SCAVENGING OF

TWO-STROKE CYCLE DIESEL ENGINES

Paul H. Schweitzer

THE MACMILLAN COMPANY : NEW YORK 1949

PRINTED IN THE UNITED STATES OF AMERICA

A small part of this material has previously appeared in *Diesel Power and Diesel Transportation.* Copyright, 1942, 1943 by P. H. Schweitzer.

PREFACE

The purpose of this book is to make it *easier* to design *better* two-stroke cycle diesel engines.

The two-stroke cycle engine has gained significantly in economic importance during recent years. This growth has been most spectacular in automotive, rail, and marine propulsion, where weight is a pertinent factor. Improvement in fuel economy has been insignificant in the last two decades and future progress in this direction is strictly limited. Great strides have been made, however, in reducing engine weight and it is in this field that great opportunity for further advance still exists.

The continued efforts, that are being exerted to obtain the most power from an engine of a given size, justify a more thorough study of scavenging than has been accorded in the past. Guessing must give way to science, rules of thumb to formulas, and haphazard trials to methodical development. Too frequently in the past the customer's premises served as a proving ground for the manufacturer. To be competitive, future engines must no longer be over-dimensioned, and each one must be a finished product when it is placed in the market.

The difference between the two-stroke cycle engine and the four-stroke cycle engine lies principally in the method of scavenging. This book is limited to the treatment of scavenging, as skill in design of scavenging is considered the key to successful two-stroke cycle engines.

Literature on scavenging printed in the English language is almost nonexistent. Calculations found in foreign literature are as a rule too involved to be of real help to the practical designer; they are compositions of research men for research men, with an overabundance of Greek letters and differential calculus. The applicability of the findings is for the most part anything but obvious. As a result, the information contained in many German publications has been largely ignored even by engineers familiar with the German language.

Because of his long consulting activity, the author is keenly conscious of the difficulties which confront the designer of a two-stroke cycle engine. He also has sympathy for the development engineer whose job it is to bring the inferior performance of a prototype up to standard. It is primarily for these men, men in industry who are paid to produce engines, not theories, that this book has been written.

To accomplish this purpose, use of all higher mathematics is avoided except in appendixes which can be ignored by any reader who is interested only in the application of the derivations, rather than in the mathematics employed. By going one step further, complicated formulas have been frequently reduced to charts readily applicable to design problems. Finally, the application of the formulas and charts is illustrated with numerous numerical examples.

Tests for charging and scavenging performance described in research publications require elaborate equipment, and have been similarly ignored, except as research projects. The problem of evaluating scavenging, however, is encountered every time a new engine is developed. The author has endeavored to build a bridge between the research worker and the practical engineer by making available and simplifying advanced calculations for the use of the designer, and also by describing trouble-shooting types of tests for the benefit of the test man.

The first concept of this book was circulated in mimeographed form to a selected list of leading designers in 1940. A condensed version of it was published in twelve installments in *Diesel Power and Diesel Transportation* in 1942 and 1943. The proposed methods of calculation have been widely used since and have stood the test of time. Expanded and brought up-to-date, the same material is now presented as a book.

The author is indebted to the whole diesel industry for the data that made this book possible. General Motors Corporation, Fairbanks, Morse & Company, and Nordberg Manufacturing Company were generous with information upon which the descriptions of their engines are based. Of the foreign manufacturers, Sulzer Bros. and Petters, Ltd. were particularly helpful with data previously considered confidential.

So many individuals have helped in the preparation of this book that it is impossible to mention all of them. Most of the illustrations have been drawn by Kurt Goldman and Ray Strohm, who also have helped with the calculations. The mimeographed edition was proofread by P. M. Heldt, reviewed critically by Dr. G. W. Lewis and Otto Spiess, and encouragingly by J. W. Anderson, E. R. Brater, C. G. Curtis, T. B. Danckwortt, Emil Grieshaber, C. N. Guerasimoff, E. C. Magdeburger, R. P. McCulloch, G. F. Noltein, G. H. Sample, R. H. Schneider, F. H. Shepherd, L. T. Sherman, P. S. Vaughan, and others. The final manuscript was proofread by Dr. T. C. Tsu, R. L. Stanley, and S. A. Steiger.

P. H. SCHWEITZER

TABLE OF CONTENTS

SCAVENGING OF TWO-STROKE CYCLE DIESEL ENGINES

CHAPTER 1

PRESENT STATUS OF THE DIESEL ENGINE

1.1 The diesel engine, which was barely invented fifty years ago and which was practically unknown to the public twenty years ago, has made such tremendous strides in the last two decades that it is now being hailed as a preponderant prime mover of our age. Even before the war the yearly production of diesel horsepower was rising like a parabola due to its rapidly expanding employment in ships, locomotives, trucks, buses, tractors, and other mobile equipment. The war gave the diesel additional impetus as a result of its almost universal use in landing boats, submarines, and other light naval craft. In its heavy-duty form the diesel engine has been supplying power for generating electricity, manufacturing, pumping, irrigation, and a variety of other uses. Its more sanguine protagonists look forward to its general use in airplanes and automobiles.

In appraising the competitive position of the diesel engine against gasoline and steam engines, a deciding factor is its fuel economy. The thermal efficiency of the diesel engine exceeds that of all other heat-converting prime movers. Compared with a thermal efficiency of about 10 per cent in a reciprocating steam engine, 20 per cent in a modern steam turbine, 25 per cent in a carburetor engine, the brake thermal efficiency of the average diesel engine is about 33 per cent.

These are all rough average figures referring to full load. At part load the advantage of the diesel is still more pronounced. It is not exceptional to find a diesel-motored vehicle giving **twice the fuel mileage** of its gasoline-motored counterpart. Compared with the gasoline engine, the diesel enjoys the further advantage of burning a cheaper fuel, and one that involves less fire hazard.

The diesel engine owes its high thermal efficiency to its high compression ratio and high air-fuel ratio. The compression ratio of spark-ignition engines is limited by preignition and detonation while in the diesel a most economical ratio can be selected, which is around 14 : 1. The actual air-fuel ratio in a diesel engine varies from about 22 : 1 at full load to as lean as 80 : 1 at idling, or 50 to 450 per cent more than the theoretically correct air-fuel ratio. Both high compression ratio and high air-fuel ratio favor fuel economy. The latter becomes particularly significant at part load, when the diesel continues to take in a full charge of air, in contradistinction to the spark-ignition engine the air intake of which must be throttled in proportion to its reduced fuel intake in order to maintain an approximately constant mixture ratio to assure ignition.

DIESEL AND GASOLINE ENGINE COMPARED

1.2 The most significant difference between a carburetor engine and a compression-ignition engine lies in the nature of the charge. The carburetor engine deals with a homogeneous charge of air and fuel vapor of about 14.5 : 1 air-fuel ratio irrespective of load. In the diesel cylinder there is a heterogeneous or stratified charge, the mixture ratio of which varies from point to point, from very lean to very rich. The total air-fuel ratio is from 22 : 1 to 30 : 1 at full load, and may go beyond 80 : 1 at no load.

A stratified charge with varying mixture ratio has advantages and disadvantages; it is responsible for the good fuel economy of the diesel at part load. The disadvantage lies in the fact that either air or fuel must be wasted because during the short time available all fuel molecules cannot be expected to find all oxygen molecules. If an engine runs at 1200 rpm (many run faster), the whole expansion stroke lasts only 1/40 second. To burn the fuel efficiently, mixing should be complete in the early part of the expansion stroke. But because injection and mixing cannot start much before the beginning of the stroke, there is a time interval of the order of 1/100 second. Air and fuel cannot mix completely in such short time, even if the fuel spray is perfectly atomized and distributed.

1.3 Diesel Combustion.

Ordinary diesel fuel is a mixture of hydrocarbons. Its components are various hydrocarbon molecules, and the fuel ultimately consists of nothing but carbon and hydrogen with some impurities. The carbon-hydrogen ratio may vary between 6 : 1 and 8 : 1 (by weight), depending on the source and processing of the fuel. One pound of hydrogen requires $32/4.03 = 7.94$ pounds of oxygen for complete combustion and one pound of carbon requires $32/12 = 2.67$ pounds of oxygen. One pound of atmospheric air contains 0.232 pound of oxygen and thus one pound of hydrogen requires $7.94/0.232 = 34.2$ pounds of air, and one pound of carbon requires $2.67/0.232 = 11.5$ pounds of air for complete combustion.

One pound of an average diesel fuel that contains one weight of hydrogen to seven weights of carbon requires for complete combustion $1/8(34.2) + 7/8(11.5) = 14.3$ pounds of air. This may safely be rounded off to 14.5 pounds.

Therefore, if at the time of ignition the combustion chamber contains one pound of diesel fuel and 14.5 pounds of air, completely mixed so that each fuel particle is completely surrounded with the corresponding amount of oxygen particles, the charge burns completely, with neither fuel nor oxygen left. The combustion products are carbon dioxide and water vapor.

However, if the air-fuel ratio is chemically correct but the mixture is not perfect, the fuel particles fail to find the corresponding oxygen molecules in the short time available, and at the end of the power stroke both unburnt fuel and oxygen remain.

In a gasoline engine, the carburetor effects the mixing of fuel and air outside the cylinder, and the entering charge represents a more or less homogeneous mixture. In a diesel engine we have a heterogeneous or a stratified mixture. A gasoline engine can burn a chemically correct mixture without difficulty; a diesel engine cannot.

1.4 Excess Air.

In order to facilitate burning of the fuel, the fuel injected into the cylinder is reduced by about one-third below what would be permissible for a chemically correct mixture. Instead of using the chemically correct air-fuel ratio of 14.5 : 1, at least 50 per cent excess air is customarily used, which boosts the air-fuel ratio to $1.5 \times 14.5 = 21.8$. In this way air is wasted but fuel is saved.

Thus it is evident that the necessity for mixing fuel and air in a diesel-engine cylinder in an extremely short time means supplying an excess of air above the chemically correct ratio. The excess usually varies between 50 and 100 per cent at full load, depending on how well the engine design promotes quick and thorough mixing. If air and fuel mix well and promptly, excess required is small; if the fuel droplets fail to reach the far and hidden corners and air pockets in the cylinder, or if they fail to mix intimately with the air encountered, the necessary excess is large. In any case, less than minimum excess air means unburnt fuel, which results in high fuel consumption, smoke, sooting, and overheating.

This unavoidable excess air reduces the power that can be obtained from a diesel cylinder. If the engine cannot operate without, for example, 50 per cent excess air, it means that one-third less fuel can be burned in a diesel cylinder than in a carburetor-engine cylinder of the same size. Because both gasoline and diesel fuel have roughly the same heat energy per pound (about 19,000 Btu), the power output of a diesel runs about one-third less than that of a gasoline engine of the same size, speed, general design, and efficiency.

Diesel efficiency, however, is considerably better than that of the gasoline engine, which finds expression in 20 to 30 per cent lower specific fuel consumption. This reduces the power handicap of the diesel to $5/4 \times 2/3 = 10/12$, which means that compared with the 100 psi brake mean effective pressure of the gasoline engine the brake mean effective pressure of an unsupercharged diesel is 83 psi. A good part of the efforts of the diesel-engine designers in the last 25 years was concentrated on increasing the diesel bmep.

Because excess air is mainly responsible for the lower bmep in the diesel, it is natural that considerable effort has been exerted to reduce this margin of excess air. These efforts achieved some success, but the path was not an unbroken climb but rather one marked with ups and downs.

1.5 Evolution of the Diesel Engine.

The German engineer, Dr. Rudolph Diesel, invented in 1893 the engine that bears his name, and the first successful engine, a 25-hp unit, was completed by him in 1897. The first diesel to operate in commercial service was built in this country by the Busch-Sulzer Bros. Diesel Engine Co. in 1898. These machines, as well as thousands that followed in the next 25 years, were of the air-injection type. High-pressure air produced by a compressor blasted the fuel oil into the cylinder. While the air-injection engine was a bulky affair, weighing a couple of hundred pounds per horsepower, the high-pressure air proved an effective mixer and excess air ran relatively low.

In 1910, the Englishman McKechnie developed the airless- or solid-injection engine which 20 years later has become the dominant type. This design eliminated the bulky and troublesome air compressor, the fuel being injected under pressure produced by a small high-precision fuel-injection pump. Dispersion troubles were, however, aggravated. The tiny spray or sprays failed to mix intimately with cylinder air, making a smoky exhaust that could be cleared only by increasing excess air. This cut down power output.

Gradual improvement led to modern injectors built to tolerances closer than those for almost any other mechanical device. These produce finely atomized sprays carefully distributed over the air space. Combustion chambers are shaped to fit the spray. By these means, mixing was improved and the excess air margin reduced.

1.6 Spray Penetration.

With extremely fine sprays another trouble is encountered — lack of penetration. In atmospheric air a high-velocity jet reaches several yards easily, but in 500-psi air it cannot penetrate more than a few inches. In an incredibly short time the oil globules that start out like rockets stop dead and fall down like shot birds, devoid of energy. No further increase of injection pressure can improve the penetrative power of the spray. On the contrary, increasing the injection pressure makes the penetration less because the globule size becomes so small that penetrating power vanishes. The plain fact is that fine atomization and deep penetration are contradictory requirements, yet for good mixing both are needed; this causes difficulties in large cylinders.

