and thereby compresses it, a simple open cycle gas turbine and an exit nozzle which expands the gas and converts the thermal energy of the exit gas into kinetic energy. The increased velocity of the air thereby produces thrust.

Figure 17-3 shows the basic arrangement of the diffuser, compressor, combustion chamber, turbine and exhaust nozzle of a turbojet engine.

In this figure, it can be seen that a pressure rise in the entering air is caused by slowing it down in the diffuser. The major part of the pressure rise is accomplished in the compressor which is driven by the turbine. Addition of heat in the combustion chamber at essentially

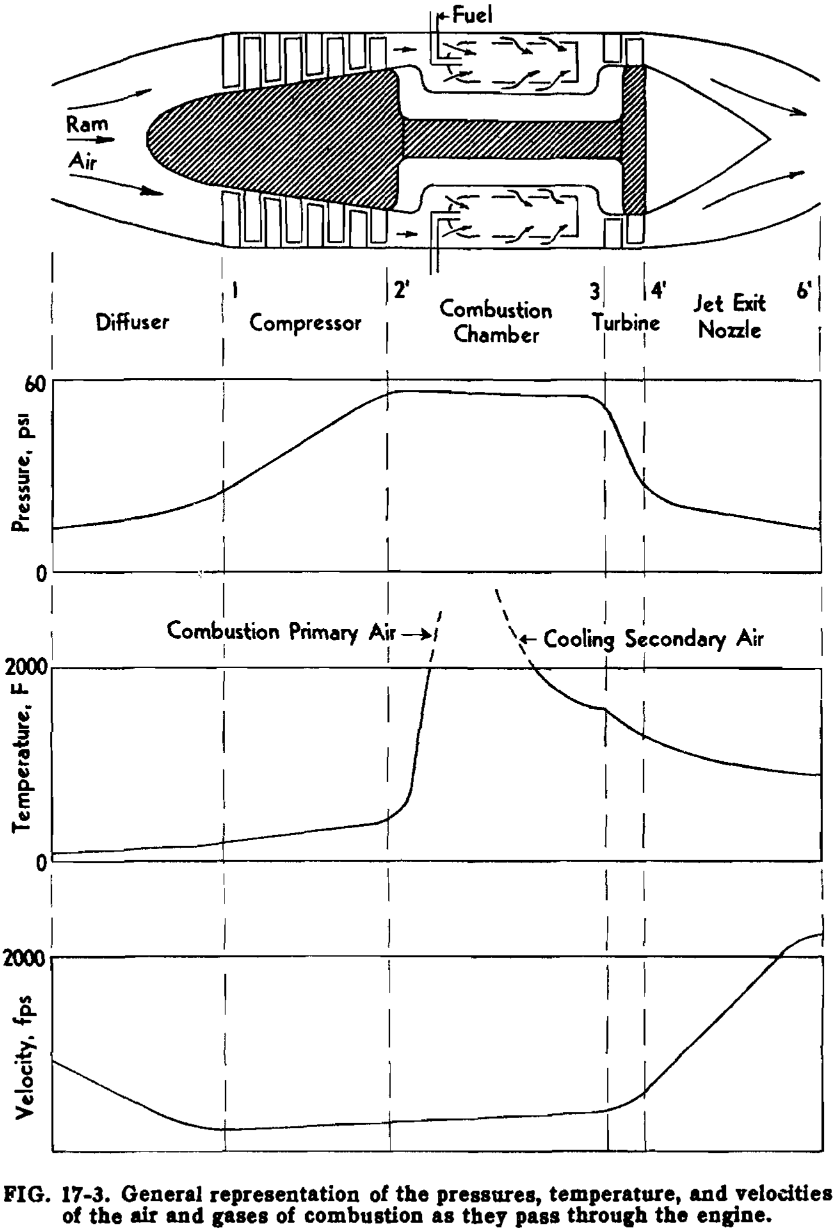


FIG. 17-3. General representation of the pressures, temperature, and velocities of the air and gases of combustion as they pass through the engine.

constant pressure causes expansion of the air. In order to fix the permissible turbine inlet temperature only a certain amount of heat may be added during the combustion process. The heated air then expands through the turbine thereby increasing its velocity while losing pressure. The turbine extracts enough energy to drive the compressor and the necessary auxiliary equipment. The hot gas is then expanded through the exit nozzle and the energy of the hot gas is converted into as much kinetic energy as is possible. This change in velocity of the air passing through the engine multiplied by the mass flow of the air is the change of momentum which produces thrust.

17-6. Development of the Turbojet Engine. Frequently the discussion of certain aspects of the jet engine are almost pointless without comparison to a reference plane. The reciprocating aircraft engine, as the recent prime mover of all aircraft, is admirably suited to serve as such a plane. For example, in the thrust equation, 17-1, if we interpret V_j to indicate the velocity of the air leaving the propeller, this equation applies equally as well to the propeller driven aircraft as to the turbojet. The main difference between the two being that the propeller aircraft accelerates a large mass of air with a small change in velocity while the turbojet accelerates a small mass of air with a large change in velocity. Inspection of the propulsive efficiency equation, 17-4, leads to the deduction that the propulsive efficiency of the propeller aircraft is greater than that of the turbojet since the ratio V_j/V_0 more closely approaches unity. Why then have we turned to the turbojet as the better means of propulsion? The answer lies in the limitations of the reciprocating engine at high speeds.

At the outset of World War II, the military required and demanded simpler engines of higher output and lighter weight, with less frontal area, and with reduced profile drag in order to obtain aircraft with higher speeds. These needs could not be filled by the then existing techniques and the reciprocating aircraft engine which is in itself a masterpiece of engineering ingenuity and work. Largely due to military requirements in the past, it has been developed to a high peak of efficiency, light weight, and reliability as compared to the reciprocating engines of two decades ago. However, the power ratings of this type of engine have reached the point where any further increase probably can be accomplished only by the addition of more cylinders, which rapidly increases the complexity and weight, or by a major improvement in fuels. The need for higher power ratings is coupled with the demand for both higher speeds and larger aircraft carrying greater pay loads. As the flight speeds are increased, the power required to overcome the drag of the airplane rapidly rises. In the reciprocating engine with its large frontal area and required air cooling, which causes a large nacelle profile drag, an increase in speeds above 300 mph gives a very rapid rise to the drag (Fig. 17-15) and therefore the power required to overcome it. Also the efficiency of the present day propellers drops rapidly above speeds of 400 to 450 mph (Fig. 17-15) due to the breakdown of the streamline flow around the blade section as the propeller tips approach and exceed sonic velocities. An increase in the power rating of the engine is accompanied by an increase in the size, weight, and number of cylinders of the engine, which in turn causes added complications to maintenance, control, accessories, and installation. The reciprocating aircraft engine is restricted to power ratings below 5000 hp and speeds below 500 mph, unless there is a major improvement in fuels and in propeller design. Eor these reasons then, the scientist and engineer turned to the atmospheric jet engine.

Prior to this great impetus given the research and development of the turbojet during World War II, the history of this engine fairly closely follows that of the gas turbine as discussed in Article 16-1.

In 1930, Air Commodore Whittle designed and patented a turbojet engine. In 1936, work was begun in England on an axial flow and a centrifugal flow turbojet engine which was successfully run and tested in 1937. At the same time that England was carrying out the design and research, the Germans were making rapid progress in the field of jet propulsion engines. The Germans in August 1939 flew the first airplane, the Heinkel 178, powered by a turbojet engine, HeS-3. However, this particular engine was later abandoned due to low thrust and short life. For this reason, the flight of the turbojet powered British airplane, Gloster E 28/29, in May 1942, gave rise to England's claim to have flown the first successful aircraft of its type. Thus, the turbojet engine has been developed during the past decade from the first basic experimental models to the widespread use of this engine in modern aviation.

The needs and demands being fulfilled by the turbojet engine are:

(1) Low specific weight—one-fourth to one-half of the reciprocating engine.
(2) Relative simplicity—no unbalanced forces or reciprocating parts.
(3) Small frontal area reduced air cooling problem—less than one-fourth the frontal area of the reciprocating engine giving a large decrease in nacelle drag and consequently giving a greater available excess thrust or power, particularly at high speeds.
(4) Not restricted in power output—engines can be built with greatly increased power output over that of the reciprocating engine without the accompanying disadvantages.
(5) Higher speeds can be obtained—not restricted by a propeller to speeds below 500 mph.

17-7. Turbojet Engine Cycle. The basic cycle for the turbojet engine is the Joule or Brayton cycle. The temperature-entropy diagram for a typical turbojet engine using the following simplifying assumptions is shown in Fig. 17-4. The simplifying assumptions are those which were used in the calculation of the open cycle gas turbine, Article 16-4, and, in addition, the following three assumptions:

(1) The entering atmospheric air is diffused isentropically from velocity V_0 down to zero ($V_1 = 0$), i.e., the diffuser has an efficiency of 100 per cent (process 0 to 1 in Figs. 17-3 and 17-4).
(2) The hot gases leaving the turbine are expanded isentropically in the exit nozzle, i.e., the efficiency of the exit nozzle is 100 per cent (process 4′ to 6′ in Figs. 17-3 and 17-4).
(3) The turbine extracts only enough energy from the hot gases to drive the compressor; therefore, the turbine work is assumed to be equal to the compressor work, $wk_{Tu} = wk_C$ ($h_3 - h_{4'} = h_{2'} - h_1$ in Figs. 17-3 and 17-4).

In the ideal diffuser, the atmospheric air entering at a velocity V_0 is diffused isentropically down to zero velocity at state 1 ($V_1 = 0$) which

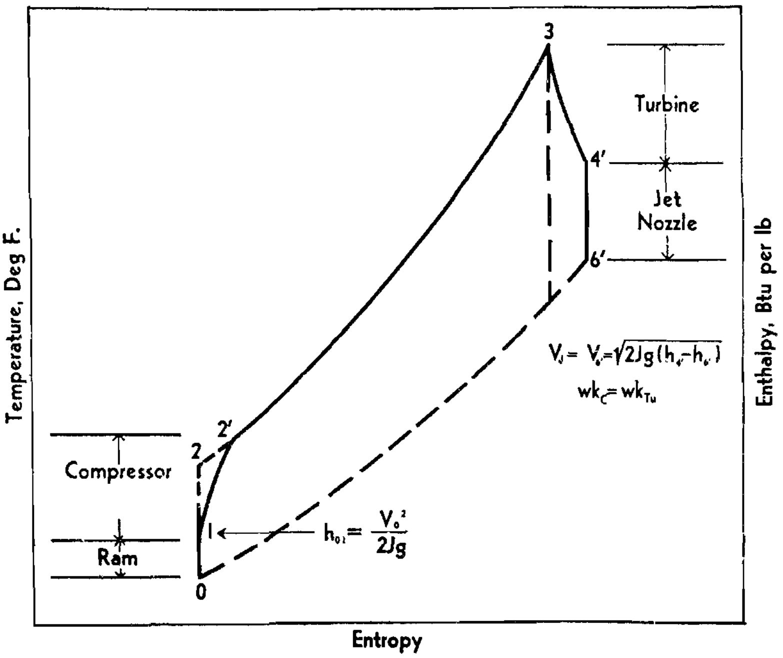


FIG. 17-4. Temperature-entropy diagram of a typical turbojet engine cycle.

is the entrance to the compressor. In the actual diffuser the process is irreversible adiabatic and the air leaves the diffuser at a velocity between 200 and 400 ft per sec. The process in the diffuser, process 0 to 1 in Figs. 17-3 and 17-4 is termed ram compression. Ram compression is the transformation of the kinetic energy of the entering air into pressure energy. The rise in the ram pressure ratio, p_1/p_0, with increase in Mach number[1] of the vehicle is shown in Fig. 17-5.

From Fig. 17-5, it may be determined that the ram acts in effect like a compressor. At a Mach number of 2.0, it is found that the ideal ram pressure ratio is 8.0. At this high a Mach number it becomes economical to go to the ram jet engine and do away with the mechanical compressor and turbine wheels.

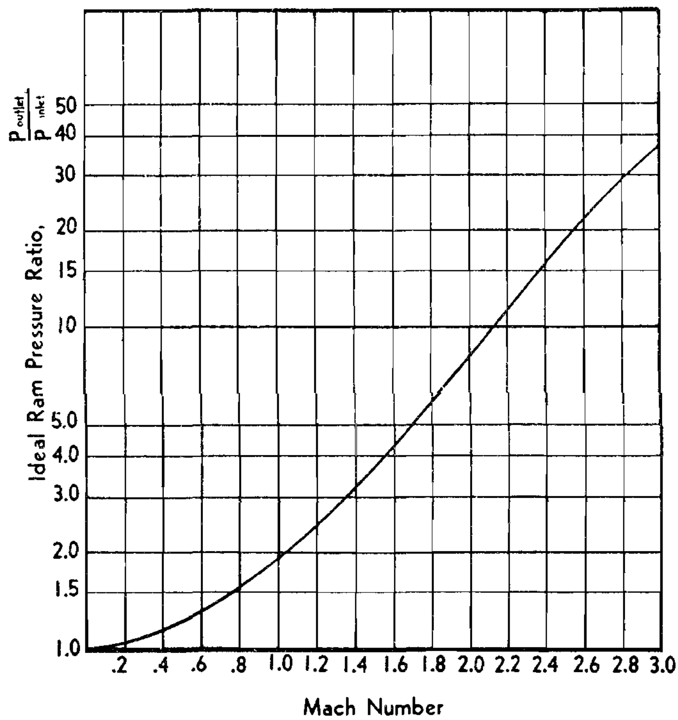


FIG. 17-5. Ideal ram pressure ratio vs. Mach number of vehicle calculated for sea level conditions.

[1] Mach number is the ratio of the flight speed of the airplane or missile to sonic velocity, where sonic velocity, a, may be determined from

$$a = \sqrt{gkRT_0} \text{ (ft/sec).}$$

The general flow energy equation applies to the turbojet cycle. The flow energy equation from thermodynamics between states 0 and 1 is

$$\frac{V_0^2}{2gJ} + h_0 + q_{01} = \frac{V_1^2}{2gJ} + h_1 + \frac{wk_{01}}{J}$$

Since in an ideal diffuser $V_1 = 0$, $q_{01} = 0$, $wk_{01} = 0$, the enthalpy at state 1 from the flow energy equation becomes

$$h_1 = h_0 + \frac{V_0^2}{2gJ} = h_0 + \frac{V_0^2}{50{,}000} \qquad \text{(Btu/lb)} \qquad (17\text{-}5)$$

where h_0 = enthalpy of the atmospheric air in Btu per lb.

The air leaving the diffuser enters the compressor where the pressure is raised to p_2. The actual compressor work from equation 16-9 is given as

$$wk_C = \frac{(h_2 - h_1)}{\eta_c} = h_{2'} - h_1 \qquad \text{(Btu/lb)}. \qquad (17\text{-}6)$$

The heat supplied to the air in the combustion chamber from equation 16-11 becomes

$$q_S = (h_3 - h_{2'}) \qquad \text{(Btu/lb)}. \qquad (17\text{-}7)$$

From the simplifying assumptions, the work of the turbine is equal to the work of the compressor. Therefore,

$$wk_{Tu} = wk_C = h_{2'} - h_1 = h_3 - h_{4'} \qquad \text{(Btu/lb)} \qquad (17\text{-}8)$$

The enthalpy at state 4′, the exit from the turbine and the entrance to the jet exit nozzle, may then be calculated from

$$h_{4'} = h_3 - wk_C = h_3 - wk_{Tu} \qquad \text{(Btu/lb)} \qquad (17\text{-}9)$$

The exit or jet velocity, V_j (state 6′), of the gases relative to the exit nozzle, when it is assumed that velocity $V_{4'} = 0$, may be determined from the flow energy equation where

$$h_{4'} = h_{6'} + \frac{V_j^2}{2gJ} \cdot \qquad (17\text{-}10)$$

Therefore, V_j may be calculated from

$$V_j = V_{6'} = [2gJ(h_{4'} - h_{6'})]^{\frac{1}{2}} \qquad \text{(ft/sec)}. \qquad (17\text{-}11)$$

The thermal efficiency of the turbojet cycle is the net work, which is termed propulsive power in this case, developed by the cycle divided by the total heat supplied. The propulsive power (PP), from Article 17-4, was defined as the difference between the kinetic energies of the entering and leaving air, i.e.,

$$PP = \frac{w_a}{2g}(V_j^2 - V_0^2) \qquad \text{(ft-lb/sec)}.$$

The total rate flow of the heat supplied (Q_S) to the cycle in terms of foot-pounds per second may be written

$$Q_S = w_a J(h_3 - h_{2'}) \qquad \text{(ft-lb/sec)}.$$

The thermal efficiency of the turbojet engine cycle becomes

$$\eta_t = \frac{PP}{Q_S} = \frac{V_j^2 - V_0^2}{2gJ(h_3 - h_{2'})}. \tag{17-12}$$

The thermal efficiency of the turbojet cycle may also be expressed in terms of enthalpies. The jet exit velocity (V_j) may be expressed in terms in enthalpies, equation (17-11),

$$\frac{V_j^2}{2gJ} = h_{4'} - h_{6'}.$$

The velocity of the entering air (V_0) relative to the diffuser may also be expressed in terms of enthalpies, equation (17-5),

$$\frac{V_0^2}{2gJ} = h_1 - h_0.$$

In order to obtain the net energy transferred in terms of the entire cycle, the turbine work ($h_3 - h_{4'}$) and compressor work ($h_{2'} - h_1$) must be considered. Combining the above, the net energy gained of the entire cycle is then,

$$\begin{aligned} \text{net energy gained} &= \left(\frac{V_j^2}{2gJ} + wk_{Tu}\right) - \left(\frac{V_0^2}{2gJ} + wk_c\right) \\ &= (h_3 - h_{6'}) - (h_{2'} - h_0). \end{aligned}$$

The thermal efficiency expressed in terms of enthalpies is

$$\eta_t = \frac{(h_3 - h_{6'}) - (h_{2'} - h_0)}{h_3 - h_{2'}}. \tag{17-13}$$

A simple rearrangement of this equation shows

$$\eta_t = \frac{(h_3 - h_{2'}) - (h_{6'} - h_0)}{h_3 - h_{2'}}$$

or

$$\eta_t = \frac{q_S - q_R}{q_S}$$

the basic thermal efficiency equation for any heat engine.

Knowing w_a, V_0, V_j, and Q_S, the various performance variables of the turbojet engine may be calculated. Example problems for the turbojet engine are given in Appendix A.

17-8. Turbojet Components. The basic components of the turbojet engine, with the exception of the diffuser and the exhaust system, are identical with those of the open cycle gas turbine. However, certain of these will be discussed in greater detail due to their enormous affect on the overall efficiency of the turbojet engine.

The *diffuser* of a turbo-jet engine is normally provided by the airplane manufacturer. This important item must provide the greatest possible pressure rise by slowing the incoming air and converting its kinetic energy into pressure. In our simplifying assumptions the complete conversion of kinetic energy to pressure is assumed. This would, of course, not allow the entering air to reach the compressor while in reality the air leaves the diffuser with a velocity from 200 to 400 ft/sec.

From Fig. 17-5 it is seen that as the vehicle reaches fairly high Mach Numbers, the diffuser plays an increasingly larger part in accomplishing the necessary pressure rise. In fact, at velocities in excess of Mach 1.5, the ram compression alone is sufficient to operate the engine, and the compressor and its turbine may be dispensed with. This is actually done in the ram jet engine.

The shape, area, and location of the actual air inlet in an airplane is highly important and, therefore, various studies are underway to obtain optimum designs for high speed aircraft. Variable area entrance diffusers are being developed for new aircraft in order to maintain high diffuser efficiency for both high and low speed operation.

The *centrifugal compressor* operates by taking air into vanes located at the hub of the rotating disc and then forcing it outward by the rapidly rotating blades. The centrifugal force accelerates the air to high velocity at which time it reaches the compressor diffuser. There, the air is slowed and its kinetic energy converted into pressure. Fig. 17-6 shows pressure ratio vs tip speed for a typical centrifugal compressor.

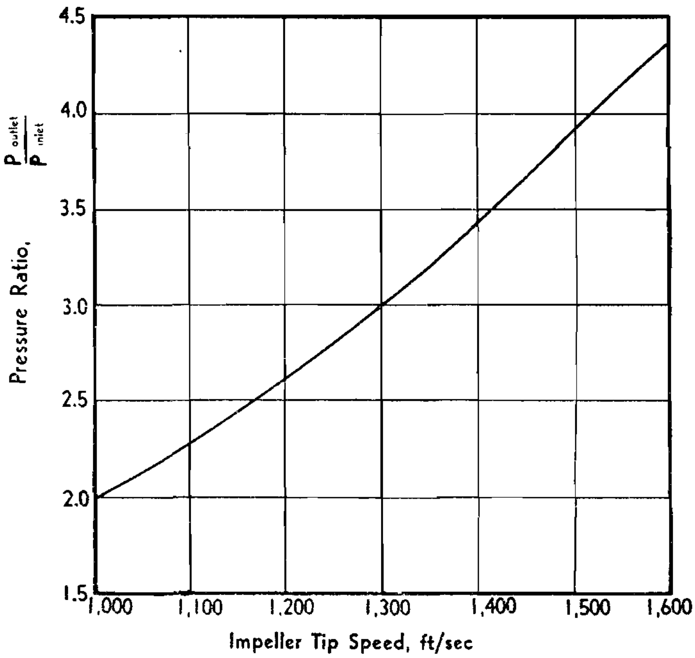


FIG. 17-6. Pressure ratio vs. tip speed for a centrifugal compressor (courtesy of W. R. Hawthorne and S.A.E.).

One of the primary disadvantages of this compressor is the lack of straight-through air flow, i.e., the air which leaves the compressor in a radial direction must then change direction so it parallels the axis of the engine. This leads to the use of ducting with the attendant pressure losses.

Although the centrifugal compressor is rugged and reliable, the excessive diameters involved make its use undesirable in engines with thrust ratings over about 6,000 pounds.

The *axial compressor* is essentially a reaction turbine operated as an air pump. It consists of a rotor with several rows of air-foil blades and a stator with rows of stationary blades.

Its construction is complex and expensive. Due to the elevated temperatures involved at the discharge end, the air-foil blades there currently must be fabricated of heat resisting steel, while at the suction ends aluminum is employed. Inasmuch as aircraft compressors must be as light as possible, a continuing research is being conducted to find light, strong, heat and corrosion resisting metals for use in the turbojet.

The major advantages of the axial compressor, i.e., straight-through air flow, capable of multi-staging and small frontal area, outweigh its sensitivity and fragility. Therefore, it is the current choice for use in turbojet engines of high thrust output.

A variation of the axial compressor, the *twin-spool* (dual-spool or co-axial) compressor, has two or more sections, each revolving at or near the optimum speed for its pressure ratio and volume of air flow. This system is relatively complex but is the only means currently available to obtain pressure ratios of 10:1 or greater. In order to achieve ultimate power, such pressure ratios are necessary for turbojet engines with thrust ratings in the region of 10,000 lbs. of thrust or greater. The J-57 (see Fig. 17-7) engine, the most powerful engine produced at present, and the J-67, an even more powerful engine which is currently being developed, incorporate the use of this compressor.

New developments in compressors include the *mixed-flow compressor* which is essentially a centrifugal compressor with the rotating blades curved at the ends to cause the exit air to leave axially.

The J-44, a small (1,000 lb. thrust), expendable, turbojet engine used for target and guided missile power plants uses this type of compressor.

The *supersonic compressor* is essentially an axial type compressor with very high rotative speeds so that the airfoil compressor blades are moving much faster than the local speed of sound. It's performance is such that a single stage supersonic compressor can approximate the performance of some multistage sub-sonic compressors in use today. Multistage supersonic compressors hold great promise for the future but as yet are not sufficiently developed to be of practical use.

Combustion Chambers. The design of combustion chambers is one of the most difficult parts of jet engine design. For example, present day combustion chambers have heat inputs of the order of 6,000,000 BTU per cubic foot of combustion chamber per hour. When we consider that the average input to the firebox of a naval steam boiler is in the order of 60,000 BTU per cubic foot per hour, we can better appreciate the complex problems involved.

The criteria for a well designed combustion chamber are many. The most important of these are:

(1) Flame stability.
(2) Low pressure losses.
(3) High combustion efficiency.
(4) Low carbon deposits.
(5) Light weight.
(6) Satisfactory life.

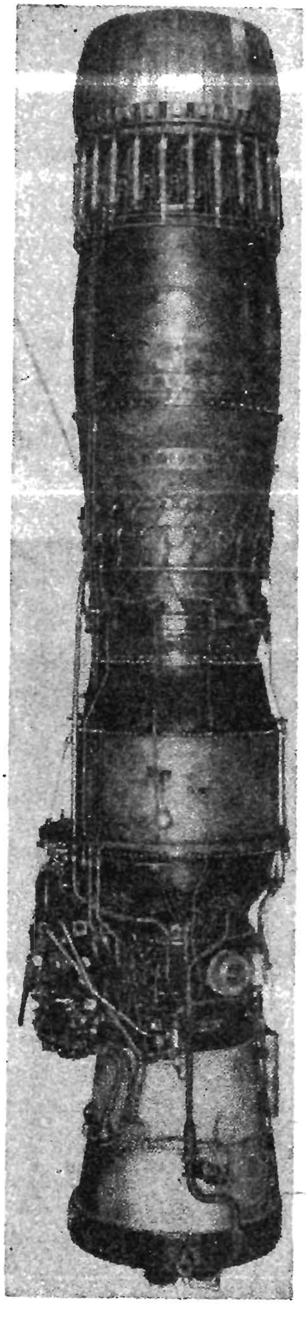


FIG. 17-7. Turbojet engine, J-57, designed and built by Pratt and Whitney Aircraft (courtesy of Pratt and Whitney Aircraft).

(7) Minimum size or frontal area.
(8) Thorough blending of primary and secondary air to provide uniform temperature to the turbine.

The major problem of the combustion chamber is that of flame stability, i.e., the flame must remain in the chamber and remain steady. If it does not, the phenomenon known as a "blowout" or "flameout" occurs. The cause for a flameout may be one of two, either the flame is blown out of the exit of the chamber in much the same manner that the flame may be blown away from a candle wick, or the flame is extinguished by excessively rich A/F mixtures. Complicating the problem is the

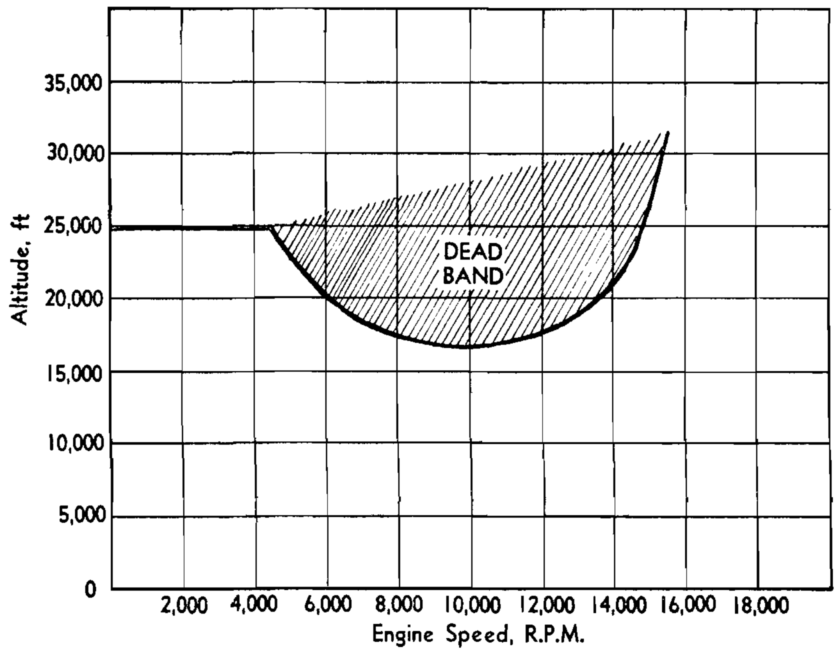


FIG. 17-8. The dead band chart for a typical turbojet engine. Blowout failures occur in Dead Band (courtesy of *Aviation Week*).

high speed of the air passing through the chamber (150 to 400 ft/sec). This speed is not constant but varies with the change in power output of the engine. Figure 17-8 shows the "dead band" region, as regards altitude and engine RPM, where flameouts occurred in early combustion chambers.

The combustion chambers are classified by their physical shape and by the type of airflow through them. The two basic types being the

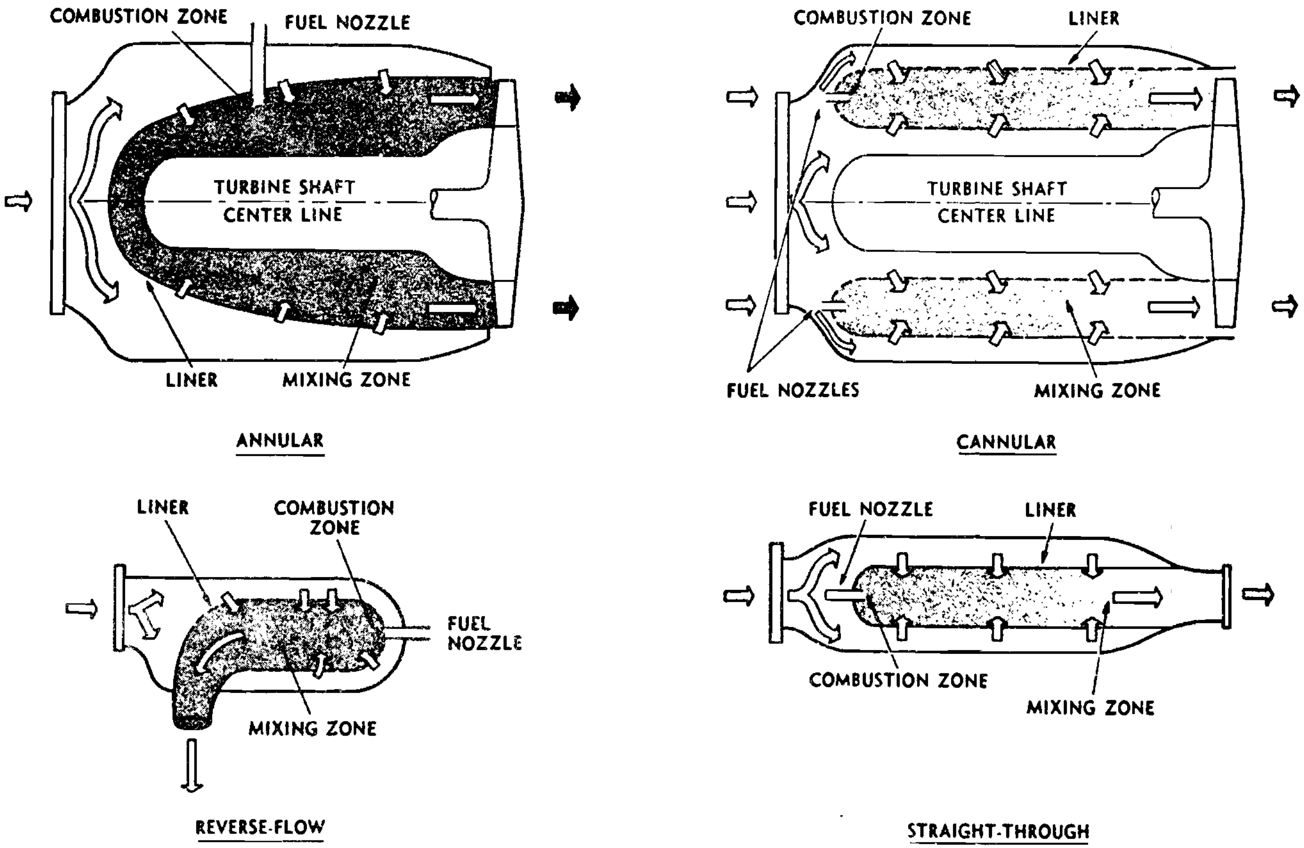


FIG. 17-9. Types of Combustion Chambers (Courtesy General Electric Company).

"can" and the "annular" construction. The annular type combustion chamber consists of one toroidal shaped chamber with straight-through air flow where the entire combustion for the engine is accomplished. By virtue of the fact that it requires no additional ducting, where pressure losses are likely to occur, it is theoretically the best of the two types. However, it has certain disadvantages. For example, each prototype engine requires that the annular chamber be designed to fit its particular needs and, once installed, it is difficult of access and maintenance. Practical considerations then favor the use of the can type chamber. It can be easily replaced in the engine and, once a good design has been achieved, it can be used in almost any turbojet engine. All that is required is to provide a sufficient number of chambers to handle the required flow of air. Figure 17-9 illustrates these two basic types of chambers as well as certain of their variations. The "cannullar" type shown incorporates some of the advantages of both types.

Turbines. Turbines of the impulse, reaction and a combination of both types are used to drive the compressor and some auxiliaries of the turbojet engine. In general, it may be stated that those engines of relatively low thrust and simple design employ the impulse type, while those of large thrust employ the reaction and combination types.

By far the greatest problem in turbine design lies in the fact that available materials for the fabrication of the buckets and vanes seriously limit the permissible inlet temperature. A great deal of research has been done in order to find a satisfactory solution to the problem. As a result, various schemes of bucket cooling, ceramic coatings and others have been developed. At present, however, none of these have proved completely successful although some have promise.

Normally a single stage turbine is used to drive the compressor. In the twin-spool compressor engines there is a turbine for each compressor. In the J-57 engine one turbine is double staged, giving a total of three turbine stages that are extracting energy from the gases. Increasing the static thrust rating of turbojet engines means larger compressors which in turn require more power to operate them. As a result, multistage turbines will probably become more common in turbojet engines as the demand for higher thrust is increased.

Exhaust System. The exhaust nozzle is the place where the thermal energy of the gas is converted into kinetic energy. Regardless of how efficiently the other components have fulfilled their individual function, the ultimate thrust of the engine can not be achieved unless the exhaust nozzle is properly designed.

The exhaust system must fulfill three basic requirements:

(1) Straighten the gas flow to an axial direction after it leaves the turbine. (Most of this is done in the exhaust cone.)
(2) Confine the exhaust gases and direct the flow to a practical discharge point.
(3) Increase the discharge velocity to as high a value as possible. (This is done by expanding the gas through a nozzle.)

A nozzle is a flow passage where the static pressure decreases and the velocity increases in the direction of flow.

At any section across a nozzle the continuity equation applies:

$\frac{1}{v} \times A \times V =$ weight of gas moving across any nozzle section.

$$A = \frac{wv}{V} \qquad (17\text{-}14)$$

where

A = Section Area
w = weight rate of flow of gas (lb/sec)
v = specific volume of gas (ft^3/lb)
V = Velocity of gas (ft/sec)

By rearrangement,

$$V = \frac{wv}{A}.$$

From inspection of this equation it becomes apparent that if the exit area of the nozzle were reduced to minute size and "w" were held constant, V (in this case equal to V_j) and, therefore, the kinetic energy could reach infinity. In reality, however, there is a limit imposed by the peculiarities of supersonic flow.

As the area of the throat of the nozzle is decreased, the velocity of the exiting gases increases until the critical velocity, i.e., Mach 1, is reached. Beyond that point a further decrease in area will result in back pressure in the engine. In an actual engine this increase in back pressure would cause the turbine to develop less power and the compressor would slow down, thereby decreasing the mass of air until a new stable operating condition was obtained.

The ratio of pressures in the nozzle $P_r = \frac{P \text{ nozzle inlet}}{P \text{ nozzle outlet}}$ is also a determining factor in obtaining maximum V_j. Since the velocity along a streamline will increase as the pressure decreases, the greater the pressure ratio the greater the exit velocity theoretically achievable. Here too supersonic flow imposes a limit and, beyond the critical velocity, a further increase in the pressure ratio does not result in an increase of V_j.

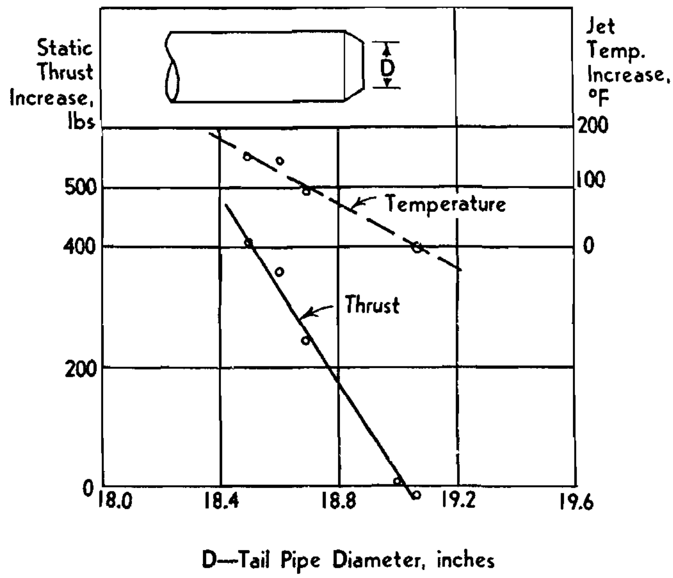
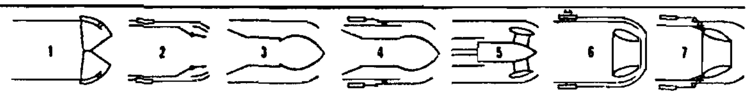


FIG. 17-10. Effect of tail-pipe nozzle diameter on static thrust and jet temperature of a jet engine (courtesy Clarence L. Johnson and I.A.S.).

From the foregoing paragraphs it may be deduced that given a fixed maximum pressure at the nozzle entrance and a fixed outside pressure, there is only one throat area that gives optimum thrust. Figure 17-10 illustrates the variation of thrust of a typical turbojet engine as the function of the nozzle throat area. In order, then, for the exhaust system to best meet the entire range of conditions encountered by the engine, it should incorporate a variable exit nozzle. This is actually done on high performance engines. Figure 17-11 shows a variety of the variable area nozzles.

Although the variable area nozzle permits the turbojet to operate at maximum efficiency over a wide range of power output, its utilization embodies certain disadvantages. The cost, weight, and complexity of the exhaust system are increased markedly. Due to these disadvantages then, the simple, fixed jet nozzle is used whenever possible.



NOZZLES	1	2	3	4	5	6	7
TYPE	CLAMSHELL	FINGER OR IRIS	CENTER PLUG	CENTER PLUG Movable Shroud	ANNULAR RING	ANNULAR RING	ANNULAR RING Movable Shroud
Aerodynamic Efficiency Internal	High	High	High	High	High	High	High
Aerodynamic Efficiency External	Poor	Fair to Good	Good	Good	Good	Fair to Good	Good
Weight, lb	200	200	200	200 +	200	200 −	200 +
Actuating Force, lb	800–1000	4000	2500 vented 4500 unvented	10,000–15,000	4500	4500	10,000–15,000
Actuator Travel for 65% Area Change, in.	3.5	5	22	22	15–26	15–26	15–26

FIG. 17-11. Various types of variable-area nozzles for turbojet engines. Figures are merely representative for one size of engine (courtesy General Electric Co.).

17-9. Turbojet Engine Performance. The effect of the variation of pressure ratio, atmospheric air temperature, component efficiencies, and turbine inlet temperature on the performance of a simple open cycle gas turbine as discussed in Chapter XVI apply equally as well for the turbojet engine. However, the turbojet engines have the additional variables of speed and altitude which must be taken into account in discussing their performance characteristics.

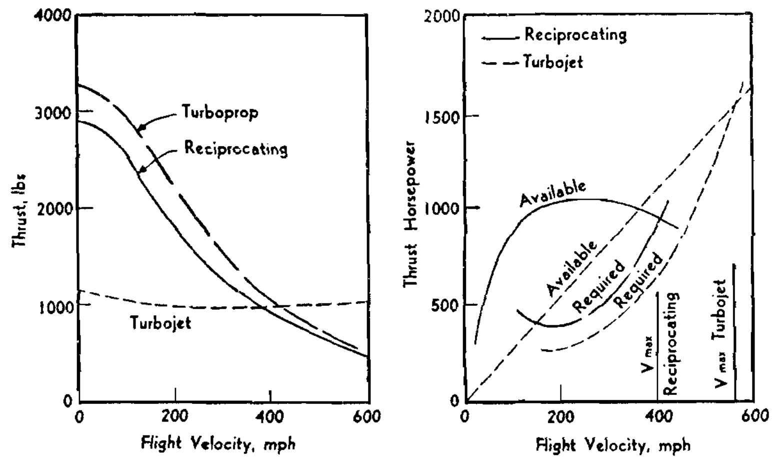


FIG. 17-12. General curves of variation of thrust, thrust horsepower available and thrust horsepower required with flight speed for a typical turbojet engine and a reciprocating engine having the same power at 375 mph.

The turbojet engines are rated on a basis of thrust in contrast to the rating of reciprocating engines on a basis of horsepower. The thrust of the turbojet engine is affected very little by the forward flight speed, i.e., the thrust is nearly constant over the subsonic speed ranges as shown in Fig. 17-12. Figure 17-12 also gives a comparison of a turbojet engine and a reciprocating engine on a basis of equal thrust and power at 375 mph. The thrust drops slightly and then increases to remain approximately constant. The basic equation for thrust

$$T = \frac{w_a}{g}(V_j - V_0)$$

indicates that either the weight rate of air flow or the jet exit velocity or both must be increased as the flight speed is increased in order to maintain a nearly constant thrust. Both w_a and V_j increase but the effect of the increase in V_j is small compared to w_a. As the flight speed increases, there is an increase in the pressure and density of the air at

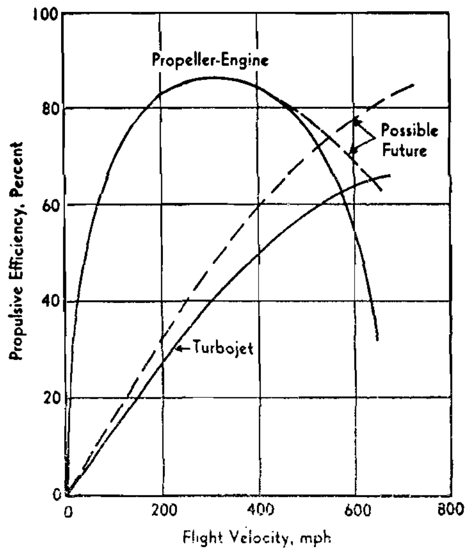


FIG. 17-13 Variation of the propulsive efficiency with flight speed.

the compressor inlet due to the ram effect that gives an increase in the weight rate of flow of air through the engine. From Fig. 17-5 it is found that the increase in the ram effect is relatively small in the lower half of the subsonic speed range where V_0 increases at a faster rate, and in the upper half it increases rapidly at a faster rate than V_0. This effect causes the dip and then increase in the thrust over the subsonic flight speed range.

The thrust of a turbojet engine is easily measured on a test stand as compared to the horsepower which is zero on the test stand since the flight speed is zero. This in addition to the nearly constant thrust with

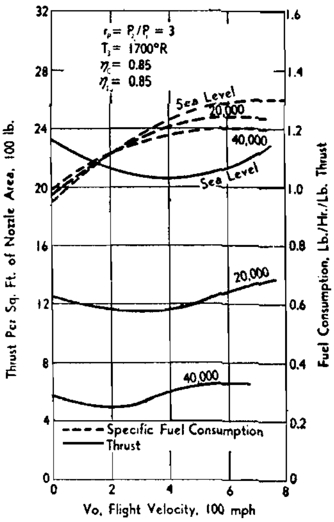


FIG. 17-14(a). Effect of flight velocity and altitude on the thrust and fuel consumption of a turbojet engine (reproduced from M. J. Zucrow, Trans. A.S.M.E., May 1946).

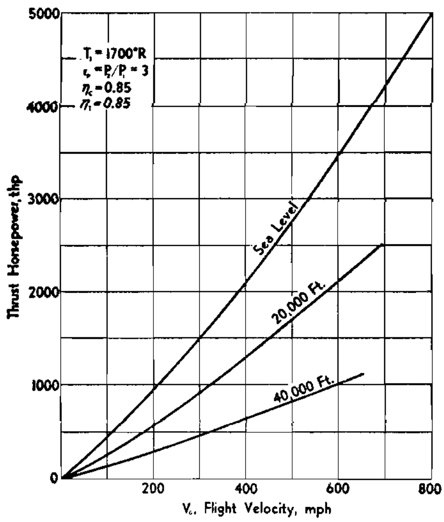


FIG. 17-14(b). Variation of thrust horsepower with flight velocity at different altitudes for a turbojet engine (reproduced from M. J. Zucrow, Trans. A.S.M.E., May 1946).

flight speed offers a better basis of comparison and rating of the turbojet engines than does the horsepower. Since the thrust remains nearly constant with variation in flight speed, the thrust horsepower,

$$\text{thp} = \frac{TV_0\,(\text{mph})}{375},$$

will vary directly with the flight speed, V_0, as shown in Fig. 17-12. It is found from the thrust horsepower equation that one pound thrust equals one horsepower at a flight velocity of 375 mph.

Figure 17-12 indicates that the horsepower of the reciprocating engine, which is the basis used in rating this type of engine, is nearly constant over its effective speed range and that the thrust is high in the low speed range but decreases rapidly with an increase in speed. The curves show that the reciprocating engine has much higher thrust and power available for take-off and in the lower speed ranges for climb than the turbojet engine. With aircraft of equal weight, the turbojet engine aircraft will require a much longer take-off run. This disadvantage is somewhat obviated by increasing the thrust of a turbojet engine by means of thrust augmentation (Article 17-11).

The drop-off in the horsepower curve for the reciprocating engine at high speeds is due to the decrease in the propulsive efficiency, Fig. 17-13. Above 400 to 450 mph the propulsive efficiency of the present day propeller decreases due to the compressibility effect on the propeller tips which causes a breakdown of the air flow and a rapid loss in propeller efficiency. This, coupled with the rapid increase in drag of the nacelle and the cooling system as the flight speed of a reciprocating aircraft engine is increased (Fig. 17-15) restrict the present day reciprocating aircraft to speeds below 500 mph. It is noted that the general curve of power required in Fig. 17-12 for the turbojet engine is below that of the reciprocating engine due to the smaller frontal diameter of the turbojet engine (approximately one-fourth to one-half that of the reciprocating engine) and due to the fact that the turbojet engine has no cooling drag (Fig. 17-15). This gives the turbojet aircraft the advantage of being able to obtain higher flight speeds, V_{max}, than reciprocating engine aircraft. The maximum velocity obtainable is the speed where the power required curve crosses the power available curve.

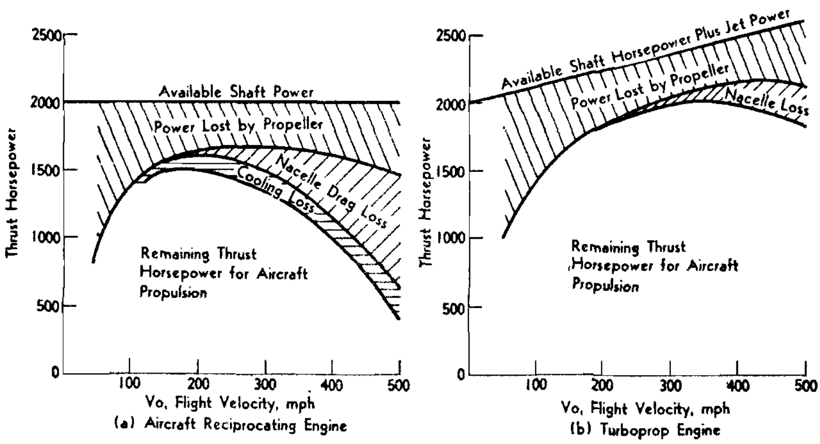


FIG. 17-15. Variation in the thrust horsepower available for aircraft propulsion for a reciprocating engine and a turboprop engine as the flight velocity is increased (reproduced from F. W. Godsey and C. D. Flagle, ref. 17-6).

The propulsive efficiency of the reciprocating engine is excellent in the lower speed ranges while that of the turbojet engine is low as shown in Fig. 17-13. The turbojet engine propulsive efficiency increases with the increase in flight speed which indicates that it is most efficient and economical to fly the turbojet engine at high speeds.

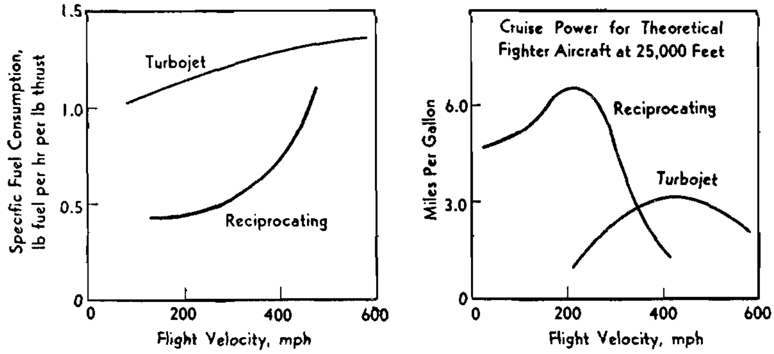


FIG. 17-16. General comparison of the variation of the specific fuel consumption and miles per gallon with flight velocity of a turbojet engine and a reciprocating engine.

The fuel consumption of the turbojet engine is high compared to the fuel consumption of the reciprocating engine in their respective most economical cruise-speed ranges (Figs. 17-16 and 17-17), where the cruise-speed range of the present day turbojet engine aircraft is in the vicinity of 450 to 500 mph as in contrast to the cruise-speed range in the vicinity of 200 mph for the reciprocating engine aircraft. The specific fuel consumption, lb fuel per lb thrust per hr, of the turbojet engines is in the range of 1.0 to 1.3 (see Table 17-1 for typical engines) while those of the reciprocating engines are in the range of 0.4 to 0.5. The *variation in the specific fuel consumption with speed and altitude* for a turbojet engine is shown in Fig. 17-14(a). The decrease in specific fuel consumption with increase in altitude is caused by a decrease in the atmospheric air temperature which produces better component efficiencies and better economy (Fig. 16-23). As the power settings and flight speed are increased above the cruise settings and speed, the reciprocating engine fuel consumption climbs rapidly so that in the higher subsonic speed range the economy of a turbojet engine is as good or surpasses that of the reciprocating engine. This is illustrated in the general curves of Fig. 17-16.

The variation in the thrust and thrust horsepower of a turbojet engine with an increase in altitude is shown in Figures 17-14 and 17-16.

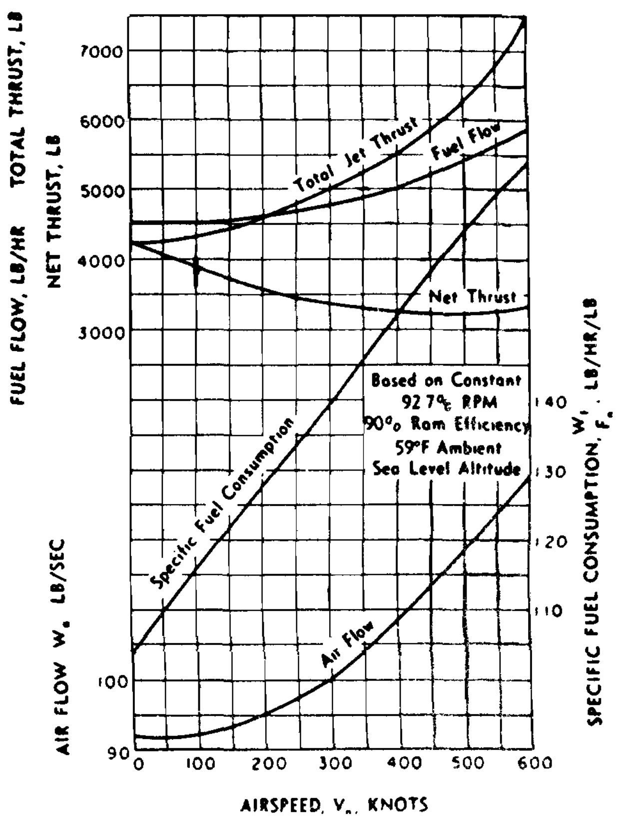


FIG. 17-17. Effect of airspeed of turbojet engine performance (courtesy General Electric Co.).

Figure 17-18 compares the variation in power with altitude for various types of propulsion engines. Although the power of the turbojet engine decreases with altitude, due to the decrease in the atmospheric density and weight rate of flow of air through the engine, the power required to overcome the drag of the aircraft normally decreases at a greater rate as the altitude is increased. Thus, a higher speed and a better economy (Fig. 17-14) may be obtained by flying the turbojet at high altitudes. The operating altitude of the turbojet engine is limited by the altitude "blowout" characteristics of the fuel, combustion chamber, and compressor.

The reciprocating engine has distinct advantages over the turbojet engine in take-off power and cruise economy. However, it is restricted in power output and flight speed, while the turbojet engine is not so

Table 17-1

(1) Turbojet Engines (Ref. 17-13)[5]—United States

Company	Model	Thrust Take-off Static (lbs)	Weight (lbs)	Frontal Area (sq ft)	Specific Weight (lb/lb thrust)	Thrust Frontal Area (lb/sq ft)	Specific Fuel Consumption (lb/lb-T-hr) (cruise)	Take-off RPM	Compressor		Turbine Inlet Temperature (deg. F)	Thrust Take-off Water Injection (lbs)	Combustion Chamber Design	Air Flow (lb/sec)
									Pressure Ratio	Type				
Allison	J33	4600	1790	13.9	0.39	331	1.12	11,750	4.4	One Stage Centrifugal	1550	5400	14 S.T.	86
	J35	5000[1]	2260	7.5	0.45	667	1.05	7,800	5.0	Eleven Stage Axial	1725		8 S.T.	85
Boeing	500	180	115		0.64		1.22	36,000	3.0	One Stage Centrifugal	1500		2 S.T.	3.25
General Electric	J31[2]	1600	850	9.4	0.53	170	1.2	16,000	3.8	One Stage Centrifugal	1470		10 Counter	33
	J33[2]	4000	1850	14.7	0.46	270	1.18	11,750	4.1	One Stage Centrifugal	1492		14 S.T.	79
	J35	4000	2400	7.5	0.60	533	1.2	7,700	5.0	Eleven Stage Axial	1500		9 S.T.	
	J47	5200	2525	7.4	0.48	703	1.03	7,950	5.0	Twelve Stage Axial	1600	6000	8 S.T.	90
Pratt and Whitney	J42[3]	5000	1700	13.4	0.34	373	1.06	12,300	4.3	One Stage Centrifugal		5750	9 S.T.	88
	J48[4]	6250	2000	13.6	0.32	450	1.00	11,000		One Stage Centrifugal		7000	9 S.T.	
Westinghouse	J30	1600	718	2.0	0.45	800	1.15	17,000	3.8	Ten Stage Axial	1500		Annular	30
	J34	3000	1200	3.1	0.40	968	1.0	11,500	4.0	Eleven Stage Axial	1400		Annular	55

(2) TURBOPROP ENGINES (REF. 17-13)[5]—UNITED STATES

Company	Model	SHP	Jet Thrust (lbs)	Equivalent[6] Shaft Horsepower (eshp)	Weight (lb)	Frontal Area (sq ft)	Specific Weight (lb/eshp)	Specific Fuel Consumption (lb/eshp-hr)	Compressors Pressure Ratio	Compressors Type	Take-off RPM	eshp per Frontal Area	Combustion Chamber
Allison	T38	2550	520	2750	1225	4.3	0.44	0.63	6.3	Seventeen Stage Axial	14,000	640	8 S.T.
	T40[7]	5100	1040	5500	2500	3.1	0.44	0.63	6.3	Two Seventeen Stage Axial	14,000	1774	Two 8 S.T.
General Electric	T31[8]	2250	600	2480	2000	7.7	0.81	0.55	5.5	Fourteen Stage Axial	13,000	322	9 Counter
Boeing	502	115			202		1.76	1.25	3.0	One Stage Centrifugal	36,000		2 S.T.

(3) RECIPROCATING ENGINES (REF. 17-13)[5]—UNITED STATES

Company	Model	Take-off Horsepower	Weight (lb)	Frontal Area (sq ft)	Specific Weight (lb/shp)	shp per Frontal Area	Specific Fuel Consumption (lb/shp-hr)	RPM	Cylinders	Compression Ratio	BMEP (max) (psi)	Take-off Horsepower Water Injection
Pratt and Whitney	R4360	3250	3520	15.0	1.01	223	0.43	2700	28	7.0	235	3500
	R2800	2300	2390	15.2	0.96	151	0.42	2800	18	6.75	255	2500
Wright	R3350	2500	2915	16.9	1.04	148	0.48	2800	18	6.5	238	

1. With afterburner 6000 lb-thrust.
2. No longer being manufactured by General Electric.
3. Manufactured under license from Rolls Royce.
4. With afterburner and water injection approximately 9000 lb-thrust.
5. See Reference 17-13 for foreign engines.
6. See Appendix C for equivalent horsepower.
7. T40 has two of the T38 turbojet power sections, i.e., it is a dual turboprop.
8. Sectionalized T-31 in Model Room internal combustion laboratory, U. S. Naval Academy. This engine is no longer manufactured.

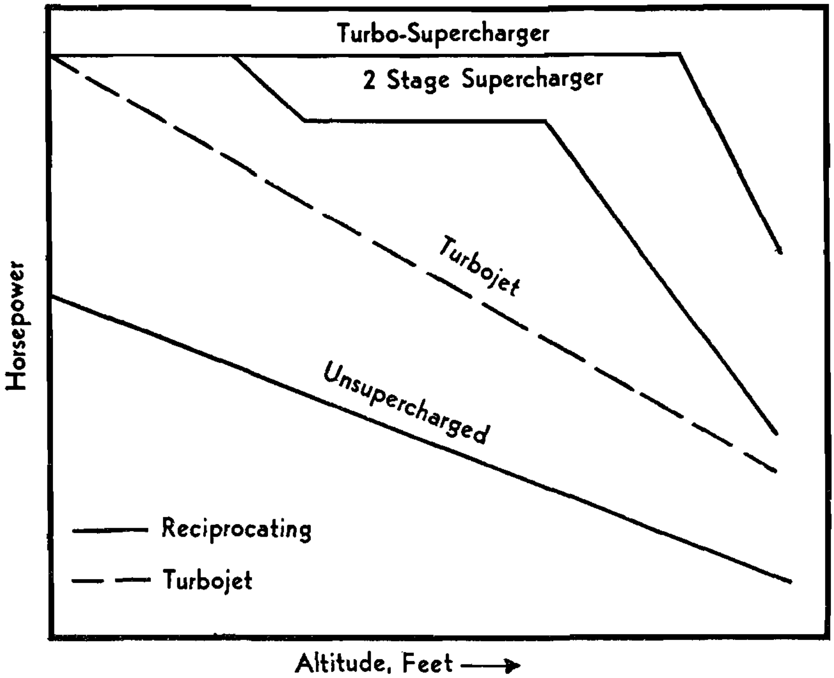


FIG. 17-18. Effect of altitude on the power available.

restricted. The operating speed range of the turbojet engine in which it shows great advantage over other types of propulsion engines is from around 500 mph up to around 1000 mph.

17-10. Other Aspects of the Turbojet Engine. In some of the preceding articles of this chapter, some of the problems of turbojet design have been mentioned. These represent, however, only a few of the large number which are confronted in the operating engine. A detailed discussion of each of the problems is prohibited by space considerations and by the purview of this text. The following, then, is presented merely to highlight some of the more important considerations.

Installation. In the typical aircraft using a single turbojet engine, the aircraft is literally wrapped around the engine. This configuration gives rise to varied access difficulties when normal repairs are attempted. On the other hand, the relatively light weight and mode of propulsion of the turbojet permit flexibility not enjoyed by reciprocating engines. For example, the inlet diffuser may incorporate a radome in its design without detrimental effect on the engine, or the engine may be hung below the wing of the aircraft in the popular "pod" type of installation. The "pod" helps to overcome the difficulty of "access" and permits relatively quick interchange of entire engines.

Accessories. Original turbojet engines incorporated only those accessories that were necessary for the engine itself, i.e., starting, ignition, fuel control, and lubrication systems. As the engine was further developed, however, the increase in the size and complexity of the aircraft itself proceeded apace. Currently, some high performance turbojet aircraft utilize power boost for controls, elaborate electronic systems and other accessories to the extent that their total weight may equal one-third of the weight of the engine. The design problems involved in providing the proper arrangement and power for such a large array of fringe systems becomes enormous. In certain large aircraft, small, self-contained gas turbine plants are used to provide power to the accessory systems.

Lubrication. The great temperatures at which modern turbojets operate cause normal petroleum base lubricating oils to break down. For this reason, the engineer turned to synthetic lubricants as a solution to the problem. In doing so, however, it was found that the synthetic lubricants themselves created new problems in that they often dissolved or distorted the available materials used for seals and ducting. Another of the lubricating problems posed by the turbojet is that of adequate bearing lubrication. Investigation reveals that the bearing located at the compressor inlet may be $-50°$ F, while the temperature at the turbine bearings may be well over 1,000° F. In order to surmount the formidable obstacles of satisfactory lubrication, new lubricants and new materials for seals and ducting had to be developed.

Instrumentation. In order to successfully operate the turbojet engine, and to prevent it from damaging itself, suitable instrumentation must be provided. Consider for a moment the array of instruments that confront the pilot of a modern jet aircraft. Each of these instruments pertaining to the engine plays its part in permitting the achievement of optimum performance and serve also as warning devices should malfunctioning occur. To mention but a few, such instruments may include the engine speed (RPM), the turbine inlet temperature, fuel flow gages, torquemeters, etc. To fulfill their purpose, each of these must be held to close tolerances of accuracy.

17-11. Thrust Augmentations. One of the disadvantages of the turbojet engine is the relatively small power available for take-off and climb compared to the reciprocating engine. Methods of improving the thrust of the turbojet engine not only for take-off and climb but also for combat and emergency power requirements of military aircraft, have become increasingly important.

The large extra thrust required to reduce the take-off run of heavily loaded turbojet engine aircraft may be provided for by the conventional jet assisted take-off (JATO) with the use of solid or liquid propellant rockets (Articles 18-5, 6). However, this method is not as readily adaptable for providing extra power for climb, combat emergencies, and carrier wave-offs as other types of thrust augmentation.

From the thrust equation, it is found that the means of increasing the thrust are to either increase the weight rate of flow of air (w_a) or increase the jet exit velocity (V_j) or increase both variables. Two different methods of thrust augmentation are currently used. These methods are:

(1) Water or liquid injection.
(2) Afterburner or tail-pipe burning (turbojet with afterburner or reheater).

In the past, other systems were developed and tried, but they have not proven satisfactory.

In the liquid injection method, water, alcohol, or a combination of water and alcohol is sprayed into the ram air stream at the compressor inlet. The evaporation of the liquid extracts heat from the inlet air. The liquid in excess of the saturation limit at the compressor inlet is evaporated during the compression process which further cools the air. The cooling gives an increase in both the pressure ratio and the air flow which is reflected throughout the engine as an increase in the

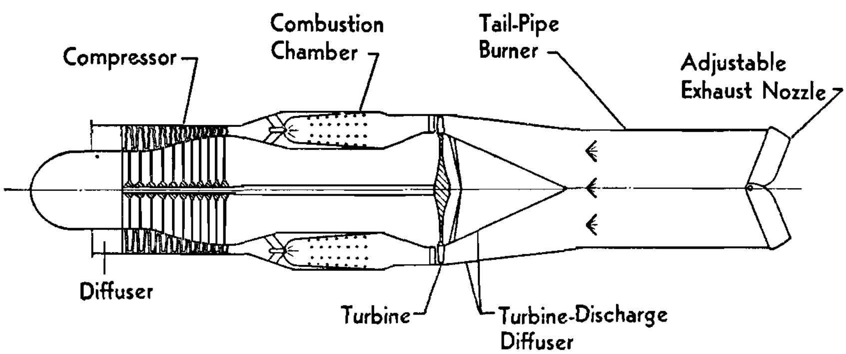


FIG. 17-19. Turbojet engine with afterburner or with tail-pipe burning (courtesy of NACA).

weight rate of flow of the working medium and also as an increase in the exit nozzle pressure. This results in an increase in the thrust of the engine. Water-methanol injection will produce approximately a 12 to 15 per cent increase in the take-off thrust of a turbojet engine (Table 17-1).

A schematic diagram of a turbojet engine equipped for tail-pipe burning or after burning is shown in Fig. 17-19. Tail-pipe burning consists of introducing and burning fuel between the turbine and the exit nozzle which has the same effect as adding a reheater to the cycle. Tail-pipe burning raises the exit temperature which results in an increased jet velocity which, in turn, produces an increase in the thrust. Since the burner inlet velocities must be sufficiently low to support stable combustion and to avoid excessive pressure losses, a diffuser is provided between the turbine outlet and the tail-pipe burner inlet. Flame holders which provide the stagnation regions and turbulence necessary for combustion are located down-stream from the fuel injection nozzles. A variable area exit nozzle is used in order that the engine also may operate as a straight turbojet. Since the turbojet with tail-pipe burning consists of a turbojet and essentially a ram jet, the engine is sometimes designated as a turbo-ram-jet engine. Tail-pipe burning is not only an augmentation device for improving the take-off and high speed performance of an airplane, but the turbo-ram-jet engine also may be considered a distinct type of engine for flight at supersonic speed.

The thrust per sq ft of frontal area of a turbojet with afterburner is about 1.3 to 1.4 times as great as that of a turbojet engine at zero flight speed and about 4 times greater at a Mach number of 2. The specific fuel consumption, lb/hr/lb thrust, of a turbo-ram-jet engine is about 2.0 to 2.5 times that of the turbojet engine at zero flight speed and 1.5 times greater at Mach 2. The thrust per weight of a turbo-ram-jet engine is about twice that of the turbojet engine.

In an analysis made by the NACA, Flight Propulsion Laboratory, Cleveland, (17-1), it was determined that the load carrying capacity of an airplane with a turbojet with afterburner engine was greater than that of a turbojet engine at its best supersonic range at 1400 mph. This greater range and load carrying capacity is due to the fact that the turbojet with afterburner has a high thrust, a high thrust per sq ft of frontal area, and a high thrust per unit engine weight at supersonic speeds where the drag increases rapidly. This increase in range was obtained with a turbojet engine with afterburner having only a slightly higher fuel consumption than that of the turbojet engine at these speeds.

Although both the water injection and afterburner are in use at the present time, the disadvantages of the water injection system is apparently causing a shift toward the exclusive use of the afterburner. Foremost of these disadvantages is the increased weight which must be carried. Thus, the aircraft would be penalized throughout its flight duration, except when the augmentation was used. This system is also limited in the duration of its application by the amount of water carried. On the other hand, the afterburner utilizes the primary fuel of the engine, thus obviating the necessity for additional liquid, and is available as long as the supply of fuel should last.

17-12. Turboprop Cycle. From the preceding articles, it is seen that the turbojet engine can compete with the reciprocating engine in propulsive efficiency and specific fuel consumption only at speeds in excess of 500 mph, but at these speeds the range of the turbojet is about one half that of the range at the best cruising speed of the reciprocating engine. At speeds below 500 mph, the propulsive efficiency of the turbojet is low due to the very high kinetic energy losses in the exhaust gases. It was natural for the aeronautical engineers and the scientists to seek a method that could utilize this energy. At the same time, a method was sought to combine the advantages of the turbojet engine, i.e., low specific weight, small frontal area and nacelle drag, simplicity, and lower vibration, with the advantages of the propeller, i.e., high propulsive efficiency at speeds below 500 mph and high power available for take-off and climb. Furthermore, since the reciprocating engine was close to its zenith in its maximum power output, unless

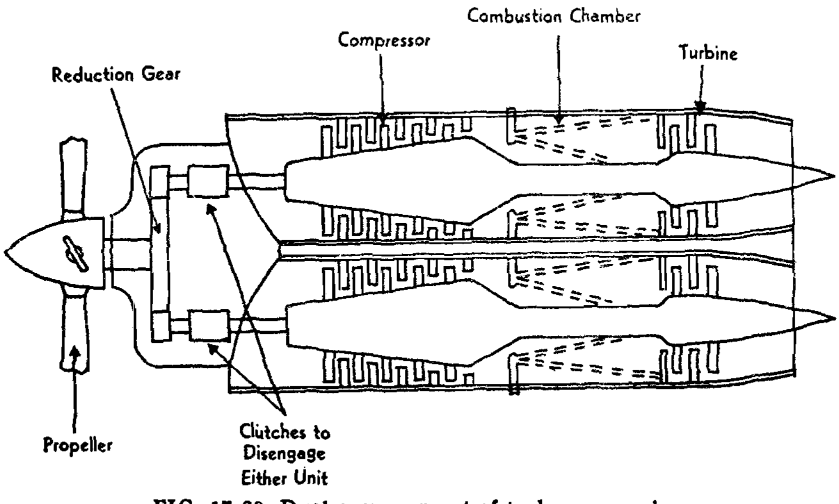


FIG. 17-20. Dual arrangement of turboprop engine.

radical developments were made, it was necessary to produce an engine capable of much higher maximum output in the lower subsonic speed range to meet the demands of both commercial and military aviation. This led to the development of the turboprop engine.

The turboprop[2] or propjet engine, Fig. 17-20, consists of a geared propeller attached to a turbojet unit. The turbine has been modified by adding additional stages in order to extract more energy from the hot gases of combustion so as to provide added power to the propeller shaft before the gases are exhausted through the jet exit nozzle. In the

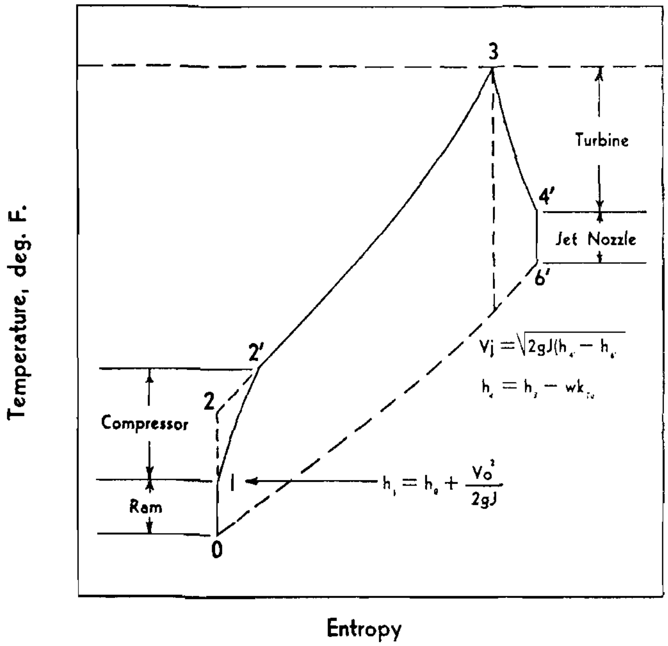


FIG. 17-21. Temperature-entropy diagram for a turboprop engine.

present turboprop engines, approximately 80 to 90 per cent of the power developed is used to drive the shaft and the remaining 10 to 20 per cent is used to obtain thrust from the jet. The control of the unit is governed by changing the propeller pitch and the quantity of fuel burned in the combustion chamber.

The turboprop engine cycle, Fig. 17-21, is the same as that of the turbojet engine cycle, except the turbine expansion process is greater.

[2] A sectionalized T-31 turboprop engine is on display in the model room of the Internal Combustion Engines Laboratory, Department of Marine Engineering, U. S. Naval Academy.

The energy delivered to the propeller, assuming no losses in the reduction gear is expressed

$$\text{prop work} = wk_{Tu} - wk_c$$
$$= (h_3 - h_{4'}) - (h_{2'} - h_1) \qquad \text{(Btu/lb)}. \qquad (17\text{-}15)$$

The horsepower developed by the propeller of the turboprop engine may be found from

$$\text{Prop hp} = \frac{J\eta_g\eta_{\text{prop}}w_a\left[(h_3 - h_{4'}) - (h_{2'} - h_1)\right]}{550} \qquad (17\text{-}16)$$

where

η_g = reduction gear efficiency
η_{prop} = propeller efficiency.

The thrust produced by the jet exit gases may be calculated from

$$T = \frac{w_a}{g}(V_j - V_0) \qquad \text{(lbs)}.$$

The thermal efficiency of the turboprop engine cycle in terms of the enthalpies at the state points, as shown in Fig. 17-21, may be calculated from equation (17-13).

Regenerators, intercoolers, and reheaters may be added to the turboprop cycle. The effect of the addition of these elements on the performance characteristics of the turboprop engine will be the same as those shown in Chapter XVI for the addition of these elements to a simple open cycle gas turbine. However, the addition of these elements adds weight to the engine and will decrease the thrust/weight ratio of the engine. Also, the addition of an intercooler will increase the drag considerably. Much research is being done towards the design and development of a light weight regenerator having good characteristics, high efficiency and low pressure losses.

17-13. Turboprop Performance. The variation of the thrust of the turboprop engine with flight velocity is shown in Fig. 17-12. The turboprop, then, has good take-off characteristics and good climb characteristics in the lower subsonic speed range due to the large thrust and power available. The variation in the propulsive efficiency of the turboprop engine with flight velocity will be a little lower than the propeller engine shown in Fig. 17-13. Since the thermal efficiency of the turbojet and the turboprop are the same, the over-all efficiency of the turboprop, i.e., the fuel economy, will be superior to the turbojet engine at speeds below about 550 mph.

One of the principal immediate advantages of the turboprop engine over the reciprocating engine is the high thrust per sq ft of frontal area. Furthermore, since the turboprop has reduced cooling losses, a considerable reduction in drag is realized which becomes more pronounced as the flight speed is increased. Figure 17-15 shows the relationship between the thrust horsepower available and the flight speed for a reciprocating engine and turboprop engine. The high drag due to the nacelle drag losses coupled with the cooling losses of a reciprocating engine at speeds above 500 mph leaves little thrust horsepower available to propel the airplane. With the turboprop, the nacelle drag losses are much smaller and are almost offset by the increase in the jet thrust so that the turboprop has a large thrust horsepower available at high subsonic speeds, 500 to 600 mph.

Although the addition of the reduction gear and propeller adds weight to the turboprop over that of the turbojet the specific weight ratio of the turboprop engine is half that of the reciprocating engine at the lower altitudes. As the altitude is increased, the thrust of the turboprop decreases while that of the turbosupercharged reciprocating engine is constant up to altitudes around 40,000 ft (see Fig. 17-18). Therefore, the difference in the specific weight ratio between the reciprocating engine and turboprop engine decreases with increase in altitude, and the reciprocating engine may have a lower ratio at altitudes above 35,000 ft.

The results of an analysis made by the NACA Flight Propulsion Laboratory on a turboprop engine, (17-1), which had compressor and turbine efficiencies of 80 per cent, a turbine inlet temperature of 1500° F, and a compressor pressure ratio of 12, showed that the specific fuel consumption of the turboprop was about the same as that of a good present day reciprocating engine. Also, as the compressor and turbine efficiencies are improved and turbine inlet temperature raised, the thermal efficiencies of the turboprop will be improved, which will result in a better fuel economy than that obtained by a reciprocating engine.

The specific fuel consumption, thrust per lb weight, and thrust per sq ft of frontal area for a turboprop compared to the other types of propulsion engines are shown in Table 17-1. By considering the above three factors, it may be shown that the turboprop has the possibility, in the future, of becoming superior to the reciprocating engine from the standpoint of lower weight of installation, lower drag, higher take-off thrust, higher miles per gallon of fuel, longer range, and heavier pay loads for both short and long ranges. The speed range of operation of the turboprop is between 0 and 550 mph, and as the propeller is improved, the speed range will be increased up to a Mach number between 0.8 and

0.9. The above avantages added to the simplicity, ease of maintenance, small vibration and noise level, and possibility of higher power per unit of the turboprop engine may cause the turboprop to become the major engine in the field of propulsion below speeds of 600 mph and may confine the reciprocating engine to small light airplanes.

The turboprop engine is adaptable to multiple power arrangements wherein a number of turbojet type of units are attached to drive a common gear box and propeller. This means that a basic engine unit is capable of multiples of 2, 3, or 4 times the power output of a single unit simply by use of a new gear box and engine control system. This arrangement has the additional advantages of being able to produce high power outputs for take-off climb, and high speed runs for large air-

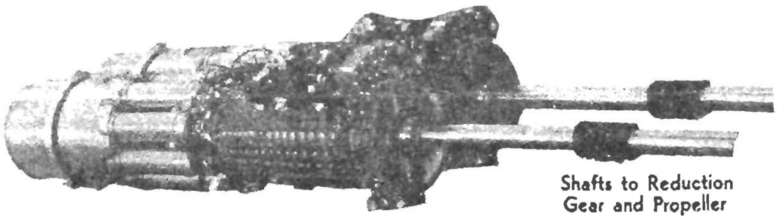


FIG. 17-22(a). Allison T-40 Turboprop showing dual arrangement.

planes and to obtain good fuel economy at cruising speeds by cutting out one of the turbojet units while driving the propeller through the other unit which is operating at the optimum RPM and output for best fuel economy.

Much research, development, and experimentation will be required in the future before the turboprop will reach its desired peak performance and service operation.

17-14. Ram Jet. A French engineer, Rene Lorin, is credited with inventing and patenting the first ram jet. Lorin's patent, obtained in 1913, was followed by several other patents during the past few decades. It was not until the advent of the high speed wind tunnel, however, that the U. S. Navy sponsored research team developed a workable ram jet at Johns Hopkins University.

The ram jet, which has been referred to in the past as athodyd (aerothermo-dynamic ducts), Lorin tube, or "flying stovepipe," is a steady combustion or continuous flow engine. It has the simplest construction of any propulsion engine, consisting essentially of an inlet diffuser, a combustion chamber, and an exit nozzle or tailpipe (Fig. 17-23).

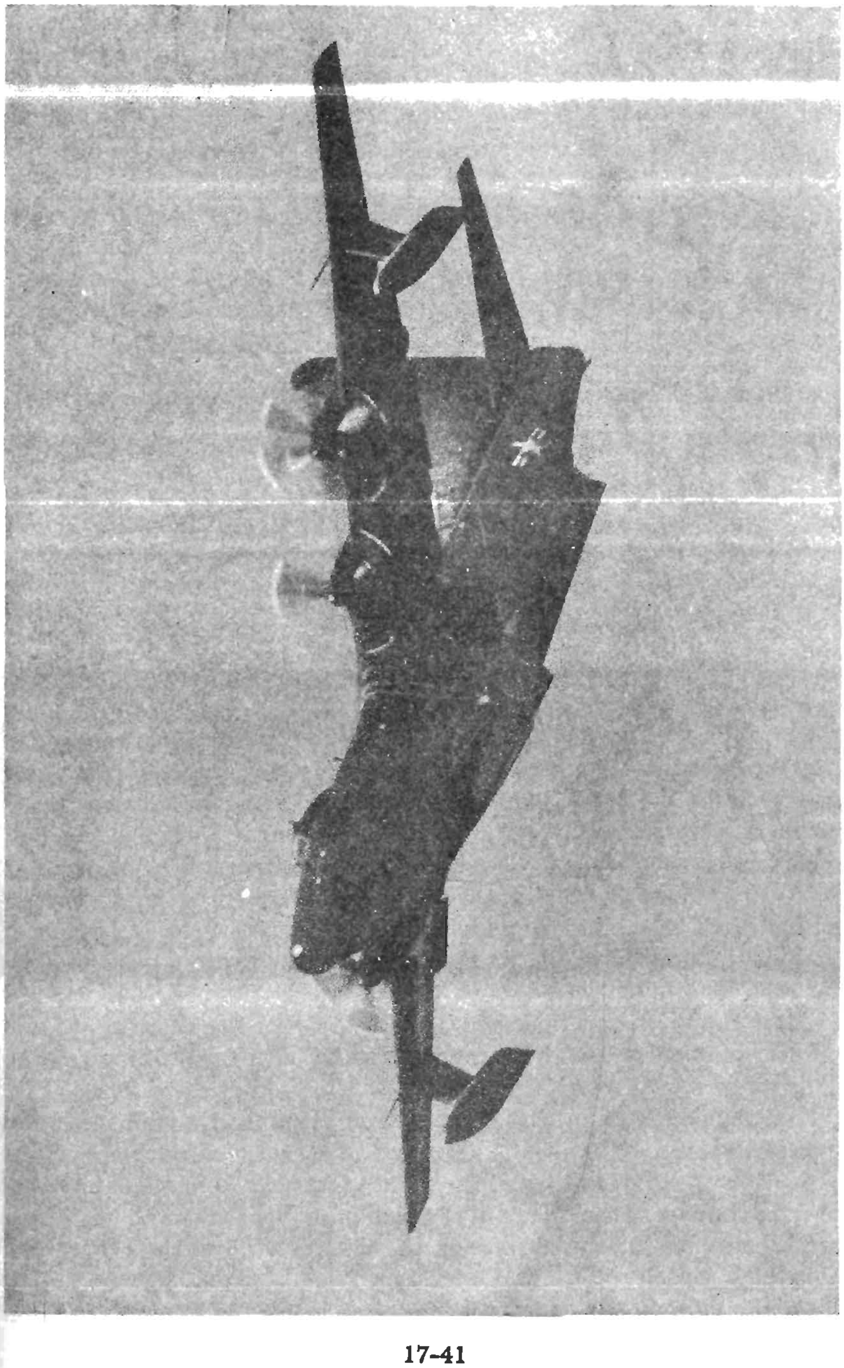


FIG. 17-22(b). Navy P5Y powered by Allison turboprop engines.

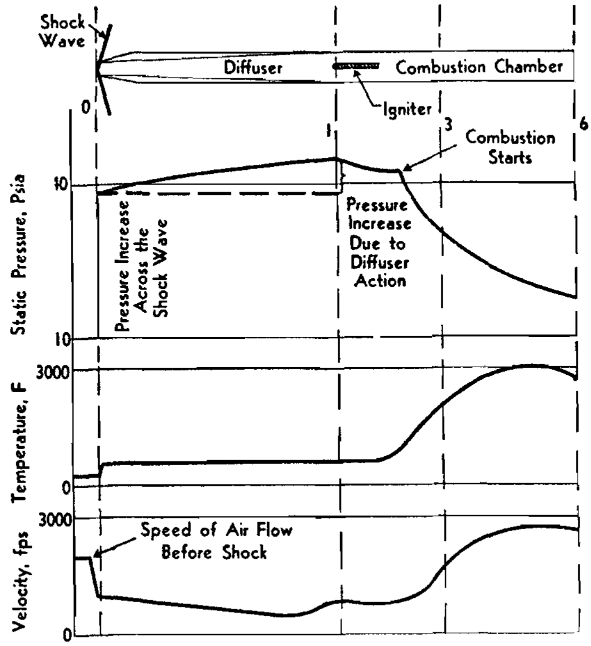


FIG. 17-23. Supersonic ram jet engine.