1.7 Turbulence.

Another approach to the objective of good mixing lies in the use of air turbulence. Instead of making the spray go to the air, the Englishman Ackroyd conceived the idea of making the air go to the spray. Designers have since devised a variety of means of imparting to the combustion air a motion that would enable it to seek out the oil particles rather than wait passively for them.

One way consists of inducing the air to perform a rotary motion in the cylinder by admitting it tangentially. Another is a separate *swirl chamber* into which the piston squeezes the air, which swirls around while the fuel is injected. Still another answer lies in starting out with a preliminary explosion in a *precombustion chamber*, depending on the ensuing agitation to mix the cylinder air with the fuel. A similar effect is obtained by allowing some of the combusion air to accumulate in an internal *air-storage cell*, and to blow out from it at the opportune moment to feed the combustion in the same manner as the oxygen jet of an oxyacetylene burner.

Thus in recent years the classical open-chamber engine of Diesel gained such companions as the swirl chamber, the precombustion chamber, the air-cell, and the energy-cell engines, each contributing in its own way to the ultimate goal of the diesel engineer — perfect mixing of air and fuel in the shortest possible time.

Still, excess air cannot be dispensed with. In some designs it can be cut to about 25 per cent if we are willing to pay the price for it. The price is increased fuel consumption because of higher mechanical and thermal losses. Any scheme that employs high turbulence for mixing air with fuel must provide the high-velocity air, and this requires power. Turbulent air also loses more heat to the cylinder walls and leaves less for useful work. As a result, *divided-chamber* engines may show about 10 per cent higher fuel consumption, and the total power output for a given cylinder size runs about the same as in the classical open chamber. Nevertheless, the divided-chamber engine proved popular in small sizes because of its better speed flexibility and because it avoids excessively small nozzle orifices. Coarser spray from a single-hole nozzle proves adequate when air turbulence performs the major duty of mixing.

1.8 Increasing Output.

It is not the amount of fuel that limits the engine power, but the amount of oxygen that can be squeezed into the cylinders every minute. Because air is the only practical source of oxygen, ability to take in air measures the success of the engine. Without increasing engine size three ways are open materially to increase air-taking capacity: (1) fill cylinders more often by increasing rotative speed; (2) squeeze more air into the same cylinder space by putting it under pressure, or supercharging; (3) fill cylinders more often by cutting out idle strokes, which means using the *two-cycle* principle. In the course of the development of the diesel engine none of these paths has been left unexplored.

1.9 Engine Speed.

Early diesels ran like steam engines, 100–200 rpm. The submarine gave the first incentive to step up speed gradually to 400, 600, and 800 rpm. The modern automotive diesel engine runs up to 2000–3000 rpm, but still falls short of gasoline engine speeds.

Two factors stand in the way of further increasing the speed of the diesel engine. One of them is that which is responsible for excess air — the mixing process. To insure clean combustion, the air and fuel must be mixed in the early part of the power stroke of the piston. At 2400 revolutions per minute, 30 degrees crank angle corresponds to 1/480 second, and to mix air and fuel intimately in such a short period of time is difficult. The carburetor engine has no such problem because the fuel and air are mixed in the carburetor outside of the cylinder, and a uniform mixture is fed into the cylinder.

The other factor that has a bearing on speeding up the diesel engine is the *ignition lag.*

1.10 Ignition Lag.

Diesel ignition is frequently described as taking place instantaneously upon injecting the fuel into the cylinder, the condition being that the temperature of the cylinder air exceeds the self-ignition temperature of the fuel. There is, however, a delay of one or more thousandths of a second. It might be thought that a delay of that magnitude could not be serious, but at 2400 revolutions per minute, 0.002 second corresponds to 28.8 degrees crank rotation. That is so long that it is likely to prevent ignition altogether and cause *missing.* Even before this happens, combustion knock occurs. It may reach such proportions that it is not only a nuisance, but becomes intolerable.

The longer the fuel hesitates to ignite, the faster and more violently it burns when it lets go. That becomes evident when it is realized that fuel accumulates in the cylinder during the ignition delay period. The injection period may last 10 to 30 degrees. If the entire fuel charge or a major portion of it is already dispersed in the cylinder and vaporized when ignition occurs, all of that fuel ignites practically instantaneously and explodes with a violence as if thousands of spark plugs had set it off at once.

The diesel combustion knock is similar to detonation in gasoline engines and it is as destructive if not checked at an incipient stage. Yet significant differences exist.

Almost every factor that causes a gasoline engine to knock is likely to make a diesel engine run smoother, and what is anti-knock in a gasoline engine is pro-knock in a diesel. High compression ratio is considered a primary cause of gasoline engine knock, while a rough going diesel can be made smooth simply by raising its compression ratio. High temperatures of intake air or cylinders make a gasoline engine knock, but have the opposite effect in a diesel. Anti-knock dopes for gasoline, like tetraethyl lead, are pro-knock in the diesel, while ethylnitrate is the reverse. Fuels of paraffin base as they come from eastern crudes, consisting mostly of saturated straight-chain hydrocarbons, are the worst knocking fuels in spark ignition engines and the smoothest, most desirable fuels for diesel operation. The aromatics of the west coast and cracked fuels, on the other hand, burn roughly in diesel engines, but are premium fuels for carburetor engines. High altitude or low intake air density causes a diesel engine to knock, while supercharging suppresses the knock. With carburetor engines the reverse is true.

The behaviors of compression-ignition and spark-ignition engines with regard to knocking are so completely opposite that doubt has been raised as to the identity of the two phenomena. Nevertheless, in essence the knocking process is always the same. In both the spark-ignition and the diesel engine the knock is accompanied by a very fast burning and an excessive rate of pressure rise. While the critical rate varies somewhat with cylinder size and other factors, ordinarily both gasoline and diesel engines run smoothly if the maximum rate of pressure rise is less than 30 pounds per square inch per degree of crank rotation. Both are likely to knock if the pressure rise is above

50 pounds per degree and are almost certain to knock if it exceeds 100 pounds per degree. It has also been established that in both types of engines the knock is preceded by spontaneous ignition of a portion of the combustible charge in the cylinder. But here the similarity ends.

In a spark-ignition engine the rapid pressure rise takes place and the knock occurs at the last part of the combustion. It is the *last gas* to burn that knocks. The first portion of gas ignited by an electric spark certainly could not knock, because only the gas adjacent to the spark plug ignites, which is of minute quantity. The propagation of combustion is a gradual process for a while. The flame front expands and in so doing compresses the unburnt portion of the mixture. Finally a point may be reached at which the unburnt mixture is compressed to its self-ignition temperature. Then this charge ignites in its entire mass causing a very rapid pressure rise and a sharp knock frequently called *detonation.*

In a diesel engine, on the contrary, the combustion **begins** with spontaneous ignition. Here this is the **"legal"** method of ignition. But self-ignition need not necessarily cause detonation, as thousands of smooth running diesel engines testify. The determining factor is how much fuel ignites simultaneously. The greater the amount of fuel igniting in the cylinder simultaneously, the steeper is the pressure rise and the more violent is the knock.

It is at this point that ignition lag enters the picture. The fuel injected does not ignite instantaneously. It takes one to four milliseconds to ignite the fuel charge already in the cylinder. The length of ignition lag depends on the quality of the fuel and temperature and pressure in the cylinder. The longer the ignition lag, the more fuel accumulates in the cylinder before ignition. When finally ignition occurs nearly all of this accumulated fuel ignites simultaneously. Therefore, the longer the ignition lag the more severe the knock.

This difference in ignition characteristics explains the divergent behaviors of gasoline and diesel engines. All factors which reduce ignition lag have, therefore, anti-knock effect in a diesel engine. High compression ratio, high intake air temperature, and pressure all accelerate ignition, and therefore shorten ignition lag and suppress knock. By the same token these very factors speed up combustion in the gasoline engine, produce spontaneous ignition in the last portion of the charge, and cause detonation. High octane number fuels are slower burning and thereby prevent detonation. High octane fuels would do very poorly in diesel engines where high cetane fuels are desired. Any fuel that has a high octane number has a low cetane number and vice versa.

1.11 Supercharging.

The conventional way of charging the cylinder with air is by suction produced by the piston. If the air is aspirated into the cylinder by the piston, the amount of air charge is limited; it can never exceed the displacement volume. However, if the charge is forced into the cylinder under pressure, the amount of the charge can exceed the displacement volume.

Supercharging applied to spark-ignition engines was found to promote detonation and its use has been largely restricted to aeroengines at high altitudes.

The diesel does not object to the increase of the compression pressure in the same way the spark-ignition engine does. Raising of the compression pressure does not cause combustion knock in a diesel engine; if anything, it makes the combustion smoother.

Two methods of supercharging are being used — the mechanically driven supercharger and the turboblower.

The mechanically driven supercharger may be a reciprocating, a rotary, or a centrifugal blower driven by the engine or by a separate motor. The blower has capacity such that it fills the cylinder with air to a pressure that exceeds atmospheric pressure by several pounds per square inch.

If the cylinder charge is, for example, 25 per cent more than that of the unsupercharged engine, the gross power output is also 25 per cent more, roughly speaking.

The economical limit of mechanical supercharging is around 30 per cent, giving engine bmep of a little over 100 psi. If this limit is exceeded, the blower drive absorbs relatively too much power and the law of diminishing returns sets in. Fuel economy also becomes poorer at high supercharge. The supercharge pressure is usually between 3 and 5 psi in commercial mechanically supercharged engines.

It occurred to the French engineer, Rateau, to use a turbine driven by the exhaust gas to drive a centrifugal blower to supercharge an aircraft engine. Sanford A. Moss, of the General Electric Company, perfected this method, which has been used extensively in our combat planes for high altitude supercharging. The Swiss engineer, Alfred J. Buchi, applied the idea to diesel engines.

The turbine of the Buchi supercharger is driven by the exhaust gases that discharge from the exhaust valves at an elevated pressure and temperature. This is accomplished without appreciably raising the exhaust back pressure. Although the cylinder release pressure is high, the mean pressure in the exhaust pipe is seldom more than 2 psig. By the proper timing and combining of the exhaust impulses of a multicylinder engine, Buchi conveys a good part of their energy to the turbine wheel without creating an undesirable back pressure in the engine. The exhaust gas at a temperature of about 1000 F impinges upon the turbine blades made of special heat resisting alloy. The turbine drives a centrifugal blower mounted on the same shaft, which feeds the engine with air at 3 to 6 psig pressure.

To get the full benefit of supercharging, the cylinders must be scavenged of residual gases between cycles. This is accomplished by *valve-overlap.* The exhaust valve is kept open for a certain length of time after the inlet valve opens, and the blast of fresh air supplied by the blower blows out the foul gases and fills the cylinder with pure fresh air. Because several cylinders are open to the exhaust manifold simultaneously, the grouping of the exhaust ports and the sequence and timing of the exhaust surges must be worked carefully to avoid the bucking of one cylinder by the exhaust of another cylinder.

A power increase of 50 per cent giving 130 psi bmep is not uncommon with Buchi supercharging with no increase in brake specific fuel consumption.

The more recently introduced constant pressure turbocharging operates with considerable exhaust back pressure to the engine, which is overcome by correspondingly higher supercharge pressure.

1.12 Two-Stroke Cycle.

Carburetor engines operate usually on the four-stroke cycle. One stroke is used to fill the cylinder with fresh charge, the next to compress that charge. At the end of the compression stroke ignition takes place, followed by the expansion or power stroke. Finally, the fourth stroke is used to push the burnt gases out of the cylinder. Since two strokes constitute one engine revolution, the cylinder is filled up only at every second revolution. From the standpoint of feeding air to the engine, every other revolution is wasted.

It occurred to the British engineer, Dugald Clerk, as early as 1878, to build an engine that would utilize every engine revolution for filling the cylinder, thereby inventing the two-stroke cycle engine. In a four-stroke cycle engine the charge is sucked into the cylinder by the piston. In a two-stroke cycle engine there is no suction stroke and the charge must be pressed into the cylinder by a pump or blower. The addition of a low-pressure air compressor or blower is a matter of small consequence compared to doubling the number of power strokes, yet the two-stroke cycle engine made little headway before the advent of the diesel engine. The reason is simple.

In a two-stroke cycle engine both exhaust and intake must take place in a short time while the piston is at, or near, bottom center. The two processes overlap; in fact, the fresh charge is used to scavenge the cylinder of the burnt gases. In scavenging with a mixture of fuel and air, fuel is wasted. If the mixing of fuel and air is performed outside the engine, in a carburetor, this can hardly be avoided. In consequence, two-stroke cycle gasoline engines have been used only in small sizes, as in outboard engines for small motorboats, etc., where fuel economy is of secondary importance.

In a diesel engine where pure air is charged to the cylinder, scavenging can be done with air without wasting fuel. This makes the two-stroke cycle for the diesel practicable. Indeed, the most significant recent advances in diesel engineering pertain to the two-stroke cycle engine.

In this country, the two-stroke cycle diesel engine has exceeded the production of the four-stroke cycle types for some time and during the war the two-stroke cycle emerged as the dominant type.