Since the ram jet has no compressor, it is dependent entirely upon ram compression. Figure 17-5 shows the increase in the ideal ram pressure ratio as the Mach number is increased. The ram pressure ratio increases very slowly in the subsonic speed range. Thus, the ram jet must be "boosted" up to a speed of over 300 mph by a suitable means, such as a turbojet or a rocket, before the ram jet will produce any thrust and must be boosted to even higher speeds before the thrust produced exceeds the drag. The ram jet should be operated within a reasonable range of its diffuser design speed.

After the ram jet is "boosted," the velocity of the air entering the diffuser is decreased and is accompanied by an increase in pressure. This creates a "pressure barrier" at the after end of the diffuser. The fuel that is sprayed into the combustion chamber through injection nozzles is mixed with the air and ignited by means of a spark plug. The expansion of the gases toward the diffuser entrance is restricted by the pressure barrier at the after end of the diffuser; consequently, the gases are constrained to expand through the tailpipe and out the exit nozzle at a high velocity. The NACA Flight Propulsion Laboratory found from photographs of a ram jet being tested in the wind tunnel that the pressure barrier was not completely effective and that there was a definite pulsation created in the combustion chamber which affected the air flow in front of the diffuser. Further research is being conducted to determine the flow phenomena that actually exist in a ram jet.

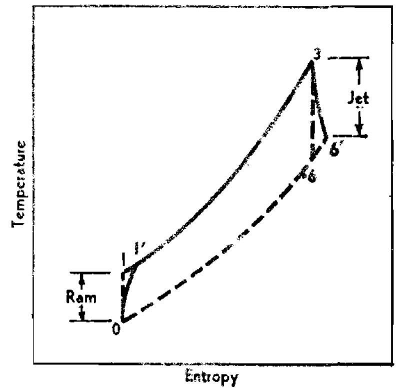


FIG. 17-24. T-s diagram of ram jet engine.

Since the ram jet engine has no turbine, the temperature of the gases of combustion is not limited to a relatively low figure as in the turbojet engine. Air fuel ratios of around 15 to 1 are used. This produces exhaust temperatures in the range of 3500 to 4000° F. Extensive research is being conducted on the development of hydrocarbon fuels that will give 30 per cent more energy per unit volume than current aviation gasolines. Investigations are underway to determine the possibility of using solid fuels in the ram jet and in the afterburner of the turbojet engine. If powdered aluminum could be utilized as an aircraft fuel, it would deliver over two and one half times as much heat per unit volume as aviation gasoline, while some other metals could deliver almost four times as much heat.

The temperature, pressure, and velocity of the air during its passage through a ram jet engine at supersonic flight are shown in Fig. 17-23.

The cycle for an ideal ram jet, which has an isentropic entrance diffuser and exit nozzle, is the Joule cycle as shown by the dashed lines in Fig. 17-24. The difference between the actual and ideal ram jet is due principally to losses actually encountered in the flow system. The sources of these losses are:

(1) Wall friction and flow separation in the subsonic diffuser and shock in the supersonic diffuser.

(2) Obstruction of the air stream by the burners which introduces eddy currents and turbulence in the air stream.
(3) Turbulence and eddy currents introduced in the flow during burning.
(4) Wall friction in the exit nozzle.

By far, the most critical component of the ram jet is the diffuser. Due to the peculiarities of streamline flow, a diffuser which is extremely efficient at a given speed may be quite inadequate at another velocity.

Because of the simplicity of the engine, the ram jet develops greater thrust per unit engine weight than any other propulsion engine at supersonic speed with the exception of the rocket engine. The thrust per unit frontal area increases both with the efficiency and the air flow through the engine; therefore, much greater thrust per unit area is obtainable at high supersonic speeds. General performance of a ram jet engine in the subsonic speed range would have a specific fuel consumption between 6 and 8 lbs fuel per lb thrust-hr and a specific weight between 0.1 and 0.2 lb per lb thrust. The supersonic ram jet engine has a specific fuel consumption between 2.5 and 4.0 and a specific weight between 0.1 and 0.4. The best performance, then, of the ram jet engine is obtained at flight speeds of 1000 to 2200 mph.

17-15. The Pulse Jet Engine. The pulse jet engine is somewhat similar to a ram jet engine. The difference is that a mechanical valve arrangement is used to prevent the hot gases of combustion from going out the diffuser.

The German inventor, Paul Schmidt, patented principles of the pulse jet engine in 1930. It was developed by Germany during World War II, and was used as the power plant for the V-1 weapon or "buzz bomb."

The turbojet and ram jet engine are continuous in operation, and are based on the constant pressure heat addition Brayton cycle. The pulse jet is an intermittent combustion engine, and it operates on a cycle similar to a reciprocating engine and may be better compared with an ideal Otto cycle rather than the Joule or Brayton cycle.

The compression of incoming air is accomplished in a diffuser, the air passes through the spring valves and is mixed with fuel from a fuel spray located behind the valves. A spark plug is used to initiate combustion, but once the engine is operating normally, the spark is turned off and residual flame in the combustion chamber is used for ignition. The engine walls also may get hot enough to initiate combustion.

The mechanical valves, which were forced open by the entering air, are forced shut when the combustion process raises the pressure within the engine above the pressure in the diffuser. The combustion products, since they cannot expand forward, move to the rear at high velocity. When the combustion products leave, the pressure in the combustion chamber drops and the high pressure air in the diffuser forces the valves open and fresh air enters the engine.

Since the products of combustion leave at a high velocity, there is a certain scavenging of the engine caused by the decrease in pressure occasioned by the exit gases. There is a stable cycle set up in which alternate waves of high and low pressure travel down the engine. The alternating cycles of combustion, exhaust, induction, combustion, etc., are related to the acoustical velocity at the temperature prevailing in the engine. Since the temperature varies continually, the actual process is complicated, but a workable assumption is that the tube is acting similar to a quarter wave length organ pipe. The series of pressure and rarefaction waves move down it at the speed of sound for an assumed average temperature.

The frequency of the combustion cycle may be calculated from the following expression:

$$= \frac{a}{4L} \qquad \text{(cycles/sec.)}$$

where,

$a = \sqrt{kgRT}$
L = length of engine (from valves to exit).

A serious limitation placed upon pulse jet engines is the mechanical valve arrangement. Unfortunately, the valves used have resonant frequencies of their own, and under certain conditions, the valves will be forced into resonant vibration and will be opening when they should be shutting. This limitation of valves also limits the engine because the gas goes out the diffuser when it should go out the tail pipe.

Despite the apparent noise, and despite the valve limitation (which future designs may avoid), pulse jet engines have several advantages when compared to other thermal jet engines.

1. The pulse jet is very inexpensive when compared to a turbojet.
2. The pulse jet produces static thrust and produces thrust in excess of drag at much lower speeds than a ram jet.

The potential of the pulse jet is considerable and its development has just begun. Future research may well bring about a wide application in its use.

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EXERCISES

17-1. State the two laws on which the propulsion of a vehicle through a fluid medium is based.

17-2. Give the basic difference between the two fundamental types or jet propulsion engines.

17-3. List the five types of atmospheric jet engines.

17-4. Name the two types of rocket engines.

17-5. A turbojet engine has a forward speed of 620 mph, produces 3000 lbs of thrust, and uses 85 lbs of air per second.

Calculate:

(a) Relative exit velocity, V_j, ft/sec.
(b) Thrust horsepower
(c) Propulsive efficiency

Ans:

(a) 2050 ft/sec.
(b) 4960 hp
(c) 61.4%

17-6. A turbojet engine flying at a speed of 650 mph has an exit jet velocity of 1610 ft per sec and consumes air at the rate of 69 lbs per sec.

Calculate:

(a) Thrust
(b) Thrust horsepower
(c) Propulsive efficiency

Ans:

(a) 1405 lbs
(b) 2433 hp
(c) 74.5%

17-7. List five needs or demands that are being fulfilled by turbojet engines.

17-8. Why is a reciprocating engine restricted in forward speed and power output? What are these limits?

17-9. Draw a schematic diagram and a T-s diagram of a turbojet engine, label all components, process lines, and state points.

17-10. What is "ram" effect?

17-11. Given a turbojet engine with the following data:

$h_0 = 10$ Btu/lb (entrance to diffuser)
$h_1 = 17$ Btu/lb (exit from diffuser)
$h_3 = 430$ Btu/lb (entrance to turbine)
$h_{4'} = 370$ Btu/lb (exit from turbine)
$V_j = 2420$ ft/sec
$w_a = 50$ lb/sec

Calculate:

(a) Flight velocity, V_0 (ft/sec)
(b) Enthalpy at exit from compressor, state 2′ (Btu/lb)
(c) Enthalpy at exit from jet nozzle, state 6′ (Btu/lb)
(d) Thrust (lb)

Ans:

(a) 592 ft/sec
(b) 77 Btu/lb
(c) 253 Btu/lb
(d) 2830 lbs

17-12. How does the thrust and thrust horsepower of a turbojet engine vary with flight velocity? Compare these with a reciprocating engine

17-13. How does the specific fuel consumption, thrust, and thrust horsepower of a turbojet engine vary with altitude? Compare these with a reciprocating engine.

17-14. Name three methods of thrust augmentation.

17-15. Which of the four methods of thrust augmentation appears to be the best? Why?

17-16. What is meant by Mach number?

17-17. What features of the turbojet engine and the reciprocating engine are combined to give the turboprop engine its highly desirable characteristics?

17-18. Given the following data for a turboprop engine (Fig. 17-21):

$w_a = 38$ lb/sec
$V_0 = 1080$ ft/sec
$h_0 = 34$ Btu/lb
$h_{2'} = 134$ Btu/lb
$h_3 = 331$ Btu/lb
$h_{4'} = 208$ Btu/lb
$h_{6'} = 174$ Btu/lb

Calculate:

(a) Jet exit velocity, V_j (ft/sec)
(b) Enthalpy at exit to diffuser, state 1 (Btu/lb)
(c) Horsepower delivered to propeller assuming no loss in the reduction gear
(d) Thrust developed by the jet exit gases

Ans:

(a) 1304 ft/sec
(b) 57.3 Btu/lb
(c) 2480 hp
(d) 264 lbs

17-19. Given the following data for a turboprop engine (Fig. 17-21):

State point	h Btu/lb	State point	h Btu/lb
0	22.8	3	398
1	37.9	4	313
2	80.4	4′	323
2′	86	6′	292

$w_a = 38$ lb air/sec
$V_j = 1245$ ft/sec
$V_0 = 868$ ft/sec
Reduction gear efficiency = 100%
(a) Draw T-s diagram, label all points and indicate the engine component in which each process occurs.

Calculate:

(b) Thrust developed by jet gases, lbs.
(c) Energy delivered to propeller in Btu/lb.
(d) Thermal efficiency.

Ans:

(b) 445 lbs
(c) 26.9 Btu/lb
(d) 13.7%

17-20. Explain why a turboprop engine having the same available power output at 100 mph as a reciprocating engine has a greater available thrust horsepower at 500 mph.

17-21. What are the basic construction differences between a turboprop and a turbojet engine?

17-22. Draw a schematic diagram of a ram jet engine and label all components.

17-23. Describe the operation of a ram jet engine.

17-24. Why must a ram jet engine be boosted to speeds around 400 mph?

17-25. What are the advantages and disadvantages of ram jet engines?

17-26. Draw a schematic diagram of a pulse jet engine.

17-27. Describe the operation of a pulse jet engine.

17-28. What are the advantages and disadvantages of pulse jet engines?

CHAPTER XVIII

ROCKET ENGINES

Rocket type engines, as it was brought out in Chapter XVII, differ from other thermal jet engines in that its propellant carries both the fuel and the oxidizing agent. As a result, this type of engine is independent of the atmospheric oxygen that other thermal jet engines must rely upon. From this point of view, rocket engines are most attractive. There are, however, other operational facts that make this engine less useful. The purpose of this chapter, therefore, is to discuss the fundamentals of rocket engine construction, its performance and application in view of other type thermal engines previously covered in this text.

Also, as it was indicated in Chapter XVII, rocket type engines are *classified as to the type of propellant used in them. Confining, therefore, all of these* engines into two major groups: one type belonging to the group that utilizes *liquid* type propellants, and the other group that uses *solid* type propellants. The fundamental theory that governs the operation of rocket engine may be equally applied to both the liquid and the solid type of propellant rocket engine.

The theory of operation of this engine will be developed at first in order to have a clearer understanding of some of the operational parameters that will be used later in articles describing different types of rocket engines.

18-1. Brief History. The jet propulsion action of the rocket has been recognized for centuries. Recorded history indicates that rockets were first used by the Chinese and Arabs in the thirteenth and fourteenth centuries. Since this early beginning, the use of rockets has oscillated between their use in wartime as a weapon and their use in peacetime for signaling or pyrotechnic displays. It is interesting to note that rockets capable of carrying incendiary explosives two miles were used in the War of 1812. However, their employment as a weapon was practically eliminated during the latter part of the nineteenth century by the advances made in artillery, which had greater range and accuracy. Although the rocket was employed only to an insignificant extent in World War I, marked advances were made by the research that was undertaken at that time. In World War II, the rocket became a major offensive weapon employed by all warring powers. Rockets and rocket powered weapons have advanced to a point where they may be used effectively in military operations.

18-2. Application of Rocket Engines. Rocket propulsion, at this time, should not be regarded as a competitor of existing means for propelling airplanes, but as a source of power for reaching objectives unattainable by other methods. The Germans applied rocket propulsion to the high subsonic speed, short duration ME-163 interceptor and to the V-2 missile. In this country, the rocket engine is being used and is under experimental test for an increasing number of applications. Some of these applications are: (1) artillery barrage rockets; (2) anti-tank "bazooka" rockets; (3) all types of guided missiles; (4) aircraft launched rockets; (5) jet assisted take-off for airplanes; (6) engines for long range, high speed guided missile and pilotless aircraft; and (7) the main and auxiliary propulsion engines on transonic airplanes such as the D-558.

It will be repeated again that the rocket engine differs from the other jet propulsion engines in that the entire mass of the gases in the jet is generated from the propellants carried within the engine. Therefore, it is not dependent on the atmospheric air to furnish the oxygen for combustion. However, since the rocket carries its own oxidizer, the propellant consumption is very high.

18-3. Basic Theory of Operation. Figure 18-1 shows a schematic diagram of the basic components of a liquid bi-propellant (see also Fig. 17-1) rocket engine. It consists of an injection system, a combustion chamber, and an exit nozzle. The oxidizer and fuel burn in the combustion chamber producing a high pressure. The pressure produced is governed by the weight rate of flow of the propellants, the chemical characteristics of the propellants, and the cross-sectional area of the nozzle throat. The gases are ejected to the atmosphere at supersonic speeds through the nozzle. The nozzle converts the pressure energy of the gases into kinetic energy. The reaction to the ejection of the high velocity gases produces the thrust of the rocket engine.

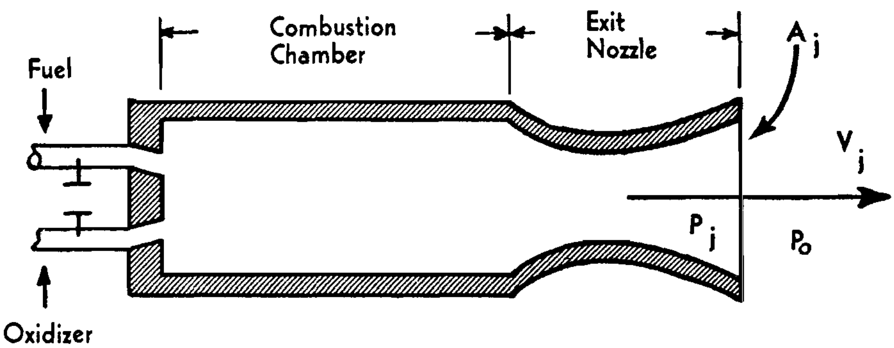

FIG. 18-1. Components of a liquid bipropellant rocket motor.

The thrust developed is a resultant of the pressure forces acting upon the inner and the outer surfaces of the rocket motor. The resultant internal forces acting on the engine are

$$\text{Resultant Internal Forces} = \frac{w_p V_j}{g} + p_j A_j \qquad \text{(lbs)}$$

where

w_p = weight rate of flow or weight rate of propellant consumption, lbs per sec
V_j = Exit or jet velocity relative to nozzle, ft per sec.
p_j = Exit static pressure, lb per sq ft.
A_j = Exit area, sq. ft.

The resultant external forces acting on the rocket motor are $p_0 A_j$ lbs, where p_0 is the atmospheric pressure in lb per sq ft. The thrust which is a resultant of the total pressure forces, becomes

$$T = \frac{w_p V_j}{g} + A_j(p_j - p_0) \qquad \text{(lbs)} \qquad (18\text{-}1)$$

Equation 18-1 shows the effect of atmospheric pressure on the thrust of a rocket engine, i.e., the lower the atmospheric pressure the higher the thrust. Therefore, maximum thrust will be obtained when $p_0 = 0$, i.e., *The rocket engine thus obtains its maximum thrust when operating in a vacuum.*

In testing a rocket engine, the thrust, the total propellant consumption, and the total time are readily measured. It is convenient, then, to express the thrust in terms of the weight rate of flow of propellants and an "effective jet velocity," V_{je}, i.e., express the thrust

$$T = \frac{w_p}{g} V_{je} \qquad \text{(lbs)} \qquad (18\text{-}2)$$

The **"effective jet exit velocity"** is a hypothetical velocity[1] used for convenience in test work and it is defined from equations (18-1) and (18-2) as

$$V_{je} = V_j + \frac{gA_j}{w_p}(p_j - p_0) \qquad \text{(ft/sec)}$$

[1] Determined by computations from the known thrust and rate of propellant consumption.

The "effective jet velocity" has become an important parameter in rocket engine performance.

The thrust power, TP, developed by a rocket engine is defined as the thrust multiplied by the flight velocity, or

$$TP = TV_0 = \frac{w_p}{g} V_{je} V_0 \qquad \text{(ft-lb/sec)}$$

The propulsive efficiency, η_p, is the ratio of the thrust power to propulsive power supplied. The propulsive power is the thrust power plus the kinetic energy lost in the exhaust, i.e.,

$$\text{K.E. Loss} = \frac{w_p}{2g}(V_{je} - V_0)^2 \qquad \text{(ft-lb/sec)}$$

Therefore, the propulsive efficiency may be expressed as

$$= \frac{TP}{TP + \text{K.E. Loss}} = \frac{(w_p V_{je} V_0/g)}{(w_p V_{je} V_0/g) + [w_p(V_{je} - V_0)^2/2g]}$$

$$= \frac{2(V_0/V_{je})}{1 + (V_0/V_{je})^2} \qquad (18\text{-}3)$$

Specific Impulse, I_{sp}, has become an important parameter in rocket engine performance and is defined as the thrust produced per unit weight rate of propellant consumption

$$I_{sp} = \frac{T}{w_p} = \frac{V_{je}}{g} \qquad \text{(lb-sec/lb)} \qquad (18\text{-}4)$$

Specific impulse, with the units of pounds of thrust produced per pound of propellant burned per second, gives a direct comparison as to the effectiveness among propellants. It is desirable to use propellants with the greatest possible specific impulse, since this allows a greater useful load to be carried for a given overall rocket weight.

18-4. Rocket Engine Performance. From the analysis of equation 18-2 it can be seen that the thrust of a rocket engine depends on the *effective jet velocity* and the *weight rate of propellant consumption.* Also the same equation indicates that the rocket is completely independent of the forward flight velocity. Again, equation 18-1 shows that as the altitude increases the thrust of the rocket engine also increases because the atmospheric pressure (p_0) decreases. The variation in the thrust

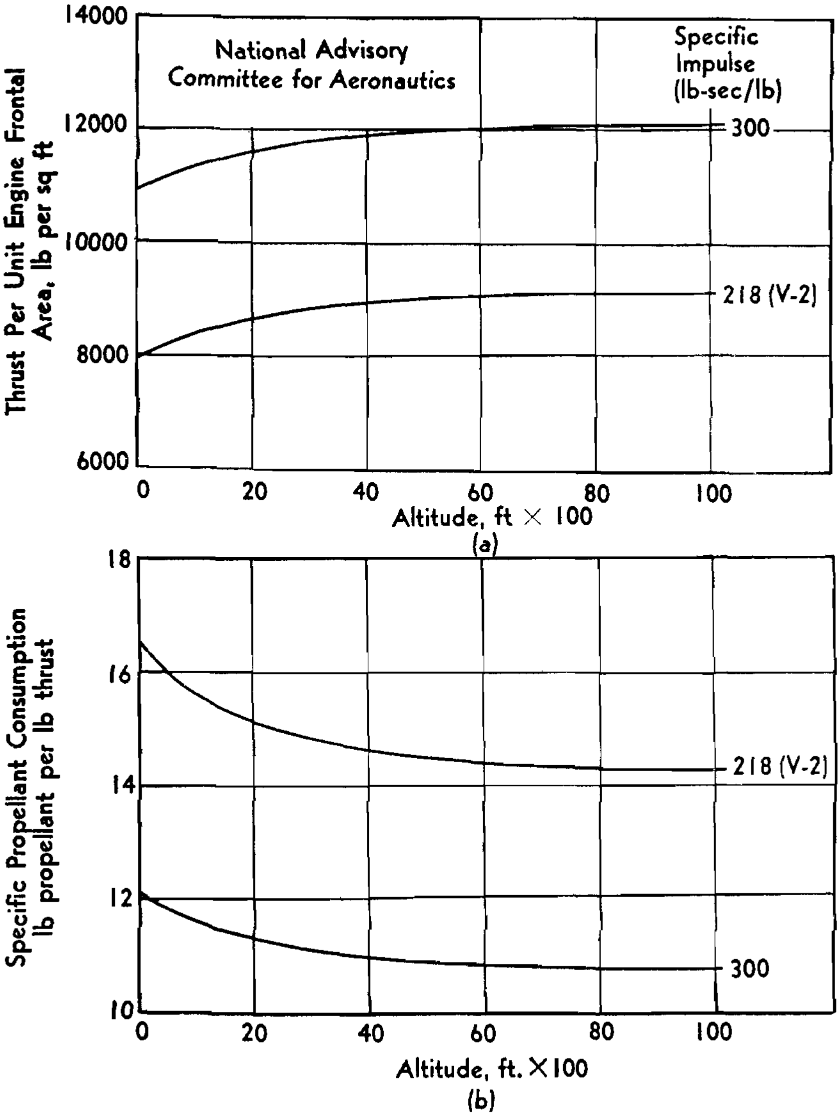

FIG. 18-2. (a) Variation of the thrust per unit engine frontal area and (b) specific thrust propellant consumption with altitude for two values of specific ***impulse (courtesy of NACA).***

per unit frontal area and the specific propellant consumption with altitude for two different values of specific impulses are shown in Fig. 18-2. A specific impulse of 218 lb-sec per lb was obtained by the German V-2 rocket using liquid oxygen and alcohol. Additional information

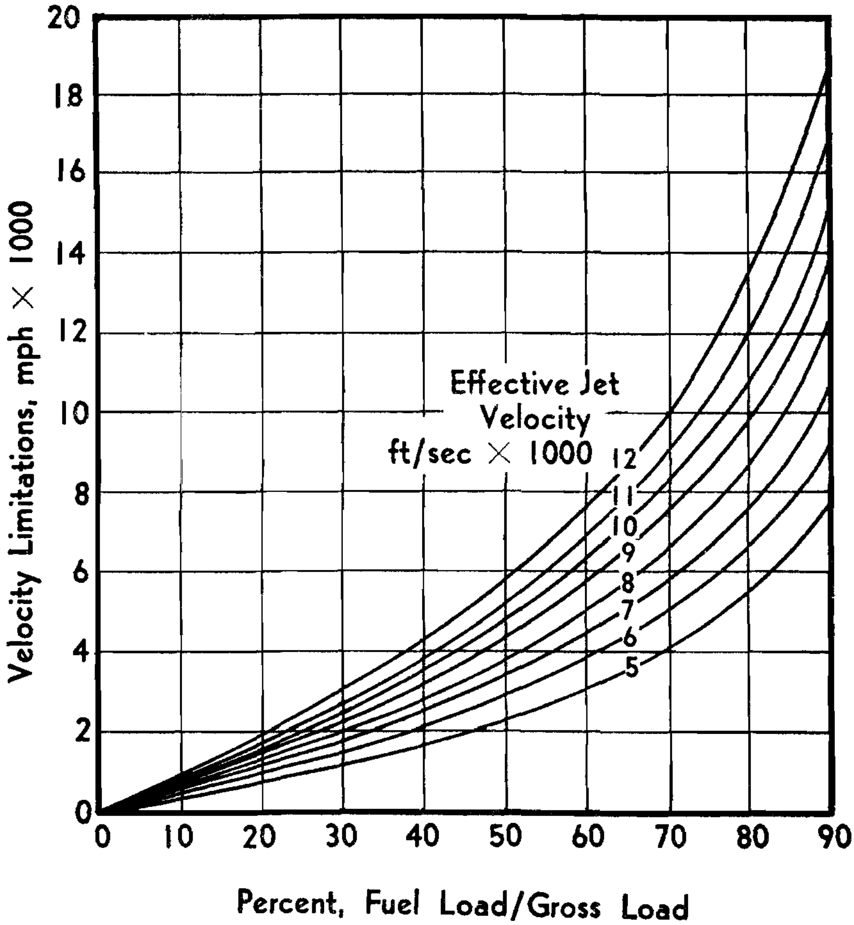

FIG. 18-3. Flight velocity limitations set by fuel load and effective jet velocity (courtesy of NACA).

concerning V-2 rocket will be given at the end of this article while the specific impulses of some of the propellants are shown in Tables 18-1 and 18-2.

The jet velocity attained by the rocket engine is a direct function of the characteristics of the propellants. The velocity is determined to a great extent by the molecular weight, the temperature, pressure, and the specific heat of the gases of combustion within the combustion chamber. Table 18-2 shows the representative exhaust velocities and specific impulses attained by some liquid bi-propellants. The flight velocity attainable is limited by and is a function of the effective jet velocity (Fig. 18-3). The flight velocity is increased with an increase in the ratio of fuel load to gross load and the effective jet velocity.

The thrust power, the thrust horsepower, and the propulsive efficiency of rocket engines vary directly with the flight speed of the missile. Since the jet velocity is very high, the propulsive efficiency is low in the subsonic speed range. Figure 18-4 indicates that the rocket engine can not compete, as far as efficiency is concerned, with other

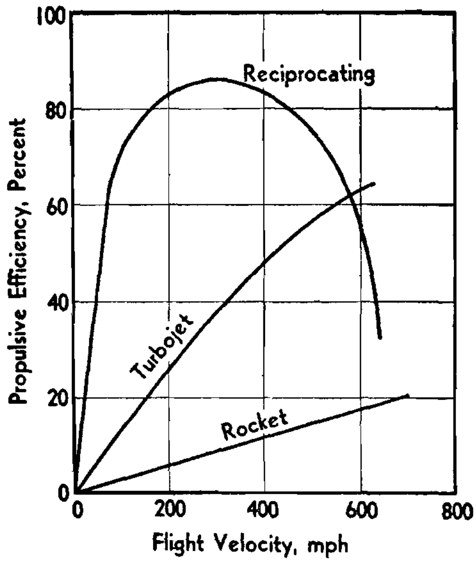

FIG. 18-4. Comparison of the propulsive efficiencies of a reciprocating, a turbojet, and a rocket engine at the same altitude.

forms of propulsion engines in the subsonic speed range. However, it is the only engine that can be used for speeds of above 2500 mph and for altitudes above 100,000 feet. The rocket engine also has a place in the subsonic speed range where highly concentrated output of power is desired for short intervals of time.

The rocket engine consumes its supply of propellants rapidly resulting in a very short duration of the powered thrust. Depending on the type, the duration of thrust varies from a split second up to a few minutes. Therefore, with rocket powered missiles, such as the German V-2 and Viking, it is necessary to attain an extremely high velocity at the end of the powered thrust so that the momentum of the missile will carry it over a relatively large range in the form of a ballistic trajec-

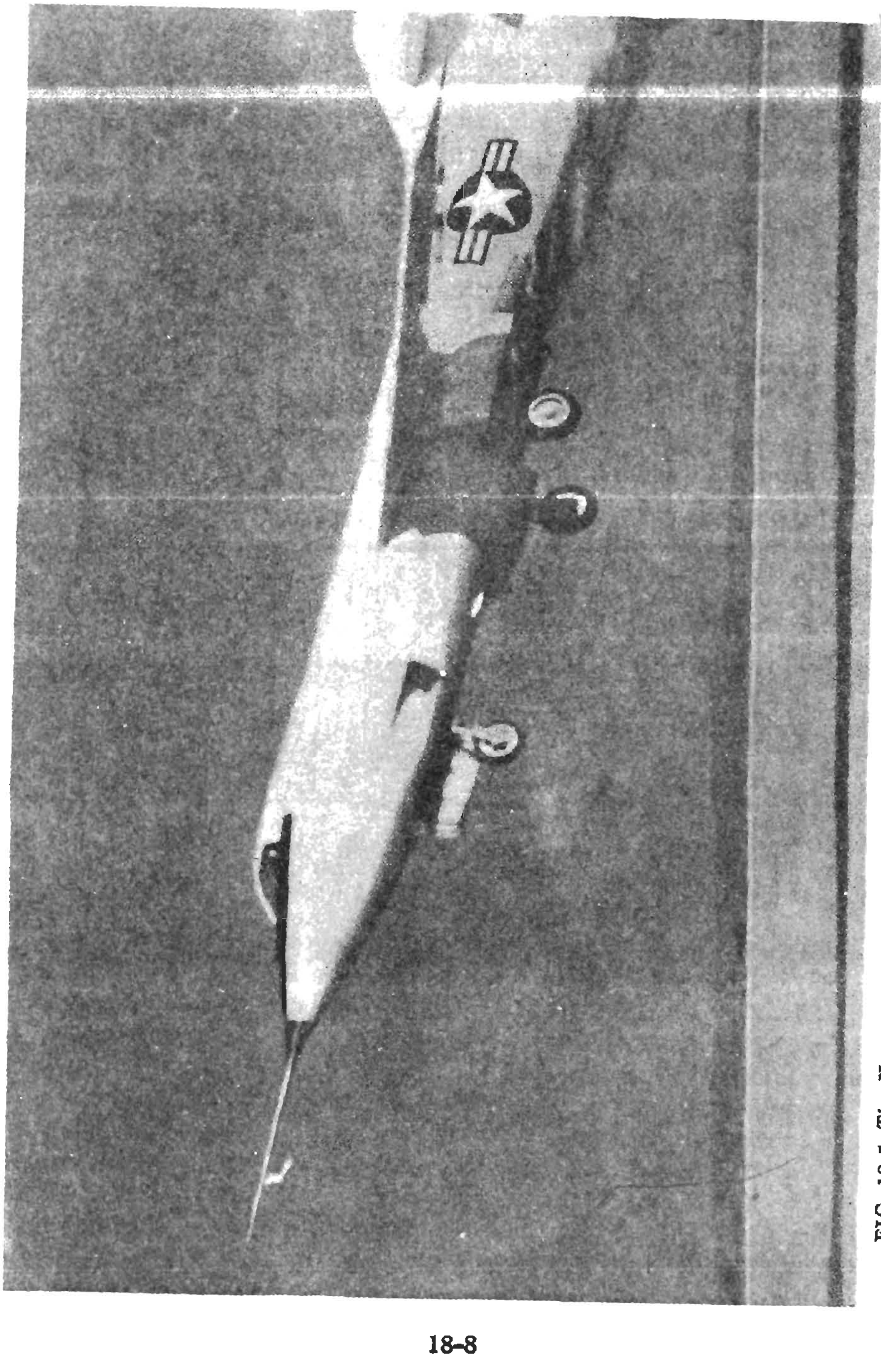

FIG. 18-5. The Navy supersonic airplane D-558 showing a JATO take-off. The D-558 contained a J-34 turbojet engine and four liquid bipropellant rocket engines.

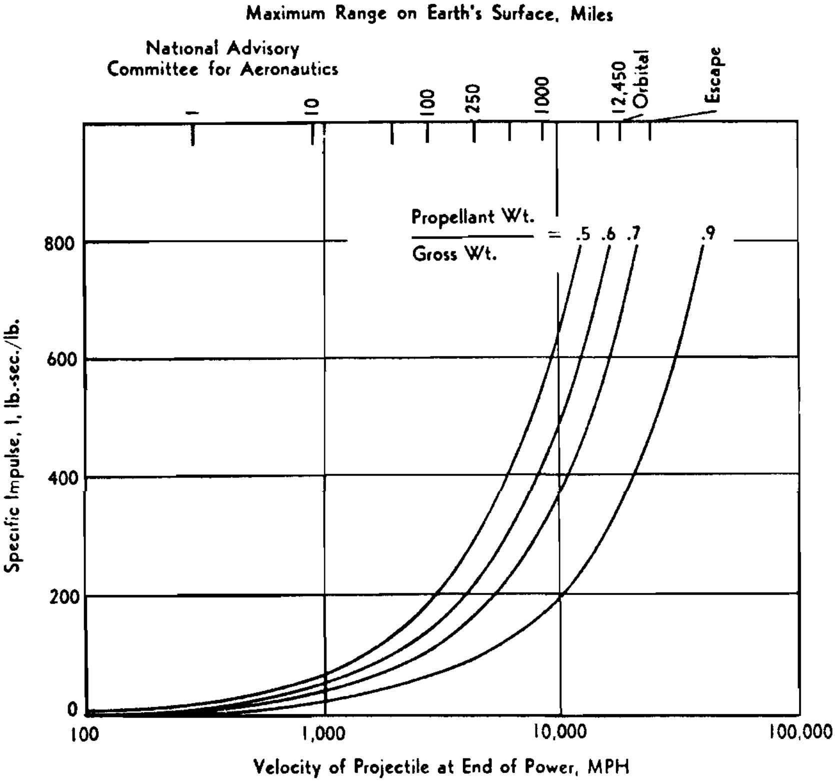

FIG. 18-6. Variation of range and velocity of projectile at end of power with the specific impulse and ratio of propellant weight to gross weight assuming no drag and a ballistic trajectory (courtesy of NACA).

tory. Figure 18-6 shows the variation in range and velocity of projectile at end of the power thrust as a function of the specific impulse and the ratio of propellant weight to gross weight, assuming no drag and a ballistic trajectory. At the top of the graph is a scale of maximum range on the earth's surface.

The curves of Fig. 18-6 indicate the importance of increasing the specific impulse to improve both flight velocity and range. A large increase in range may also be gained by increasing the ratio of the propellant weight to the gross weight of the rocket; however, there is a practical upper limit of the value of this ratio for a single rocket. A possible means of increasing the ratio and the range is the step-rocket where two or more rockets are joined as a unit. The rockets of this unit are arranged to burn consecutively, and each step is discarded or detached when the propellant of that step is exhausted. With the propellants available now, it would be possible, in the final step of the step-rocket to attain a velocity of 18,000 mph. A velocity this high would permit a flight to any point on the earth's surface or the establishment of a permanent orbit around the earth.

An example of the performance of a rocket is the published data on the German V-2 rocket. The complete missile weighed 24,000 lbs. The propellants weighed 18,500 lbs (77 per cent of the total weight of the rocket); liquid oxygen 11,000 lbs and alcohol 7,500 lbs. The thrust developed was 48,000 lbs. The propellants (about 3000 gallons) were consumed in 71 seconds. The velocity of the missile at the end of the powered thrust, i.e., at the end of 71 seconds, was 5300 ft per sec at an altitude of 22 miles. The missile continued to climb and at the top of its ballistic trajectory the missile had an altitude of 68 miles. The horizontal range attained was 200 miles.

An increase in the specific impulse will increase the thrust per unit frontal area and decrease the specific propellant consumption, Fig. 18-2. The large thrust per unit frontal area and small weight per lb thrust as shown in Fig. 18-3 indicates the compactness of a rocket engine. This, together with the simplicity of a rocket engine, constitutes one of the major advantages of this type of power plant. These advantages are obtained at a high cost of propellant consumption, which is due to a large extent to the rocket carrying not only its own fuel but its own oxidizer.

The rocket engine for airplane propulsion is applicable to the high speed, short range category where low engine weight, compactness, and simplicity of the engine outweigh the disadvantages of high propellant consumption. Also, it may be used as an auxiliary power augmentator for jet assist take-off, high rate of climb, and high speed runs (see Fig. 18-5) of short duration. For projectiles and guided missiles (like the Navy Viking), the rocket motor provides an enormous thrust from a simple, compact unit. In addition, it has the ability to provide propulsion at very high altitudes and propulsion outside the earth's atmosphere.

18-5. Solid Propellant Rockets. The solid propellant rocket differs from other engines in that total mass of fuel is stored and burned within the combustion chamber. There is no fuel supply system.

The solid propellant rocket motor consists of a seamless tube, usually made of steel, closed solidly at one end. The open end holds the nozzle which may be a single or multi-orifice type. Usually, small projectile rockets (less than 4 in. in diameter) use single orifice nozzles, while larger ones use multi-orifice nozzles. The single orifice nozzles have their axis parallel to the axis of the motor, while the axis of the orifices of the multi-orifice nozzle may be at an angle to the axis in order to rotate the projectile, thus providing spin stabilization. It should be noted, however, that JATO and large missile booster units are often single nozzle type engines. The solid propellant rockets are divided into two main types according to the amount of surface area exposed to burning. These two types are restricted burning and unrestricted burning.

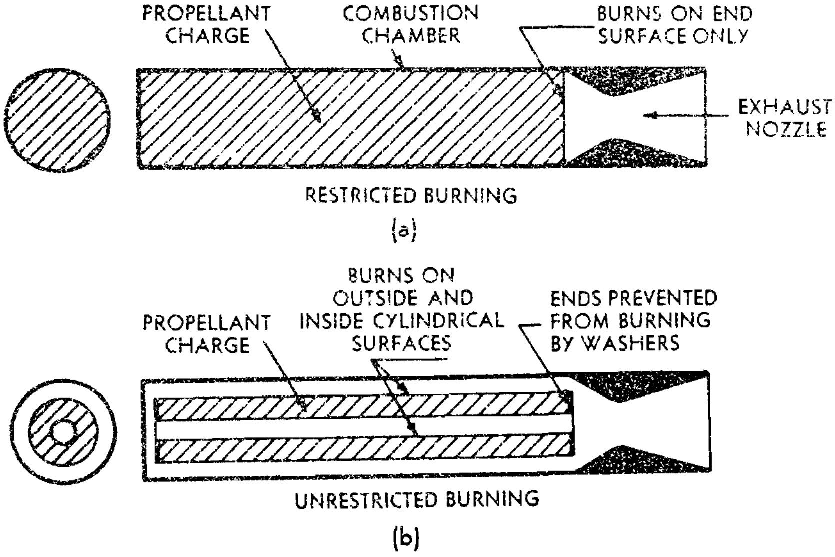

FIG. 18-7. Component parts of a solid propellant rocket.

The restricted burning rocket is one in which the propellant is constrained to burn on only one surface, Fig. 18-7(a). The manner in which a cigarette burns is similar to the type of burning of a restricted burning rocket. For example, the charge in a JATO unit is poured in while liquid and fits the chamber tightly so that it is only free to burn on one end. The unrestricted burning rocket is essentially free to burn on all surfaces at the same time, Fig. 18-7(b). The restricted burning rocket delivers a small thrust for a relatively long period while the unrestricted burning rocket delivers a relatively large thrust for a short period. Table 18-1 shows an approximate comparison between the two types of solid propellant rockets.

As the thrust increases, the duration of flight of a rocket decreases. This is due primarily to the fact that available materials for the construction of the combustion chamber and nozzle will not withstand for long the high temperatures and enormous stresses to which they must be exposed. The combustion chamber pressures range from 600 to 2000 psi and the chamber temperatures from 3000 to 5000° F. There is no way to cool this type of rocket, except for the small amount of cooling obtained from the air stream.

The solid propellant charge contains all the material necessary for combustion, i.e., fuel and oxidizer. The charge may be homogeneous, as for example, ballistite, which consists essentially of nitroglycerin and nitrocellulose. These compounds contain the oxygen necessary for combustion. The charge may be a heterogeneous mixture of fuel and oxidizer. The oxidizers may be chlorates, perchlorates, or nitrates. The fuels, which also act as a binder to give the charge desirable strength

TABLE 18-1

SOLID PROPELLANT ROCKETS

Type	Restricted	Unrestricted
Thrust	100–3000 lbs	500–50,000 lbs
Duration	4–120 sec	.05–8 sec
Specific impulse	160–200 lb-sec/lb	160–200 lb-sec/lb
Applications	JATO Airborne missiles	Aircraft rockets Artillery rockets Anti-aircraft rockets Boosters

characteristics, may be asphalt or oil mixtures, waxes or plastics. In the heterogeneous charge, the oxidizer is usually about 75 per cent of the weight of the propellant while the remaining 25 per cent is the fuel.

It is of interest to note, that under severe cold conditions, the propellant grain will not burn hot enough to keep itself ignited, and a process known as "chuffing" may occur. In "chuffing," the propellant alternately catches fire from the hot motor walls and goes out. Since this may occur with the armed round skidding in random directions on the ground near the launching position, it may be a cause of some concern. At the other extreme, a rocket projectile exposed to the hot noon time sun on a summer day may get the propellant so hot that, when fired, burning is almost instantaneous. Motor failure results. While war rockets are designed to operate under widely varying conditions without failure, the effect of temperature on the propellant must always be realized.

The basic requirements of a solid propellant are:

1. *Burning rate.* Should remain nearly constant regardless of pressure or temperature.
2. *Mechanical strength.* Should have adequate compressive and impactive strength, especially at low temperatures. Compressive strength is required to prevent grain break-up while burning. Impactive strength is required to prevent grain break-up while handling. A broken grain would have greater surface area, would release gas at a greater rate, and would probably, at best, be inaccurate; at worst, cause a motor explosion.
3. *Toxicity.* For ease of manufacture the propellant should be as non-toxic as compatible with satisfactory performance.
4. *Specific Impulse.* Should be as high as practicable. Solid propellants now vary from 100 to 220 pounds of thrust per pound of propellant burned per second. Great improvement is not likely in the near future.

18-6. Liquid Propellant Rockets. Liquid propellant rockets, as the name implies, utilizes liquid propellants which are stored in the containers outside the combustion chamber. The basic theory of operation of this type of rocket is same as that for solid propellant rocket. This theory was described in Article 18-3.

Liquid propellant rockets were developed in order to overcome some of the undesirable features of the solid propellant rockets such as short duration of thrust, and no provisions for adequate cooling or control of the burning after combustion starts. In the liquid propellant rocket, the propellants in a liquid state are injected into a combustion chamber, burned, and exhausted at a high velocity through the exit nozzle. Also, the liquid propellant is used to cool the rocket engine by circulation of fuel around the walls of the combustion chamber and around the nozzle. The cooling of motor has three basic purposes which are as follows:

1. Maintains metal strength by keeping combustion chamber walls and nozzle surfaces cool enough to prevent failure.
2. Reduces heat losses by using heat lost to motor walls to preheat fuel.
3. May allow the use of less expensive or non-critical materials for combustion chambers.

The maximum duration of an uncooled rocket motor is about 25 seconds when the same type of fuel is used as in cooled rocket motor. The use of liquid cooled rocket motors allows them to be operated as long as fuel lasts. Certain liquid fuels, however, such as hydrogen peroxide, burn at such temperatures that no cooling is needed. (See Chapter XIX.) In general, though, most liquid propellant rocket motors are regeneratively cooled.

Another basic advantage of liquid propellants for rocket use is that the size of the combustion chamber may be reduced for a given thrust, as compared to the solid propellant rocket. The use of solid propellants normally requires that all of the propellant be in the combustion chamber at the start of burning. The importance of this may be seen from the amount of fuel used in a large rocket such as the V-2 missile. The V-2 carries about 10 tons of propellant and burns it in about 71 seconds. Putting that much solid propellant in a rocket would require too large and too heavy a combustion chamber.

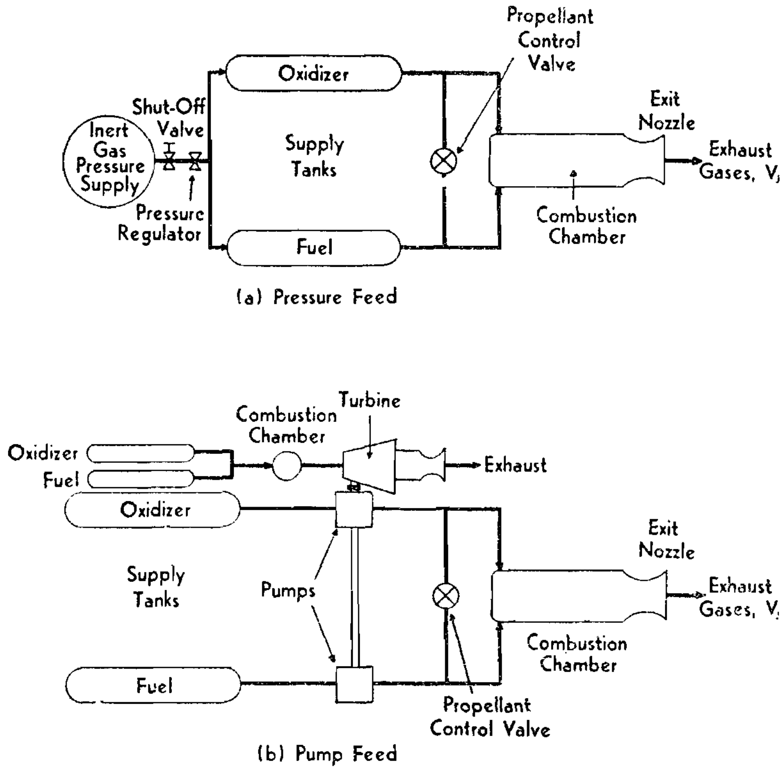

FIG. 18-8. Liquid bipropellant rocket systems.

As it was pointed out previously, an additional advantage of a liquid propellant rocket over that of a solid propellant engine is the ability to discontinue combustion in the combustion chamber at any time.

A liquid propellant rocket consists of three major components as follows:

1. Rocket motor
2. Propellant system
3. Controls

Figure 18-8 shows schematic diagrams of two liquid bipropellant rocket systems.

The *rocket motor* consists of an exit nozzle, Fig. 18-1, a combustion chamber, propellant injectors and an ignition system. As it was stated previously, most liquid fuel rocket motors are cooled by circulating fuel around the engine walls. The cooling fluid may move axially, or may circulate in a helical path around the motor. Normally, *axial flow* is only used for large size motors, because the amount of fuel used for propulsion of small motors is not sufficient to fill the practical size passages that can be manufactured. *Helical flow*, on the other hand, is usually used in small motors, since it produces too much pressure drop to be practical when considerable quantities of fuel are used as in large size motors.

Film cooling is another method of cooling in which a thin layer of fuel covers the exposed wall surfaces from excessive heat. The thin film of fuel may enter the combustion chamber and nozzle through a series of holes in the motor walls or walls of a special porous material that may be used. This method, however, reduces the efficiency of the engine, because some of the fuel used for cooling is not burned. The best way to arrange for the maintenance of the protective film is not yet known.

The combustion chamber is usually of cylindrical shape (Fig. 18-1) with one end closed and the other end terminating at the entrance to the exit nozzle, which is usually of the DeLaval type. There are no obstructions on the walls between the combustion chamber and the nozzle. In the liquid propellant rocket, the combustion chamber pressures vary from 300 to 750 psia, and combustion chamber temperatures from 3000 to 6000° F. The exhaust gas velocities vary from 5600 to 12,000 ft per sec.

Injection of the fuel and oxidizer into the combustion chamber is accomplished through injectors, which have the same function as those in a compression ignition engine, i.e., atomize and mix the propellants so that a homogeneous fuel-oxidizer mixture results which can be readily vaporized and burned.

At present, there are three basic types of injectors (Fig. 18-9):

1. Impinging stream injector.
   The fuel and oxidizer are injected through a number of separate holes so arranged that the fuel stream from one hole intersects the oxidizer stream from another hole and both break up into small droplets. This works well with hypergolic[2] propellants.
2. Spray injectors.
   The injector delivers a conical, cylindrical or sheet spray pattern. The spray patterns from fuel and oxidizer nozzles intersect to atomize and mix.
3. Non-impinging injector.
   A series of holes through which the fuel and oxidizer are in-injected and the streams do not intersect. The mixing is done by turbulence and diffusion. It is useful for non-hypergolic fuel combinations.

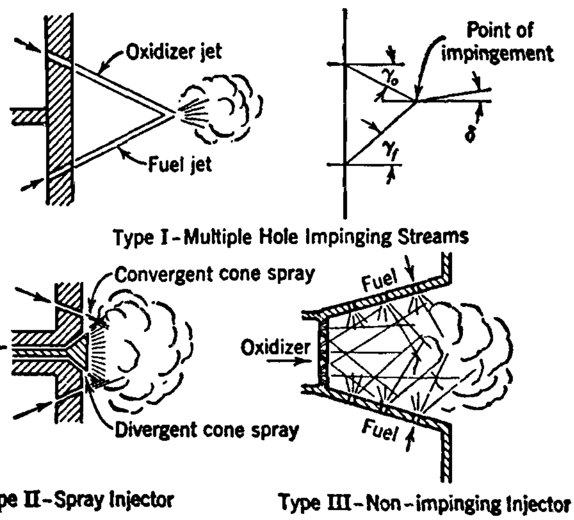

FIG. 18-9. Injector types. (Reprinted with permission from Sutton "Rocket Propulsion Elements," copyright 1950, John Wiley & Sons, Inc.).

As it was pointed out previously, fuels of the hypergolic type ignite spontaneously upon contact with the oxidizer. In this case, therefore, there is no need for ignition system that would start the combustion.

[2] Ignite spontaneously upon contact of fuel with oxidizer.

There are other types of liquid fuels, however, that need a source of energy for initiation of combustion. These can be a spark plug type ignition similar to our present automobile ignition system, powder charge igniter, introduction of small amount of fuel into the combustion chamber that has characteristics of hypergolic fuel or a use of catalyst (see Chapter XIX) that will initiate combustion upon coming into contact with operating fuel such as hydrogen peroxide.

The *propellant system* employs either a pressure feed or a pump feed to transfer fuel from a storage tank to the combustion chamber.

In the *pressure feed* system, Fig. 18-8(a) the pressure exerted by the inert gas stowed under high pressure forces the fuel and the oxidizer through the proportioning valves or orifices that regulate the fuel-oxidizer ratio into the combustion chamber against the combustion pressure. The pressure feed system is simple, inexpensive and reliable. It is limited, however, to small or short duration rockets, because the weight of tanks becomes prohibitive when this system is used in large rockets. The weight of tanks capable of carrying several tons of propellants and being pressurized to pressures of 300 to 750 psia would be greater than the more complex pump feed system described below.

In the *pump feed* system, Fig. 18-6(b), a pump is utilized to force the propellants into the combustion chamber. This feed system is adaptable to rockets of high power and long duration. The pumps are driven by relatively small turbines which may or may not have their own combustion chambers. The turbines having combustion chambers will be operated by the products of combustion of monopropellant (explained later) such as hydrogen peroxide or the main rocket fuel and oxidizer. The turbines having no combustion chambers can be driven by the combustion products bled off from the main rocket motor.

There are advantages and disadvantages to each of these three systems and the choice is based upon factors beyond the scope of this book. It may be of interest, however, to note that the World War II designed V-2 used the monofuel system, and the more recently designed and continuously modified "Viking" rocket uses a chamber bleed system.

A disadvantage in the use of the monopropellant is that the extra weight of the turbine fuel may be from 3 to 5% of the total weight. Since this is a low energy fuel, the over-all efficiency of the rocket is reduced. A disadvantage of the main fuel system is that the rocket fuels themselves are usually too powerful to be used by a turbine, while a disadvantage of the chamber bleed system is that it requires an auxiliary starting system.

The successful operation of liquid propellant rocket depends on the functioning of its controls. In general, *controls* consist of propellant metering devices, starting devices, pressure regulators, ignition, safety devices, etc. Some of these devices must be remote controlled, while others are of fixed prearranged nature. Controls of this type of rocket are rather complex, and therefore, are beyond the scope of this text.

The two main classes of liquid propellant rockets are monopropellants and bipropellants. The monopropellant contains all the elements necessary for the decomposition of the liquid into high temperature gases; thus, one storage tank is eliminated and the control system is simpler. The most common monopropellants are nitromethane and hydrogen peroxide. The bipropellant utilizes two liquids: oxidizer and fuel. The principal oxidizers now in use are liquid oxygen, nitric acid, and hydrogen peroxide.

A few of the commonly used and proposed propellants with their specific impulse are shown in Table 18-2.

TABLE 18-2

LIQUID PROPELLANTS

Oxidizer	Fuel	Specific Impulse (lb-sec/lb)	Jet Velocity (ft/sec)
(a) Bipropellant			
Liquid oxygen	Liquid hydrogen	360	12,000
Liquid oxygen	Ethyl alcohol	245	8,150
Liquid oxygen	Gasoline	235	8,100
Nitric acid	Aniline	220	7,500
(b) Monopropellant			
Hydrogen peroxide		145	
Nitromethane		220	

The choice of a liquid fuel as a rocket propellant is based upon the following considerations:

1. Specific impulse
2. Density impulse[3] $= \dfrac{\text{Specific Impulse}}{\text{Specific Volume}}$
3. Availability and expense
4. Ease of handling
5. Storage requirements

[3] Fuels of high energy content and low density (lbs/ft^3) are not desirable because of large and heavy containers that are required to store enough fuel for a given mission. What is wanted is maximum energy in a smallest package.

Table 18-3

Engine Performance

Engine type	Flight speed (mph)	Fuel Consumption (lb)/(hr)(lb)			Thrust per area (lb)/(sq ft)			Thrust per weight (lb)/(lb)		
		Altitude								
		0	30,000	50,000	0	30,000	50,000	0	30,000	50,000
Reciprocating Engine-Prop	300	0.41	0.32	0.30	120	155	140	0.42	0.39	0.28
	500	.69	.56	.53	70	90	80	.23	.26	.20
Turboprop	300	0.39	0.34	0.34	280	150	70	0.99	0.41	0.17
	500	.63	.60	.60	170	80	30	.78	.32	.14
Turbojet	300	1.07	0.97	0.94	760	300	110	2.40	0.87	0.40
	500	1.22	1.09	1.07	780	320	120	2.60	1.01	.43
	1200	1.70	1.40	1.40	1500	800	300	7.10	2.80	1.20
Turbojet with After burner	300	1.99	1.69	1.64	1290	515	215	3.31	1.28	0.35
	500	2.12	1.80	1.74	1405	600	270	3.61	1.38	.46
	1200	2.25	2.00	2.00	3100	1500	800	10.00	4.60	1.90
	1800	2.40	2.20	2.20	8000	4100	1800	22.70	17.40	7.20
Ram jet	1200	2.80	2.30	2.30	2700	1200	1400	16.40	10.80	5.90
	1800	2.60	2.10	2.10	9000	3800	1600	16.80	11.80	8.30
Rocket	1200	16.50	14.80	14.40	7900	8900	9050	27.10	29.80	30.40
	1800	16.50	14.80	14.40	7900	8900	9050	27.10	29.80	30.40

The handling, storage characteristics, and physical properties of the propellants differ materially for each liquid fuel. These properties are very important and should be well known before using a particular propellant or combination of propellants.

18-7. Comparison of the Various Propulsion Systems. Table 18-3 is a summary of the engine performance of the various types of propulsion systems. It was prepared by E. J. Manganiello of the NACA Flight Propulsion Laboratory, Cleveland. It was compiled from a group of papers on the comparison of the performance of aircraft propulsion systems which were presented by the NACA Cleveland Laboratory Staff at the March 28, 1947 meeting of the Institute of Aero-

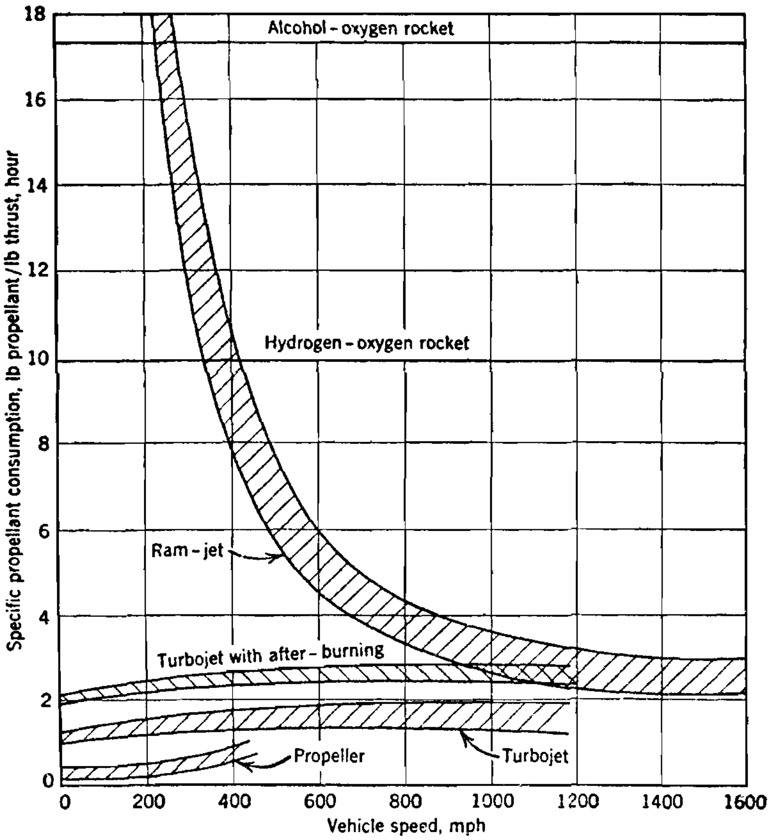

FIG. 18-10. Propellant or fuel consumption versus Flight speed for different propulsion systems. (Reprinted with permission from Sutton "Rocket Propulsion Elements," copyright 1950, John Wiley & Sons, Inc.)

nautical Sciences. The papers have been published as NACA Technical Note No. 1349 entitled "Performance and Ranges of Application of Various Types of Aircraft-Propulsion Systems" (reference 18-1).

The propulsion systems compared were: (a) conventional reciprocating engine and propeller, (b) turboprop engine, (c) turbojet engine, (d) turbojet with afterburner (turbojet with tail-pipe burning), (e) ram jet engine, and (f) rocket engine. The performance values presented are not, in general, for existing engines but were calculated on a basis of laboratory tests of the various components of the propulsion systems and reasonable assumptions. However, the values shown in Table 18-3 give a general comparison of the performance characteristics of the various propulsion systems.

The general and specific assumptions used in the preparation of Table 18-3 may be found in reference 18-1.

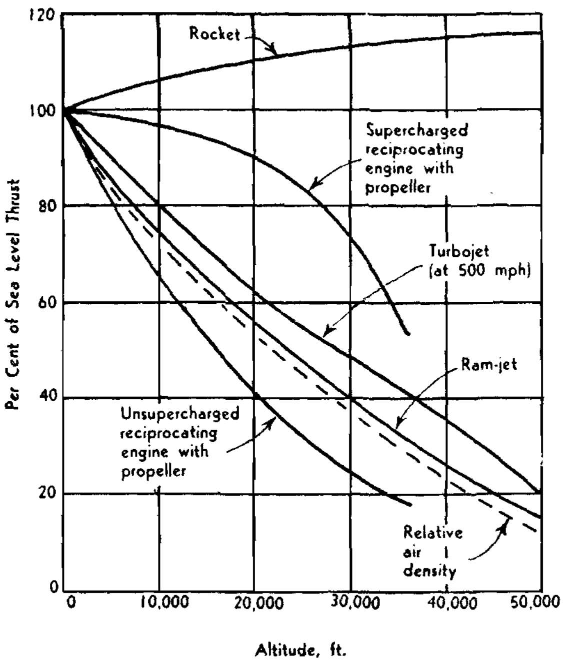

FIG. 18-11. Variation of thrust with altitude for different propulsion systems. (Reprinted with permission from Sutton, "Rocket Propulsion Elements," Copyright 1950, John Wiley & Sons, Inc.)

Figure 18-10 shows the specific propellant consumption in pounds per pound of thrust per hour versus speed for different propulsion engines. The curves in this figure indicate that the use of rocket engines to power airplanes as we know them today is not feasible because of their high fuel consumption. Also, the use of ram-jet engine is not economical at lower than 1000 mph vehicle speeds.

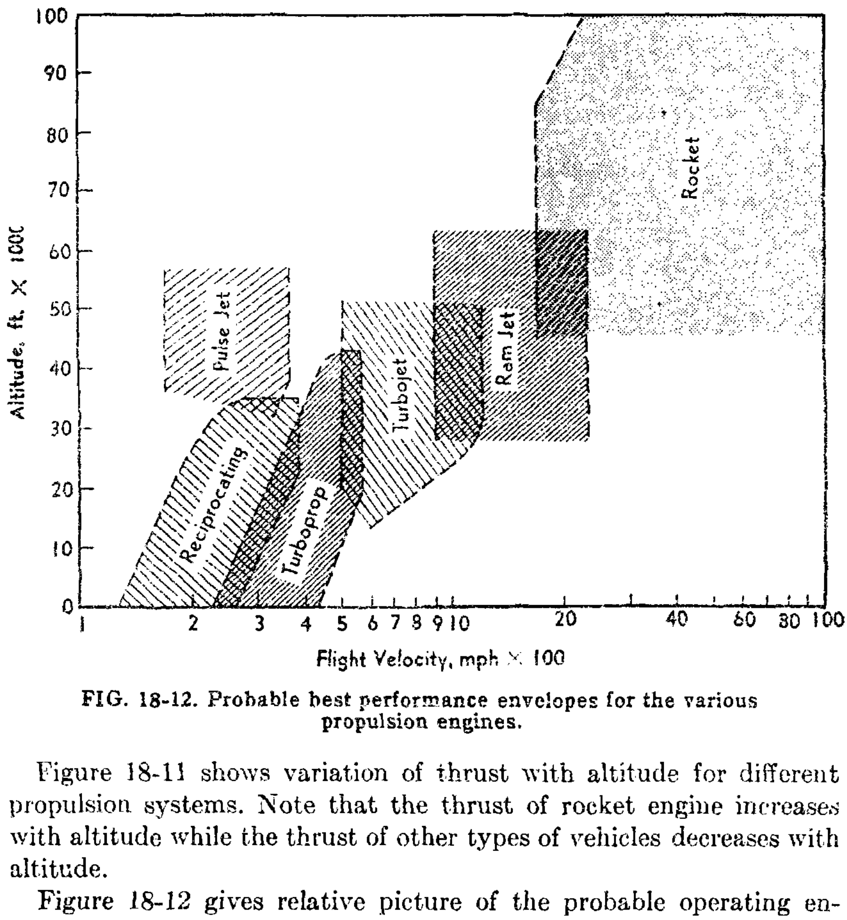

FIG. 18-12. Probable best performance envelopes for the various propulsion engines.

Figure 18-11 shows variation of thrust with altitude for different propulsion systems. Note that the thrust of rocket engine increases with altitude while the thrust of other types of vehicles decreases with altitude.

Figure 18-12 gives relative picture of the probable operating envelope of the various propulsion systems, while Fig. 18-13 presents a pictorial comparison of the performance characteristics of the various types of engines.

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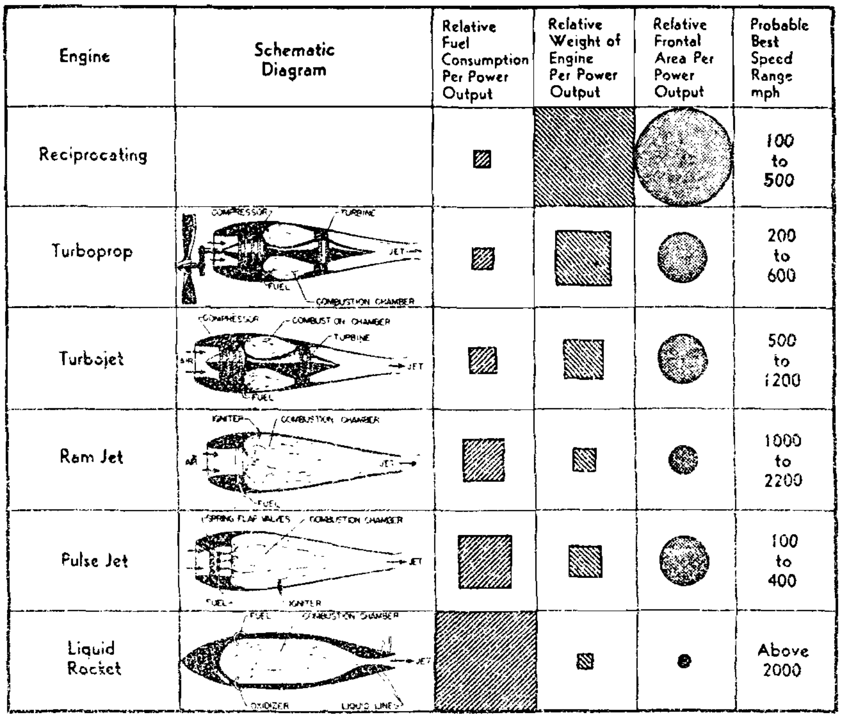

FIG. 18-13. Comparison of the relative performance characteristics of the various propulsion power plants.

18-3. W. G. A. Perring, "The Mechanism of the German Rocket Bomb (V-2)," *The Journal and Proceedings of the Institute of Mechanical Engineers* (England), June 1946

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18-6. George P. Sutton, "Rocket Propulsion Elements," John Wiley and Sons, Inc., New York, 1949.

EXERCISES

18-1. What are the advantages and disadvantages of rocket engines?

18-2. Define specific impulse.

18-3. The effective jet exit velocity of a rocket is 8000 ft per sec, the forward flight velocity is 4000 ft per sec, and the propellant consumption is 161 lbs per sec.

Calculate:

(a) Thrust
(b) Thrust horsepower
(c) Propulsive efficiency

Ans:

(a) 40,000 lbs (c) 80.0%
(b) 290,000 hp

18-4. Given the following data for a bipropellant rocket engine:

$V_0 = 2000$ ft/sec
$V_{je} = 6000$ ft/sec
$w_p = 15.6$ lbs/sec

Calculate

(a) Thrust (b) Propulsive efficiency

Ans:

(a) 2950 lbs (b) 60%

18-5. What are the differences between a restricted and an unrestricted solid rocket? List two applications of each type.

18-6. Draw a schematic diagram of a liquid bipropellant rocket engine using a pump feed. Label all components.

18-7. What are the differences between a pump feed and a pressure feed system? When should each be used?

18-8. What are three types of oxidizers in common use?

18-9. What is the difference in the use of a monopropellant and a bipropellant liquid rocket engine? Give an example of each.

18-10. Compare the various types of jet propulsion engines as to specific fuel consumption, best speed range, and relative weight.

CHAPTER XIX

HYDROGEN PEROXIDE FOR PROPULSIVE POWER[1]

19-1. History. The idea of utilizing hydrogen peroxide in propulsion engines was first proposed by the Germans in 1936. It was known that hydrogen peroxide under the proper conditions would yield gaseous oxygen which could be utilized as an oxidizer for the combustion of fuel in a heat engine. By such a process, the engine could operate completely independent of an atmospheric air supply, thus making it possible for the engine to provide propulsion power for submarines, torpedoes, and rockets. In 1936, the maximum concentration of hydrogen peroxide, i.e., the ratio of the weight of the hydrogen peroxide to the total weight of the solution, was thirty per cent. This low concentration could not be used in a propulsion engine due to the excessive quantity of liquid required to be carried to provide a sufficient amount of oxygen for the combustion of a fuel. A German peroxide manufacturer[2] developed a new distillation process which produced peroxide for commercial use with a concentration up to 85 per cent by weight. Following this development, the Germans aimed their research toward the design and construction of submarine, torpedo, and rocket engines utilizing hydrogen peroxide as a propellant.

The leading proponent of the use of hydrogen peroxide in Germany was Professor Walter. He named the 80 to 85 per cent concentration of hydrogen peroxide "Ingolin" after his son. "Ingolin" and "T-substance" were the terms used in Germany to designate hydrogen peroxide; and although these terms have been used by some scientists in this country, they have not been accepted, as yet, as common scientific terminology.

The basic fundamentals and characteristics of hydrogen peroxide and its application to propulsive engines as developed by the Germans will be discussed. However, many phases of the subject and the details of the various systems are in the classified status and cannot be set forth in this text.

19-2. Chemistry of Hydrogen Peroxide. Hydrogen peroxide (formula H_2O_2) will be discussed in terms of the per cent concentration of hydrogen peroxide by weight. For commercial applications, such as a bleach, a 30 per cent concentration is used, and, as a disinfectant, a three per cent solution is used. However, an 80 to 90 per cent concentration is required for propulsive power in submarine and rocket motors.

[1] References 19-1, 2.
[2] Electrochemische Werke, München.

Hydrogen peroxide is a clear liquid similar in appearance to water, except for a slight greenish tint, a slight but distinct odor, and the presence of minute gas bubbles throughout the solution. It is considerably heavier than water and has a specific gravity of between 1.36 and 1.40 for concentrations between 80 and 90 per cent respectively.

Hydrogen peroxide has a strong tendency to decompose to water (or steam) and oxygen, and the decomposition is accompanied by a release of heat. The chemical equation for the decomposition of hydrogen peroxide is

$$2\ H_2O_2 \rightarrow 2\ H_2O + O_2 + \text{heat}.$$

The above reaction is continuously occurring to a small extent within the solution. If a stabilizer is added and the heat released is permitted to dissipate, hydrogen peroxide under good storage conditions has a loss in concentration of about one to three per cent a year. The rate of decomposition is primarily a function of the temperature and the catalytic impurities present either in solution or on the surface of the containers. Unfortunately, almost all substances act as a catalytic agent including dust, dirt, and most metals in the form of solids or ions in the solution. Only a few materials can be classed as violently catalytic, such as the permanganates of calcium, potassium, or sodium. These compounds are used as a solid in the catalyst chamber of the hydrogen peroxide engine or as a liquid catalyst to be sprayed into a combustion chamber. When hydrogen peroxide is sprayed into the catalyst chamber, the permanganates cause an immediate and complete decomposition of the H_2O_2 accompanied by a relatively small heat release of the exothermic reaction. The heat released by concentrations of 80 to 90 per cent is between 990 and 1120 Btu per lb of hydrogen peroxide; and the resulting mixture, which contains from 62 to 58 per cent superheated steam and from 38 to 42 per cent oxygen by weight, would have a temperature between 950 and 1380° F. The resulting volumetric expansion of the decomposition of hydrogen peroxide would be large, around 5000:1.

When hydrogen peroxide is used as a monopropellant or monofuel, as above, the heat released from the decomposition, around 1000 Btu per lb, is about one-twentieth that of the heat released from the combustion of one pound of a hydrocarbon fuel. The use of hydrogen peroxide as a monopropellant or monofuel is a costly process and was used by the Germans only when the reliability and simplicity outweighed the cost. Since H_2O_2 upon decomposition provides free oxygen, its principal use therefore is to act as an oxidizer to furnish the free oxygen for the combustion of a fuel in a bipropellant system.

The contact of hydrogen peroxide with a combustible material such as wood results in the ignition of the wood, especially if some catalytic material, i.e., dust or dirt, is present. The following processes take place: the dirt causes local "hot spots" from the decomposition; the heat released from the decomposition raises the temperature of the combustible to its autoignition point; and the wood continues to burn very readily in the oxygen released from the decomposition and in the oxygen from the atmosphere. Since the decomposition supplies free oxygen, the fire cannot be extinguished by carbon dioxide or carbon tetrachloride. Water is the only liquid that will extinguish a fire fed by the O_2 from the decomposition of H_2O_2. The water reduces the concentration of the hydrogen peroxide which results in the reduction of the heat released from the decomposition and in the lowering of the temperature to below the autoignition point of the combustible.

Hydrogen peroxide alone is neither shock sensitive nor combustible so that there is no danger of reaction propagation along a stream of the liquid as there is with gasoline. It is somewhat thermal sensitive in that decomposition can be effected at elevated temperatures without the presence of a catalyst. The decomposition temperature is around 140° F, but may vary depending on the concentration and the impurities present in the solution. However, if the peroxide is mixed with a fuel such as alcohol, the resulting mixture is very sensitive to shock, heat, or catalytic material. If a liquid catalyst such as hydrazine hydrate is added to the mixture, the resulting reaction is extremely violent and has an energy release equivalent to that of nitroglycerine. Thus, if hydrogen peroxide is mixed with alcohol or glycerine, it is very unstable and dangerous to have around.

19-3. Hydrogen Peroxide Storage. Up until the past few years, it was thought that highly concentrated hydrogen peroxide would be unstable and unsafe to store for prolonged periods. The Germans found that hydrogen peroxide was safe to store as long as: (1) the impurities in the solution were small; (2) the storage tanks were properly prepared; and (3) the temperature of the liquid was observed regularly and not permitted to rise. A rise in temperature is an early indication that decomposition has increased from some contamination. The decomposition heat release must be checked by external cooling or by the addition of a stabilizer so that a further increased rate of decomposition will not result from the temperature rise. Normal decomposition may be retarded by the addition of a very small amount of stabilizer such as sodium phosphate, phosphoric acid, or 8-oxyquinoline. These compounds are especially useful if the contamination has resulted from metallic ions; but decomposition may not be checked by a stabilizer if gross contamination is present. If the temperature continues to rise and cannot be checked, the hydrogen peroxide is dumped into water.

The material most practical for permanent bulk storage containers of hydrogen peroxide is aluminum of at least 99.5 per cent purity and having a low copper content (0.03 per cent maximum). For machinery parts and high pressure piping, the austenitic stainless steels are satisfactory. Stainless steel is not recommended for storage containers because the decomposition rate of the hydrogen peroxide is somewhat higher than when in contact with aluminum. There are other aluminum alloys which may be used for storage containers and piping. Also, pyrex glass and some plastics such as polyvinyl chloride are compatible. It is necessary to keep the piping system tight and to prevent the leakage of hydrogen peroxide. Water may be used to clean and flush out the system.

19-4. Hydrogen Peroxide Manufacture. Hydrogen peroxide is manufactured by the Pietzsch method. Sulfates (SO_4) are converted into persulfates (S_2O_8) by an electrolysis process. The persulfates are precipitated from the solution and converted into hydrogen peroxide and sulfates by a high temperature steam treatment. The weak solution of hydrogen peroxide is concentrated by vacuum distillation up to the required concentration.

The cost of manufacture of hydrogen peroxide in concentrated form is high due to the electrical power, expensive equipment, and man hours required. As manufacturing facilities are increased the cost will come down but will probably remain high as compared to some of the other oxidizers. At the end of the war, the Germans could produce about 3000 tons of hydrogen peroxide a month, which was a small quantity compared to their military requirements.

The Germans were far ahead of any other country in the manufacture of hydrogen peroxide. In this country, concentrations higher than 50 per cent were a laboratory product during the war. However, concentrations of 90 per cent with fewer impurities than produced by Germany are now commercially available in this country.

Since the Germans were the first to produce high concentrations for commercial use, they pioneered the utilization of hydrogen peroxide for propulsive power. They used 80 to 85 per cent concentration in the engines of 26 types of operational war weapons and in 40 or more weapons that were in the experimental stage. Among the principal uses of hydrogen peroxide by the Germans were a monopropellant system to launch the V-1 bomb, to drive the propellant feed pumps of the V-2 bomb, and to furnish propulsive power for short duration power plants, and in a bipropellant system to supply the oxidizer for the propulsive engines of torpedoes, submarines, airplanes, and guided missiles.

19-5. Hydrogen Peroxide As a Monopropellant or Monofuel. The Germans used hydrogen peroxide as a monopropellant where the simplicity and reliability of this type of engine outweighed the disadvantage of the costly high specific propellant consumption. The principal advantages of this type of engine over the bipropellant system engines are the very simple design, avoidance of complicated flow-control devices, very compact construction, ability to undergo long storage, and reliability upon first trial run of the engine. An engine that obtained its propulsive power by the use of H_2O_2 as a monopropellant was called a cold engine because the temperature of the mixture resulting from the decomposition was only around 1200° F. This type of engine was used for short duration power plants in expendable devices, which included projectile propulsion, gliding bombs, remote control gliders, aerial torpedoes, underwater torpedoes, etc., and in reusable devices, which included take-off and landing auxiliaries, catapults, auxiliary drive for pumps, etc. Cold engine power plants were built in Germany in many sizes from 20 lbs thrust up to 130,000 lbs thrust.

In order to decompose the H_2O_2, the hydrogen peroxide must come in contact with a violent catalytic agent. The catalytic agents used in the monopropellant engines in Germany were in the form of a solid or a liquid. In some types of engines, the hydrogen peroxide may be sprayed into a catalytic chamber containing porous porcelain stones that are coated with permanganates of calcium, potassium, or sodium, while in other types of engines the catalyst in the form of a liquid and the hydrogen peroxide are sprayed directly into a combustion chamber. The liquid permanganates were termed Z-substance by the Germans.

Figure 19-1(a) shows the basic components of a monopropellant hydrogen peroxide jet propulsion engine of short duration developed by Walter using a liquid catalyst. The air flask is sealed by the diaphragm of a primer-operated rip valve. A pressure wave created by the powder charge of the primer blows out the center of the diaphragm or rip sheet; and the rest of the sheet folds back under the high pressure from the air wave and lies against the walls. The high pressure air flows through a pressure reducer, and either through sealing membranes designed to break at a certain pressure, or through check valves into the propellant and liquid catalyst tanks. This forces the propellant and liquid catalyst through their lower sealing membranes or non-return valves into the combustion chamber where the hydrogen peroxide is decomposed. The mass of the products of the decomposition then passes through the jet nozzle at a relatively high speed to produce the thrust power.

Figures 19-1(b) and 19-1(c) show two types of monopropellant hydrogen peroxide, short duration, jet propulsion engines utilizing a solid catalyst for the decomposition. In Figure 19-1(b), after the valve is opened, the H_2O_2 is fed by gravity into the catalytic chamber, decomposed, and converted into steam. Since the steam is sufficiently pure, a small part of it may be led directly back into the propellant tank, where it is utilized as a pressure feed gas. The remainder of the

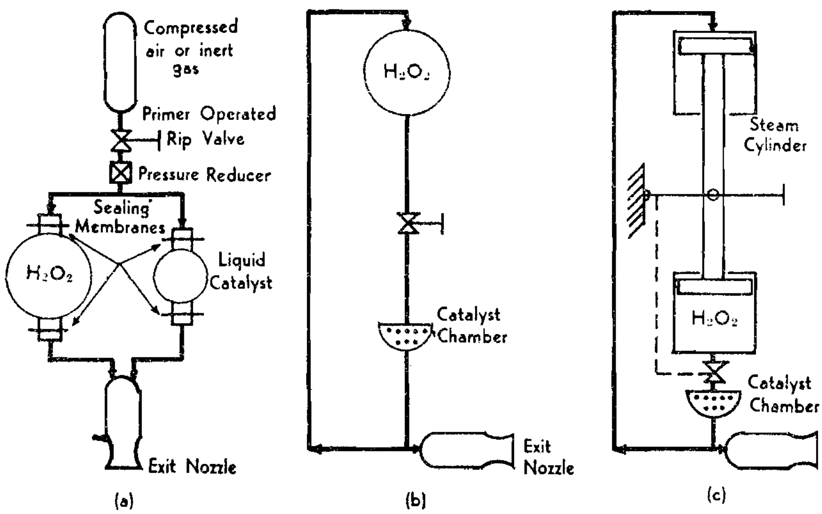

FIG. 19-1. Basic components of Walter's monopropellant hydrogen peroxide jet propulsion engines.

steam is expanded through the exit nozzle into the atmosphere. Although a very rapid spontaneous decomposition of the hydrogen peroxide occurs at 140° F, this unit utilizing steam at a high temperature is not dangerous if the entire working process is of a short duration.

A further development of this system is to separate the pressure feed steam from the propellant by utilizing a differential piston system as shown in Fig. 19-1(c). After opening the valve, a small amount of H_2O_2 is sprayed into the catalyst chamber by a hand operated pump or by other auxiliary means to start the decomposition. The forced feeding then proceeds at a rate that is dependent on the ratio of the piston diameters.

Figure 19-1 shows only the basic components of three types of Walter monopropellant engines. These types of engines are restricted to short duration, usually less than 30 seconds but may be as long as two minutes, and are a rocket type of engine. When a monopropellant engine is used in a catapult to launch missiles such as the V-1 bomb, the steam generation takes place directly in a constantly increasing cylinder space.

For bipropellant rocket engines of long duration, i.e., duration of one minute or longer, it is necessary to use the pressure pump feed system in place of the inert gas pressure system (Article 18-6). The high speed centrifugal propellant pumps are driven by a small turbine in which the working fluid is steam from the decomposition of hydrogen peroxide (Fig. 19-2). The flow rate of the propellants is regulated by varying the speed of the pumps or by throttling the flow from a constant pressure pump. The essential features of this type of pressure feed system are shown in the lower part of Fig. 19-2, i.e., in the auxiliary pump section of a rocket engine which utilizes hydrogen peroxide as a propellant.

The turbine, which is brought up to speed by means of a starting motor, drives the oxidizer and fuel pumps. From the hydrogen peroxide pressure line, a secondary flow is bled off and passes through a regulator valve into the catalytic chamber. Here the decomposition of the hydrogen peroxide produces steam which expands through the turbine that drives the propellant pumps. The turbine and the H_2O_2 pump are designed so that the pump pressure is always higher than the turbine operating pressure by an approximately constant amount which makes it possible to secure a feed of the hydrogen peroxide to the catalytic chamber throughout the entire operating load range of the rocket engine. Each throttling position of the regulator is thus associated with a certain operating condition. In a bipropellant rocket such as the V-2, where the oxidizer is not hydrogen peroxide, an auxiliary steam generation system such as shown in Fig. 19-1(a) is used except that the control and valve system is more complex and the steam that is generated is expanded through a turbine instead of through a jet exit nozzle.