The brake mean effective pressure of advanced types of two-stroke cycle engines is equal or higher than that of unsupercharged four-stroke cycle engines. True, supercharging increases the power output of four-stroke cycle engines by 30 to 50 per cent by adding only a small fraction of that percentage to the weight or space requirement. However, the two-stroke cycle engine at the same rpm **doubles the power** output by doubling the number of power strokes. Therefore, where weight or space is at a premium, the two-stroke cycle can be expected to be in demand. Because of the close relation between weight and cost, even in applications where weight as such is secondary, the two stroke cycle engine is likely to compete favorably with four-stroke cycle engines, supercharged or unsupercharged, on the basis of first cost.

Most of the older two-stroke cycle diesel engines are of the crankcase-scavenged type. The underside of the piston together with the cylinder and crankcase constitute the scavenging pump. Although its power output is low, approximately 40 psi bmep, the crankcase-scavenged engine makes a strong appeal to the user by reason of its simplicity. The absence of cylinder head valves and a separate scavenging pump permits low production and maintenance costs.

The crankcase-scavenged engine has been a success in sizes up to 100 hp per cylinder but in larger sizes it has proved to be uneconomical because of its low specific output and poorer economy of fuel and lubricating oil. Addition of a separate scavenge pump, rotary or reciprocating, made the simple port-scavenged diesel a suitable prime mover for stationary power plants and seagoing vessels. This type of engine still dominates the field in the several hundred horsepower per cylinder category.

The latest arrival in two-stroke is the *uniflow* type. Uniflow or straight-through scavenging, where the air is admitted at one end of the cylinder and the exhaust escapes at the other end, is inherently more efficient than cross or loop scavenging, where the air must make several turns in going through the cylinder.

In some uniflow engines the air is supplied by a rotary blower to ports located around the bottom of the cylinder, and the exhaust escapes through poppet valves at the top of the cylinder. Brake mean effective pressures of 85 psi are common with this arrangement, and engine weight in one production model for marine engines has been reduced to the incredibly low figure of 4 pounds per horsepower.

Another uniflow type of two-stroke cycle is the opposed-piston engine, originally invented by the German professor Junkers. Two pistons move in the same cylinder, opposite each other, and the combustion takes place in between. Fresh air is admitted through a row of ports at one end of the cylinder, and exhaust takes place through another row of ports at the other end. One piston controls the inlet and the other piston controls the exhaust. As a result the long cylinder is effectively scavenged. In brake mean effective pressure the opposed-piston engine is at par with the exhaust poppet valve engine.

The third uniflow arrangement is by sleeve or slide valves. The advantage of the two-stroke cycle sleeve-valve engine lies in the absence of poppet valves, permitting the use of simple symmetrical cylinder heads. The cylinders are short and do not require double crankshaft or long outside connecting rods as do opposed-piston engines. Its uniflow scavenging with rotary swirl permits high power output, but it presents a problem of cooling and lubrication because of the sleeve.

1.13 Two-Stroke Cycle Supercharging.

In a certain sense practically all two-stroke cycle engines are supercharged. Instead of aspirating air in, it is forced into the cylinder by a pump or blower. Usually, however, the amount of air so forced into the cylinder does not exceed the amount which the same cylinder would receive in four-stroke cycle by normal *breathing*. This is especially true of so-called *symmetrically scavenged* engines, where the exhaust ports are open for a time interval after the inlet ports are closed. Since the exhaust pipe is in free communication with the atmosphere, it is obvious that whatever supercharge has been built up in the cylinder during the early part of the charging process would largely be lost during that interval. This circumstance makes supercharging of *common* types of symmetrically scavenged engines impractical. In consequence, symmetrical scavenge has been abandoned for high output engines.

Cam-operated valves offer a simple means for overcoming such a difficulty. By opening the exhaust valves early and by closing them also early (unsymmetrically around bottom dead center), an adequate blowdown period is provided to discharge the burnt gases before the inlet ports open, together with a supercharge period during which the exhaust is already closed but the inlet is still open. Thus the cylinder can be filled to higher than exhaust pressure. A similar effect is obtained in opposed-piston engines by giving a phase advance to the exhaust crank, and in single-piston-scavenged engines by the use of supercharge valves.

The normal supercharge of two-stroke cycle engines seldom exceeds 5 psig and the economical limit is usually between 2 to 4 psig. The amount of air so delivered exceeds the cylinder displacement by 30 to 50 per cent. Nevertheless, since most of that extra air is short-circuited between the inlet and exhaust, the so-called *supercharged* two-stroke cycle engine seldom traps more than a cylinder volume of normal temperature-pressure air for combustion.

The natural trend toward increasing engine output by supercharging meets its limitation in the case of mechanically supercharged two-stroke cycle engines at a charging pressure where the power absorbed by the blower begins to exceed the power increase of the engine.

The power required to furnish the charging air increases faster than the power output because in addition to the air burnt, a considerable amount of excess air and short-circuited air (which blows through while both inlet and exhaust are open) must be compressed and introduced into the cylinder.

1.14 High Supercharge.

For moderately high supercharge an arrangement has been developed by Sulzer Bros. in Switzerland, by which low-pressure air is used for scavenging and high-pressure air is admitted only after the exhaust ports are closed. In this way the short-circuiting of high-pressure air is avoided. This arrangement permitted charging air pressures up to 6 psig and a power increase up to 30 per cent, but it was not practical to go beyond these limits. The reason is that as long as the energy in the high-pressure exhaust remains unutilized, a substantial increase of the bmep, by boosting the inlet air pressure, involves an appreciable loss in the exhaust blowdown process. In other words, it is impractical to raise the inlet pressure much without also raising the exhaust pressure, which in turn calls for a turbine to utilize that pressure.

Application of the Buchi system to the two-stroke cycle engine has engaged the attention of several inventors. Compared to its four-stroke cycle version, the problem is aggravated by two circumstances: (1) The two-stroke cycle engine requires more scavenging air and therefore greater turbine output to supply the increased amount. At the same time the exhaust temperature is lower because of dilution of the exhaust gas with the short-circuited air, which causes a decrease in the exhaust energy available to drive the turbine. (2) In the two-stroke cycle engine the charging of the cylinder requires a positive pressure difference between inlet and exhaust. During starting or at low load operation the turboblower will not provide the necessary charging pressure; therefore the machine is unable to start.

These difficulties have been partly overcome by an arrangement proposed by Curtis as early as 1924. To the turboblower a mechanically driven blower is added, which is gear-driven from the crankshaft. The turboblower forms the first stage and the mechanical blower the second stage of the compression. At start or at low load the turboblower does not contribute to the compression of the charge, but the air furnished by the mechanical blower alone is sufficient to operate the engine. At heavy load the effort largely shifts to the exhaust driven blower with the result that a handsome boost in power output is obtained without the penalty of increased specific fuel consumption. This scheme was used during the war in the German Junkers aircraft diesels. Work along this line is still in progress.

1.15 Power Gas Process.

The other method of high supercharging of two-stroke cycle engines is known as the *power gas* process and has aroused a great deal of interest lately. In this scheme the charging compressor, usually of the reciprocating type, is driven directly by the engine. The engine is supercharged to, for example, 70 psia so that the compressor absorbs the entire engine output. The engine bmep is thereby raised to perhaps 200 psi. The hot exhaust gas drives a turbine which yields all of the useful power. In this arrangement the turbine becomes the main power unit, and the engine serves merely as a combustion chamber for the gas turbine, furnishing also its own charging air on the side.

One advantage of the power gas process is the high power output obtained per cubic inch of cylinder displacement. Another advantage is the absence of mechanical connection between engine and turbine. The only connection is a pipe which leads the engine exhaust into the turbine. This makes it possible to connect any number of engine units to the turbine and so to obtain more horsepower on a single shaft than is possible with straight diesel engines. This makes the power gas process attractive for ship propulsion.

The first experiments with machinery of this type were carried out from 1927 to 1929 by Goetawerken in Sweden. A few years later such power plants were installed in ships.

One type of power gas generator that has aroused a great deal of attention is the free-piston engine compressor. In this machine, which also was originated by Junkers, the compressor piston is integrally connected to the engine piston, both reciprocating together without a crankshaft between them. Pescara in France applied the free-piston machine for power gas generation. Between

1937 and 1939 he built several of these machines in various sizes. The engines were of the opposed-piston type. Integral with the engine were the cylinders and pistons of the multistage air compressor, the whole air output of which was charged into the engine through one row of ports at one end. The exhaust gas discharged through ports at the other end was conducted into a turbine which was to deliver the useful power.

Subsequently the power gas process, which may be considered a special case of two-stroke supercharging, was thoroughly studied by Sulzer Bros. in Switzerland in both variants, i.e., with crankshaft and with free-piston power gas generators. A number of these power plants have been built and have given good performance. The reported fuel consumption was 0.35 pound per bhp-hr which is at least on a par with the fuel economy of the straight diesel and considerably better than the fuel economy of the straight gas turbine.

Prompted by the favorable results obtained abroad, several American firms are now engaged in development work on compound power plants combining the diesel engine with the gas turbine.

CHAPTER 2

PECULIARITIES OF THE TWO-STROKE CYCLE ENGINE

2.1 The performance of a two-stroke cycle engine depends to a remarkable degree on the art of scavenging and on the port design. Data of 121 modern commercial two-stroke cycle diesel engines [Danckwortt, 1939] show that the brake mean effective pressures of the various types are as follows:

	POUNDS PER SQUARE INCH	
	RANGE	AVERAGE
Opposed-piston engines	70–120	86
Uniflow engines other than opposed-piston	60–98	80
Separately scavenged engines other than uniflow	53–70	61.7
Step-piston engines	40–56	48
Crosshead engines other than step-piston	34–48	41
Crankcase-scavenged engines	28–40	37.7

The extreme values of the averages for individual types are in the ratio of 2.3 to 1, which far exceeds the range of commercial four-stroke cycle engines. Even the differences between engines of the same type are greater than can be explained by ordinary differences in workmanship and design. In most instances incorrect porting can be blamed for inferior performances.

While chances for defective design are numerous, there are errors which repeat so frequently that they sould be singled out for special attention:

1. *Short-circuiting.* The air admitted passes directly to the exhaust ports without displacing the burnt products.
2. *Insufficient exhaust lead.* There is not enough time for the burnt gases to escape through the exhaust ports and for the cylinder pressure to drop sufficiently before the inlet opens, and the burnt gases therefore enter the inlet ports in large quantities.
3. *Too-late closure of the inlet ports.* If the inlet ports are left open too long, the flow through them becomes reversed and the cylinder loses air near the end of the inlet period.
4. *Excessive exhaust-port (valve) area.* Too-large exhaust area after bottom center prevents pressure build-up in the charge before exhaust closure, which is essential to effective supercharging.

2.2 Scavenging Methods.

The first two-stroke cycle engines had inlet valves in the cylinder head, but Dugald Clerk's [Clerk, 1909] early engines already had ports for exhaust. Close to the end of the expansion stroke the piston uncovered a wide annular channel through which the exhaust gas quickly escaped. The

English engine builders, Day and Son, pioneered in 1889 the valveless engine in which the piston controls both intake and exhaust by covering and uncovering two sets of ports located near the bottom dead center position of the cylinder periphery. The absence of valves represented such an attractive simplification in engine construction that until recently most of the two-stroke cycle engines were of the valveless port-scavenged type.