19-6. Rocket Engine. Many of the German rocket missiles utilized the Walter "cold" engine, i.e., hydrogen peroxide as a monopropellant. However, due to the high specific propellant consumption, these missiles were restricted to a short duration of power, usually less than 30 seconds. In order to reduce the specific propellant consumption and increase the duration, the bipropellant system, where hydrogen peroxide provided the oxygen for the combustion of a suitable fuel, was used. Since the combustion of the fuel raised the jet exit temperature to between 3000 and 4000° F depending on the fuel, this type of system was called a "hot" engine.

The most publicized German airplane that used a hydrogen peroxide bipropellant rocket engine was the Me-163. This particular rocket engine was designated Walter 109-509, and it developed approximately 3800 lbs of thrust at peak load. The duration of power at peak load was 12 minutes. However, by alternately gliding and operating the engine at low power, the duration of flight could be extended to 40 minutes. The engine had three throttle settings for part load power that could be used to extend the duration of flight. The airplane de-

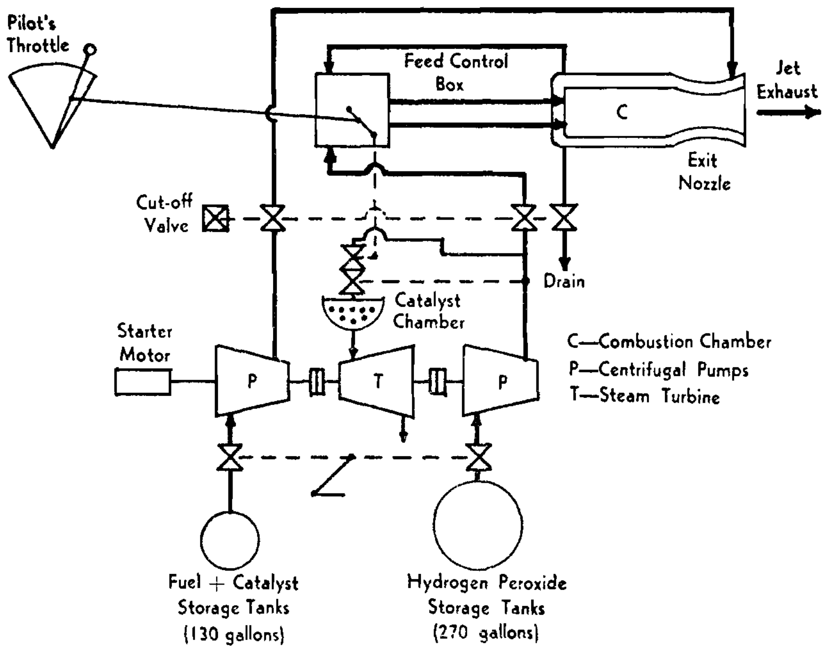

FIG. 19-2. Schematic diagram of Walter 109-509 bipropellant rocket engine used on German Me-163 and BP-20 Natter airplanes.

veloped a top speed of 550 mph and was capable of climbing to 30,000 feet altitude in 2.6 minutes. This same engine was used in the Bachem BP-20 Natter airplane that was still in the experimental stage at the end of the war.

The basic components of the Walter 109-509 rocket engine are shown in Fig. 19-2. The oxidizer of the bipropellant system was H_2O_2 of 85 per cent concentration. The fuel was methanol (methyl alcohol, CH_3OH). The fuel contained a 30 per cent solution of liquid catalyst (hydrazine hydrate, $N_2H_4 \cdot H_2O$) in order to initiate the decomposition of the H_2O_2 which in turn starts the combustion of the fuel upon the impingement of the liquid propellants in the combustion chamber. The Germans gave this mixture of the fuel and liquid catalyst the trade names of "Helmann" and T-substance.

The T-substance is circulated under pressure from the pump feed system through the combustion chamber cooling jacket to the feed control box. From the feed control box, the T-substance passes to the combustion chamber. The hydrogen peroxide under pressure passes through the feed control box into the combustion chamber. The power output of the engine was controlled by the throttle in the cockpit. Settings on the throttle were "off," "idling," and three power stages varying from 650 lbs thrust up to 3800 lbs thrust.

19-7. Torpedo Engines. The Germans used hydrogen peroxide as a monopropellant to power a jet torpedo. The jet engine had the same basic components as shown in Fig. 19-1(a). This particular torpedo, which was dropped from airplanes, developed a thrust of 1770 lbs for 106 seconds duration.

The Germans also developed another type of torpedo which was essentially wakeless and whose performance characteristics were much superior to those of any other torpedo. The propellants used in the engine of this torpedo were hydrogen peroxide, a mixture of methanol and hydrazine hydrate (T-substance), water, and a fuel that is similar to Diesel oil. This fuel was given the name of decalene by the Germans. The hydrogen peroxide and C-substance are injected into a combustion chamber to initiate the decomposition and to raise the temperature of the combustion chamber to around 1800° F. After two or three seconds, the T-substance is cut off and the decalene is injected into the combustion chamber and is burned with the free oxygen from the decomposition. The decomposition continues without a catalyst due to the high temperature in the combustion chamber. The superheated steam and products of combustion are then expanded through a turbine that is essentially a conventional steam turbine. The turbine in turn drives the propeller to furnish the propulsive power. The temperature of the steam and products of combustion leaving the combustion chamber must be cooled down to the permissible turbine inlet temperature. This is accomplished by spraying water into the combustion chamber. The water spray has the additional advantage of increasing the percentage of steam, and therefore the density of the mass entering the turbine.

This type of system results in a very dangerous situation if the arrival of the three liquids into the combustion chamber are not accurately coordinated. If any of the liquids are off-time, a serious explosion results. Reports accumulated in Germany after the war indicate that about one out of every 100 torpedoes was wrecked by explosions of this nature.

19-8. Submarine Engine. Prior to the war, Professor Walter designed, built, and operated an 80 ton experimental H_2O_2 submarine which had

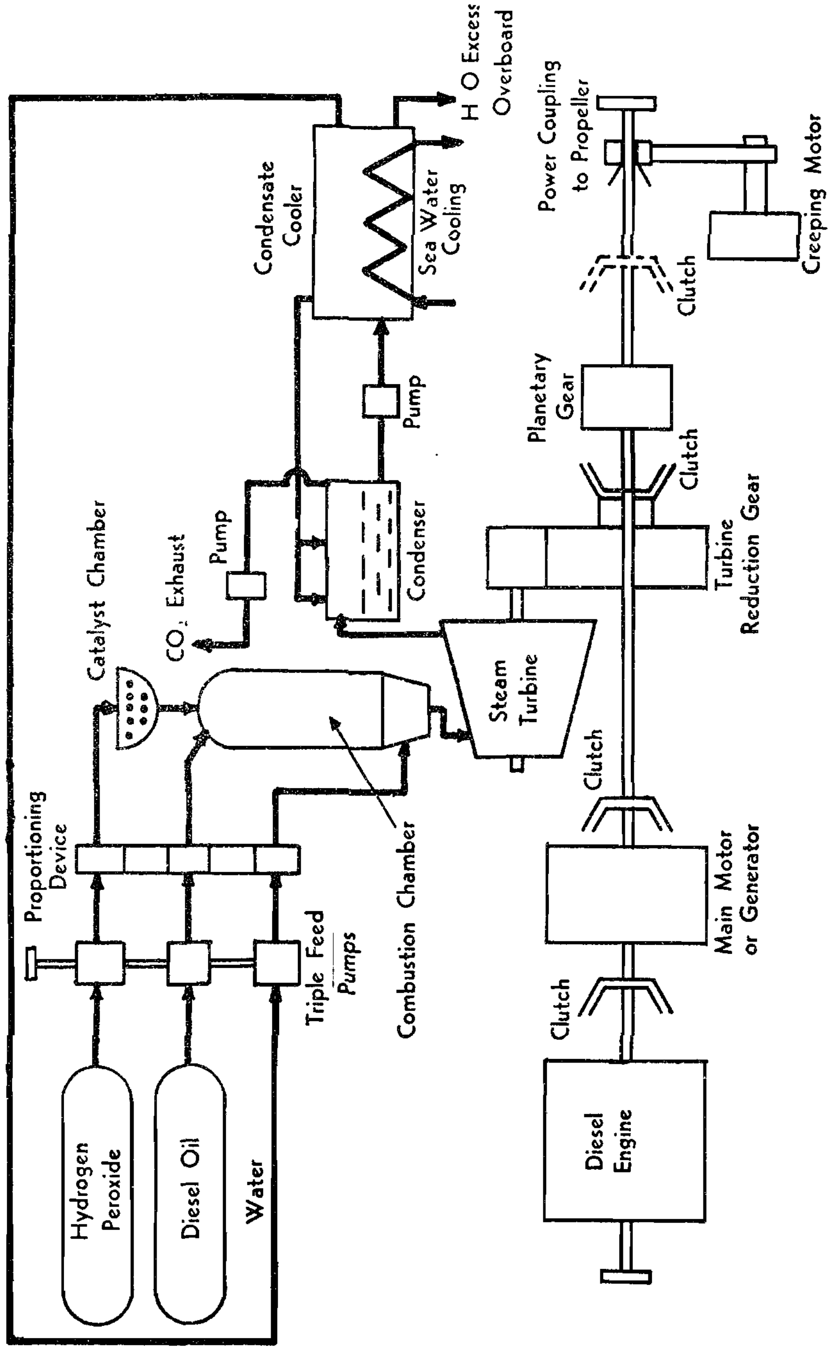

FIG. 19-3. Basic components of Walter hydrogen peroxide turbine engine for German submarine.

a submerged speed of about 25 to 26 knots. The success of this first boat led to the design and construction of nine more H_2O_2 submarines, known as the German type 17, which were used for experimental work and to train operating personnel. The German type 17 submarine was small in size, having a surface displacement of 380 tons. The H_2O_2 plant had two 2500 hp turbines which gave a submerged speed of 25 knots. These boats had a surface speed of 8.5 knots.

Under construction at the end of the war was the Walter type 26 submarine which was intended for war patrol work. These boats had a surface displacement of 900 tons and a surface speed of 11 knots. The hydrogen peroxide plant was designed with a 7500 hp turbine which produced a submerged speed of 24 knots.

Figure 19-3 shows a schematic diagram of the basic components of the hydrogen peroxide plant of the German type 26 submarine. This engine is known as the Walter turbine cycle type 26. The propellants used in this system are an oxidizer (hydrogen peroxide), a fuel (Diesel oil), and a gas coolant (water). The three propellants are delivered to the cycle under high pressure by the three high-speed single stage centrifugal pumps which are driven by a common motor. The liquids pass from the triple feed pumps to a proportioning device which meters and throttles the liquids to maintain a constant ratio of flow rates of the three liquids throughout the load range. The ratio of flow rates is 12 parts water, 9 parts hydrogen peroxide, and one part Diesel oil. The hydrogen peroxide, which is controlled by hand, acts as the master fluid and the proportioning device automatically increases or decreases the other two liquids to keep pace with it.

A cam operated starting valve sets the sequence of admission of the fluids to the cycle in starting. The hydrogen peroxide is admitted first and passes through atomizers into the catalyst chamber where decomposition takes place. The mixture of steam and oxygen then is delivered to the combustion chamber. At this time, the cooling water is allowed to circulate. The water, which passes through the walls of the combustion chamber for cooling purposes, is sprayed into the bottom of the chamber. The Diesel fuel is then sprayed into the top of the combustion chamber to burn in the free oxygen from the decomposition of the H_2O_2.

Since the combustion of the Diesel fuel creates a temperature around 3500 to 4000° F in the chamber, the cooling water is necessary to reduce the temperature of the steam and the gases of combustion to the permissible turbine inlet temperature. The chamber exit gases which consist of a mixture of carbon dioxide and steam enter the turbine at a temperature around 1100° F. The mixture of gases contains about 94 per cent steam by volume and 85 per cent by weight.

The mixture of gases leaving the combustion chamber are expanded in the turbine which operates at approximately 14,000 rpm. The turbine and its auxiliaries are more or less the same as that of a conventional steam turbine plant except in one respect. The turbine casing is steam jacketed to prevent uneven heating in starting which tends to increase the blade clearance. This permits the entrance of the steam at 1100° F without the usual warm-up period. The size of the 7500 hp turbine may be judged from the size of a 2500 hp turbine which is a little larger than a conventional automobile engine. The non-condensable turbine exhaust (CO_2) is pumped overboard by a Lysholm type positive displacement pump and is dissolved rapidly in the sea water so that no trace of a wake reaches the surface. Since the system produces excess water, some of the condensate must be pumped overboard.

Included in Fig. 19-3 are the basic components of the machinery layout of the German H_2O_2 submarines. The plant had a Diesel engine for surface and for snorkel operation. A main motor and creeping motor were provided for submerged battery drive. The main motor could serve as a generator to charge the batteries and furnish power for the auxiliaries while the submarine is being driven by either the Diesel engine or the Walter engine. The main motor was always operated as either a motor or a generator, except when the creeping motor was in use. When the creeping motor which was a high-speed, low-power motor that drove the main shaft through multiple V-belts was in use, the other gears and engines were disconnected from the propeller shaft by a main shaft clutch. When the turbine of the Walter engine was clutched in, propulsive power was furnished to the propeller through a conventional single-reduction double helical gear and a planetary gear. The Diesel and main motor were on the same shaft and drove the propeller through the planetary gear.

When starting, the auxiliary load was carried by the batteries. When the turbine took over the load from the main propulsion motor, the latter was used as a generator to supply the power for the auxiliaries. The Walter engine thus operated independently of any external power source.

In this chapter the use of hydrogen peroxide for propulsive power for various types of application has been discussed. Submarine propulsion is an ideal application for H_2O_2, since it provides the free oxygen for the combustion of fuel during submerged operation. However, any good oxidizer such as liquid oxygen or nitric acid also provides free oxygen. Therefore, with some modifications to the plants, it would be possible to use liquid oxygen or other oxidizers for the applications discussed, provided safe storage facilities, etc., are possible.

HYDROGEN PEROXIDE FOR PROPULSIVE POWER

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EXERCISES

19-1. Write the chemical equation for the decomposition of hydrogen peroxide.

19-2. What is the function of a catalyst?

19-3. Name two violent catalysts.

19-4. Describe the use of hydrogen peroxide as a monopropellant.

19-5. What are the precautions for safe stowage of hydrogen peroxide?

19-6. Draw a schematic sketch of a monopropellant engine. Label all components.

19-7. List five uses of hydrogen peroxide as a monopropellant.

19-8. What are the differences between a hot and a cold engine?

19-9. What is the function of hydrogen peroxide in a bipropellant type of engine?

19-10. Describe a torpedo engine.

19-11. Draw a schematic sketch of a hydrogen peroxide submarine plant. Label all components.

19-12. What are the advantages of a hydrogen peroxide submarine plant?

CHAPTER XX

NUCLEAR POWER FOR SHIP PROPULSION

A nuclear propulsion system is defined for the purposes of this chapter as one in which the heat energy released in nuclear fission is converted by standard thermodynamic processes to the production of useful propulsive power. The purpose of this chapter is to give the student a basic understanding of the factors which affect the production of heat energy in a nuclear power source and of the problems to be resolved in converting this heat energy to useful power.

20-1. Advantages of Nuclear Propulsion. The two primary advantages of a nuclear source of power for ships are (1) the very great quantity of energy released in the consumption of only a small amount of fuel and (2) the relative independence of such a plant from any material external to the system.

These advantages are particularly important for submarines in that they make possible the true undersea craft. By reason of its independence of the oxygen in the air, the nuclear power plant may be operated below the surface continuously at full power. Moreover, the nature of the nuclear power plant is such that it should be capable of being operated at full power over long periods of time without having to return to port for the recharging of fuel. At best, the submarine plants which depend on chemical liquid fuels for energy cannot promise more than a very short time at full power under water before the recharging of fuel or oxidizer is necessary.

For surface vessels, other advantages of nuclear power can be cited. Refueling at sea or advanced bases could be reduced or eliminated for ships with nuclear power plants. Hull arrangements might be simplified by the elimination of uptakes, smoke-pipes, fuel handling equipment, and fuel storage systems. Reballasting to compensate for weight of fuel consumed would not be necessary since this weight is a negligible fraction of the nuclear powered ship's displacement.

20-2. Energy Release from Nuclear Reactions. The energy changes in nuclear reactions are extremely large when compared with the energy changes in chemical reactions. The controlled production of nuclear power depends at present on the large energy release involved in one type of nuclear reaction, the fission or splitting of heavy nuclei such as uranium. The fissioning of one pound of uranium produces the heat equivalent of the combustion of 2300 tons of coal or 300,000 gallons of oil. This large release of energy in fission results from the conversion of approximately 0.1 per cent of the mass of the fissioned uranium into energy.

20-3. The Fission Process. In the fission process the nucleus of a heavy atom such as uranium absorbs a neutron and forms an unstable product nucleus. This unstable nucleus then splits into two more or less equal parts, called fission fragments. These fragments begin to undergo transformation by radioactive decay and the products of this decay together with the fragments are termed *fission products.*

There are two results of fission that are particularly important so far as this discussion is concerned. The first is the large amount of energy released. The major portion of this appears as kinetic energy of the fission fragments. These fission fragments are stopped by the surrounding materials (such as fuel and structural material) and their kinetic energy is released in the form of heat. Since the fission fragments are stopped and their kinetic energy released as heat practically at the point of fission, the fission process in effect produces heat directly.

The second important result is that in the average fission process, 2.5 neutrons are released. Since only one neutron is required to initiate the fission process, and since more than one neutron is released in fission, it is possible to arrange for a self-sustaining chain reaction of fissions and a continuous evolution of heat.

Although most of the energy released in fission appears instantaneously as heat energy some appears as *gamma radiation,* some as energy of the neutrons released, and some as radioactive energy of the fission products. For example, the energy released in fission of the uranium isotope of atomic weight 235 is distributed approximately as follows: 83 per cent as kinetic energy of fission fragments, 3 per cent as instantaneous gamma radiation, 3 per cent as energy of neutrons, and 11 per cent as radioactive energy of fission products.

Of the materials which undergo fission, only three have properties make them suitable as concentrated fuels for a self-sustaining neutron chain reaction. One is the *uranium isotope* of atomic weight 235 (U^{235}), mentioned above. The other two are: the *plutonium isotope* of atomic weight 239 (Pu^{239}), which is produced by nuclear processes from the most abundant uranium isotope (U^{238}); and the uranium isotope of atomic weight 233 (U^{233}), which is produced from thorium (Th^{232}).

20-4. The Self-Sustaining Chain Reaction. The assembly of fuel and other materials in which the self-sustaining chain reaction is maintained is called a *reactor.* To understand how the chain reaction is made self-sustaining, it is useful first to consider what can happen to a neutron produced by fission in the reactor. The neutron can be captured by one of the fuel nuclei and cause another fission as described above; or it can be captured without causing fission in the various other materials contained in the assembly. Alternatively, the neutron can make random collisions in the material without being captured and eventually escape from the reactor. The production of a self-sustaining chain reaction requires the attaining of a proper balance between the neutrons released in fission and the neutrons either captured or lost in the above processes.

There are many ways of adjusting this balance. First, it may be adjusted by varying the size of the reactor. The ratio of surface to volume of a given solid shape decreases as the solid grows larger in size. The number of neutrons escaping from the reactor is proportional to its surface area and the number of neutrons produced is proportional to the number of fuel nuclei, and hence the reactor volume. Therefore, the number of neutrons escaping in relation to those produced can be diminished by increasing the size of the reactor. In fact, for any given shape and material of a reactor, there is a minimum or *critical* size below which the percentage of escaping neutrons is excessive, below which the reactor will not sustain a chain reaction.

The second means of achieving the required balance is by decreasing the number of neutrons absorbed parasitically in structural and other non-fuel materials. This is accomplished by using materials which have a small tendency to absorb neutrons.

A third method depends on the fact that neutrons are much more likely to be captured by fissionable materials if the neutron energies are low. This suggests increasing the number of fissions and the number of neutrons produced by slowing down or "*moderating*" neutrons from their velocity at fission to the velocity of atoms at reactor temperature (termed "*thermal*" velocity). The slowing down of neutrons is accomplished by allowing them to collide with nuclei of certain selected elements termed "*moderators*." In this process neutrons lose some of their kinetic energy to the nuclei involved in the collision. Moderating materials will be discussed in greater detail later in this chapter.

When the self-sustaining chain reaction is established, the reactor becomes a source of heat energy. Since the fission process is to all intents and purposes independent of reactor temperature, it is possible to achieve any desired temperature level in the reactor.

20-5. Components of a Reactor for Ship Propulsion. There are several possible types of nuclear reactors based on the principles described above. Before discussing the components of a representative reactor for ship propulsion, it will be useful to review briefly those principles. It has been shown that:

(1) The absorption of neutrons by the nuclei of certain very heavy atoms causes fission and results in the liberation of large amounts of energy.
(2) The major portion of this enegy appears as kinetic energy of the fission fragments and becomes available as heat.
(3) A self-sustaining chain reaction of fissions can be maintained to provide a continuous source of heat energy at any desired temperature.

With these fundamentals in mind, the components of a reactor which can be used in developing useful mechanical power from fission may be listed as follows:

(1) Fissionable material, or fuel
(2) Moderator
(3) Structural material
(4) Reflector
(5) Coolant
(6) Biological shield
(7) Control mechanisms and control instrumentation

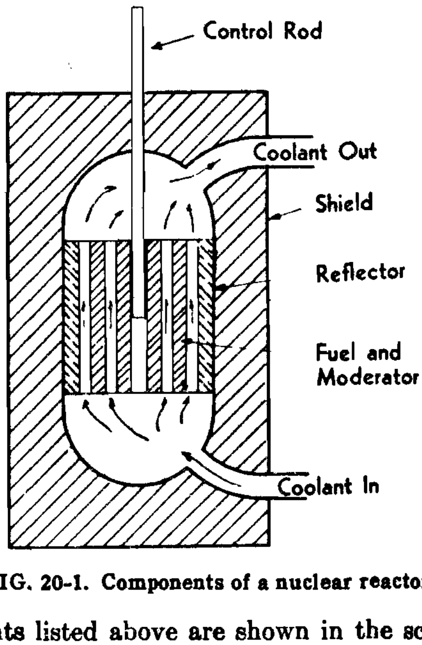


FIG. 20-1. Components of a nuclear reactor.

The components listed above are shown in the schematic diagram of a nuclear reactor in Fig. 20-1. In addition to the reactor itself, systems and components must be provided to convert the heat energy produced to mechanical energy.

20-6. Fissionable Material. The types of material which can be employed as fissionable fuels have been listed previously. Whatever the fuel used, enough must be provided so that the chain reaction can be established and maintained whenever power is needed. Also, careful consideration must be given to the manner in which the fuel is distributed in the reactor.

Heat production at any point within the reactor core depends on the number of fissions taking place at that point. The fuel must be so shaped and located that the number of fissions taking place at any point, and hence the heat production, will be as uniform as possible throughout the reactor. If a reasonably uniform distribution of heat production is not achieved, there will be localized hot spots, perhaps exceeding the melting point of the materials used. In other sections, the coolant passing through will not be heated sufficiently. That is to say, the coolant will not be used efficiently.

In reactors of the type represented in Fig. 20-1, the fissionable material is placed in assemblies called *fuel elements.* These fuel elements must be very carefully designed, taking into account considerations of heat transfer, corrosion, structural strength, physics, and metallurgy. For example, since the major portion of the energy of fission will appear first as heat within the fuel elements, the distance from the point at which a fission occurs to nearest coolant stream cannot be too great. Otherwise, the temperature at the centers of the fuel elements could become prohibitively high and cause thermal warping or cracking of the elements. Also, the elements must have sufficient surface area to transfer the heat generated to the coolant without too large a temperature difference between coolant and fuel, that is, the fuel elements must have a favorable ratio of surface to volume.

Since it is highly important to minimize the distance between the location of a fission and the coolant stream, it might be assumed that the fissionable material should be placed at the surface of the fuel element. However, highly radioactive fission products would then be circulated with the coolant. To prevent contamination of the coolant by fission products, a *protective coating* or *cladding* must separate the fuel from the coolant stream. Five major conditions must be satified by this fuel element cladding:

(1) It must not be damaged by the various types of radiation to which it is subjected as result of fission.

(2) It must have a high thermal conductivity so that the heat produced within the fuel element can be transferred to the coolant without the temperature becoming too high within the fuel element.

(3) It must not have a strong tendency to absorb neutrons.
(4) It must have adequate corrosion resistance.
(5) It must have adequate mechanical properties, such as strength and ductility, for the range of temperatures and pressures through which it may be required to operate.

20-7. Moderator. A moderator is a material which is used to slow neutrons down from the high velocities, and hence high energies, which they have on being released in the fission process. As previously stated (Art. 20-4), a neutron at thermal energy corresponding to the temperature of the reactor has a much greater chance of causing fission than the neutrons released in fission, which have very high energies. For example, the probability of neutrons producing thermal energy through fission of U^{235} is increased three hundred-fold as a result of slowing down from its energy when released in fission. Neutrons are slowed down most effectively in scattering collisions with nuclei of the light elements, such as hydrogen, deuterium, carbon (graphite), and beryllium. The heavier elements would cause the neutrons to rebound from collisions with little loss in neutron energy.

In addition to being able to slow neutrons effectively, a moderator should not capture neutrons. A tendency to absorb neutrons rules out light elements such as lithium and beryllium. The properties of a moderating material depend not only on the characteristics of the individual nucleus, but on the number of such moderating nuclei in a given volume. For example, the helium nucleus is light and does not absorb neutrons, that is, it has properties required of good moderators. In order to provide a sufficient number of nuclei, however, the helium gas would have to be so highly pressurized as to make it considerably less attractive than it would be otherwise.

The moderator can be present in a reactor as a chemical compound. For example, if hydrogen, it can be present as water which may also serve as the reactor coolant.

20-8. Reflector. To keep the critical size of the reactor, and hence the amount of fissionable material, as small as possible, it is important to conserve neutrons. This can be accomplished in part by surrounding the reactor core with a material which reflects escaping neutrons back into the core. This material is called a reflector. It should have good neutron scattering properties and preferably a small tendency to absorb neutrons. It is often a moderating material and sometimes the same material is used for both moderator and reflector. With a properly designed reflector the amount of fissionable material in a reactor core can be made smaller than that in a bare or unreflected reactor.

20-9. Coolants. The heat energy produced in the reactor core must be transferred by means of a fluid coolant to the equipment which will convert this heat energy to useful power. The choice of coolants is limited, especially when requirements peculiar to nuclear reactors are added to the usual stringent engineering requirements. A partial list of the requirements of a suitable coolant follows:

(1) Throughout the range of operating temperatures the coolant must not solidify nor undergo changes of state which drastically change heat transfer and nuclear characteristics.
(2) The coolant must be compatible with the coolant channel walls and other materials with which the coolant comes in contact. As in any heat transfer system, corrosion, erosion, and scale formation may have harmful effects on the rate of heat transfer through the coolant channel walls. Of equal importance is the fact that pitting of the channel walls may permit the highly radioactive fission products to enter the coolant stream.
(3) The heat capacity and heat transfer coefficient must be high to reduce the volume of coolant required. The volume of coolant required affects the size of the reactor. Since the reactor must be surrounded by heavy shielding material, it is important that it be kept as small as possible so as to reduce overall weight. This is particularly important for naval vessels.
(4) The tendency to absorb neutrons should be small for reasons of neutron economy already discussed.
(5) The induced radioactivity should be as low as possible. This depends on the tendency of the coolant to absorb neutrons and on the radioactive nature of the newly formed element resulting from such absorption. Coolants which interact with neutrons to form isotopes emitting high energy gamma rays are undesirable in this respect.

In selecting a coolant, it is necessary to weigh the relative importance of the favorable and unfavorable characteristics of various coolants and select the one best suited to the particular application. The characteristics of a few possible coolants are described below.

Water. A considerable amount of information is available on this coolant; its properties are reasonably well understood, and it has been widely used in power generation. When properly conditioned, it is compatible with a large number of structural materials, and it is a good moderator for thermal reactors. In addition, it has a high specific heat and a good heat transfer coefficient. On the other hand, it must be pressurized to prevent its vaporization at the high operating temperatures necessary in a power reactor. Vaporization of the coolant during its passage through the reactor is undesirable.

Heavy Water. The properties of heavy water are nearly identical with those of light water with the important exception that it absorbs neutrons less readily than light water. It is not as effective as light water in slowing down neutrons, but is attractive for applications where low neutron losses are of primary importance.

An important disadvantage of heavy water is its high cost.

Liquid Metals. Low melting point metals such as sodium, potassium, and sodium-potassium alloys may be considered for use as reactor coolants. A principal advantage of liquid metals is the ability to obtain high temperatures at low pressures without vaporization. Because of their high heat capacities and good heat transfer properties, heat exchange surfaces and quantities of coolant can be kept small. The technology associated with their use is under development but does not have the benefit of the extensive industrial background information available on water.

Organic Liquids. Like water, organic coolants contain hydrogen and therefore offer the possibility of combining the functions of moderator and coolant in one material. Some are good heat transfer fluids and have attractive thermodynamic properties, that is, have high temperatures associated with low pressures in the liquid state. However, their stability at high temperature and under irradiation is not as satisfactory as that of inorganic coolants.

Gases. Heat transfer to gases is notably poor in part because of their low density. Hence, gaseous coolants require large heat transfer surfaces within the reactor, high pressurization, and large pumping capacity. Because of the large film drop associated with heat transfer to gases, fuel elements must operate at high surface temperatures. Among the possible gaseous coolants, helium has an advantage in that it does not absorb neutrons and does not itself become radioactive. Air has disadvantages because of oxidation problems and radioactivity due to argon impurities. Radioactive impurities in any of the gases would require the shielding of components external to the reactor.

20-10. Shielding. The intensity of gamma and neutron radiation coming from the reactor core is far greater than the human body can tolerate. Hence it is necessary to surround the reactor with enough shielding material to reduce the radiation to levels which are not harmful to personnel. In general, material of high density is required to attenuate the gamma radiation. Low energy neutrons are easily captured by materials such as boron. High energy neutrons must be slowed down before they can be readily captured.

The gamma rays and neutrons from the reactor give up energy from the shield in the form of heat. Therefore, some method of cooling the shield must be provided.

Many materials which are used for reactor coolants become radioactive when bombarded by neutrons in the core. Where this is the case, it is also necessary to provide shielding for the coolant piping, pumps, and heat exchangers.

To obtain an impression of the magnitude of the shielding problem, consider that the gamma radiation intensity alone at the inside face of the shield of an existing research reactor is approximately one million times the currently accepted level for one day exposure of human beings. To reduce this radiation to acceptable levels this research reactor has a shield of concrete approximately seven feet thick.

20-11. Control. The nuclear reactor must have some control device by which the nuclear chain reaction can be started, maintained at the selected operating power level, and shut down when required.

Several methods are available for controlling the reactor; the amount of fissionable materials in the reactor core may be varied; the number of neutrons absorbed without causing fission may be regulated; or the number of neutrons leaking from the core may be adjusted. For example, mechanical methods can be used to insert neutron absorbing materials such as boron or cadmium in the core.

In addition to providing for normal operating control and shutdown of the reactor, the control system must provide for rapid, automatic shutdown in response to signals and dangerous conditions in the reactor. The devices for causing this rapid shutdown may use the same methods as are used for operational control. Controls and safety devices are usually designed to respond to measurements of neutron *"population"* which indicate reactor power level or rate of change of power level. These changes in neutron "population" will occur before other dependent effects occur, such as changes of reactor temperature and pressure.

20-12. Instrumentation. One method of measuring neutron "population" involves an instrument termed an ionization chamber, which produces a small current in a gas filled tube when exposed to neutron radiation. The sensitivity of such a chamber is due to a coating placed on the interior walls of the chamber. By careful design of the chamber, this ionization current can be made proportional to the power level of the reactor. By suitable amplification the ionization current can be used to actuate the control mechanism of the reactor so that a desired power level is automatically maintained. It can also be linked to a mechanism which will cause the reactor to shut down in the event the reactor power exceeds a certain maximum.

20-13. Conversion of Heat Energy to Useful Power. The foregoing discussion concerned the reactor itself and considerations affecting its design; it remains to discuss the methods by which the heat energy released in the reactor can be converted to useful mechanical energy. Two basic types of systems are described: one for liquid metal and water coolants, and the other for gas coolants.

Liquid Metal or Water Coolant. Figure 20-2 is a schematic diagram showing the basic components of a nuclear power plant with a liquid metal or water coolant. In passing through the reactor the coolant itself will become radioactive in varying degree depending on the nature of the coolant and the amount and type of contained impurities. It is therefore necessary to confine this radioactive coolant in a system with a high degree of leak-tightness.

Considering first water-cooled reactors, it is not practicable to attempt to design the entire plant to the standard of leak-tightness required for systems containing radioactive coolant. Leakage of radioactive coolant could not be tolerated, for example, at turbine gland

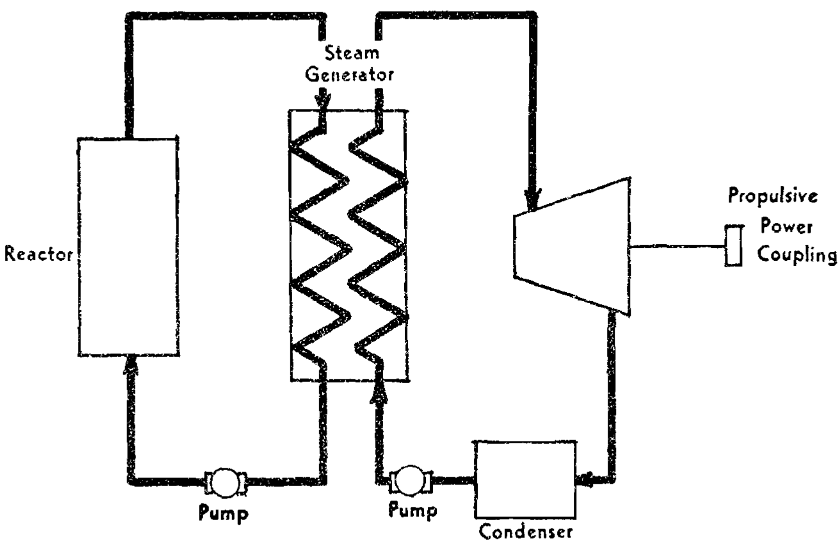


FIG. 20-2. A steam generator system utilizing nuclear energy.

seals and at pump shaft and valve stem packing. Moreover, it would be prohibitive from a maintenance point of view to have the entire plant contaminated with radioactive impurities which would be found in the steam. For these reasons, the reactor coolant is isolated in a primary loop and its heat energy transferred to water in a heat exchanger called a steam generator. By so isolating the reactor coolant, the shielding weight can be reduced to that required for the reactor and systems containing radioactive coolant.

For reactors with liquid metal coolants, a steam generator is needed for the reasons cited above. It is also needed because the properties of those liquid metals, which are attractive for power reactors, are not suitable for their use in power conversion machinery such as turbines.

The propulsion plant between the steam generator and propeller shaft can be quite conventional. Details of its design will vary depending upon the particular ship application involved.

Gas Coolant. Figure 20-3 shows one possible type of system which utilizes a gas cooled reactor. The gas is compressed, heated in the reactor, and expanded in a turbine. If the efficiencies of the components are high enough, the turbine will produce enough power for propulsion purposes over and above that required to drive the compressor. As previously mentioned, because of their low density, gaseous coolants would have to be circulated at pressures of the order of 1000 psi. Gas temperatures would have to be high to obtain adequate cycle efficiencies and fuel element temperatures would be correspondingly

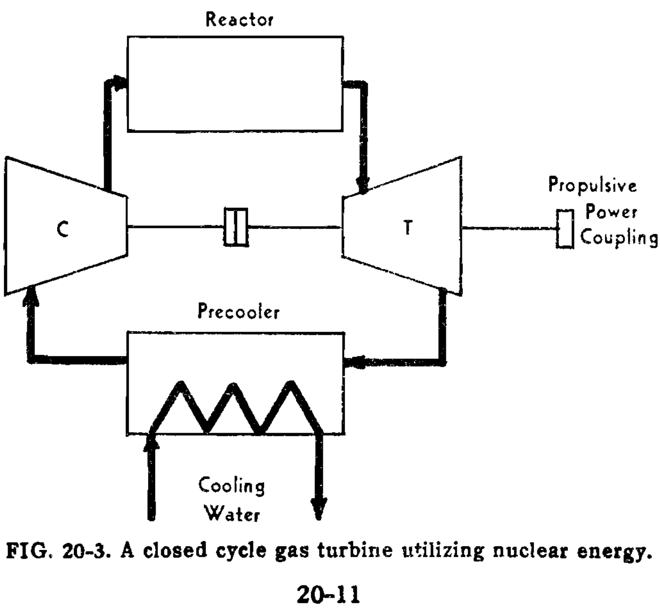


FIG. 20-3. A closed cycle gas turbine utilizing nuclear energy.

high. Becuase of the large film drop associated with heat transfer in gases, the difference between fuel element surface temperature and coolant temperature would tend to be larger than for other coolants.

20-14. Plant Component Development. Naval nuclear power plant components must meet the usual stringent standards established for all naval propulsion plants with regard to reliability, shock resistance, repairability, and simplicity. Moreover, for certain components, additional requirements must be met which preclude the use of equipment of conventional design. For example, it is important that equipment and systems containing radioactive coolant be designed for no leakage. Conventional pumps, valves, and flanges do not meet nuclear power plant standards in this respect. As a result, sizeable development efforts must be undertaken to provide equipment which does meet these standards.

Pumps which circulate coolant through the reactor and heat exchanger provide an interesting and important example of one method by which such requirements are met. In conventional practice, a centrifugal pump for high pressure use is sealed by mechanical packing around the shaft joining the impeller and the driving motor or turbine; a small leakage is accepted.

For pumping water coolant at high pressure in nuclear power plants, a new type of leak proof centrifugal pump has been developed as shown in Fig. 20-4. Its important design feature is a higher pressure barrier or can in the gap between the pump motor rotor and stator. That is to say, the rotor is inside the high pressure system, the stator is outside, and the two are electromagnetically coupled as in any induction motor.

Several difficult problems complicated the development of this pump. One was the problem of bearings. Conventional centrifugal pumps have oil lubricated bearings. Since the rotating parts of the "canned" pump are inside the coolant system, oil lubrication of bearings would result in reactor and system contamination. Bearings were therefore developed which are lubricated by the high temperature water coolant.

20-15. Materials Development. The characteristics of nuclear power plants have created new requirements which must be met by standard engineering materials. In certain cases, reactor requirements have resulted in the development of entirely new materials. An important example of the latter type of development is that associated with the production, metallurgy, and engineering use of zirconium.

Prior to 1948, very little work had been done on zirconium. Its development was undertaken because of the need for a reactor material with a small tendency to absorb neutrons and good corrosion resistance. Zirconium metal is now being produced in large quantity with both

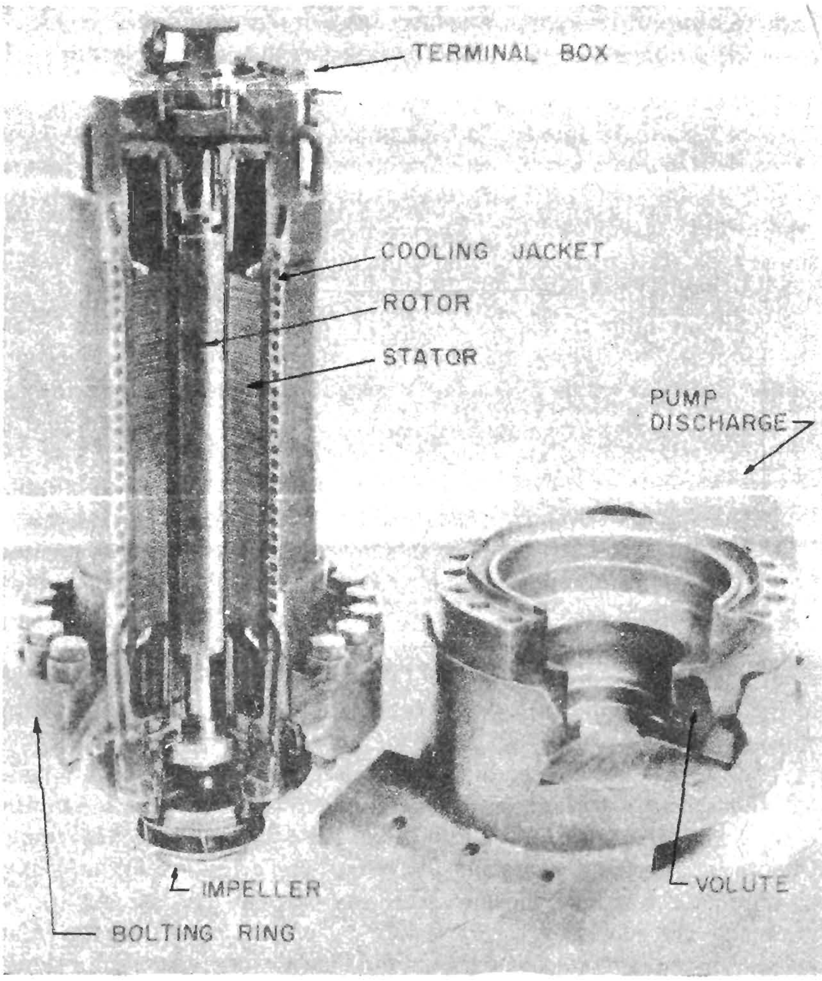


FIG. 20-4. High pressure centrifugal type pump designed for pumping water coolant in nuclear power plants.

these qualities, but not by the production methods hitherto used. New methods had to be developed to remove an impurity with a strong tendency to absorb neutrons. This impurity, hafnium, is found in all zirconium ores in small amounts (0.5 to 3 per cent) and in some ores up to 20 per cent.

Fabrication techniques were developed and zirconium alloys were investigated to improve both physical properties and corrosion resistance. As a result of this work, zirconium and its alloys are now among the leading cladding and structural materials for reactors.

20-16. Reprocessing. The nuclear reactor will not continue to operate for an indefinite period because of the following factors:

(1) There is a finite expenditure of fuel. At some point the quantity of fissionable atoms available will fall below the point where the reactor will operate.

(2) Some of the fission products formed during operation absorb neutrons to an appreciable degree. The quantity of these products increases roughly with the time the reactor has been operating. Neutron losses due to absorption in these products, together with other neutron losses, may eventually limit or prevent reactor operation.

(3) Radiation damage to materials within the reactor may, after a long period of operation, make fuel removal necessary.

The recovery of unused portions of fuel and useful by-products will be carried out by chemical reprocessing of the fuel element. Because of the intense radioactivity present in the material to be processed, reprocessing must be done by remote control equipment behind shielded walls.

20-17. Submarine Thermal Reactor. The first nuclear power plant for ship propulsion is that of the submarine, *USS Nautilus* (SSN 571). This power plant is named Submarine Thermal Reactor (STR). It is so named because the reactor is of a type in which the neutrons causing fission are, on the average, at thermal energy. The reactor coolant is water at high temperature and high pressure.

The design, development, and construction of STR was organized in two phases: the first, a land based prototype called Mark I; and the second, the *Nautilus* power plant, which is called Mark II. The entire project has been carried out as a cooperative effort of the U. S. Navy and Atomic Energy Commission.

Design and development of Mark I began in 1948. The work was carried forward by the Westinghouse Electric Corporation with the assistance of the Argonne National Laboratory. When construction of plant components was complete, the entire plant was assembled by Electric Boat Division, General Dynamics Corporation, in a section of a submarine hull at the National Reactor Testing Station near Arco, Idaho. This land-based submarine hull is shown in Fig. 20-5, inside the main building at the STR test site. The hull passes through a large "sea tank" so that it is completely submerged in water in way of the reactor compartment. The Mark I power plant with its associated propulsion equipment has been assembled in this hull in much the same

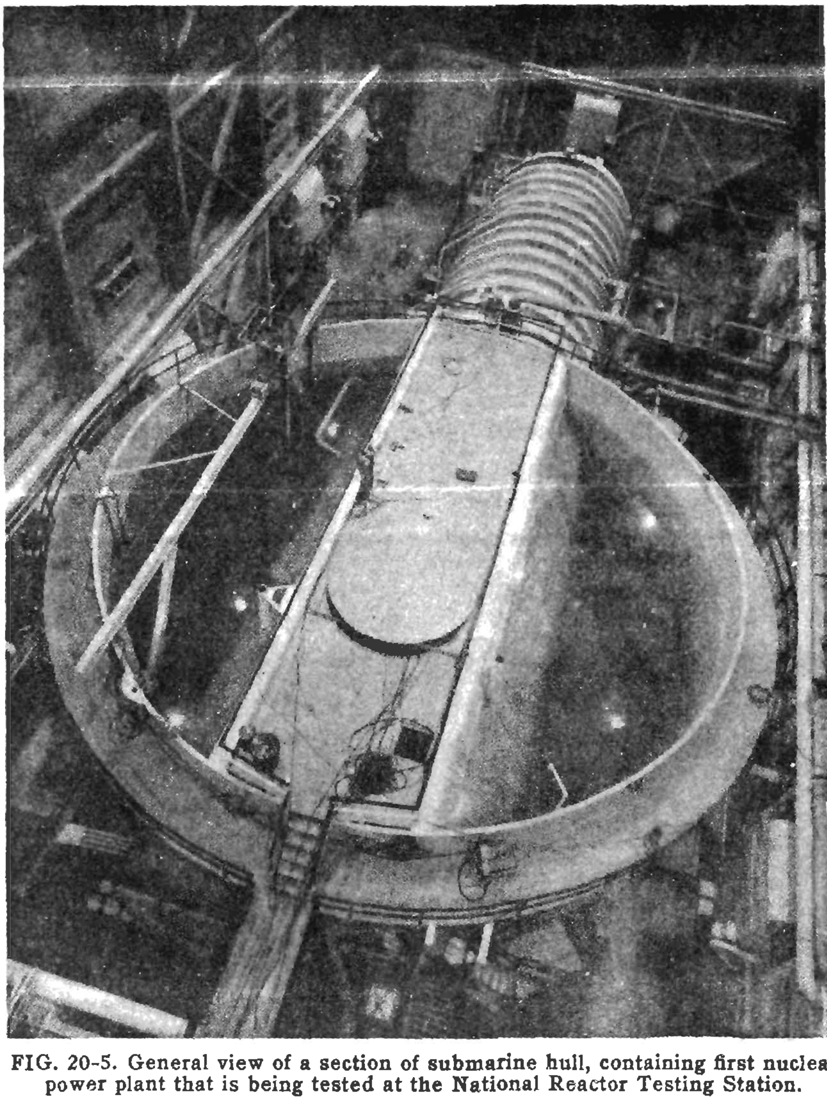


FIG. 20-5. General view of a section of submarine hull, containing first nuclear power plant that is being tested at the National Reactor Testing Station.

way that the Mark II plant is installed in the *Nautilus*. Outside the main building is an artificially-constructed cooling pond, which is used to provide the large amounts of cooling water required to operate a power plant.

Mark I began operating at full power in May 1953, and thus became the first nuclear power plant in history to produce significant quantities of power. It has been used to determine the operating characteristics

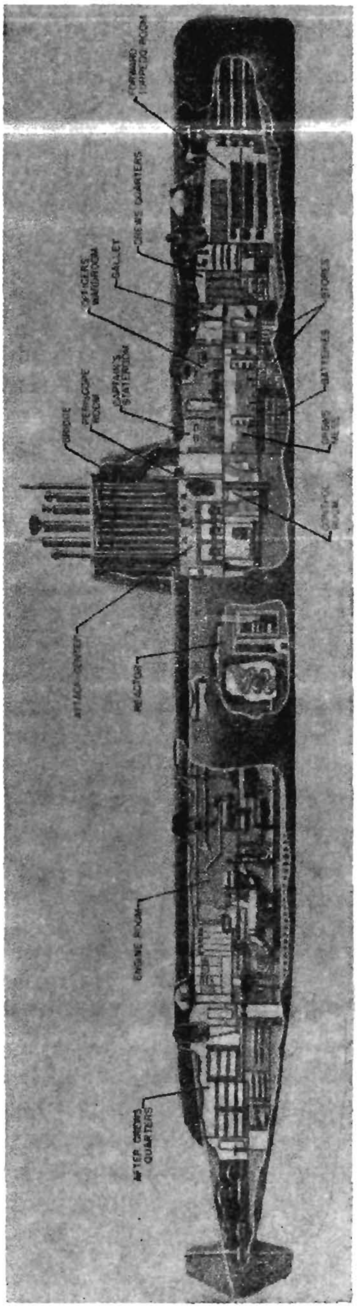


FIG. 20-6. Artist's conception of the atomic powered submarine, the *USS Nautilus*, which was launched at Groton, Connecticut, on January 21, 1954.

of this type of power plant and also to train the *Nautilus* crew. Results obtained from its operation have made it possible to improve the *Nautilus* power plant.

The keel plate of the *Nautilus* was laid by Electric Boat Division at Groton, Connecticut, on June 14, 1952, and the submarine was launched on January 21, 1954. The general features of its hull configuration and arrangement of spaces are shown in a cutaway sketch in Fig. 20-6. Its design speed is in excess of twenty knots and its endurance, exceptionally long in both the surface and submerged condition.

20-18. Submarine Intermediate Reactor. The second nuclear power plant for ship propulsion is the Submarine Intermediate Reactor (SIR), the power plant of the *USS Sea Wolf* (SSN 575). SIR is basically different from STR in that the reactor coolant is liquid sodium. Alsc, the average energy of the neutrons in SIR is in a range intermediate between thermal energy and the fast energies which neutrons have when released in fission.

Like STR, the SIR project is being accomplished in two stages: the first, a land-based prototype called Mark A; and the second, the SEA WOLF power plant, which is termed Mark B. Design and construction of both plants is being carried out by the General Electric Cmpany. Like STR, SIR is a cooperative program of the Navy and Atomic Energy Commission.

The land based prototype is located at West Milton, New York, not far from Schenectady. The *Sea Wolf* is being constructed by Electric Boat Division at Groton, Connecticut, her keel plate having been laid on September 21, 1953.

20-19. Conclusion. Progress made thus far in the field of nuclear ship propulsion is encouraging. Many formidable technical problems have been solved. Improvements of a fundamental nature in the technology of power reactors have already been made, and other potential improvements are continually being disclosed. There is a strong incentive to take advantage of these improvements in order to provide naval vessels with increasingly better tactical performance. Recognizing this, the Navy and the Atomic Energy Commission are continuing to advance the design and development of new and improved types of nuclear power plants for ship propulsion.

References

20-1. "Sourcebook on Atomic Energy," S. Glasstone, D. Van Nostrand, Inc., New York, 1950.

20-2. "The Elements of Nuclear Reactor Theory," S. Glasstone, M. C. Edlund, D. Van Nostrand, New York, 1952.

APPENDIX A

EXAMPLE PROBLEMS

A-1. Ideal Simple Open Cycle (Fig. 16-7).

a. *Given:*

$$\begin{aligned} W_a &= 1 \text{ lb air} \\ T_1 &= 60^\circ \text{ F} = 520^\circ \text{ F abs} \\ T_3 &= 1200^\circ \text{ F} = 1660^\circ \text{ F abs} \\ r_p &= p_2/p_1 = p_3/p_4 = 4 \\ \eta_C &= 1.0 \\ \eta_{Tu} &= 1.0 \end{aligned}$$

b. *Find:* iwk_C, q_S, iwk_{Tu}, and η_t

c. *Solution:*

1. Enter Air Charts with $T_1 = 520^\circ$ F abs and fi

$$p_{r_1} = 2.507, \qquad h_1 = 28.8 \text{ Btu/lb}$$

2. Then p_{r_2} is

$$p_{r_2} = p_{r_1}(p_2/p_1) = (2.507)(4) = 10.028$$

3. Enter Air Charts with p_{r_2} and find

$$T_2 = 771.2^\circ \text{ F abs}, \qquad h_2 = 89.4 \text{ Btu/lb}$$

4. Ideal compressor work per lb air, iwk_C, from equation (16-2) is

$$iwk_C = h_2 - h_1 = 89.4 - 28.8 = \mathbf{60.6} \text{ Btu/lb}$$

5. Enter Air Charts with $T_3 = 1660^\circ$ F abs and find

$$p_{r_3} = 171.6, \qquad h_3 = 316.6 \text{ Btu/lb}$$

6. Heat supplied to the air in the combustion chamber, q_S, from equation (16-3) is

$$q_S = h_3 - h_2 = 316.6 - 89.4 = \mathbf{227.2} \text{ Btu/lb}$$

7. Find p_{r_4} from

$$p_{r_4} = \frac{p_{r_3}}{p_3/p_4} = \frac{171.6}{4} = 42.9$$

8. Enter Air Charts with p_{r_4} and find

$$T_4 = 1153.4^\circ \text{ F abs}, \qquad h_4 = 184.1 \text{ Btu/lb}$$

9. Ideal turbine work per lb air, iwk_{Tu}, from equation (16-4) is

$$iwk_{Tu} = h_3 - h_4 = 316.6 - 184.1 = \mathbf{132.5} \text{ Btu/lb}$$

10. Thermal efficiency, η_t, of the ideal simple open cycle from equation (16-7) is

$$\begin{aligned} \eta_t &= \frac{iwk_{Tu} - iwk_C}{q_S} = \frac{132.5 - 60.6}{227.2} \\ &= \frac{71.9}{227.2} = \mathbf{0.316} \end{aligned}$$

11. The thermal efficiency also may be calculated from equation (16-8), *if* the mean ratio of specific heats is known. In this particular problem the mean ratio of specific heats, k, is 1.375. Therefore,

$$\eta_t = 1 - \left(\frac{1}{r_p}\right)^{(k-1)/k} = 1 - 0.685 = 0.315$$

A-2. Actual Simple Open Cycle (Fig. 16-8).

a. *Given:* Same conditions as problem A-1 except

$$\eta_C = 0.80 \quad \text{and} \quad \eta_{T_u} = 0.80$$

From problem A-1:

$T_1 = 520°$ F abs, $p_{r_1} = 2.507$, $h_1 = 28.8$ Btu/lb
$T_2 = 771.2°$ F abs, $p_{r_2} = 10.28$, $h_2 = 89.4$ Btu/lb
$T_3 = 1660°$ F abs, $p_{r_3} = 171.6$, $h_3 = 316.6$ Btu/lb
$T_4 = 1153.4°$ F abs, $p_{r_4} = 42.9$, $h_4 = 184.1$ Btu/lb
$rp = p_2/p_1 = 4$

b. *Find:* wk_C, wk_{T_u}, q_S, and η_t.

Solution:

1. Actual compressor work, wk_C, from equation (16-9)

$$wk_C = iwk_C/\eta_C = \frac{(h_2 - h_1)}{\eta_C} = \frac{60.6}{0.8} = 75.8 \text{ Btu/lb}$$

2. Enthalpy at state 2′, exit from actual compressor, is

$$h_{2'} = h_1 + wk_C = 28.8 + 75.8 = 104.6 \text{ Btu/lb}$$

3. The heat supplied in the combustion chamber, q_S, from equation (16-11)

$$q_S = h_3 - h_{2'} = 316.6 - 104.6 = 212 \text{ Btu/lb}$$

4. Actual turbine work, wk_{T_u}, from equation (16-12)

$$wk_{T_u} = iwk_{T_u}\eta_{T_u} = 132.5(0.8) = 106.0 \text{ Btu/lb}$$

5. Enthalpy at state 4′, exit from actual turbine, is

$$h_{4'} = h_3 - wk_{T_u} = 316.6 - 106.0 = 210.6 \text{ Btu/lb}$$

6. The thermal efficiency, η_t from equation (16-14)

$$\eta_t = \frac{wk_{T_u} - wk_C}{q_S} = \frac{106 - 75.8}{212} = 0.142$$

A-3. Ideal Open Cycle with Regeneration (Fig. 16-13).

a. *Given:* Same conditions as problem A-1 plus $\eta_s = 1.0$
From problem A-1:

$T_1 = 520°$ F abs, $p_{r_1} = 2.507$, $h_1 = 28.8$ Btu/lb
$T_2 = 771.2°$ F abs, $p_{r_2} = 10.028$, $h_2 = 89.4$ Btu/lb
$T_3 = 1660°$ F abs, $p_{r_3} = 171.6$, $h_3 = 316.6$ Btu/lb
$T_4 = 1153.4°$ F abs, $p_{r_4} = 42.9$, $h_4 = 184.1$ Btu/lb
$r_p = p_2/p_1 = 4$
$\eta_C = 1.0$
$\eta_{T_u} = 1.0$

b. *Find:* q_{ix}, q_S, η_t.

c. *Solution:*

1. Since $T_4 = T_5$ and $h_4 = h_5$, the heat transferred in the ideal regenerator becomes

$$q_{ix} = h_5 - h_2 = h_4 - h_2 = 184.1 - 89.4 = \mathbf{94.7} \text{ Btu/lb}$$

2. The heat added in the combustion chamber is

$$q_S = h_3 - h_5 = h_3 - h_4 = 316.6 - 184.1 = \mathbf{132.5} \text{ Btu/lb}$$

3. The thermal efficiency from equation (16-16) with $h_4 = h_5$ is

$$\eta_t = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_4} = 1 - \frac{h_2 - h_1}{h_3 - h_4} = 1 - \frac{60.6}{132.5} = \mathbf{0.542}$$

4. The thermal efficiency also may be calculated as follows using a mean $k = 1.375$:

$$\eta_t = 1 - \left(\frac{T_1}{T_3}\right) r^{(k-1)/k} = 1 - \left(\frac{520}{1660}\right)(1.46) = 0.544.$$

A-4. Actual Open Cycle with Regeneration (Fig. 16-13 and 16-14).

a. *Given:* Same conditions as problem A-2 plus $\eta_x = 0.80$.
From problem A-2:

$$\begin{aligned} h_1 &= 28.8 \text{ Btu/lb} \\ h_2 &= 89.4 \text{ Btu/lb} \\ h_{2'} &= 104.7 \text{ Btu/lb} \\ h_3 &= 316.6 \text{ Btu/lb} \\ h_{4'} &= 210.6 \text{ Btu/lb} \\ wk_C &= (h_{2'} - h_1) = 75.8 \text{ Btu/lb} \\ wk_{T_a} &= (h_3 - h_{4'}) = 106 \text{ Btu/lb} \end{aligned}$$

b. *Find:* q_x, q_S, and η_t.

c. *Solution:*

1. From equation (16-18), heat transferred in the regenerator is

$$q_x = (h_{4'} - h_{2'})\eta_x = (210.6 - 104.7)(0.80) = \mathbf{84.7} \text{ Btu/lb}$$

2. The enthalpy at state 5′, exit from cold side of regenerator and entrance to combustion chamber, becomes

$$h_{5'} = h_{2'} + q_x = 104.7 + 84.7 = 189.4 \text{ Btu/lb}$$

3. The heat supplied to the air in the combustion chamber, q_S, from equation (16-19) is

$$q_S = h_3 - h_{5'} = 316.6 - 189.4 = \mathbf{127.2} \text{ Btu/lb}$$

4. The thermal efficiency, η_t, of an actual open cycle with regeneration, equation (16-20), becomes

$$\eta_t = \frac{(h_3 - h_{4'}) - (h_{2'} - h_1)}{h_3 - h_{5'}} = \frac{106 - 75.8}{127.2} = \mathbf{0.236}$$

A-5. Regenerator Efficiency (Fig. 16-14).

a. *Given:*

$$T_{2'} = 834° \text{ F abs}, \quad h_{2'} = 104.7 \text{ Btu/lb}$$
$$T_{5'} = 1174° \text{ F abs}, \quad h_{b'} = 189.4 \text{ Btu/lb}$$
$$T_{4'} = 1256° \text{ F abs}, \quad h_{4'} = 210.3 \text{ Btu/lb}$$

b. *Find:* η_x.

c. *Solution:* From equation (16–17)

$$\eta_x = \frac{h_{5'} - h_{2'}}{h_{4'} - h_{2'}} = \frac{189.4 - 104.7}{210.3 - 104.7} = 0.803$$

A-6. Actual Open Cycle with Regeneration and Intercooling (Fig. 16-16)

a. *Given:*

$$T_1 = 520° \text{ F abs}, \ p_{r_1} = 2.507, \ h_1 = 28.8 \text{ Btu/lb}$$
$$r_p = r_{p_{21}} = p_2/p_1 = 4$$
$$T_3 = 1660° \text{ F abs}, \ p_{r_3} = 171.6, \ h_3 = 316.6 \text{ Btu/lb}$$
$$h_{4'} = 210.6 \text{ Btu/lb}$$
$$wk_{T_u} = h_3 - h_{4'} = 106 \text{ Btu/lb}$$
$$\eta_C = 0.80$$
$$\eta_{T_u} = 0.80$$
$$\eta_x = 0.80$$
$$\eta_{\text{int}} = 0.80$$

Two stage compressor with an intercooler

b. *Find:* wk_C, q_S, and η_t.

c. *Solution:*

1. From equation (16-22a), $r_{p(a1)}$ is

$$r_{p(a1)} = \sqrt{r_p} = 2$$

2. $p_{ra} = p_{r_1}(r_{p(a1)}) = 2.507(2) = 5.014$
3. Entering Air Charts with $p_{ra} = 5.014$, h_a is found to be 56.2 Btu/lb. Therefore, the ideal compressor work of the first stage is

$$iwk_{C(a1)} = h_a - h_1 = 56.2 - 28.8 = 27.4 \text{ Btu/lb}$$

4. The actual compressor work from the first stage is

$$wk_{C(a'1)} = iwk_C/\eta_C = \frac{27.4}{0.80} = 34.3 \text{ Btu/lb}$$

5. The enthalpy at the exit from the first stage compressor, state a', is

$$h_{a'} = h_1 + wk_{C(a1)} = 28.3 + 34.4 = 63.1 \text{ Btu/lb}$$

6. The enthalpy at the exit to the intercooler, state b, from equation (16-24) is

$$h_b = h_{a'} - (h_{a'} - h_1)\eta_{\text{int}} = 63.1 - (34.4)(0.80) = 35.6 \text{ Btu/lb}$$

7. Entering Air Charts with $h_b = 35.6$ Btu/lb, it is found that $p_{rb} = 3.02$, $T_b = 548.3$ F abs
8. Since $r_{p(b2)} = \sqrt{r_p}$, then

$$p_r = p_{rb}(\sqrt{r_p}) = 3.02(2) = 6.04$$

9. Entering Air Charts with p_{r_2}, h_2 is found to be 64.4 Btu/lb. Therefore, the ideal compressor work for the second stage compression is

$$iwk_{C(b2)} = h_2 - h_b = 64.4 - 35.6 = 28.8$$

10. The actual compressor work for the second stage becomes

$$wk_{C(b2')} = iwk_{C(b2)}/\eta_C = \frac{28.8}{0.80} = 36.0 \text{ Btu/lb}$$

11. The compressor work is the work of the first stage plus the work of the second stage

$$wk_C = wk_{C(a'1)} + wk_{C(b2')} = 34.3 + 36.0 = 70.3 \text{ Btu/lb}$$

12. The enthalpy at the exit from the second stage compressor, state 2′, is

$$h_{2'} = h_b + wk_{C(b2')} = 35.6 + 36.0 = 71.6 \text{ Btu/lb}$$

13. The enthalpy at the exit from the regenerator on the cold side, state 5,′ from equation (16-17) is

$$\begin{aligned} h_{5'} &= h_{2'} + (h_{4'} - h_{2'})\eta_x = 71.6 + (210.6 - 71.6)(0.80) \\ &= 182.8 \text{ Btu/lb} \end{aligned}$$

14. The heat supplied, q_S, in the combustion chamber is

$$q_S = h_3 - h_{5'} = 316.6 - 182.8 = 133.8 \text{ Btu/lb}$$

15. The thermal efficiency, η_t, of air actual open cycle gas turbine with regeneration and intercooling from equation (16-25) is

$$\begin{aligned} \eta_t &= \frac{(h_3 - h_{4'}) - (h_{a'} - h_1) + (h_{2'} - h_b)}{h_3 - h_{5'}} \\ &= \frac{106 - 70.3}{133.8} = \frac{35.7}{133.8} = 0.267 \end{aligned}$$

A-7. Actual Open Cycle with Regeneration, Intercooling, and Reheating (Fig. 16-18).

a. *Given:* From problem A-6

$$\begin{aligned} h_1 &= 28.8 \text{ Btu/lb} \\ h_{a'} &= 63.1 \text{ Btu/lb} \\ h_b &= 35.6 \text{ Btu/lb} \\ h_{2'} &= 71.6 \\ wk_C &= (h_{a'} - h_1) + (h_{2'} - h_b) = 70.3 \text{ Btu/lb} \\ \eta_C &= 0.80 \\ \eta_{Tu} &= 0.80 \\ \eta_x &= 0.80 \\ \eta_{int} &= 0.80 \\ h_3 &= h_d = 316.6 \text{ Btu/lb} \\ p_{r_3} &= p_{rd} = 171.6 \\ r_p &= p_2/p_1 = 4 \\ r_{p(a1)} &= r_{p(ba)} = 2 \end{aligned}$$

Two stage turbine with reheater.

b. ***Find:*** wk_{Tu}, q_S, and η_t.

c. *Solution:*

1. Assume $r_{p(3c)} = r_{p(d4)} = 2$
2. $p_{rc} = \dfrac{p_{r3}}{r_{p(3c)}} = \dfrac{171.6}{2} = 85.8$
3. From Air Charts, $h_c = 244.4$ Btu/lb and $T_c = 1387.6°$ F abs.
4. Actual turbine work of the first stage is

$$wk_{Tu(3c')} = (h_3 - h_c)(\eta_{Tu}) = (316.6 - 244.4)(0.80) = 57.7 \text{ Btu/lb}$$

5. The enthalpy at the exit from the first stage turbine, state c', is

$$\begin{aligned} h_{c'} &= h_3 - wk_{Tu(3c)} = h_3 - (h_3 - h_c)\eta_{Tu} \\ &= 316.6 - 57.7 = 258.9 \text{ Btu/lb} \end{aligned}$$

6. Since $T_3 = T_d$, the heat supplied in the reheater, process c' to d, becomes

$$q_{S2} = h_d - h_{c'} = 316.6 - 258.9 = 57.7 \text{ Btu/lb}$$

7. $p_{r4} = \dfrac{p_{rd}}{r_{p(d4)}} = \dfrac{171.6}{2} = 85.8$
8. From Air Charts, $h_4 = 244.4$ Btu/lb and $T_4 = 1387.6°$ F abs.
9. Actual work of the second stage turbine is

$$\begin{aligned} wk_{Tu(d4')} &= (h_d - h_4)(\eta_{Tu}) = (316.6 - 244.4)(0.80) \\ &= 57.7 \text{ Btu/lb} \end{aligned}$$

10. Work of turbine is the work of the first stage plus the work of the second stage.

$$wk_{Tu} = wk_{Tu(3c')} + wk_{Tu(d4')} = 57.7 + 57.7 = 115.4 \text{ Btu/lb}$$

11. The enthalpy at the exit to the second stage turbine and the entrance to the regenerator on the hot side, state $h_{4'}$, is

$$h_{4'} = h_d - wk_{Tu(d4')} = 316.6 - 57.7 = 258.9 \text{ Btu/lb}$$

12. The enthalpy of the exit from the regenerator on the cold side, state $h_{5'}$, from equation (16-17) is

$$\begin{aligned} h_{5'} &= h_{2'} + (h_{4'} - h_{2'})\eta_u = 71.6 + (258.9 - 71.6)(0.80) \\ &= 221.4 \text{ Btu/lb} \end{aligned}$$

13. Heat supplied in first combustion chamber, process $5'$ to 3, is

$$q_{S1} = h_3 - h_{5'} = 316.6 - 221.4 = 95.2 \text{ Btu/lb}$$

14. The thermal efficiency of an actual open cycle gas turbine with regeneration, intercooling, and reheating from equation (16-30) becomes

$$\begin{aligned} \eta_t &= \frac{[(h_3 - h_{c'}) + (h_d - d_{4'})] - [(h_{a'} - h_1) + (h_{2'} - h_b)]}{(h_3 - h_{5'}) + (h_d - h_{c'})} \\ &= \frac{115.4 - 70.3}{95.2 + 57.7} = \frac{45.1}{152.9} = 0.295 \end{aligned}$$

A-8. Ideal Turbojet Engine (Fig. 17-3 and 17-4).

a. *Given:* $V_0 = 660$ mph $= 968$ fps
$T_0 = 520°$ F abs, $p_0 = 14.7$ psi, $h_0 = 28.8$ Btu/lb
$T_3 = 1700°$ F abs, $p_{r_3} = 188.4$, $h_3 = 327.4$ Btu/lb
$r_p = p_2/p_1 = 4$
$iwk_C = iwk_{T_u}$
$\eta_C = 1.0$, $\eta_{T_u} = 1.0$, $\eta_{\text{diffuser}} = 1.0$, $\eta_{\text{exit nozzle}} = 1.0$
$V_1 = 0$
$w_a = 40$ lb/sec

b. *Find:* V_j, T, thp, η_p, η_t, and η.

c. *Solution:*

1. Enthalpy at station 1, h_1, from equation (17-5) is

$$h_1 = h_0 + \frac{V_0^2}{2Jg} = 28.8 + \frac{(968)^2}{50{,}000} = 28.8 + 18.75 = 47.55 \text{ Btu/lb}$$

2. Enter Air Charts with h_1 and find

$$p_{r_1} = 4.09, \qquad T_1 = 598° \text{ F abs}$$

3. $p_1 = \dfrac{p_{r_1}}{p_{r_0}}(p_0) = \left(\dfrac{4.09}{2.507}\right)(14.7) = 23.97$ psi

4. Since $N_m = \dfrac{V_0}{a_1} = \dfrac{V_0}{\sqrt{gkRT_0}} = \dfrac{968}{1115} = 0.868$, where $k = 1.4$, the ideal ram pressure ratio, p_1/p_0, may be checked on curves of Fig. 17-5.

5. $p_2 = p_3 = p_1(r_p) = (23.97)(4) = 95.8$ psi.
6. $p_{r_2} = r_p(p_{r_1}) = 4(4.09) = 16.36$.
7. Enter Air Charts with p_{r_2} and find

$$h_2 = 117.1 \text{ Btu/lb}, \qquad T_2 = 884.8° \text{ F abs.}$$

8. Ideal compressor work per lb of air, iwk_C, is

$$iwk_C = h_2 - h_1 = 117.1 - 47.55 = 69.55 \text{ Btu/lb.}$$

9. Heat supplied in combustion chamber per lb of air, q_S, is

$$q_S = h_3 - h_2 = 327.4 - 117.1 = 210.3 \text{ Btu/lb}$$

10. Since $iwk_C = iwk_{T_u}$, the enthalpy at station 4 is

$$h_4 = h_3 - iwk_{T_u} = 327.4 - 69.55 = 257.85 \text{ Btu/lb}$$

11. Enter Air Charts with h_4 and find

$$p_{r_4} = 98.6, \qquad T_4 = 1439° \text{ F abs}$$

12. $p_4 = \dfrac{p_{r_4}}{p_{r_3}}(p_3) = \dfrac{98.6}{188.4}(95.8) = 50.1$ psi

13. Since $p_6 = 14.7$ psi, p_{r_6} is

$$p_{r_6} = p_{r_4}\left(\frac{P_6}{P_4}\right) = 98.6\left(\frac{14.7}{50.1}\right) = 28.96$$

14. Enter Air Charts with p_{r_6} and find

$$h_6 = 154.7 \text{ Btu/lb}, \qquad T_6 = 1036.2° \text{ F abs}$$

15. Jet velocity, V_j, from equation (17-10), when $\eta_{T_n} = 1.0$ and $V_4 = 0$, is

$$V_j = \sqrt{2gJ(h_4 - h_6)} = 223.7\sqrt{257.85 - 154.7} = 2270 \text{ ft/sec}$$

16. The thrust, T, from equation (17-1) is

$$T = \frac{w_a}{g}(V_j - V_0) = \frac{40}{32.2}(2270 - 968) = 1619 \text{ lb}$$

17. The thrust power is

$$\text{Thrust power} = \frac{w_a}{g}(V \quad - V_0)V_0 = TV_0 = 1{,}567{,}192 \text{ ft-lb/sec}$$

18. The thrust horsepower, thp, from equation

$$\text{thp} = \frac{TV_0}{375} = \frac{1619(660)}{375} = 2845 \text{ hp}$$

19. The propulsive efficiency from equation (17-4) is

$$\eta_p = \frac{2}{1 + V_j/V_0} = \frac{2}{1 + (2270/968)} = 0.597$$

20. The ideal thermal efficiency, η_t, from equation (17-12) is

$$\eta_t = \frac{(V_j^2 - V_0^2)}{2gJ(h_3 - h_2)} = \frac{(2270)^2 - (968)^2}{2(32.2)(778.2)(210.3)} = 0.401$$

21. The overall efficiency, η, from equation

$$\eta = \frac{\text{thrust power}}{\text{JQs}} = \left(\frac{\text{propulsive power}}{\text{JQs}}\right)\eta_p = \eta_p\eta_t = (0.597)(0.401) = 0.239$$

A-9. Turbojet Engine (Fig. 17-3 and 17-4).

a. *Given:*
$h_0 = 9.6$ Btu/lb
$h_1 = 16.5$ Btu/lb
$h_2 = 71.0$ Btu/lb
$h_{2'} = 83.0$ Btu/lb
$h_3 = 426.2$ Btu/lb
$h_4 = 350.4$ Btu/lb
$h_{6'} = 248.1$ Btu/lb
$w_a = 40$ lb/sec

b. *Find:* V_0, V_j, T, thp, η_p, η_t, and η.

c. *Solution:*

1. Find V_0 from equation 17-5;

$$V_0 = \sqrt{2gJ(h_1 - h_0)} = \sqrt{2(32.2)(778)(16.5 - 9.6)} = 588 \text{ fps}$$
$$= 588\left(\frac{3600}{5280}\right) = 401 \text{ mph}$$

2. From equations (17-8) and (17-9), $h_{4'}$ is

$$h_{4'} = h_3 - wk_{T_a} = h_3 - wk_C = h_3 - (h_{2'} - h_1)\ 426.2 - 66.5$$
$$= 359.7 \text{ Btu/lb}$$

3. Therefore, V_j, from equation 17-10, becomes

$$V_j = \sqrt{2gJ(h_{4'} - h_{6'})} = 223.7\sqrt{111.6} = \mathbf{2364} \text{ fps}$$

4. From equation (17-1), the thrust is determined as

$$T = \frac{w_a}{g}(V_j - V_0) = \frac{40}{32.2}(2364 - 588) = \mathbf{2206} \text{ lbs}$$

5. The thp from equation

$$\text{thp} = \frac{TV \text{ (fps)}}{550} = \frac{2206(588)}{550} = \mathbf{2358} \text{ hp}$$

6. The propulsive efficiency from equation (17-4) is

$$\eta_p = \frac{2}{1 + V_j/V_0} = \frac{2}{1 + (2364/588)} = \mathbf{0.40}$$

7. The thermal efficiency from equation (17-12) becomes

$$\eta_t = \frac{V_j^2 - V_0^2}{2gJ(h_3 - h_{2'})} = \frac{(2364)^2 - (588)^2}{50{,}000(343)} = \mathbf{0.306}$$

8 The overall efficiency of the turbojet engine is

$$\eta = \eta_t\eta_p = (0.306)(0.40) = \mathbf{0.12}$$

A-10. Turbojet Engine.

a. *Given:* A turbojet engine is cruising at an altitude of 20,000 feet in steady flight at 620 mph. The atmospheric pressure and temperature is 6.75 psia and 448° F abs. The engine consumes 85 lbs air per second and produces 3500 lbs of thrust.

b. *Find:* thp, V_j, and η_p.

c. *Solution:*

1. The thp is

$$\text{thp} = \frac{TV_0}{375} \text{ (mph)} = \frac{3500(620)}{375} = \mathbf{5790} \text{ hp}$$

2. The jet exit velocity from equation (17-1) becomes

$$V_j = \frac{Tg}{w_a} + V_0 = \frac{3500(32.2)}{85} + 909 = \mathbf{2234} \text{ fps}$$

3. The propulsive efficiency from equation (17-4)

$$\eta_p = \frac{2}{1 + (V_j/V_0)} = \frac{2}{1 + (2234/909)} = \mathbf{0.578}$$

A-11. Turboprop Engine (Figure 17-21).

a. *Given:*
$r_p = 4$
$V_0 = 1000$ ft/sec
$h_0 = 29$ Btu/lb
$h_{2'} = 129$ Btu/lb
$h_3 = 327$ Btu/lb
$h_{4'} = 203$ Btu/lb
$h_{6'} = 169$ Btu/lb
$w_a = 40$ lb air/sec

b. ***Find:***

1. wk_C in Btu/lb
2. Horsepower developed by propeller, assuming $\eta_g = 1.0$ and $\eta_{prop} = 1.0$.
3. V_j.
4. Thrust developed by jet exit gases.

c. ***Solution:***

1. Find h_1 from

$$h_1 = h_0 + \frac{V_0^2}{2gJ} = 29 + \frac{(1000)^2}{50,000} = 29 + 20 = 49 \text{ Btu/lb}$$

2. Compressor work, wk_C, is

$$wk_C = (h_{2'} - h_1) = 129 - 49 = 80 \text{ Btu/lb}$$

3. Propeller horsepower from equation (17-16) becomes

$$\begin{aligned}\text{Prop hp} &= \frac{J\eta_g\eta_{prop}}{550}(w_a)(h_3 - h_{4'}) - (h_{2'} - h_1)\\ &= \frac{778}{550}(1.0)(1.0)(40)[(327 - 203) - (129 - 49)\\ &= 2490 \text{ hp}\end{aligned}$$

4. The jet exit velocity is

$$V_j = \sqrt{2gJ(h_{4'} - h_{5'})} = 223.7\sqrt{203 - 169} = 1304 \text{ ft/sec}$$

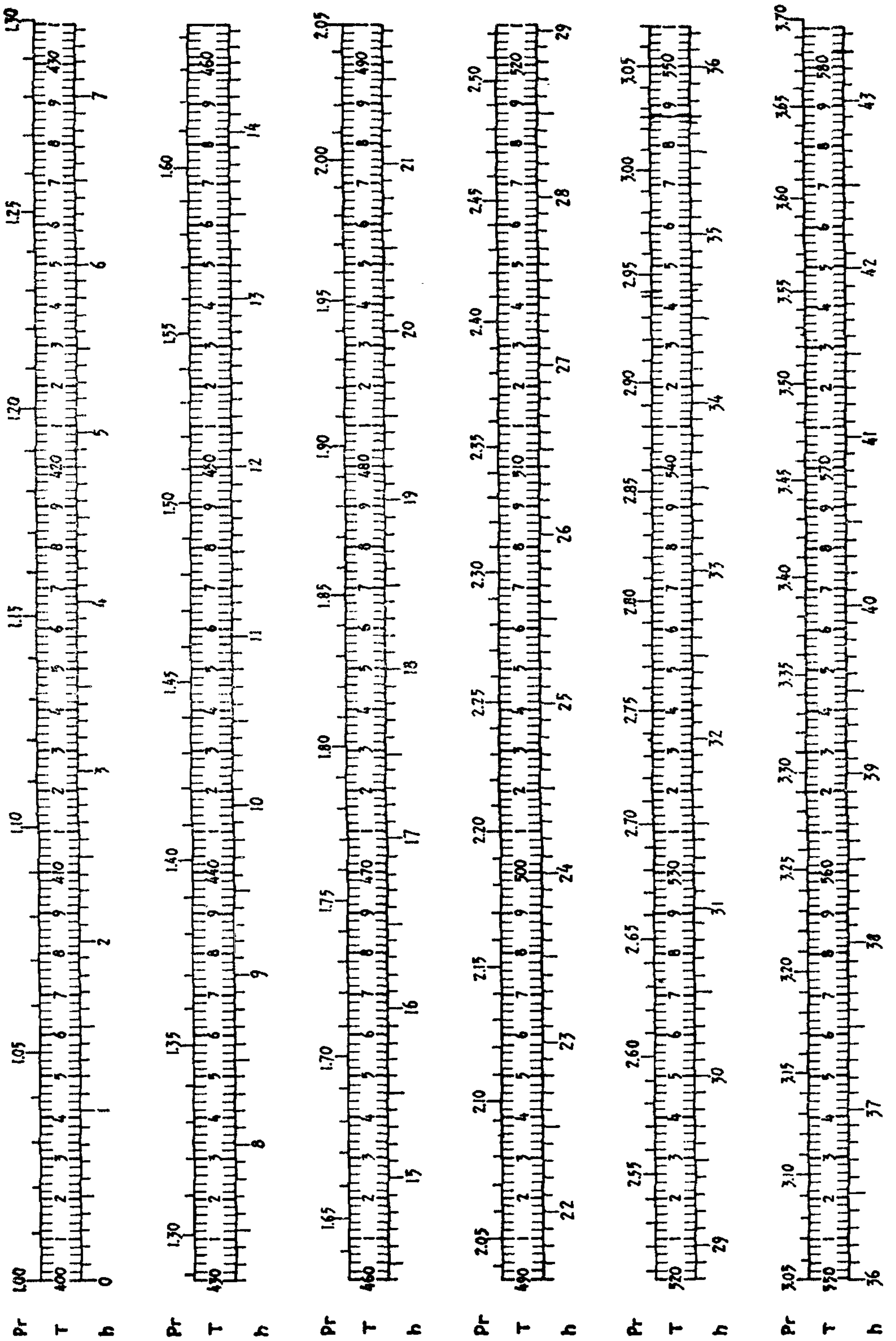
5. The thrust developed by the jet gases