Because the ports are controlled by the piston only, the points of port opening and closing are symmetrical with respect to bottom center (see Fig. 2–1). For instance, if the inlet and exhaust

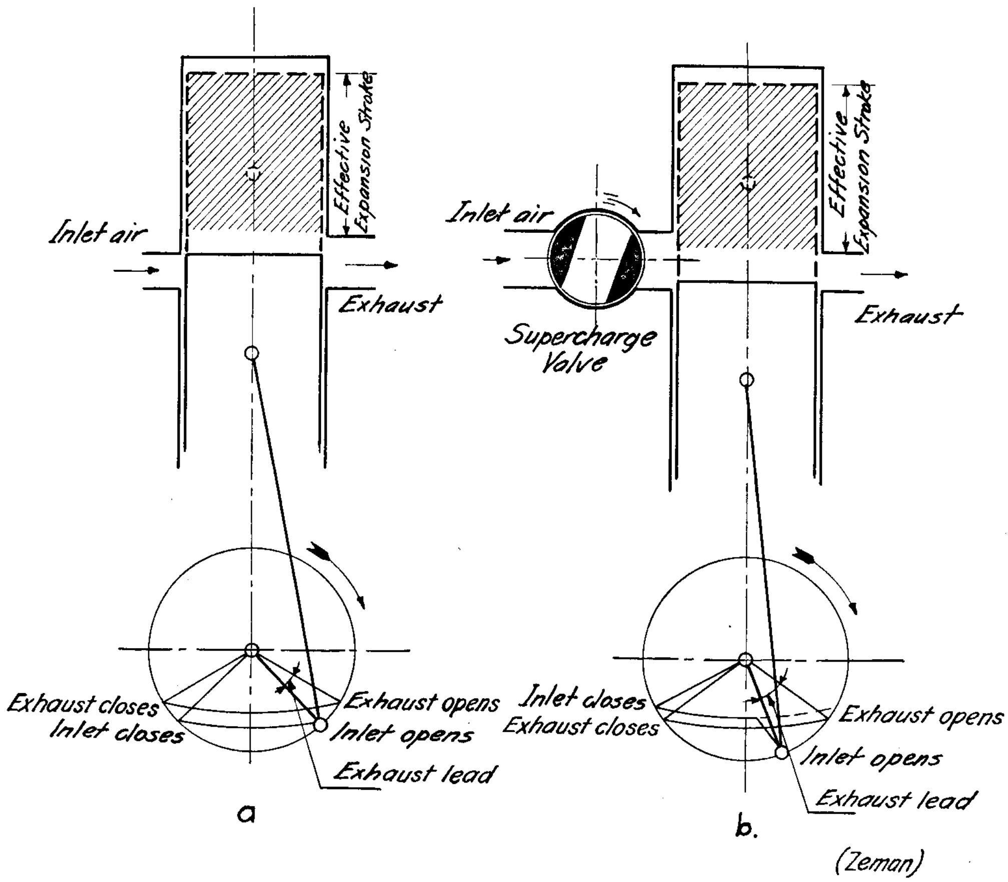

Fig. 2-1. *a.* Symmetrical and *b.* Unsymmetrical Scavenging. Note the early exhaust opening and late inlet closing made possible with supercharge valve. (J. Zeman, *Zweitakt Dieselmaschinen*, Springer, 1935. Copyright, 1935 by Julius Springer in Vienna.)

ports open 50 and 65 degrees before bottom center, respectively, they must close 65 and 50 degrees after bottom center. The exhaust ports must, of course, be higher than the inlet ports, so that the exhaust gas may escape through the exhaust ports instead of blowing into the inlet ports. This necessitates an exhaust lead of so many crank degrees, as shown in Fig. 2–1*a*. The inevitable consequence is that on the upstroke the exhaust ports close later than the inlet ports. If the exhaust lead is 15 crank degrees, the exhaust closure lags the inlet closure also by 15 crank degrees. The result is that no matter what pressure the cylinder is charged with through the inlet ports, for a certain period the inlet ports are closed while the exhaust ports are still open, permitting air of higher than atmospheric pressure to escape. No supercharge of any consequence is possible with symmetrical scavenge. Furthermore, the effective piston displacement is reduced to the swept volume above the exhaust ports as shown by the shaded area in Fig. 2–1*a*. This method of scavenging, nevertheless, continues to be popular because of its simplicity.

In order to increase the air charge, Sulzer in 1912 pioneered unsymmetrical scavenging, shown diagrammatically in Fig. 2–1*b*. Here the inlet port is made higher than the exhaust port, but an auxiliary valve keeps it closed until after the exhaust port has been uncovered and the cylinder pressure has dropped below the scavenge pressure. On the upstroke the inlet port can remain open after the exhaust is closed, making supercharging possible. The supercharge valve may be mechanically actuated or it may be an automatic valve; it may be in the intake port (Sulzer) or in the exhaust port (M.A.N.)*; but in each case it effects an appreciable pressure rise during the early part of the compression stroke. Such a pressure rise is impossible with symmetrical scavenging. A supercharge valve may not produce an actual supercharge, that is, a charge exceeding that corresponding to the total piston displacement, but for the sake of brevity we shall use the word *supercharge* for a charge which exceeds that obtainable with symmetrical scavenging. A supercharge valve such as that shown in Fig. 2–1*b* is only one of several possible arrangements for obtaining unsymmetrical scavenging. Others are poppet-valve engines, opposed-piston engines, and sleeve- or slide-valve engines. Fig. 2–2 shows in comparison the timing diagrams of a symmetrically and an unsymmetrically scavenged engine.

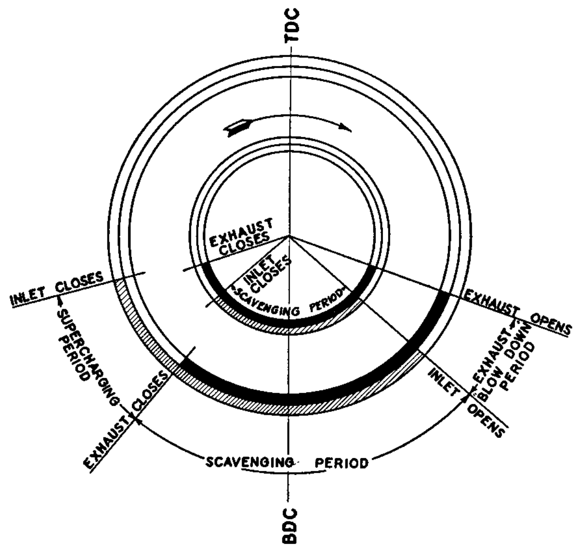

Fig. 2-2. Timing Diagrams. Inner diagram, symmetrical scavenge. Outer diagram, unsymmetrical scavenge.

Four-stroke cycle engines obtain their charges by aspiration during the suction strokes. The two-stroke cycle engine has no suction stroke and the air must be pumped in. In the crankcase-scavenged engine the engine piston sucks the air into the crankcase and pumps it from there into the cylinder (Fig. 2–3).

In the separately scavenged engine a separate pump or blower furnishes the air (Fig. 2–4). In the crankcase-scavenged engine the air charge cannot be as large as the piston displacement; in the separately scavenged engine there is no such limitation.

A third classification of two-stroke cycle engines is based on the air flow. The most common arrangement is cross scavenging, illustrated in Fig. 2–5 at *a*. The incoming air is directed upward, and then down to force out the exhaust gases through the oppositely located exhaust ports. With loop or reverse scavenging, the fresh air first sweeps across the piston top, then up, down, and out through the exhaust. In the M.A.N. type of loop scavenge (Fig. 2–5*b*) the exhaust and inlet ports are on the same side, the exhaust above the inlet. In the Schnuerle type, Fig. 2–5*c*, the ports are side by side. The Curtis type of scavenging (Fig. 2–5*d*) is similar to the Schnuerle type, except that upwardly directed inlet ports are placed also opposite the exhaust ports.

The most perfect method of scavenging is the uniflow method, where the fresh air charge is admitted at one end of the cylinder and the exhaust escapes at the other end. The air flow is from

* Abbreviation for Maschinenfabrik Augsburg Nuernberg.

end to end, and little short-circuiting between the intake and exhaust openings is possible. The three available arrangements for uniflow scavenging are shown in Fig. 2–6. A poppet valve is used at *a* to admit the inlet air or for the exhaust, as the case may be. In *b* the inlet and exhaust ports

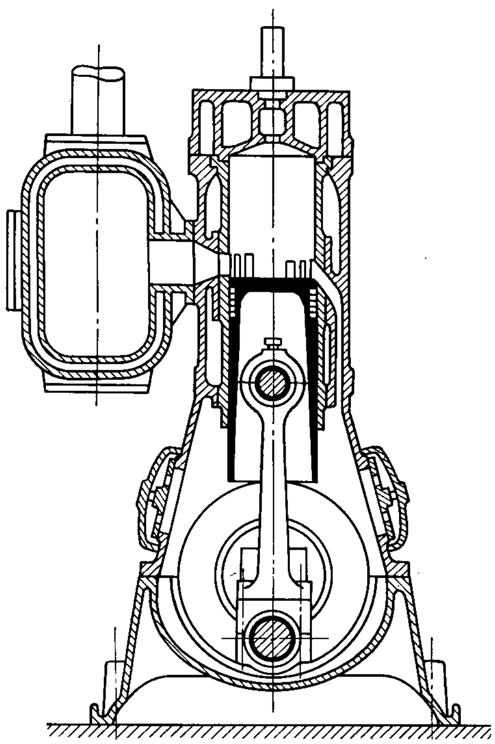

Fig. 2-3. Crankcase-Scavenged Engine (Sulzer).

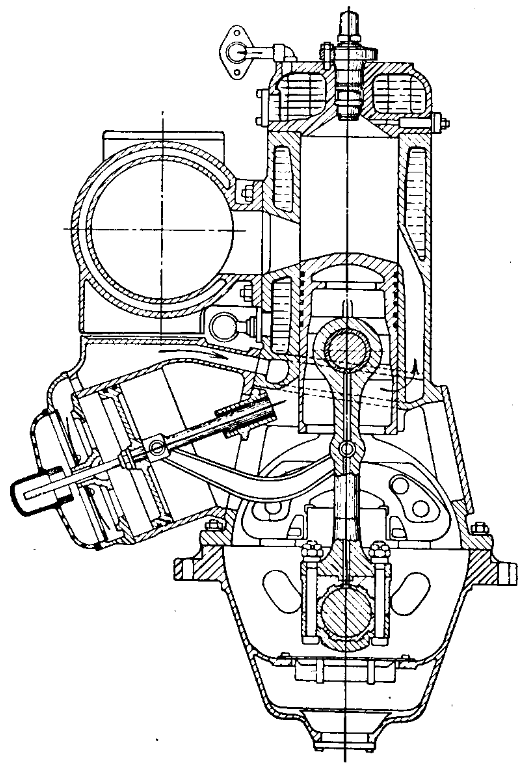

Fig. 2-4. Separately Scavenged Engine (Deutz).

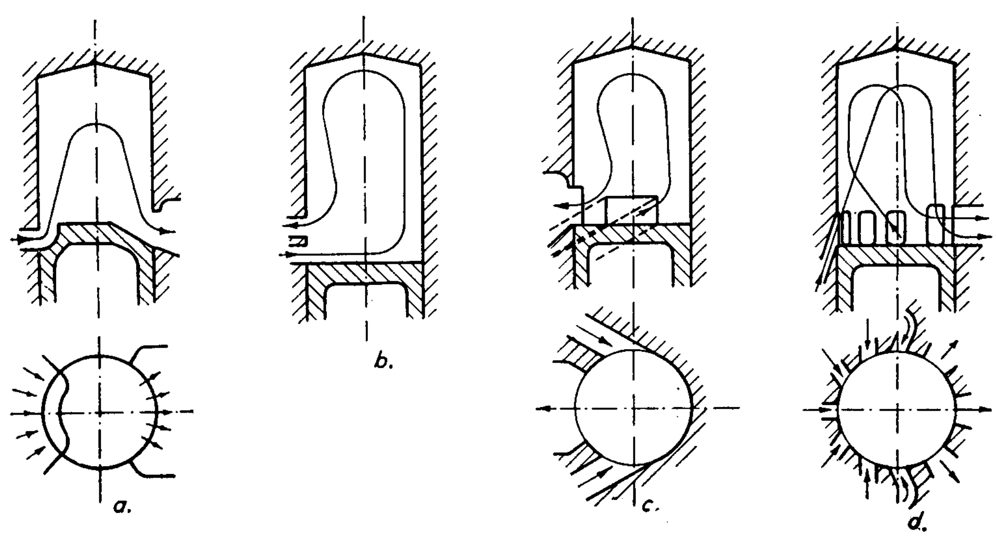

Fig. 2-5. Methods of Scavenging. *a.* Cross scavenging; *b.* Loop scavenging, M.A.N. type; *c.* Loop scavenging, Schnuerle type; *d.* Loop scavenging, Curtis type.

are both controlled by separate pistons that move in opposite directions. In *c* the inlet and exhaust ports are controlled by the combined motion of piston and sleeve. In an alternative arrangement one set of ports is controlled by the piston and the other set by a sleeve or slide valve. All uniflow systems permit unsymmetrical scavenging and supercharging.

The enumerated methods of scavenging permit a great number of combinations:

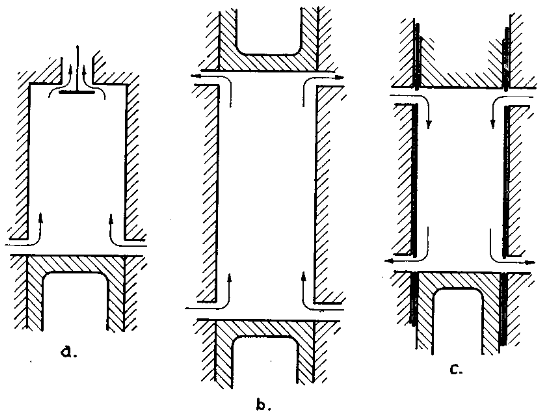

Fig. 2-6. Uniflow Scavenging. *a.* Poppet valve; *b.* Opposed piston; *c.* Sleeve valve.

1. Cross, crankcase-scavenged.
2. Cross, crankcase-scavenged, supercharge valve.
3. Cross, separately scavenged.
4. Cross, separately scavenged, supercharge valve.
5. Loop, crankcase-scavenged.
6. Loop, crankcase-scavenged, supercharge valve.
7. Loop, separately scavenged.
8. Loop, separately scavenged, supercharge valve.
9. Uniflow, poppet valve, crankcase-scavenged.
10. Uniflow, poppet valve, separately scavenged.
11. Uniflow, opposed piston, crankcase-scavenged.
12. Uniflow, opposed piston, separately scavenged.
13. Uniflow, slide valve, crankcase-scavenged.
14. Uniflow, slide valve, separately scavenged.

Altogether there are 14 arrangements, not counting subvarieties such as the use of a rotary blower, pump, step-piston, etc. Nos. 2 and 6 are irrational, as there is no justification for providing extra mechanism for supercharging when there is no surplus air available with which to supercharge. Of the other arrangements, Nos. 1, 3, 4, 7, 8, 10, 12, and 14 represent current commercial practice. Nos. 5, 9, 11, and 13 are not believed to be in production. Uniflow engines have always been built as high-output engines which, of course, cannot be the case with crankcase scavenging. No. 5, however, seems to be perfectly justified and has probably been overlooked thus far. Figure 2–7 shows a number of arrangements, more of which have been collected by Venediger [Venediger, 1934].

2.3 Short-Circuiting.

The ideal engine scavenges the cylinder of all residual products of combustion, fills it with uncontaminated fresh air, and in so doing wastes no fresh air through the exhaust. The ideal engine, therefore, has 100 per cent purity of charge and no loss from short-circuiting. Of course, neither of these perfections can be realized, and in practice they are mutually exclusive. A more complete removal of exhaust gases requires an increased volume of scavenging air. How good can the scavenging be—how does it affect the engine output—what means are available to approach closer to the ideal?

In ideal scavenging the scavenge air acts as a piston in pushing the burnt gases out of the cylinder without actually mixing with them. This condition can be approximated with uniflow scavenging, where the scavenge air enters the cylinder at one end and the exhaust leaves at the other end. Even then some mixing is unavoidable. Even if mixing were prevented, it would be impossible so to time the closure of the exhaust that all of the burnt gases escaped but none of the fresh air was lost. Therefore an appreciable amount of fresh air is always wasted through the exhaust. Nevertheless, the uniflow is the best possible arrangement for obtaining an air charge of high purity with relatively little loss of fresh air.

In most engines other than the uniflow type, the scavenging process consists largely of diluting the exhaust gases with sufficient fresh air to obtain a reasonably pure charge at the beginning of the compression period. During the process an almost complete mixing takes place. This is not

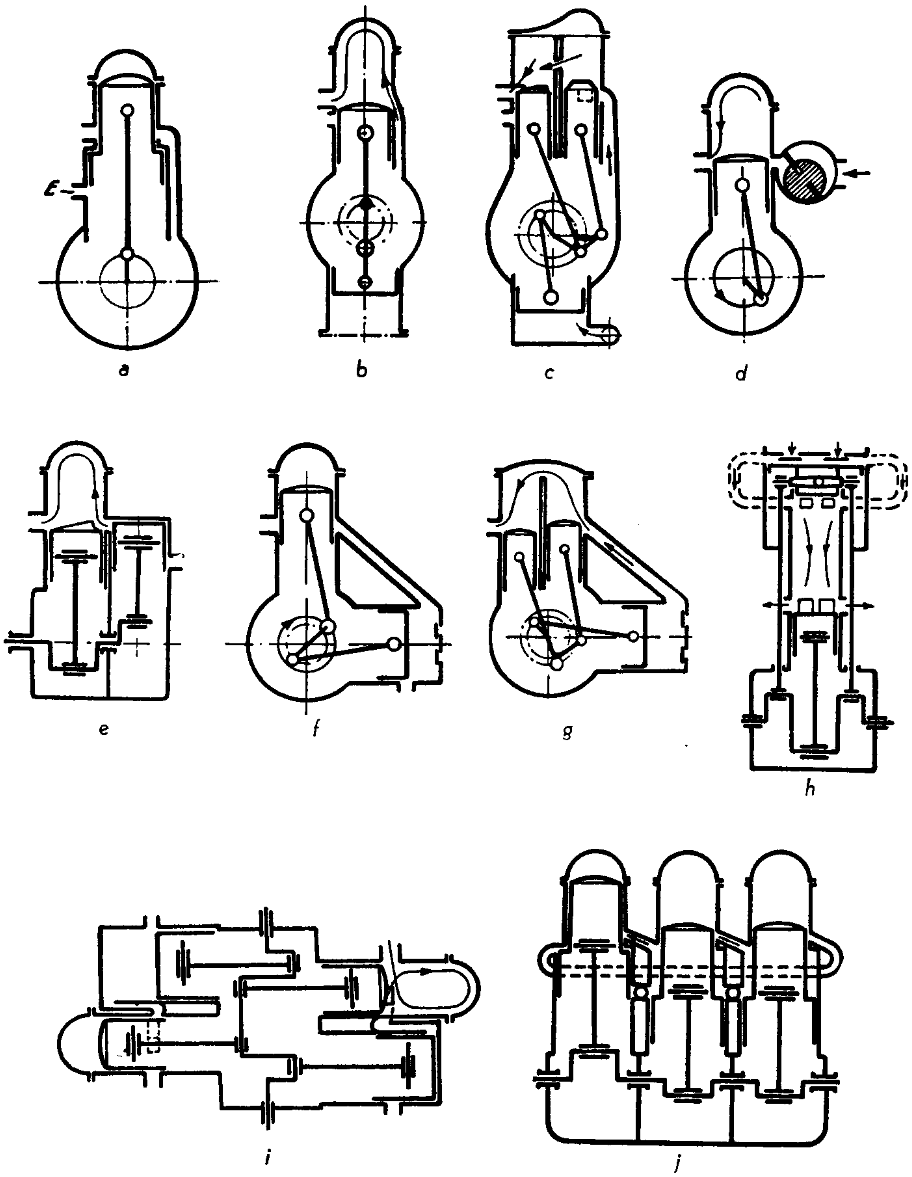

Fig. 2-7. Two-Stroke Cycle Engine Arrangements. *a*. Crankcase-scavenged, step piston; *b*. Crankcase-scavenged, auxiliary piston, *c*. Crankcase-scavenged, auxiliary piston, U cylinder, uniflow, *d*. Blower-scavenged, *e*. Pump-scavenged, *f*. Pump-scavenged, pump at rectangle, *g*. Pump-scavenged, U cylinder, *h*. Pump-scavenged, opposed piston, *i*. Pump-scavenged, opposed cylinders, *j*. Step piston, pump-scavenged, suitable for 3-cylinder engines. (After Venediger. Copyright, 1947 by Franckh'sch Verlagshandlung, Stuttgart.)

the worst possible case, however. As an extreme it can be imagined that the fresh air entering the cylinder leaves through the exhaust ports without either mixing with the exhaust gases or displacing them. Such would be a case of complete short-circuiting, which is to be avoided.

Between the two extremes — nonmixing and complete displacement (perfect scavenge), and nonmixing and no displacement (complete short-circuiting) — every combination of mixing and displacement is possible. Perfect mixing represents an intermediate case. The fresh air introduced successively dilutes the residual gases, and if sufficient air is used, at the completion of the scavenging process the cylinder content is only slightly contaminated.

In an actual engine the scavenging may be better or worse than that resulting from perfect mixing. During the scavenging process, in some engines the exhaust contains more fresh air than does the cylinder charge. In that case the scavenging performance is worse than that of straight dilution. In other engines the exhaust contains less fresh air than does the new charge. This is evidence that in addition to dilution, the fresh air displaces some residual gases and the scavenging performance is then better than that of straight dilution.

The success of the scavenging process depends on the scavenging arrangement. It was mentioned that uniflow scavenging is the least likely to cause short-circuiting. With cross scavenging and loop scavenging, it has been found that the efficiency of scavenging may or may not exceed the efficiency of perfect mixing. In most cross-scavenged engines the scavenging is somewhat worse than that of perfect mixing, while well-designed loop-scavenged engines generally exceed the scavenging efficiency of perfect mixing. In either case scavenging is effected mainly by the scavenge air diluting the residual gases in the cylinder before either leaves through the exhaust. This does not mean, however, that straight dilution can be taken for granted. How short-circuiting can be minimized so as to improve the efficiency of the scavenging is a matter of skill in design.

It is impossible to select one single index number that would describe the effectiveness of scavenging of a particular engine. With any scavenging arrangement, the more fresh air supplied to the engine the greater is the amount retained in the cylinder. With an infinite amount of delivered air, scavenging becomes perfect. The merit of any scavenging arrangement can better be described by a curve. If the air remaining in the cylinder is plotted against the air delivered to the cylinder at every cycle, a rising curve is obtained, and the higher the curve lies the better the scavenging. The actual curve may be compared with theoretical curves that correspond to *perfect scavenge* and *perfect mixing*. How these theoretical curves can be obtained will be described in the following chapters.

CHAPTER 3

TERMINOLOGY

3.1 In the absence of a terminology that is commonly accepted or at least commonly understood, it seems necessary to explain and define the terms used in this book relating to scavenging. This chapter with a list of nomenclature at its end should always be consulted when the meaning of a term is uncertain to the reader, but it may be omitted at the first reading of the book.

In any internal combustion engine the power output is largely determined by the number of oxygen molecules that can be packed into the cylinder each minute. Since air is the common supply of oxygen, the maximum amount of fuel burnt is controlled by the amount of air available, and the power output varies in direct ratio thereto, provided the thermal efficiency is unchanged.

3.2 Volumetric Efficiency.

For normally aspirated four-stroke cycle engines the **volumetric efficiency** has served satisfactorily as a measure of success in filling the cylinder with air. Volumetric efficiency has commonly been defined as the ratio of the weight of air actually aspirated to that which under normal temperature-pressure conditions fills the displacement volume or swept volume, which is equal to piston area times stroke. By referring to NTP (60 F temperature and 14.7 psia pressure) conditions, this can be more conveniently expressed as ratio of volumes

$$\eta_{vol} = \frac{V_{asp}}{V_{disp}},$$

the volume of aspirated air per cycle at normal temperature and pressure divided by the displacement volume (piston area times stroke). Volumetric efficiency expresses the air-taking ability of the engine, and power output is considered proportional to it.

Experimental determination of volumetric efficiency is simple. An air receiver of ample size is placed in the inlet passage of the engine, and a flowmeter measures the amount of air passing into the receiver. Because all air entering the receiver is delivered to the engine cylinder, the flowmeter gives V_{asp}, from which η_{vol} can be calculated.

During the suction stroke, a normally aspirated four-stroke cycle engine acts as a pump and its volumetric efficiency is merely its pumping efficiency; it shows how closely the pump delivery approaches the absolute limit, which is the piston displacement.

The advent of the two-stroke cycle and the supercharged four-stroke cycle engine upset these simple relations. The engine no longer acts as its own pump; instead, a separate pump or blower is added, with a displacement generally larger than the engine displacement, which tends to make the *volumetric efficiency* larger than unity. To make things worse, the amount of air delivered to the engine is no longer equal to the amount of air trapped in the cylinders. During the scavenging period, when the intake and exhaust ports (or valves) are both open, a portion of the air entering through the intake escapes through the exhaust port, and therefore is not present in the cylinder during the following compression stroke. This is equally true in two-stroke and supercharged four-stroke cycle engines with the customary valve overlap.

If volumetric efficiency is to serve as a measure of the success of filling the cylinder with air, a flowmeter measuring the amount of air delivered to the engine can no longer give the information from which it can be determined.

The term *volumetric efficiency* nevertheless survived, but in the hands of various authors it assumed a multiplicity of meanings.

The most frequently used definition of volumetric efficiency for a two-stroke cycle engine is

$$\epsilon_{vol} = \frac{V_{ret}}{V_{disp}},$$

where V_{ret} denotes the amount of air **retained** in the cylinder (cu in. NTP), and V_{disp} the displacement volume (cu in.). This definition, however, is open to serious objections:

1. While in a normally aspirated four-stroke cycle engine the air retained is equal to the air delivered, which latter lends itself to easy experimental determination, in a two-stroke cycle engine, the determination of the air retained is so complicated that it is beyond the facilities of an ordinary engine laboratory (see sections 16.6–16.17).

2. A more serious objection to the conventional definition is that it leads to a mathematical paradox. It has been pointed out by the author [Schweitzer, 1943] that on the basis of this definition an improvement in scavenging (that reduces the relative amount of residual gases) would result in a **lower** power output (see section 4.1).

3. If the conventional definition is restricted to four-stroke cycle engines, the status of the supercharged four-stroke cycle engine with appreciable valve overlap is uncertain, because in spite of being a four-stroke cycle engine, it is scavenged somewhat like a two-stroke cycle engine, a considerable portion of the air being short-circuited.

3.3 Definitions.

In the following the use of the term *volumetric efficiency* is avoided altogether, for other than normally aspirated four-stroke cycle engines. For the two-stroke cycle (and supercharged four-stroke cycle) engines a consistent system of definitions can be derived by the following reasoning:

Figure 3–1 is a diagrammatic representation of the charging process. The hatched areas represent pure air and the crosshatched areas represent combustion gases. The widths of the channels represent the quantity of the gases expressed by volume at NTP conditions.*

In a two-stroke cycle engine, air is supplied to the cylinders either by a separate pump or blower, or by the piston of the working cylinder acting as a pump. Depending on the relative capacity of the blower, the air delivered, V_{del}, may be either more or less than the displacement volume, V_{disp}. The ratio of the two

$$L = \frac{V_{del}}{V_{disp}} \tag{3–1}$$

is termed **delivery ratio.** It is determined experimentally by flow measurement.