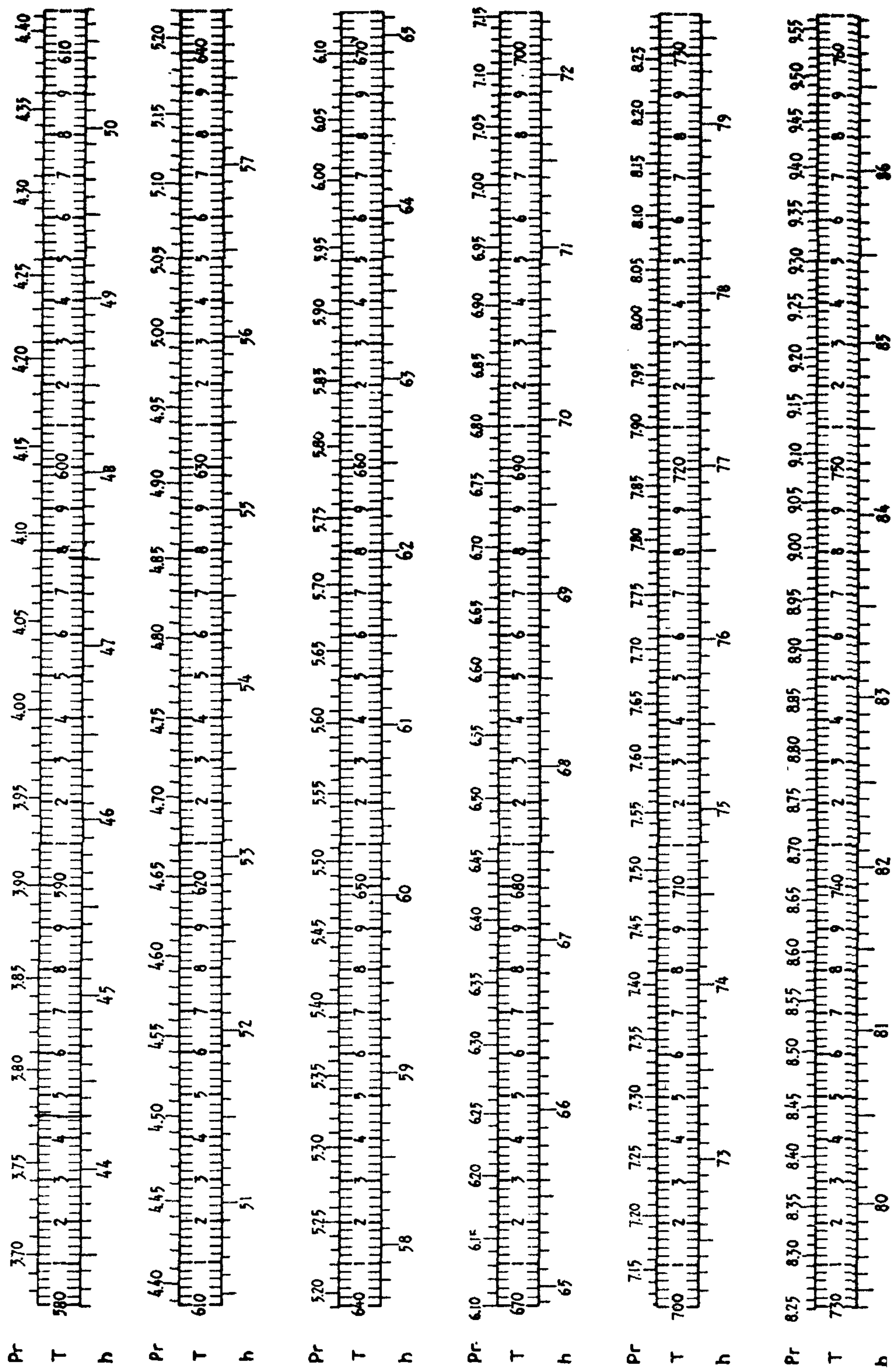
$$T = \frac{w_a}{g}(V_j - V_0) = \frac{40}{32.2}(1304 - 1000) = 378 \text{ lbs}$$

APPENDIX B

AIR CHARTS

Courtesy of U. S. Navy

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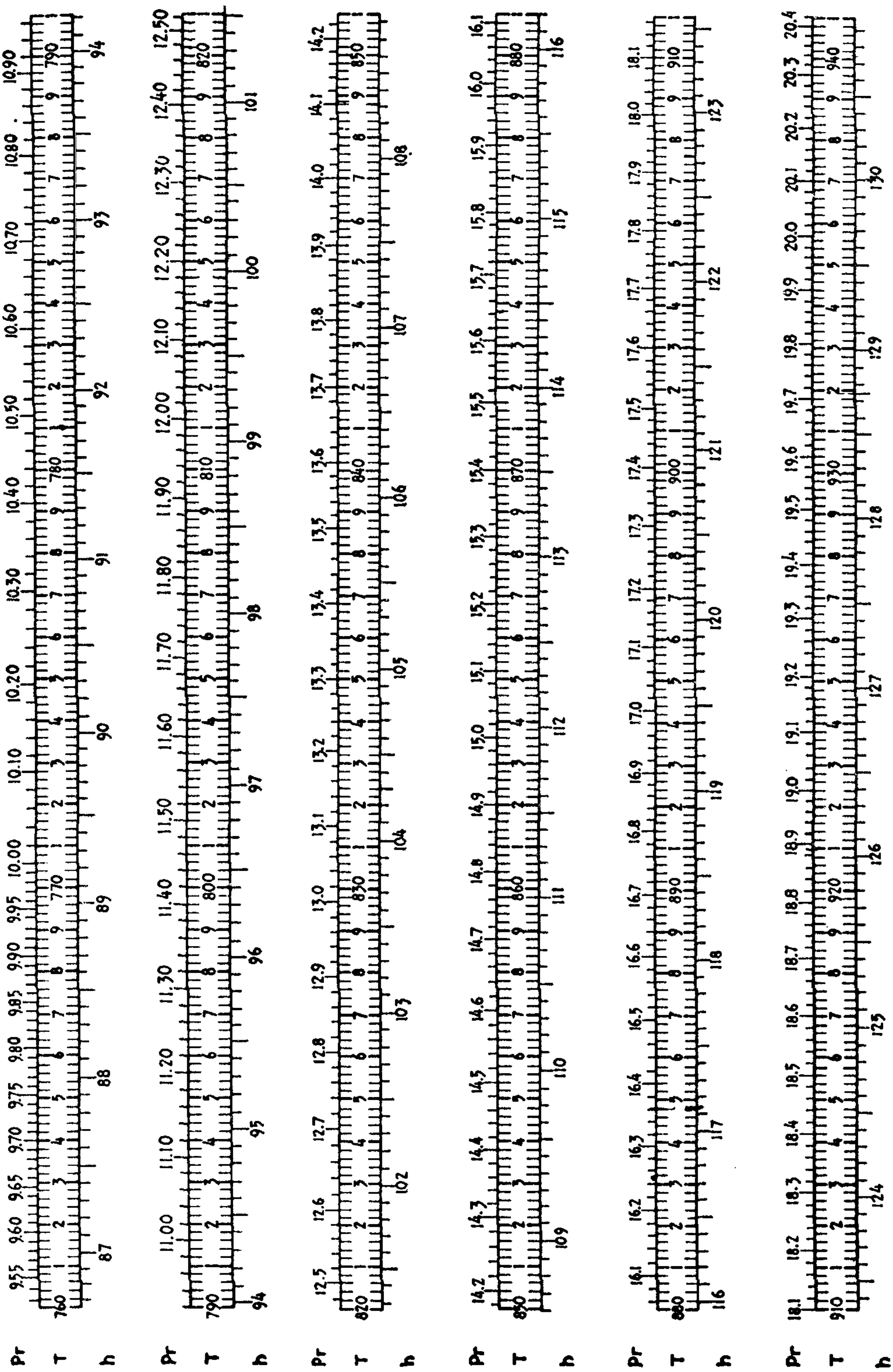

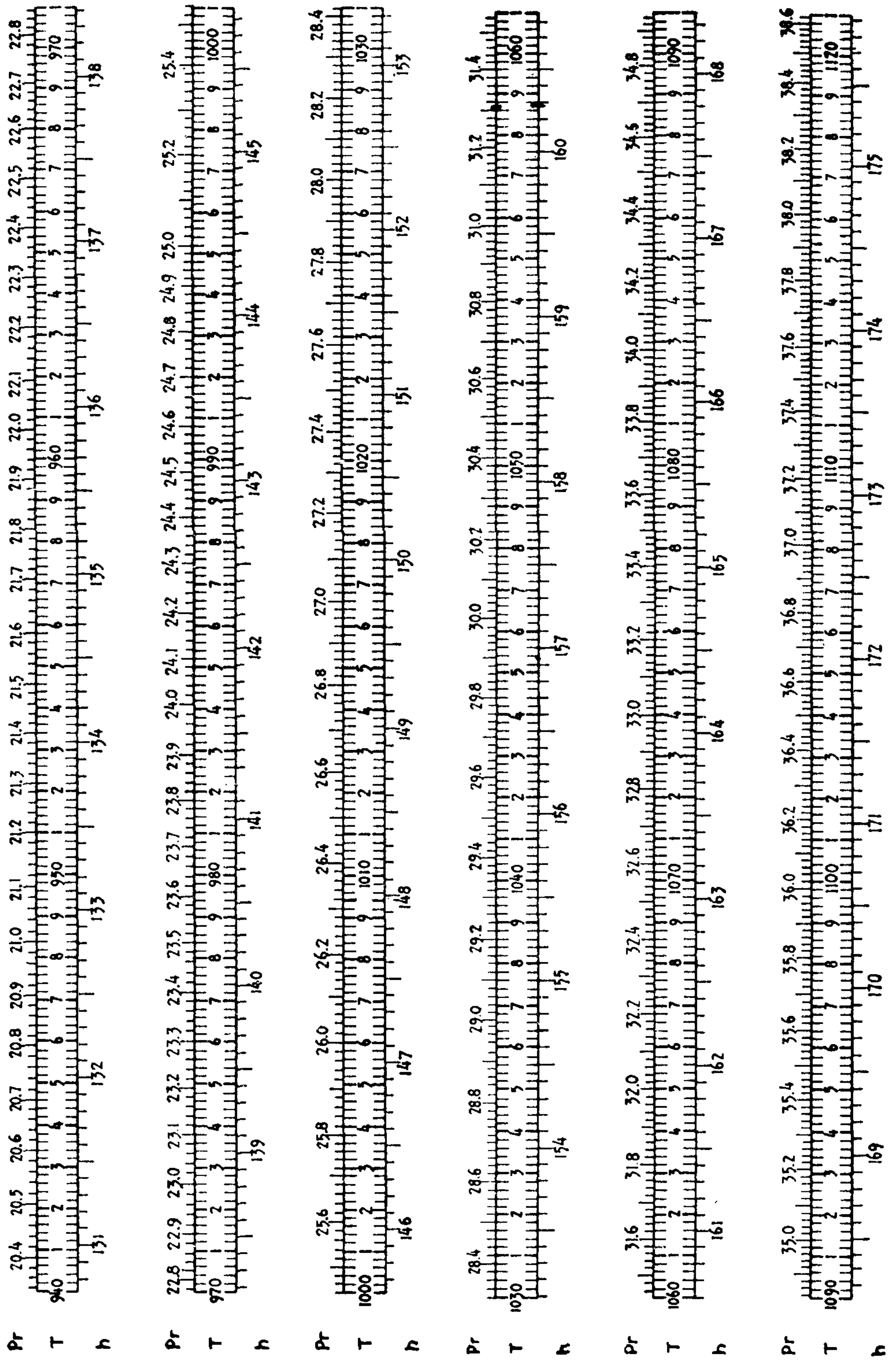
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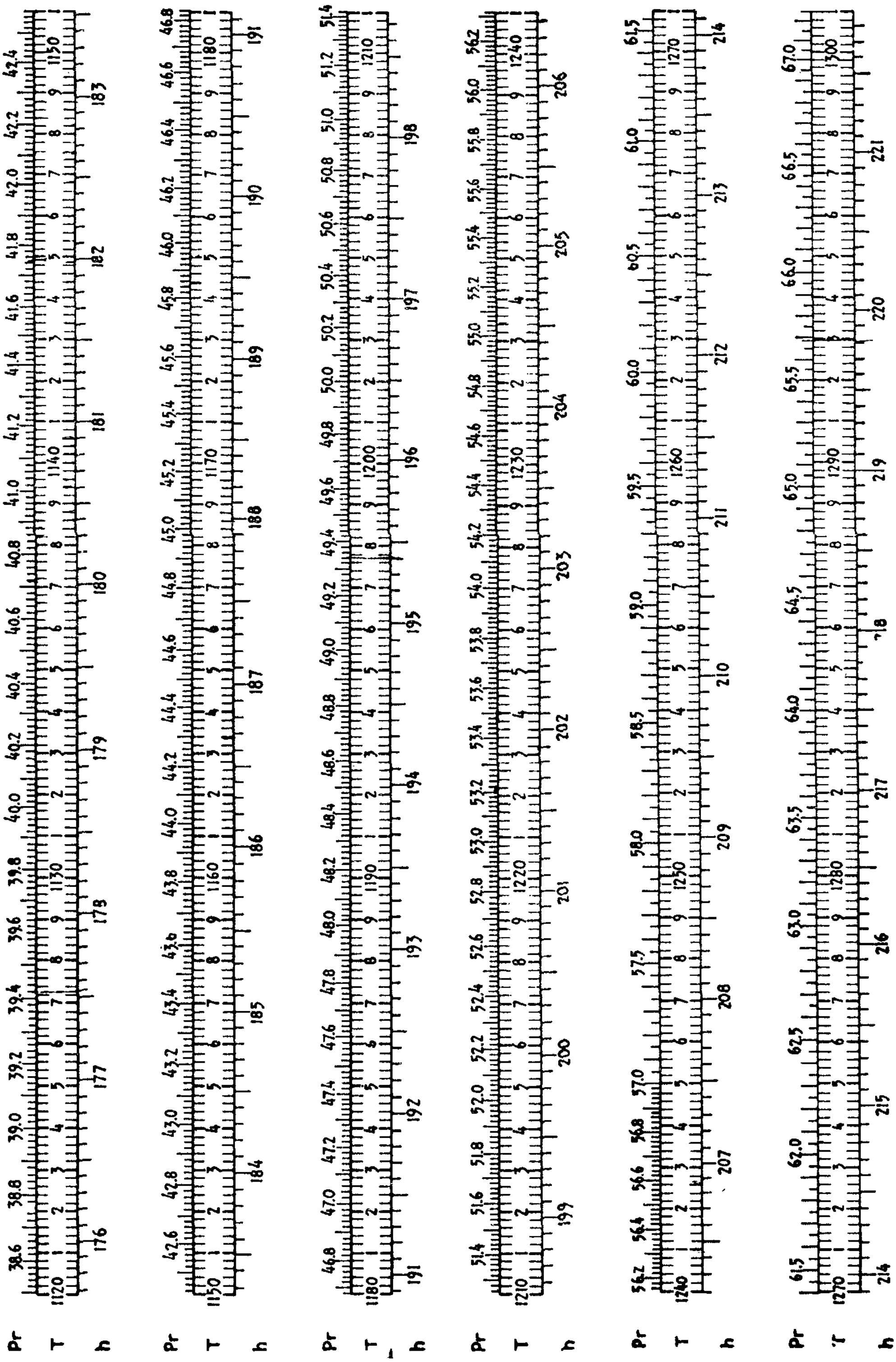
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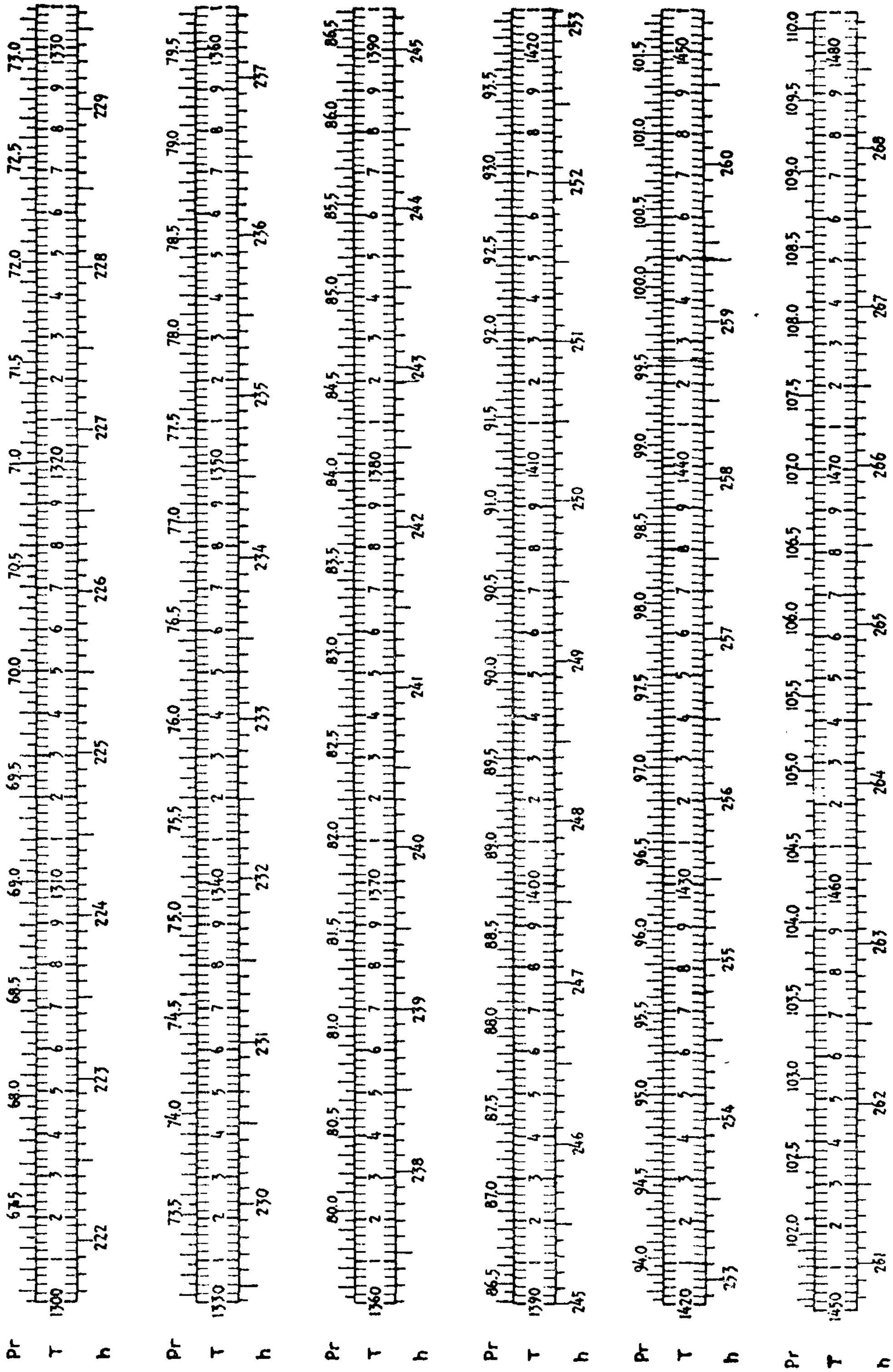

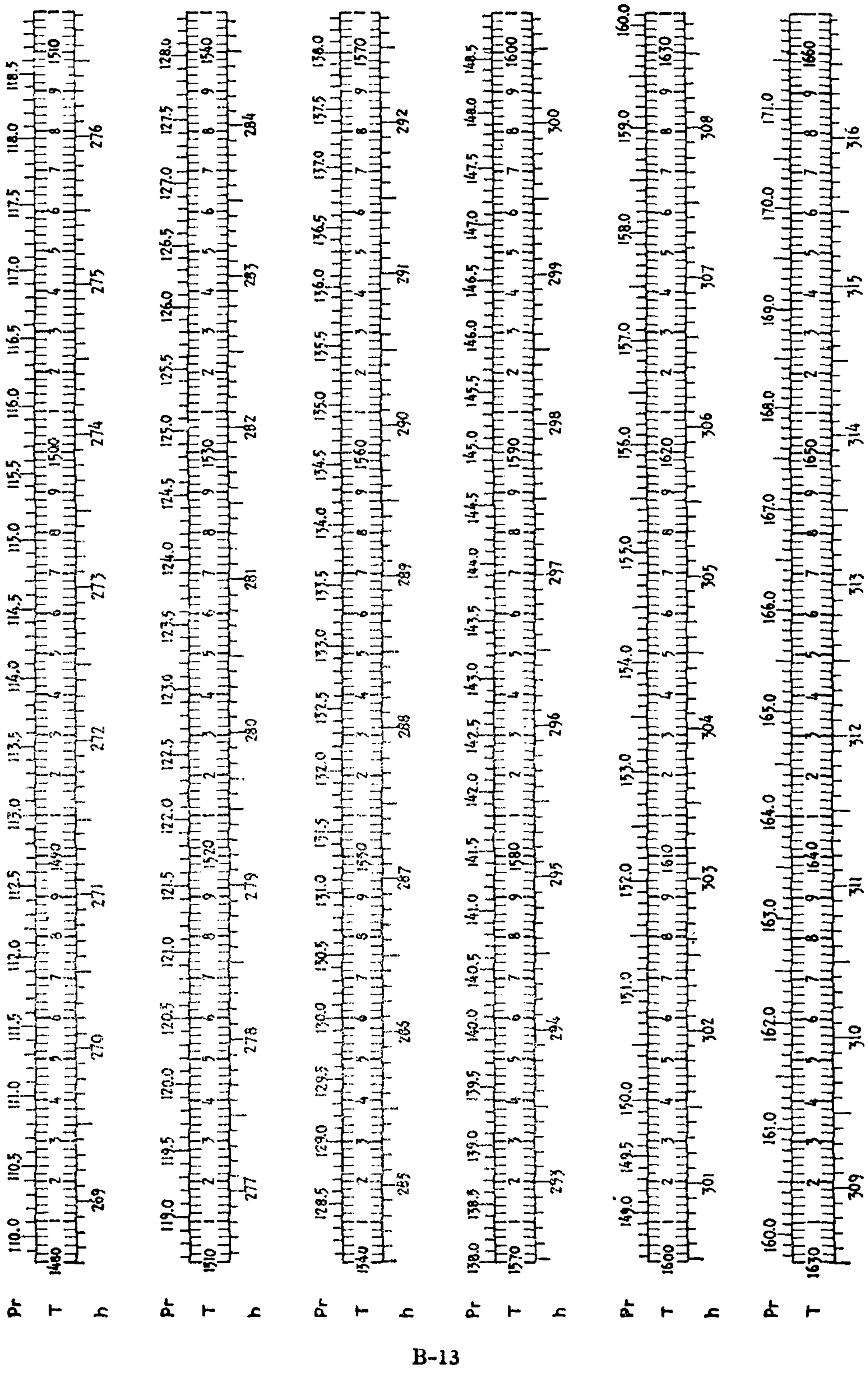

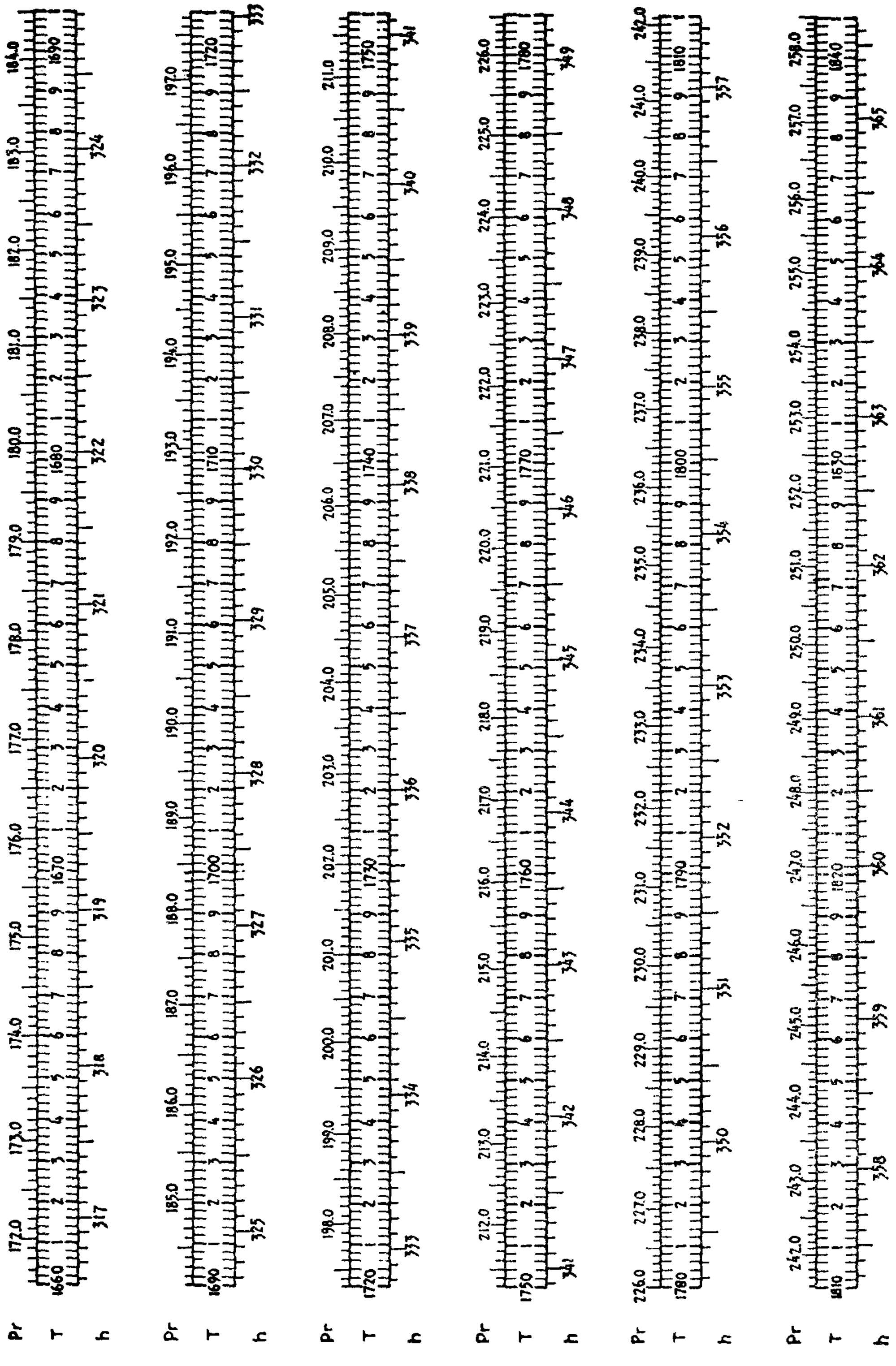

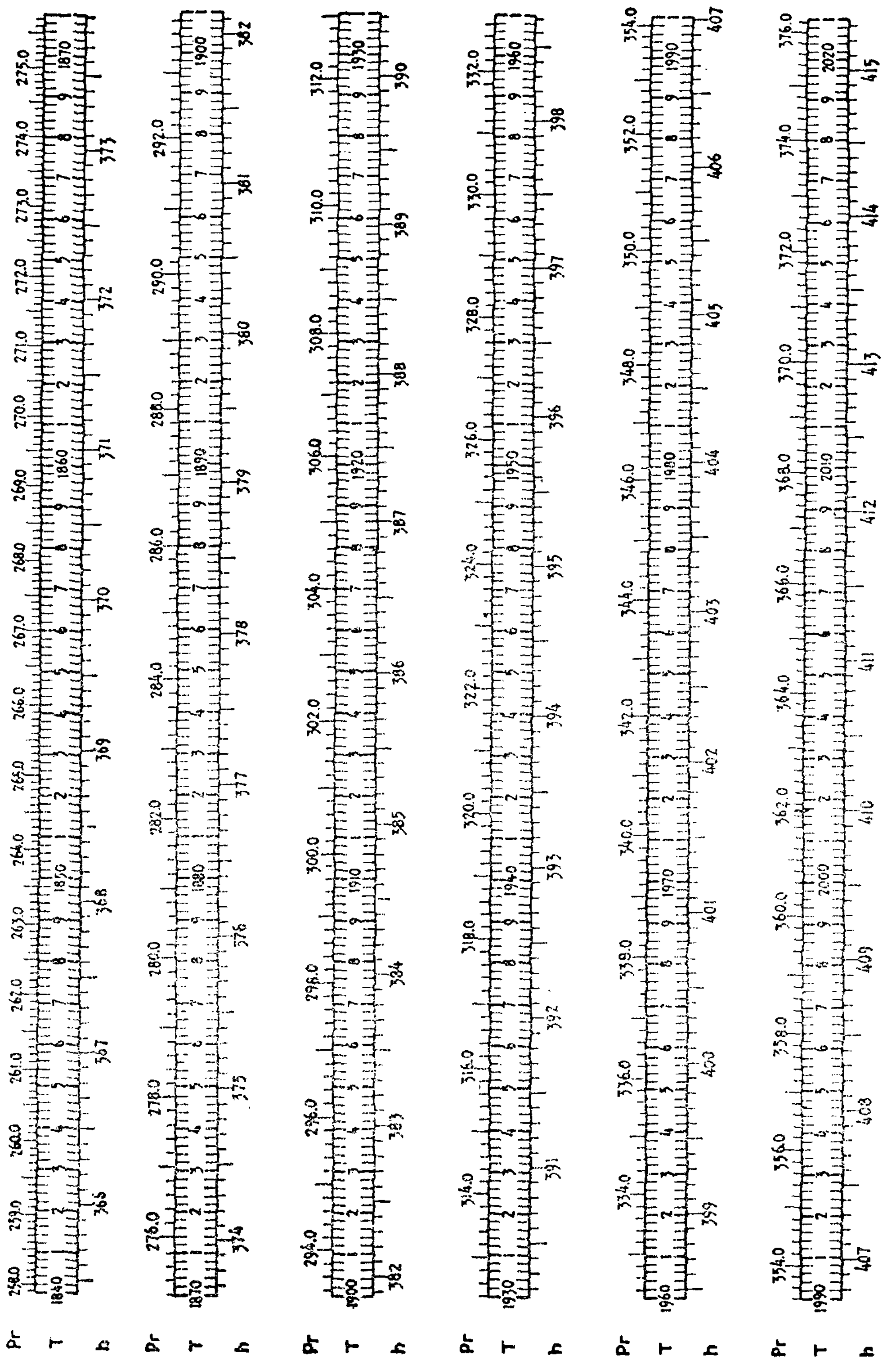
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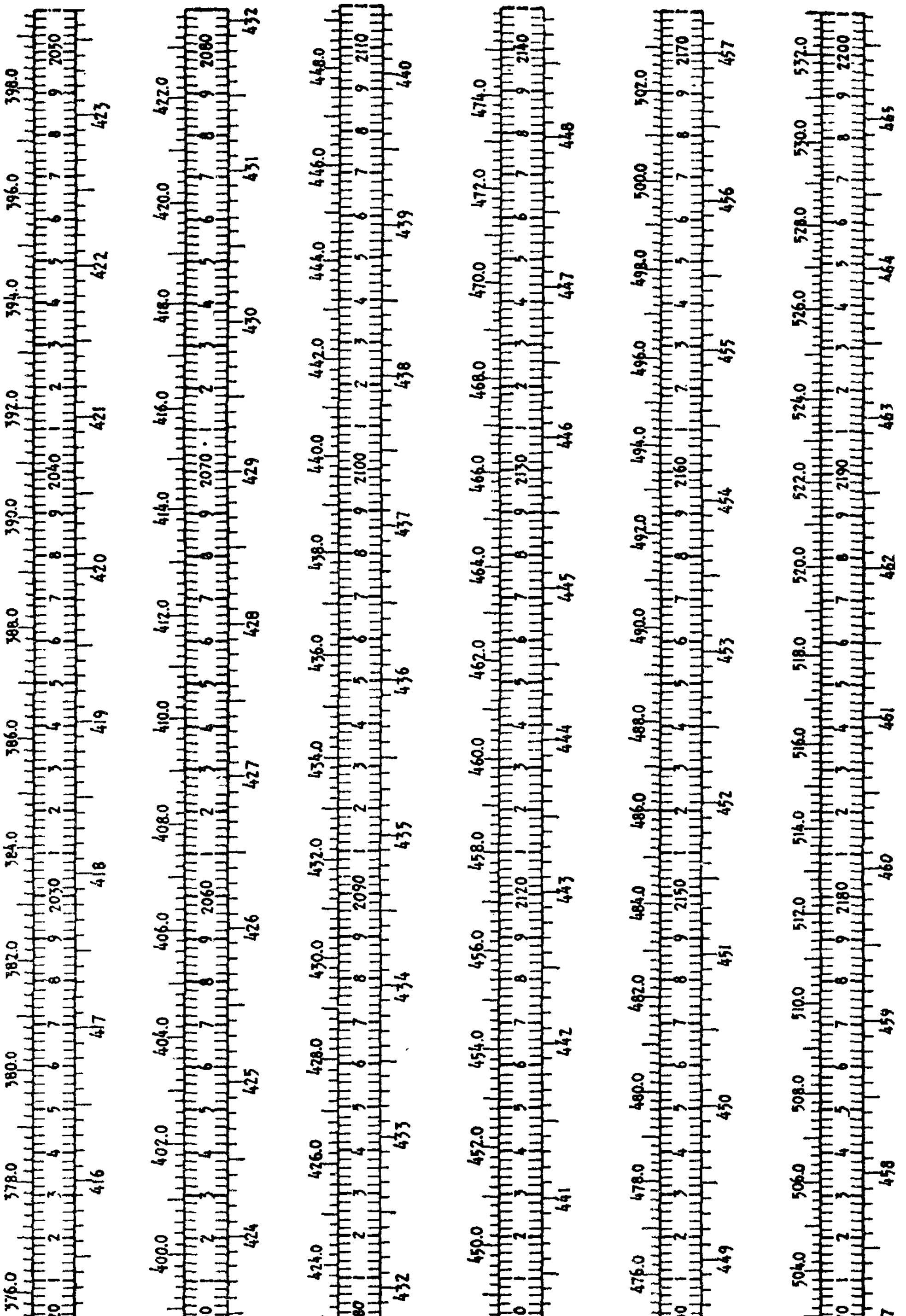

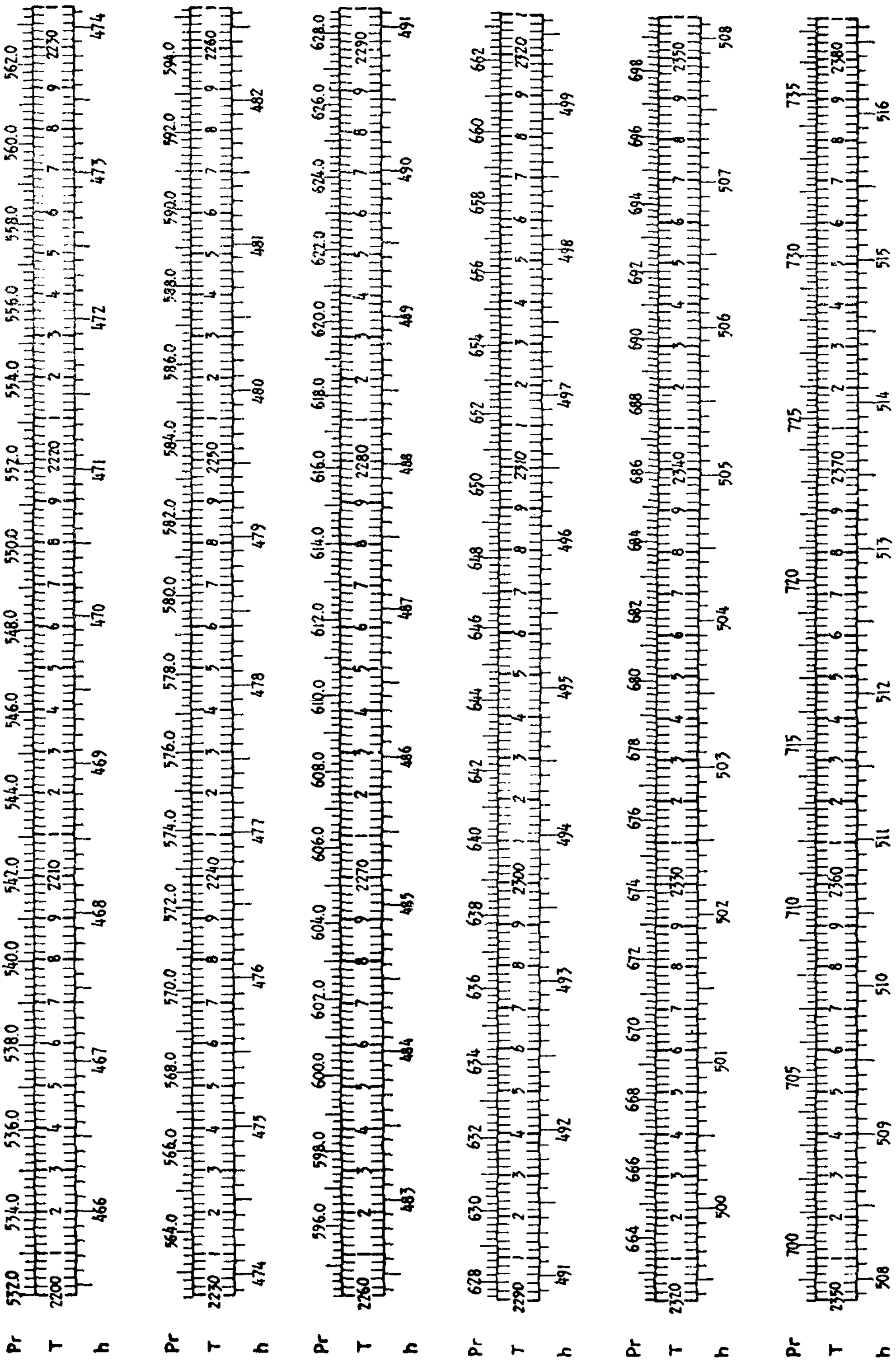

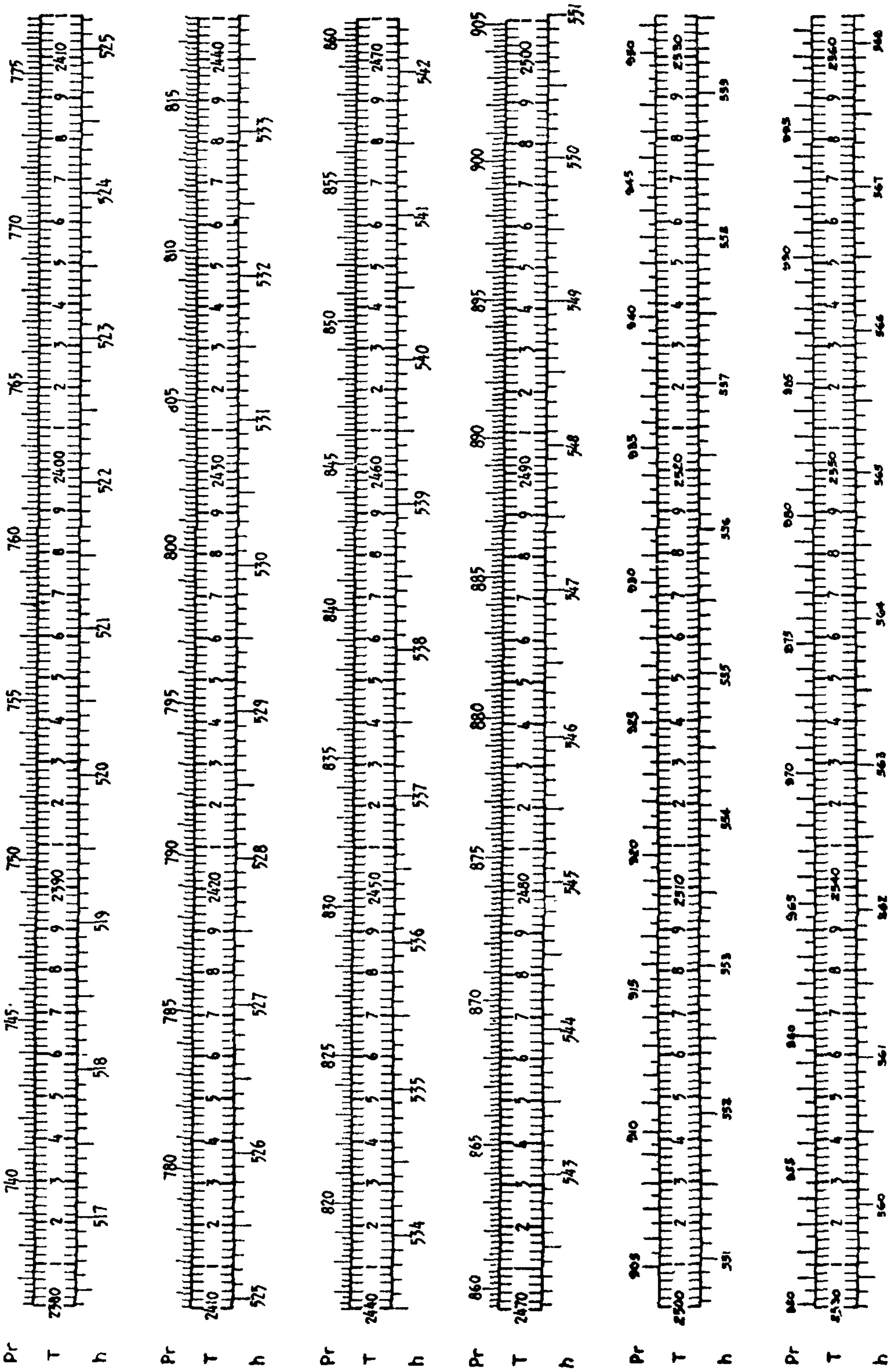
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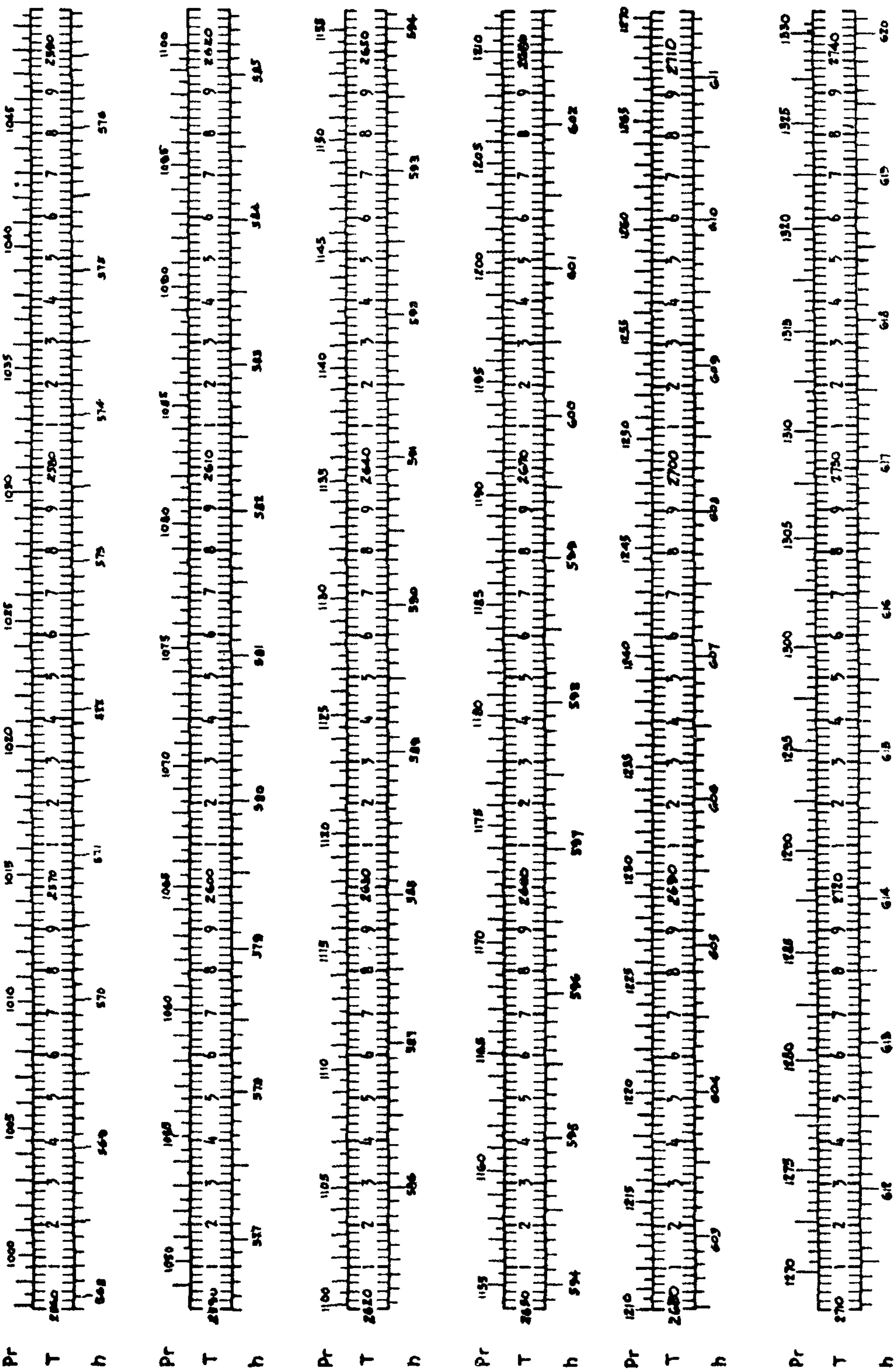