* The significant measure of the air quantity is its weight (or number of mols if various gases are being dealt with). However, the volume of air under NTP conditions may be considered as a *weight* measure and most of the subsequent equations represent relations between weights of air. They are easier to visualize than if the same relations were expressed in pounds of air.

The delivery ratio is a measure of the air quantity supplied to the cylinder relative to the engine displacement. It is predominantly controlled by the capacity of the scavenging

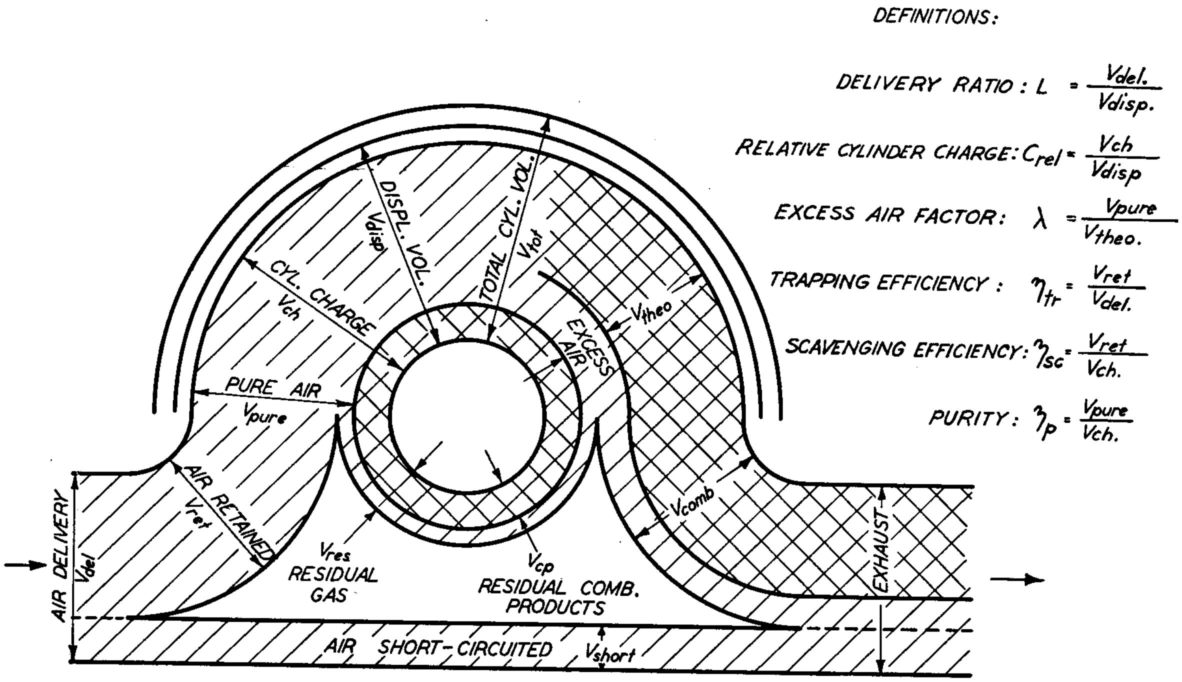

Fig. 3-1. Diagram Showing the Charging Process of an Internal Combustion Engine. The diagram represents either a two-stroke cycle engine or a four-stroke cycle engine with considerable valve overlap. In a four-stroke cycle engine without valve overlap, the air short-circuited V_{short} is zero, and the area below the dash line is missing. *All volumes refer to N.T.P. conditions.*

blower, and usually lies between 1.2 and 1.5 except in crankcase-scavenged engines, where it naturally is less than one.

In either case the air delivery V_{del} is split into two parts: the short-circuited air, V_{short}, which leaves through the exhaust port (valve) without remaining in the cylinder, and the retained air, V_{ret}, which is trapped in the cylinder and participates in the subsequent combustion.

The **trapping efficiency**

$$\eta_{tr} = \frac{V_{ret}}{V_{del}} \tag{3-2}$$

indicates what portion of the air delivered is retained in the cylinder, the rest being *wasted* through the exhaust. Short-circuiting is naturally equal to $(1 - \eta_{tr})$. The trapping efficiency, therefore, is a measure of the success of trapping the supplied air with a minimum of waste. Naturally, it is controlled by the geometry of the intake and exhaust ports (or valves) and their timing overlap.

The experimental determination of the trapping efficiency is discussed in section 16.6.

The air retained, V_{ret}, together with the residual gas, V_{res}, remaining in the cylinder after scavenging constitutes the **cylinder charge** V_{ch}. This charge may again be more or less than the displacement volume and the **relative cylinder charge**

$$C_{rel} = \frac{V_{ch}}{V_{disp}} = \frac{V_{ret} + V_{res}}{V_{disp}} \tag{3-3}$$

may be either more or less than unity, depending largely on the scavenging pressure and the port heights. **The relative cylinder charge is a measure of the success in filling the cylinder, irrespective of the composition of the charge.** Its experimental determination is simple in principle, involving only measurement of the pressure and temperature of the cylinder charge at one point of the compression stroke. The actual direct determination, however, involves difficulties, because of the rapid change of the respective quantities.

The cylinder charge may also be considered as composed of pure air and residual combustion products. By denoting

$$P = \frac{V_{pure}}{V_{disp}}$$

as **pure air ratio** and

$$\frac{V_{cp}}{V_{disp}}$$

as **residual combustion products ratio,**

$$C_{rel} = P + \frac{V_{cp}}{V_{disp}}.$$

During combustion, part of the air (or, rather of the oxygen in the air) contained in the cylinder charge burns, while the remainder, **the excess air**, is not involved in the attendant chemical reactions. Part of this excess air escapes through the exhaust with the combustion products, and another part, $V_{res} - V_{cp}$, remains in the cylinder and participates in the subsequent cycle, where V_{cp} represents combustion products in the residual gas. Therefore, the cylinder charge consists of **three** parts: the retained portion of the air delivered, part of the combustion products from the preceding cycle, and part of the excess air from the preceding cycle.

As the cylinder charge V_{ch} is contaminated by V_{cp} combustion products

$$\eta_p = \frac{V_{pure}}{V_{ch}} = \frac{V_{ch} - V_{cp}}{V_{ch}} = 1 - \frac{V_{cp}}{V_{ch}} \tag{3–4}$$

represents the **purity** of the charge and V_{cp}/V_{ch} constitutes the **pollution.** It should be realized that V_{pure} is more than that part of the air delivered which is retained in the cylinder; it includes some pure air contained in the residual gas remaining in the cylinder from the previous cycle.

The purity of the charge is a measure of the success of scavenging the cylinder from the combustion products of the preceding cycle. It is largely controlled by the shape of the combustion chamber and the scavenging arrangement (cross, loop, or uniflow). Experimental determination of the purity is relatively simple, consisting only of analysis of a gas sample taken during the compression stroke. The difficulty lies in securing a representative sample (see section 16.10).

In German technical literature the term **scavenging efficiency** (*Spuelwirkungsgrad*) has been widely used [Neumann, 1930]. It is defined as

$$\eta_{sc} = \frac{V_{ret}}{V_{ch}} = \frac{V_{ret}}{V_{ret} + V_{res}}. \tag{3–5}$$

Scavenging efficiency is a term somewhat similar to purity and **expresses the measure of the success in clearing the cylinder of residual gases from the preceding cycle.**

The useful fresh charge divided by the displacement volume is the **charging efficiency** defined as

$$\eta_{ch} = \frac{V_{ret}}{V_{disp}}. \tag{3–6}$$

Charging efficiency is a measure of the success of filling the cylinder with fresh air. Naturally

$$\eta_{ch} = L\eta_{tr} \tag{3-7}$$

as can be seen from equations (3–1) and (3–2). The charging efficiency is an important index as it most directly affects the power output of the engine. Except for the "left-over" air $\eta_{ch} = P$.

In all of the above listed definitions volumes corresponding to **NTP conditions** have been used. NTP or normal temperature and pressure means 60 F temperature and 14.7 psia pressure. Since one cubic foot of such air weighs 0.076 pound, all formulas are equally valid for air quantities expressed in pounds. Furthermore, since the NTP density of the combustion products differs very little from that of air [Schweitzer and DeLuca, 1942], in all formulas and in Fig. 3–1 the volume symbols could be replaced by weights without appreciably affecting the validity of the relations. This is an advantage of calculating everything on NTP basis.*

However, on certain occasions (as for instance in dealing with high supercharge) it is more convenient to express the air supplied to the **cylinder** in terms of cylinder displacement under **inlet pressure and temperature** conditions. In such a case, in order to avoid misunderstanding, the term **scavenge ratio** is used,

$$R_s = \frac{\text{Weight of air delivered}}{\text{Weight of air of } V_{disp} \text{ at inlet density}} \tag{3-8}$$

* In spite of the advantage of basing indexes upon NTP conditions, they are frequently based upon other conditions, such as ambient pressure and temperature (*Deutsche Kraftfahrforschung, Reichs Verkehrsministerium, Zwischenbericht* No. 103/104 Ueber 2. *Tagung des Arbeitskreises fuer Zweitaktfragen*, 1941), inlet air pressure and temperature [Taylor and Taylor, 1938], or exhaust pressure and inlet temperature [Taylor, 1940]. It should be pointed out that in many cases the numerical results are the same, no matter what base is used because only the **ratio** of two air volumes of the same conditions is being dealt with, as for instance in L, η_{tr}, and η_{ch}. In some cases, as η_{sc}, there is a small difference because of the presence of residual gas. Strictly ($V_{ch} = V_{ret} + V_{res}$) cannot be calculated without knowing the amount and composition of the residual gas left in the cylinder after the charging process has been completed. The volume of the combustion products (including water vapor) is 2 to 4 per cent greater than the volume of inlet air [Schweitzer and DeLuca, 1942] but the combustion products seldom constitute more than 20 per cent of the cylinder charge. Therefore, the difference is only a fraction of one per cent.

Indexes containing the cylinder volume are also frequently based on other than the displacement volume. *Total volume, effective volume* and *total effective volume* are sometimes used instead of displacement volume used in this book.

Total volume is equal to $V_{disp}/(1-{}^1/\epsilon)$ where ϵ is the compression ratio. Therefore delivery ratio expressed on this basis [Taylor and Taylor, 1938] is some 7 per cent smaller.

Effective volume is equal $V_{disp}(1 - h_e)$ where h_e is the relative length of the exhaust ports.

Total effective volume is equal to $\frac{V_{disp}}{1-\frac{1}{\epsilon}}(1 - h_e)$.

With these variations every index where cylinder volume figures, like L, C_{ret}, and η_{ch} may be calculated in four different manners and as many different values obtained for them [Venediger, 1947].

Although some justification may be found for any of the four methods of calculation, the displacement volume is obviously the simplest quantity to describe the capacity of an engine cylinder. Bore and stroke are always known. To obtain the *total* volume the knowledge of the compression ratio, to obtain the *effective volume* the knowledge of the port height is needed, which data frequently are unavailable. Neither is it evident that the *total* volume more truly represents the cylinder capacity than the displacement volume. The relevant part of the cylinder is the one swept by the piston. In four-stroke cycle engines displacement volume is used almost exclusively in computations. There is no good reason to deviate from this procedure in case of two-stroke cycle engines.

An argument in favor of the use of the total volume is that with displacement volume the charging efficiency may become, in some rare cases, greater than one. This however is not a weighty argument and if it were a good one it would apply equally to supercharged four-cycle engines.

For the sake of simplicity and conformity with four-stroke cycle procedure in this book all indexes and calculations refer to *displacement* volume throughout.

where the denominator is the weight of air that fills the displacement volume under **cylinder** inlet temperature pressure conditions. Naturally

$$R_s = L\frac{\rho_{NTP}}{\rho_i}. \tag{3-9}$$

For instance, if the inlet pressure is 5 atmospheres (5 × 14.7 psia) and the inlet temperature is 60 F, an air delivery large enough to fill 1.2 displacement volumes may be described more conveniently by the *scavenge ratio* of 1.2 than by the delivery ratio of 6.