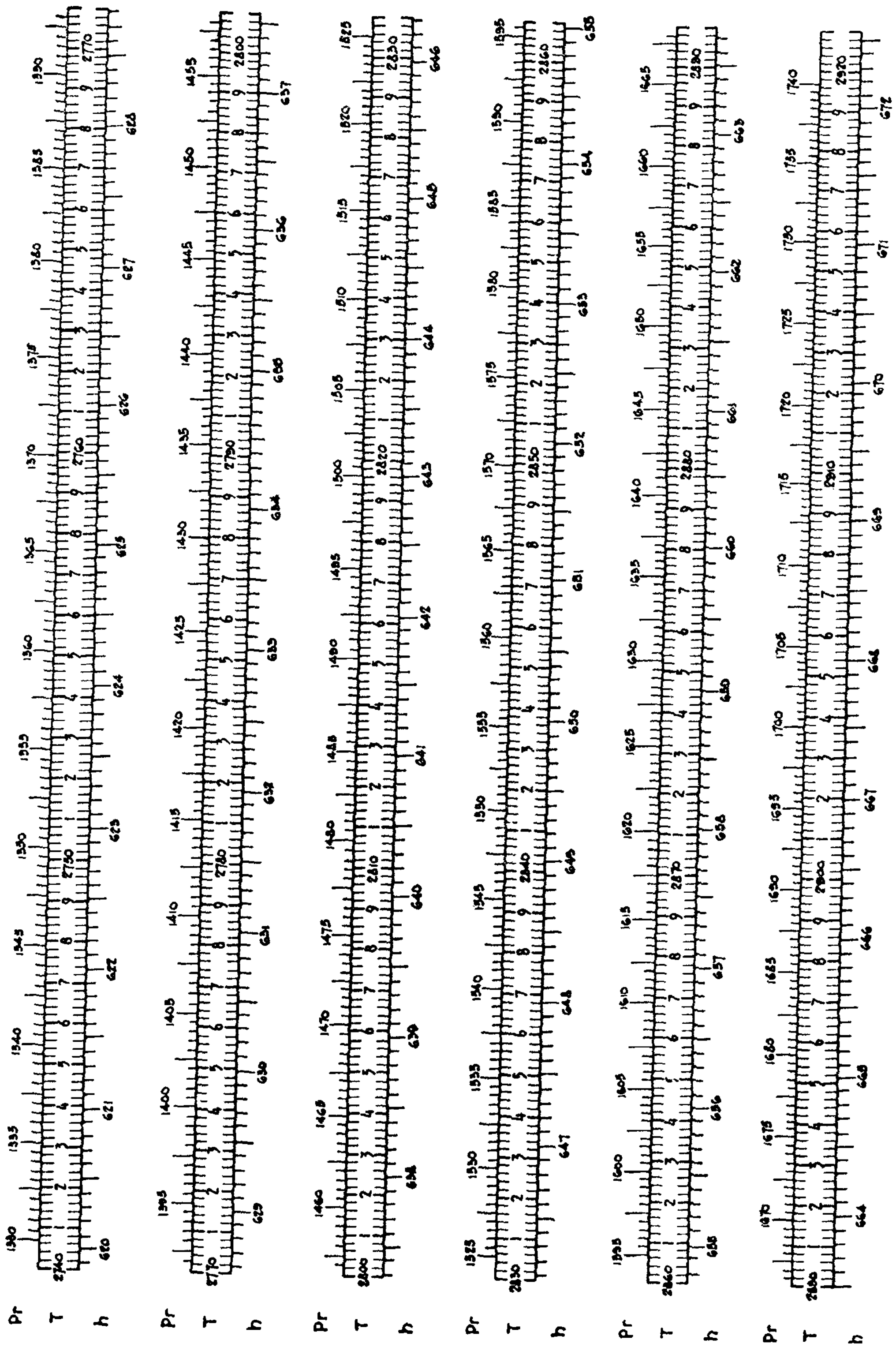
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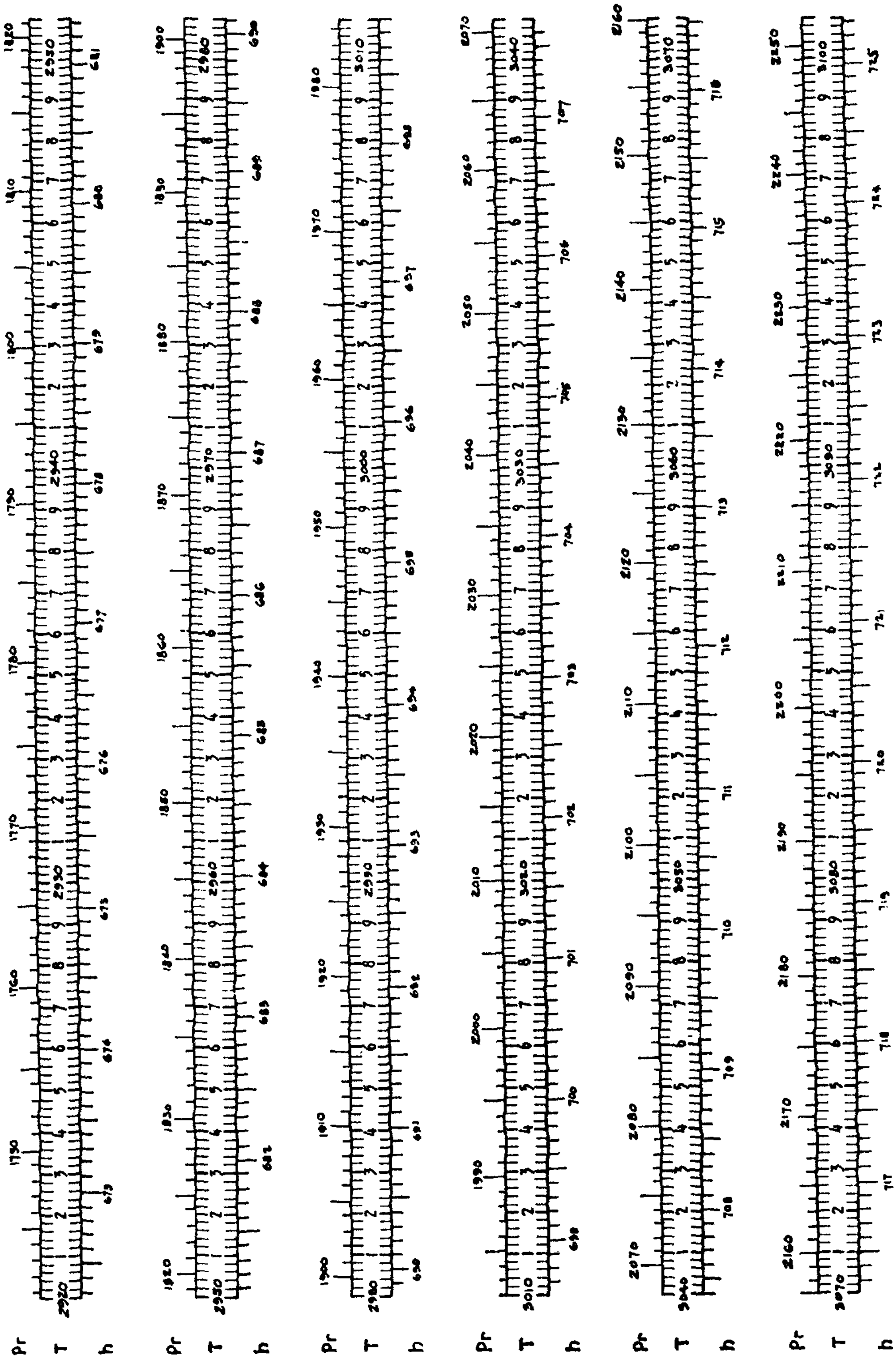
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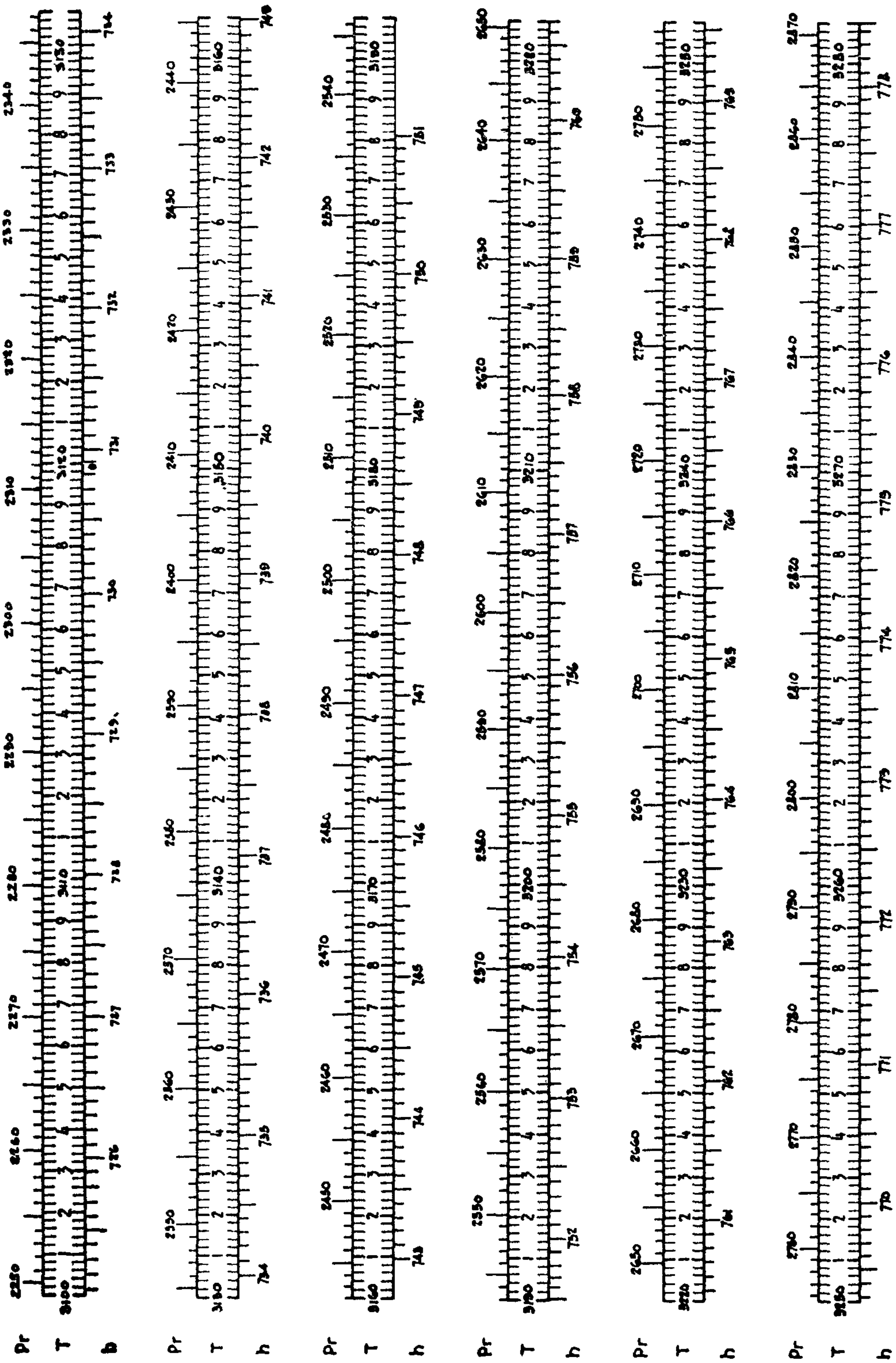

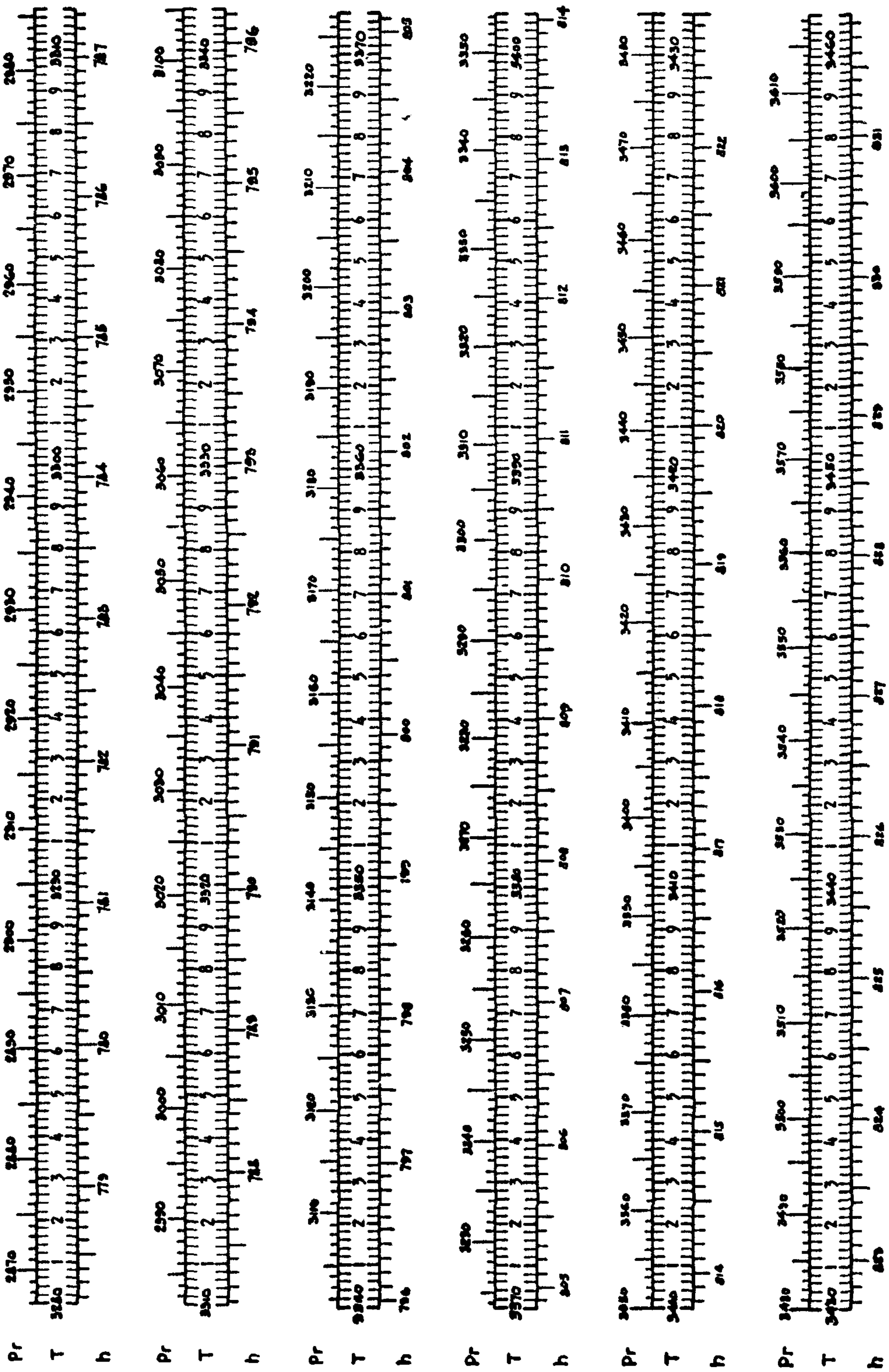
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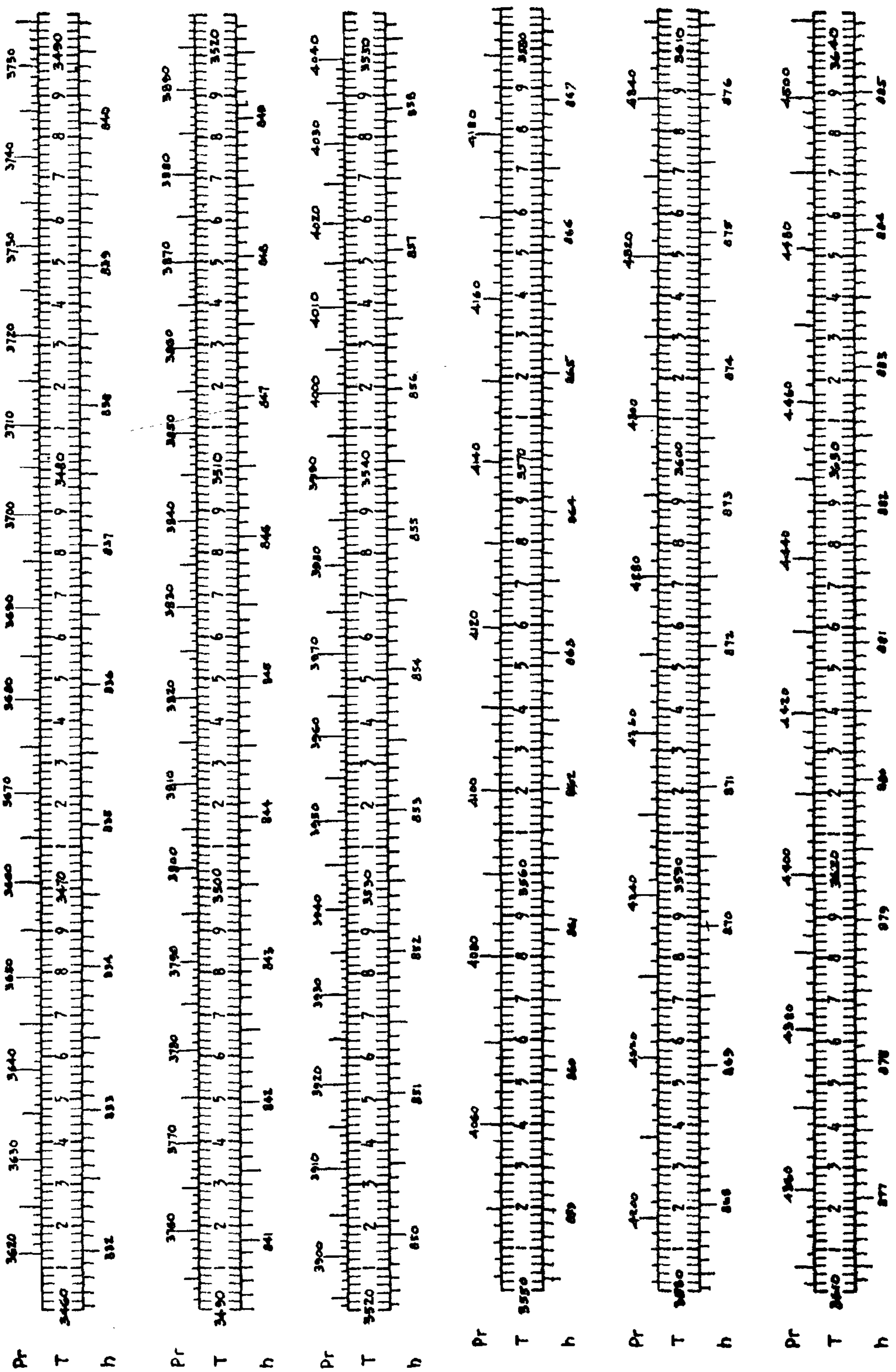

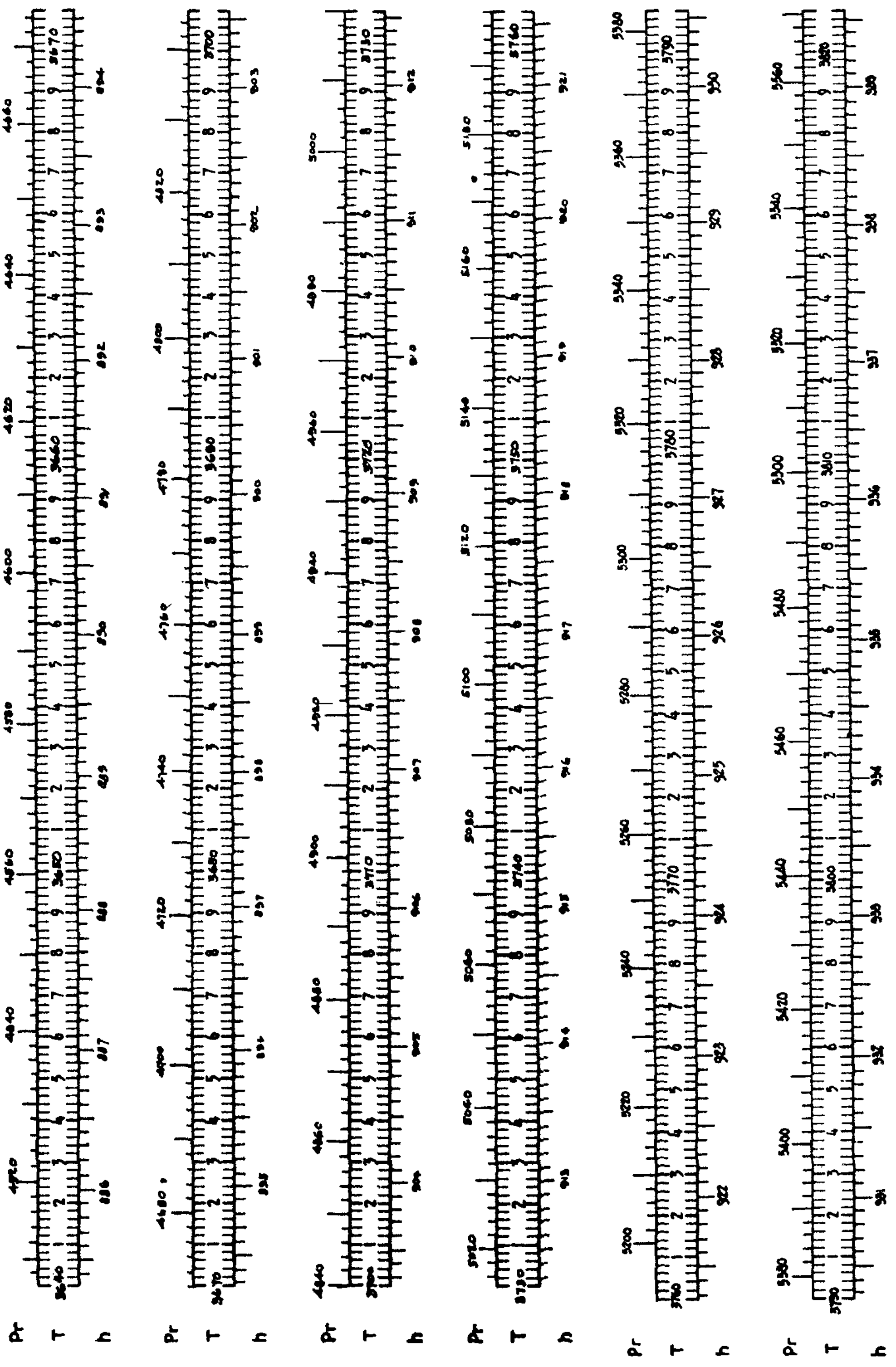

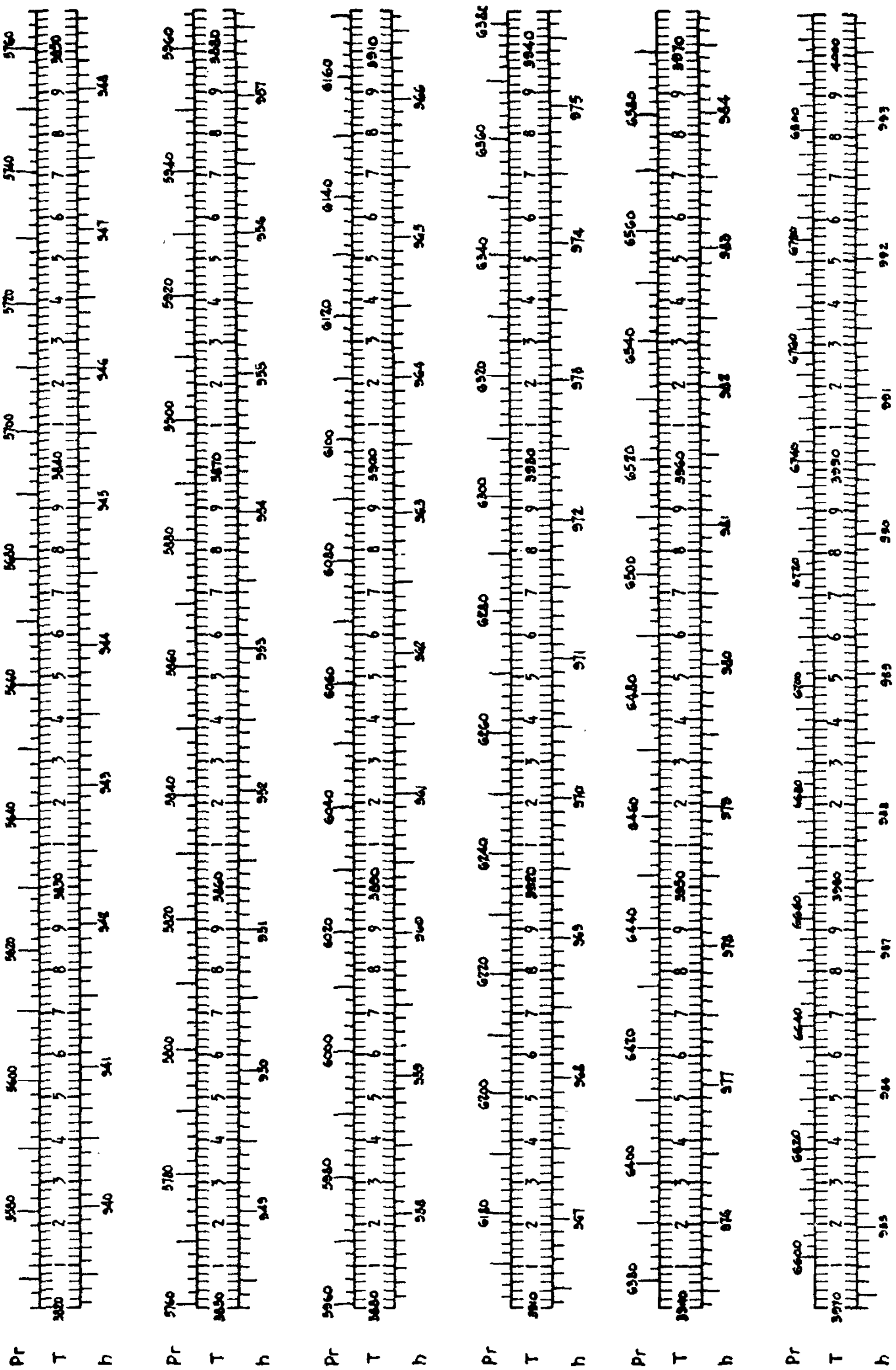
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T	h	pr	T	h	pr	T	h	pr	T	h	pr
2500	550.67	903.6	3100	725.04	2249	3700	903.26	4840	4300	1084.32	9378
2520	556.41	934.2	3120	730.92	2312	3720	909.25	4956	4320	1090.39	9573
2540	562.15	965.6	3140	736.81	2376	3740	915.25	5073	4340	1096.47	9771
2560	567.90	997.9	3160	742.71	2442	3760	921.25	5192	4360	1102.55	9972
2580	573.65	1031.0	3180	748.61	2509	3780	927.25	5314	4380	1108.63	10176
2600	579.41	1065.0	3200	754.51	2577	3800	933.26	5439	4400	1114.72	10384
2620	585.17	1099.8	3220	760.42	2647	3820	939.27	5565	4420	1120.81	10595
2640	590.94	1135.6	3240	766.33	2719	3840	945.28	5694	4440	1126.89	10810
2660	596.72	1172.3	3260	772.25	2792	3860	951.30	5825	4460	1132.98	11028
2680	602.50	1210.0	3280	778.17	2867	3880	957.32	5959	4480	1139.08	11250
2700	608.28	1248.6	3300	784.09	2943	3900	963.34	6095	4500	1145.18	11475
2720	614.07	1288.1	3320	790.02	3021	3920	969.36	6234	4520	1151.28	11704
2740	619.87	1328.6	3340	795.95	3101	6940	975.39	6375	4540	1157.38	11986
2760	625.67	1370.1	3360	801.88	3182	3960	981.42	6518	4560	1163.48	12171
2780	631.48	1412.7	3380	807.81	3265	3980	987.45	6664	4580	1169.59	12411
2800	637.29	1456.3	3400	813.75	3350	4000	993.48	6813	4600	1175.70	12655
2820	643.11	1500.9	3420	819.69	3436	4020	999.52	6965	4620	1181.81	12902
2840	648.93	1546.6	3440	825.64	3524	4040	1005.56	7119	4640	1187.92	13153
2860	654.75	1593.5	3460	831.59	3614	4060	1011.60	7276	4660	1194.03	13408
2880	660.58	1641.5	3480	837.54	3705	4080	1017.65	7435	4680	1200.15	13667
2900	666.42	1690.6	3500	843.50	3798	4100	1023.70	7597	4700	1206.27	13929
2920	672.26	1740.8	3520	849.46	3894	4120	1029.75	7762	4720	1212.39	14195
2940	678.11	1792.2	3540	855.43	3991	4140	1035.80	7929	4740	1218.51	14466
2960	683.96	1844.8	3560	861.40	4090	4160	1041.86	8100	4760	1224.64	14741
2980	689.82	1898.7	3580	867.37	4191	4180	1047.92	8274	4780	1230.77	15020
3000	695.68	1954	3600	873.34	4294	4200	1053.98	8450	4800	1236.90	15303
3020	701.54	2010	3620	879.32	4399	4220	1060.04	8629			
3040	707.41	2068	3640	885.30	4506	4240	1066.11	8811			
3060	713.28	2127	3660	891.28	4615	4260	1072.18	8997			
3080	719.16	2187	3680	897.27	4726	4280	1078.25	9186			

APPENDIX C

STEPS IN THE DEVELOPMENT OF EQUATIONS

1. **Thermal Efficiency of an Ideal Simple Open Cycle Gas Turbine.** From equation (16-7) and Figure (16-8), the thermal efficiency is given as

$$\eta_t = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{(h_3 - h_2) - (h_4 - h_1)}{h_3 - h_2}$$

$$= 1 - \frac{h_4 - h_1}{h_3 - h_2} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{T \ \ - T_1}{T_3 - T_2}$$

By virtue of the isentropic process between states 3 and 4 and between states 1 and 2, it is found that

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{(k-1)/k} \quad \text{and} \quad \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k}$$

and since $p_3 = p_2$ and $p_4 = p_1$, then

$$\frac{T_3}{T_4} = \frac{T_2}{T_1}$$

By rearranging terms and subtracting unity from each side of the equation

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

which becomes

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

and finally

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

The thermal efficiency becomes

$$\eta_t = 1 - \frac{T_1}{T_2}$$

Since there is an isentropic process between states 1 and 2,

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{(k-1)/k} = \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

and

$$\eta_t = 1 - \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

APPENDIX D

LETTERS ASSIGNED AIRCRAFT ENGINE MANUFACTURERS

Manufacturer's Name	*Letter Symbols*
Aerojet Engineering Corporation	AJ
Allis-Chalmers Manufacturing Company	AC
Allison Division, General Motors Corporation	A
Bell Aircraft Corporation	BA
Bodine Soundrive Company	BD
Chevrolet Motor Company, Division General Motors Corp.	C
Chrysler Corporation	D
Continental Aviation and Engineering Corporation	T
DeLaval Steam Turbine Company	DL
Elliot Company	EE
Ford Motor Company	F
Frederic Flader Company	FF
General Electric Company	GE
Globe Aircraft Corporation	GA
G. M. Giannini and Company	GN
Harvey Machine Company, Incorporated, Aviation Division	HM
Joshua Hendy Iron Works	JH
Kaiser Fleetwing, Incorporated	FW
Lockheed Aircraft Corporation	LA
Menasco Manufacturing Company	MN
Marquardt Aircraft Company	MA
McDonnell Aircraft Corporation	MD
Northrop Hendy Company	NA
Packard Motor Car Company	V
Pratt and Whitney Aircraft Div. United Aircraft Corp.	P
Radioplane Company	RP
Ranger Aircraft Engine Division Fairchild Engineering and Airplane Corporation	R
Taylor Turbine Corporation	TT
West Engineering Company	WS
Westinghouse Electric Corporation	WE
Wright Aeronautical Corporation Division Curtiss-Wright Corporation	W

SYMBOLS USED FOR TYPE AND MODEL DESIGNATION OF AIRCRAFT JET ENGINES

A system of symbols for type and model designation of the various aircraft jet propulsion engines has been developed and is in use by the Navy and the Air Force. The first part of the symbol consists of a letter (or letters) to indicate the basic type of engine. The letters are as follows:

Letter	*Basic Type*
J	Turbojet engine
T	Turboprop engine
PJ	Pulse jet engine
RJ	Ram jet engine
R	Rocket engine

STEPS IN DEVELOPMENT OF EQUATIONS

2. Thermal Efficiency of an Ideal Open Cycle Gas Turbine with Ideal Regeneration (Article 16-7).

The thermal efficiency is given as

$$\eta_t = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_4} = 1 - \frac{(h_2 - h_1)}{(h_3 - h_4)}$$

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

By virtue of an isentropic process between states 3 and 4, it is found that

$$T_4 = T_3\left(\frac{p_4}{p_3}\right)^{(k-1)/k}$$

and since $p_2 = p_3$ and $p_4 = p_1$, then

$$T_4 = T_3\left(\frac{p_1}{p_2}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$

By virtue of an isentropic process between states 2 and 1, it is found

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{(k-1)/k} = T_1(r_p)^{(k-1)/k}$$

Substituting in the thermal efficiency equation

$$\eta_t = 1 - \frac{T_1(r_p)^{(k-1)/k} - T_1}{T_3 - T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}} = 1 - \left(\frac{T_1}{T_3}\right)\left[\frac{(r_p)^{(k-1)/k} - 1}{1 - \left(\frac{1}{r_p}\right)^{(k-1)/k}}\right]$$

$$= 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}$$

3. Equivalent Shaft Horsepower of Turboprop Engines.

The total thrust horsepower of a turboprop engine is the sum of the thrust horsepower developed by the propeller and by the jet. Therefore

$$\text{Total thp} = \eta_{\text{prop}}\,\text{shp} + \frac{TV_0}{375}$$

where η_{prop} = propeller efficiency
V_0 = airplane velocity, mph
T = Thrust developed in jet
shp = shaft horsepower applied to propeller

The equivalent horsepower of the engine is the total thp divided by the propeller efficiency. Dividing the above equation by η_p, the equivalent hp becomes

$$\text{eshp} = \frac{\text{Total thp}}{\eta_{\text{prop}}} = \text{bhp} + \frac{TV}{375(\eta_{\text{prop}})}$$

In rating turboprop engines, the equivalent horsepower was obtained assuming V_0 at 115 mph and η_{prop} at 0.80.

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The letters "X" and "Y" may be prefixed to the basic letter to signify an experimental engine and a service test engine respectively. The numerals that follow the letters are assigned by the two military services and begin with the numeral 30 for the Navy and 31 for the Air Force. These numbers are arbitrary and do not represent any characteristics of the units involved. The even numbers above 30 are assigned progressively by the Bureau of Aeronautics to engines sponsored by the Navy; and the odd numbers are assigned progressively by the Air Materiel Command to engines sponsored by the Air Force. The second part of the designation consists of a dash and a letter (or letters) which indicate the manufacturer of the engine.

The third part of the designation consists of a dash and a numeral indicating the model number. These model numbers are assigned to jet engines in the same manner as now applied to reciprocating aircraft engines; that is, odd numbers indicate Air Force models and even numbers Navy models. Air Force model numbers for each type of jet engine begin with one and continue with consecutive odd numbers and Navy numbers begin with two and continue with consecutive even numbers. All even model numbers are assigned by the Bureau of Aeronautics, including those applied to Air Force approved engine types. All odd numbers are assigned by the Air Materiel Command, including those applied to Navy approved engine types.

A given engine design has only one type and model designation for both Services. For example, should the Navy desire to use an engine bearing Air Force type and model numbers, the Navy uses those numbers without change, provided no changes are made in the engine, for all designation purposes. Further, should the Air Force desire to use a Navy approved type of engine, but require minor production changes to the Navy model of that type, the Air Force uses the Navy type designation and assigns its own model designation, which will begin with one and will continue with consecutive odd numbers, to the modified engine regardless of the Navy model number.

The following hypothetical examples illustrate the arrangement and significance of the system of symbols:

J30-A-2	First Navy Model of First Navy Turbo-Jet Type. (Made by Allison)
J31-W-1	First AF Model of First AF Turbo-Jet Type. (Made by Wright Aeronautical)
J31-GE-2	First Navy Model of First AF Turbo-Jet Type. (Made by General Electric)
T34-P-3	Second AF Model of Third Navy Turbo-Prop Type. (Made by Pratt and Whitney)
RJ35-T-6	Third Navy Model of Third AF Ram Jet Type. (Made by Continental Motors)
PJ36-RP-7	Fourth AF Model of Fourth Navy Pulse Jet Type. (Made by Radio Plane)
XJ34-BA-2	First Navy Model of Third Navy Turbo-Jet Type. (Experimental Status) (Made by Bell Aircraft)
YRJ35-LA-2	First Navy Model of Third AF Ram Jet Type. (Restricted Service Status) (Made by Lockheed Aircraft)

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