3.4 Relations Between Scavenging Terms.

By definition, scavenging efficiency is

$$\eta_{sc} = \frac{V_{ret}}{V_{ch}} \tag{3-10}$$

and purity of charge is

$$\eta_p = \frac{V_{pure}}{V_{ch}}. \tag{3-11}$$

From Fig. 3–1 the following relation is obtained:

$$\frac{V_{cp}}{V_{res}} = \frac{V_{theo}}{V_{comb}} \tag{3-12}$$

which expresses the fact that the pollution of the gas remaining in the cylinder (before scavenging begins) is the same as that of the gas which passes to the exhaust. If V_{comb} is equal to V_{ret}, which has been shown [Schweitzer and DeLuca, 1942] to be true within 5 per cent, it follows that

$$\frac{V_{cp}}{V_{res}} = \frac{V_{theo}}{V_{ret}} \tag{3-13}$$

and also

$$V_{ch} = V_{ret} + V_{res} \tag{3-14}$$

$$V_{pure} = V_{ch} - V_{cp}. \tag{3-15}$$

The excess air factor is the amount of pure air trapped divided by the theoretically correct amount needed for combustion

$$\lambda = \frac{V_{pure}}{V_{theo}}. \tag{3-16}$$

Equations (3–4), (3–10), (3–13), (3–14), (3–15), and (3–16) are six equations with the following six unknowns: V_{ch}, V_{ret}, V_{res}, V_{pure}, V_{theo}, and V_{cp}. Procedure for solution is as follows: From (3–10) and (3–14)

$$\eta_{sc} = \frac{V_{ret}}{V_{ret} + V_{res}} = \frac{1}{1 + \dfrac{V_{res}}{V_{ret}}}. \tag{3-17}$$

From (3–14), (3–15), and (3–16)

$$\lambda V_{theo} = V_{ret} + V_{res} - V_{cp}. \tag{3-18}$$

From (3–4), (3–14), and (3–15)

$$V_{cp} = V_{ch}(1 - \eta_p) = (V_{ret} + V_{res})(1 - \eta_p). \tag{3-19}$$

From (3-13), (3-14), (3-15), and (3-16)

$$\lambda V_{cp} \frac{V_{ret}}{V_{res}} = V_{ret} + V_{res} - V_{cp} \tag{3-20}$$

or

$$V_{cp} = \frac{V_{ret} + V_{res}}{1 + \lambda \dfrac{V_{ret}}{V_{res}}}. \tag{3-21}$$

From (3-19) and (3-21)

$$1 - \eta_p = \frac{1}{1 + \lambda \dfrac{V_{ret}}{V_{res}}}. \tag{3-22}$$

From (3-17) and (3-22)

$$\frac{1}{\eta_{sc}} - 1 = \lambda \left(\frac{1}{\eta_p} - 1\right). \tag{3-23}$$

From (3-23)

$$\eta_{sc} = \frac{1}{1 + \lambda\left(\dfrac{1}{\eta_p} - 1\right)}, \tag{3-24}$$

which expresses the relation between the scavenging efficiency and purity. The scavenging efficiency is smaller than the purity except for $\lambda = 1$, when they are equal. For instance, for $\lambda = 1.5$ and $\eta_p = 0.9$, $\eta_{sc} = 0.86$. Because the difference is small, and such strict analysis uncommon, the two quantities frequently have been confused in the literature [Sulzer, 1933; Carter, 1946].

Another useful relation can be obtained from equations (3-1), (3-2), (3-3), and (3-5):

$$C_{rel}\eta_{sc} = L\eta_{tr}, \tag{3-25}$$

which permits the computation of any of the four indexes if the other three are known. Of course from (3-6) both products are equal to η_{ch}.

From equation (3-25)

$$\eta_{tr} = \eta_{sc}\frac{C_{rel}}{L} \tag{3-26}$$

and since C_{rel} does not differ much from unity (except for superatmospheric engines [see Chapter 15]), the trapping efficiency is approximately equal to the scavenging efficiency divided by the delivery ratio. With increasing air delivery the scavenging efficiency always increases and the trapping efficiency ordinarily decreases, as is shown in section 3.9.

PERFECT SCAVENGE AND PERFECT MIXING

3.5 To judge the thoroughness of engine scavenging it is useful to have a standard of comparison, theoretically arrived at. It is obvious that scavenging improves if more air is used and with an infinite amount of air complete scavenging can be obtained. Therefore, both the actual scavenging and the reference standard must refer to the same amount of air delivered.

3.6 Perfect Scavenge.

It is possible to envisage a scavenging process that takes place very slowly without any change in pressure and temperature of the cylinder content. The clearance volume, which is approximately 7 per cent in a diesel engine, is ignored and furthermore the scavenge air and the residual gas are assumed to have the same density.

If the scavenge is 100 per cent perfect, all air delivered to the cylinder is retained in the cylinder. By plotting the amount of pure air remaining in the cylinder against the amount of air delivered in the cylinder, a 45-degree straight line is obtained.

Scavenging, however, can be even better than that. To illustrate, if the engine is running at light load, the cylinder content at the beginning of the scavenging period consists largely of air. Even at full load, however, a diesel engine never uses all air available in the cylinder. The *excess air* remains as pure air (or oxygen) diffused in the combustion gases after combustion has been completed.

With an excess air factor of λ, for each pound of air that has burned, $(\lambda - 1)$ pound of air is left over after combustion, and is available for use in the following cycle even before scavenging begins.

Perfect scavenge can then be visualized as introduction of a certain amount of fresh air which is retained completely by the cylinder while an equal amount of residual gases is discharged from the cylinder. The amount of air introduced and the amount of pure air remaining in the cylinder may be expressed in terms of displacement volume. The amount of air introduced is then equal to the delivery ratio L, and the amount of pure air remaining is equal to the pure air ratio P. L may be smaller, equal to, or larger than one, but with perfect scavenge, there is no point in introducing more than one displacement volume of air because the cylinder cannot hold more than one cylinder volume of air.

If the excess air is zero (carburetor engine) the combustion uses up all air in the cylinder, and during the subsequent scavenging process the fresh air displaces nothing but combustion products. The scavenge being perfect, the amount of air contained equals the amount of air delivered, $P = L$. With excess air present before combustion, the cylinder content includes some air (or corresponding amount of oxygen) after combustion, when scavenging begins.

In order to establish the equilibrium relation between the pure air content and the amount of air delivered, the respective amounts may be designated as:

Before scavenging x_0 total charge; y_0 air
After scavenging x_1 total charge; y_1 air
After combustion x_2 total charge; y_2 air

everything expressed in cubic inches on NTP basis.

Naturally, the amount of mixture in the cylinder is always the same

$$x_0 = x_1 = x_2 = V_{disp}.$$

It is also obvious that after combustion and before scavenging the air content is the same

$$y_2 = y_0.$$

Because by definition $\lambda = y_1/(\text{air burnt})$, combustion uses up y_1/λ air and leaves

$$y_2 = y_1 - \frac{y_1}{\lambda} = y_1\left(1 - \frac{1}{\lambda}\right). \tag{3–27}$$

air in the cylinder. This is part of the mixture x_2 and the concentration of pure air in the mixture is

$$\frac{y_2}{x_2} = \frac{y_1}{x_0}\left(1 - \frac{1}{\lambda}\right).$$

During the scavenging period (0–1), V_{del} cubic inches (NTP) of air are introduced and simultaneously V_{del} mixture is released. In the mixture released, there are

$$V_{del}\frac{y_1}{x_0}\left(1 - \frac{1}{\lambda}\right)$$

cubic inches of air. Therefore, during the scavenging process we gain V_{del} and lose $V_{del}\, y_1/x_0(1 - 1/\lambda)$ air, and end up with

$$y_1 = y_0 + V_{del} - V_{del}\frac{y_1}{x_0}\left(1 - \frac{1}{\lambda}\right) \tag{3-28}$$

cubic inches of air. During combustion, part of this air is burnt, and the amount of air left over is, according to (3–27),

$$y_2 = y_1\left(1 - \frac{1}{\lambda}\right)$$

or since

$$y_2 = y_0$$

$$y_0 = y_1\left(1 - \frac{1}{\lambda}\right). \tag{3-29}$$

From equations (3–28) and (3–29)

$$y_1 = y_1\left(1 - \frac{1}{\lambda}\right) + V_{del} - V_{del}\frac{y_1}{x_0}\left(1 - \frac{1}{\lambda}\right)$$

$$V_{del} = y_1\left[\frac{V_{del}}{x_0}\left(1 - \frac{1}{\lambda}\right) + \frac{1}{\lambda}\right]$$

$$y_1 = \frac{V_{del}\,\lambda}{1 + \dfrac{V_{del}}{x_0}(\lambda - 1)}. \tag{3-30}$$

By definition

$$\frac{y_1}{x_0} = \frac{V_{pure}}{V_{disp}} = P$$

and

$$\frac{V_{del}}{x_0} = \frac{V_{del}}{V_{disp}} = L,$$

so

$$P = \frac{L\lambda}{1 + L(\lambda - 1)}. \tag{3-31}$$

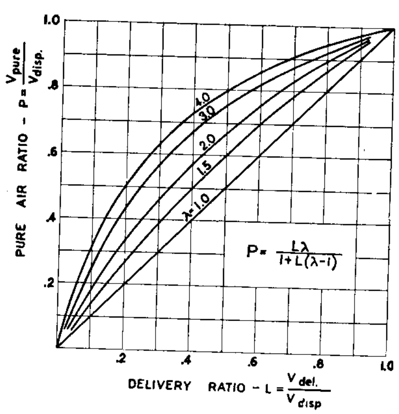

Fig. 3-2. Perfect Scavenge. Air contained in vs air delivered to cylinder in terms of displacement volume.

This equation gives the quantity of pure air remaining in the cylinder after perfect scavenge. It is shown in graphical form in Fig. 3–2. It will be noticed that except for ($\lambda = 1$), P is greater than L; that means the amount of air available is greater than the amount delivered. This is quite natural because **left-over air** is already in the cylinder when scavenging begins.

The height of any ordinate up to the line of ($\lambda = 1$) represents retained air from the fresh charge, while the piece above that represents the left-over air from the preceding cycle.

3.7 Perfect Mixing.

The ideal of perfect scavenge can, of course, never be attained, nor even closely approached.

In the actual engine some of the fresh air introduced escapes through the exhaust without displacing any residual gas. Some of the fresh air mixes with the residual gas and some displaces residual gas in the manner of perfect scavenge. The second reference standard for scavenging is termed *perfect mixing.*

With perfect mixing, all fresh air introduced into the cylinder mixes instantaneously with the existing cylinder content to form a uniform mixture. A corresponding amount of the mixture simultaneously leaves the cylinder. With perfect mixing, direct short-circuiting is zero. In this respect perfect mixing is better than the actual process. On the other hand, perfect mixing is not nearly as good as perfect scavenge, in which all fresh air introduced displaces residual gas and remains in the cylinder.

Perfect mixing may be considered a more suitable reference than perfect scavenge, inasmuch as an actual engine may have scavenge that is better, or one that is worse than perfect mixing.

The case of perfect mixing has been treated by Hopkinson [Hopkinson, 1914], who, without considering any excess air, obtained the equation

$$y = 1 - e^{-z} \tag{3–32}$$

where z is the amount of air introduced and y the amount remaining in the cylinder. The relation can be generalized to hold for any amount of excess air by reasoning similar to that followed in the preceding section.

During the scavenging process, the cylinder at any instant contains y amount of pure air in a mixture of x_0 (cu in. NTP). While an infinitesimal amount of dV_{del} is introduced into the cylinder, an equal amount of gas mixture leaves the cylinder. Since the composition of the outgoing gas is the same as the instantaneous composition of the cylinder content (perfect mixing), the outgoing gas contains

$$y\frac{dV_{del}}{x_0} \text{ cu in. (NTP) air.}$$

Consequently the increase of the air content is

$$dy = dV_{del} - y\frac{dV_{del}}{x_0},$$

which can be written as

$$dV_{del} = \frac{dy}{1 - \dfrac{y}{x_0}}. \tag{3–33}$$

This is a simple differential equation, the solution of which is

$$V_{del}\Big|_0^1 = -x_0 \log_e\left(1 - \frac{y}{x_0}\right)\Big|_0^1.$$

At the beginning of the scavenging period $V_{del} = 0$ and $y = y_0$. At the end of the scavenging period $V_{del} = V_{del}$ and $y = y_1$. Equation (3–29) still holds, by which

$$y_0 = y_2 = y_1\left(1 - \frac{1}{\lambda}\right)$$

therefore

$$V_{del} = -x_0\left\{\log_e\left(1 - \frac{y_1}{x_0}\right) - \log_e\left[1 - \frac{y_1\left(1 - \dfrac{1}{\lambda}\right)}{x_0}\right]\right\} \tag{3–34}$$

or

$$\frac{V_{del}}{x_0} = \log_e \frac{1 - \frac{y_1}{x_0}\left(1 - \frac{1}{\lambda}\right)}{1 - \frac{y_1}{x_0}}.$$

By definition

$$\frac{y_1}{x_0} = \frac{V_{pure}}{V_{disp}} = P$$

and

$$\frac{V_{del}}{x_0} = \frac{V_{del}}{V_{disp}} = L$$

therefore

$$L = \log_e \frac{1 - P\left(1 - \frac{1}{\lambda}\right)}{1 - P}. \tag{3–35}$$

Equation (3–35) gives the relation between the air delivered and the pure air contained in the cylinder with perfect mixing, and is shown in graphical form in Fig. 3–3. Again the height up

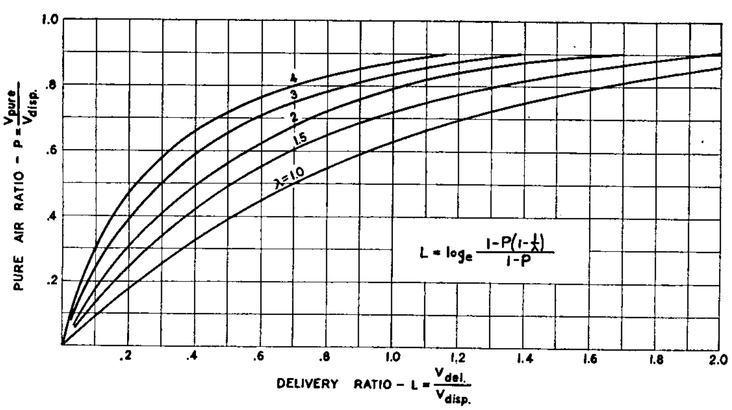

Fig. 3-3. Perfect Mixing. Air contained in vs air delivered to cylinder in units of displacement volume.

to the line ($\lambda = 1$) represents retained air and above left-over air. The line corresponding to ($\lambda = 1$) corresponds to Hopkinson's equation.

Equation (3–35) shows that the greatest amount of air that can be retained by perfect mixing is one cylinder volume, and an infinite amount of air must be expended to obtain that result.

Comparing Fig. 3–2 and 3–3, it is evident that the difference between perfect scavenge and perfect mixing is great when the excess air factor is small, but when the excess air factor is high (low engine load), the difference resulting from perfection of scavenging process is small. This has been borne out by practical experience, which shows that in lightly loaded engines the art of scavenging is unimportant.

The difference between the amounts of pure air is small also if the amount of air delivered is small. In fact, with very small amounts of air used, perfect scavenge and perfect mixing give identical results.

3.8 Trapping Efficiency.

Figure 3–4 shows the **trapping efficiency** as defined by equation (3–2) for perfect scavenge and perfect mixing. It has been obtained from Fig. 3–2 and 3–3 by plotting P/L vs L, using the curves marked ($\lambda = 1.0$). Trapping efficiency is the percentage of the delivered air which is retained in the cylinder. Left-over air is not counted in the air **retained.** Therefore, irrespective of the excess air factor, only the curves corresponding to ($\lambda = 1$) are to be considered.

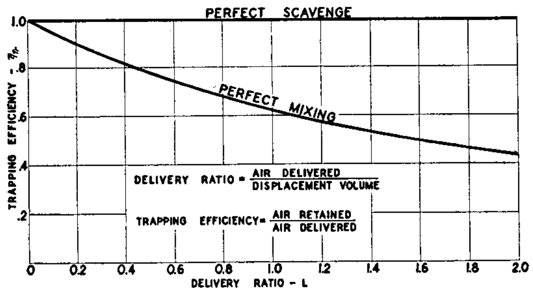

Fig. 3-4. Trapping Efficiency vs Delivery Ratio.

The trapping efficiencies of most two-stroke cycle engines would fall between that of perfect mixing and perfect scavenge, but of many of them would be lower than either. The actual amount of air available for combustion is more than trapping efficiency times delivery ratio. It depends on the excess air ratio as indicated by Fig. 3–2 and 3–3.

3.9 Supercharge.

A much used and seldom defined term is *supercharge.* A four-stroke cycle engine is supercharged if the air charge is increased by means of a blower or a similar device over the amount that the engine would receive by normal aspiration. This definition is inapplicable to the two-stroke cycle engine which lacks suction by the piston and is as a rule charged by a blower or equivalent. By such definition all two-stroke cycle engines (ignoring blowerless *Kadenacy engines*) would be supercharged.

In this book a two-stroke cycle engine is designated as *supercharged* if the pressure of the charge immediately after closure of the ports is substantially higher than the ambient pressure. This is usually accomplished by compressing the air before it is fed to the cylinders **and** by keeping some (or all) inlet ports open for some time after the exhaust ports are closed. In this way a charge of superatmospheric pressure is trapped in the cylinder and its potential power output is correspondingly increased.

If the **exhaust** pressure of the engine is substantially above the atmospheric or the ambient pressure, as for instance due to the back pressure caused by an exhaust gas turbine, the engine is called a *superatmospheric* engine. From these definitions it follows that superatmospheric engines are always supercharged, but supercharged engines are not always superatmospheric.

NOMENCLATURE

3.10 Aside from terms relating to charging and scavenging, the engine terms used in this book are fairly well established and no attempt is made to define them unless their meaning is doubtful.

A nomenclature explaining most of the symbols used in the book is listed below and should be used whenever the meaning of a symbol is not clear. This table is repeated on a fold-out at the end of the book.

NOMENCLATURE

SYMBOL	CONCEPT	UNITS USUALLY USED
A	reciprocal of mechanical equivalent of heat	Btu per ft-lb
A	area in general	sq in.

SYMBOL	CONCEPT	UNITS USUALLY USED
$A_{em} = \frac{1}{\alpha_e}\int A_e d\alpha$	mean exhaust area	sq in.
A_h	total air consumption	lb per hr
A_i	the uncovered inlet-port area at any one instant	sq in.
$A_{im} = \frac{1}{\alpha_i}\int A_i d\alpha$	mean inlet area (normal to the direction of flow)	sq in.
$A_m = \frac{1}{\alpha}\int A d\alpha$	mean blowdown area	sq in.
a	velocity of sound in an exhaust column	in. per sec, ft per sec
B	blowdown time-area	sq in.-deg
b_e	width of (rectangular or rhomboid) exhaust ports	in.
b_i	width of (rectangular or rhomboid) inlet ports	in.
bhp	brake horse power	hp
bmep	brake mean effective pressure	psi
gross bmep	brake mean effective pressure when power to drive scavenging blower is furnished separately	psi
net bmep	brake mean effective pressure calculated by deducting from the gross bmep the blower mep	psi
blower mep	engine bmep that corresponds to hp input of blower	psi
C	discharge coefficient	dimensionless
c_c	coefficient of contraction	dimensionless
c_p	specific heat at constant pressure	Btu per lb per deg F
c_v	specific heat at constant volume	Btu per lb per deg F
c_v	coefficient of velocity	dimensionless
$C_{rel} = \frac{V_{ch}}{V_{disp}}$	relative cylinder charge	dimensionless
D	cylinder diameter (bore)	in.
F	cross sectional area of pipe	sq in.
$F_a = h_m \alpha$	blowdown area	per cent degree
F_h	total fuel consumption	lb per hr
$F_i = h_{im}\alpha_i$	inlet area	per cent degree
F_o	area of an orifice	sq in.
f	specific fuel consumption	lb per bhp-hr
G	weight of air per second passing through an opening	lb per sec
G_r	weight of air at NTP in receiver	lb

SYMBOL	CONCEPT	UNITS USUALLY USED
G_{del}	weight of air delivered by blower during one engine revolution	lb
g	acceleration of gravity	ft per sec per sec
h	height of ports	in.
h_e	relative height of rectangular or rhomboid exhaust ports in relation to stroke	in. per in.
h_i	relative height of rectangular or rhomboid inlet ports in relation to stroke	in. per in.
h_{im}	mean relative height of rectangular or rhomboid inlet ports	in. per in.
h_m	relative blowdown port height	in. per in.
k	ratio of specific heats c_p/c_v	dimensionless
L	length of pipe	in.
$L = \frac{V_{del}}{V_{disp}}$	delivery ratio	dimensionless
mep	mean effective pressure	psi
mip	mean indicated pressure	psi
n	number of crank revolutions per minute number of power strokes per minute	min^{-1} min^{-1}
NTP	normal temperature pressure conditions (60 F, 14.7 psia)	
$P = \frac{V_{pure}}{V_{disp}}$	pure air ratio	dimensionless
p_0	ambient pressure normally used in the calculations, 14.5 psia	psia
p_1	initial absolute pressure of air	psia
p_2	final absolute pressure of air	psia
p_{av}	average pressure in receiver, gage	psig
p_b	blower mep	psi
p_e	absolute pressure in cylinder at the instant the exhaust port opens (expansion end pressure)	psia
p_i	absolute pressure in cylinder at the instant the inlet port opens (blowdown pressure)	psia
p_m	absolute pressure in the contracted section	psia
p_{max}	maximum absolute pressure in receiver	psia
p_{min}	minimum absolute pressure in receiver	psia

SYMBOL	CONCEPT	UNITS USUALLY USED
p_{sc}	scavenge pressure, gage	psig
R	gas constant for air	ft per deg F
R_e	gas constant for combustion gases	ft per deg F
$R_s = L\frac{\rho_{NTP}}{\rho_i}$	scavenge ratio	dimensionless
$r = \frac{W_{pure\ act}}{W_{fuel}}$	actual air-fuel ratio	lb per lb
$r_{th} = \frac{W_{pure\ th}}{W_{fuel}}$	theoretical (chemically correct) air-fuel ratio	lb per lb
$\mathcal{S}$	supercharge time-area	sq in.-deg
S	scavenge factor	dimensionless
s	length of stroke of engine	in.
T_1	absolute temperature in container 1	deg R
T_2	absolute temperature in container 2	deg R
T_e, t_e	temperature of combustion gases at instant exhaust port opens	deg R, deg F
T_m	absolute temperature in the contracted section	deg R
t	time	sec
t_{sc}	temperature of scavenge air	deg F
V_{ch}	amount of cylinder charge expressed in volume NTP	cu in.
V_{comb}	amount of combustion gas in the cylinder expressed in volume NTP	cu in.
V_{cp}	amount of residual combustion products in the cylinder expressed in volume NTP	cu in.
V_{del}	amount of air delivered to the cylinder per cycle expressed in volume NTP	cu in.
V_{disp}	displacement volume	cu in.
V_e	cylinder volume at instant exhaust port opens (expansion end volume)	cu in.
V_p	volume of the exhaust port	cu in.
V_{pure}	amount of pure air in cylinder during compression expressed in volume NTP	cu in.
V_r	volume of receiver	cu in.
V_{res}	amount of residual gas in the cylinder expressed in volume NTP	cu in.
V_{ret}	amount of fresh air retained in the cylinder expressed in volume NTP	cu in.

SYMBOL	CONCEPT	UNITS USUALLY USED
V_{short}	amount of air short-circuited per cycle expressed in volume NTP	cu in.
V_{th}	amount of air theoretically required per cycle for combustion expressed in volume NTP	cu in.
V_{tot}	total cylinder volume	cu in.
v_m	specific volume in the contracted cross section	cu in. per lb
w_i	mean inlet velocity	ft per sec
z_0	period of exhaust column vibration	sec
α	crank angle	deg
α	exhaust lead, crank angle	deg
α_1	instant of inlet port opening, crank angle after top center	deg
α_2	instant of inlet port closing, crank angle after top center	deg
α_e	exhaust duration, crank angle	deg
$\alpha_i = \alpha_2 - \alpha_1$	inlet duration, crank angle	deg
$\beta = \dfrac{b_i}{D \times \pi}$	relative width of rectangular or rhomboid inlet port	in. per in.
η_b	blower efficiency	dimensionless
$\eta_{ch} = \dfrac{V_{ret}}{V_{disp}}$	charging efficiency	dimensionless
$\eta_p = \dfrac{V_{pure}}{V_{ch}}$	purity	dimensionless
$\eta_{sc} = \dfrac{V_{ret}}{V_{ch}}$	scavenging efficiency	dimensionless
$\eta_{tr} = \dfrac{V_{ret}}{V_{del}}$	trapping efficiency	dimensionless
$\lambda = \dfrac{V_{pure}}{V_{th}}$	excess air factor	dimensionless
$\lambda' = \dfrac{V_{ret}}{V_{th}}$	alternative definition of excess air factor	dimensionless
μ	flow coefficient	dimensionless
ρ	density	lb per cu ft
τ	time	sec
ψ	Nusselt coefficient	dimensionless
ψ_e	discharge coefficient	dimensionless

CHAPTER 4

EFFECT OF SCAVENGING ON BRAKE MEAN EFFECTIVE PRESSURE

4.1 Excess Air and Brake Mean Effective Pressure.

Since the power output of an engine is controlled by its air-taking ability, a mathematical relation must exist between the brake mean effective pressure of a two-stroke cycle engine and the thoroughness of its scavenging. This relationship can be derived [Schweitzer, 1943] as follows:

The brake horsepower output of an engine can be expressed as

$$\text{(4–1)} \qquad \text{bhp} = \frac{\dfrac{V_{disp}}{12}\text{bmep}\,\dfrac{2n}{c}}{33{,}000},$$

where V_{disp} is the displacement volume in cubic inches, n the revolutions per minute, and c the number of strokes per cycle. Naturally, the horsepower is also equal to the total fuel consumption divided by the specific fuel consumption

$$\text{(4–2)} \qquad \text{bhp} = \frac{F_h}{f} = \frac{V_{pure}\rho\dfrac{2n}{c}60}{r\,f\,1728} = \frac{V_{pure}\rho\dfrac{2n}{c}60}{r_{th}\lambda\,f\,1728},$$

where V_{pure} is the volume of pure air (under NTP conditions) in the cylinder before combustion (see Fig. 3–1) r, the actual and r_{th} the theoretical (chemically correct) air-fuel ratio, $\lambda = r/r_{th}$ the excess air factor, and ρ the weight of one cubic foot of air under NTP conditions = 0.0765 pound. From (4–1) and (4–2)

$$\text{(4–3)} \qquad \text{bmep} = 13{,}750\,\frac{0.0765}{\lambda f r_{th}}\frac{V_{pure}}{V_{disp}} = \frac{1050}{\lambda f r_{th}}\frac{V_{pure}}{V_{disp}}.$$

On the other hand, from (3–3) and (3–4)

$$\text{(4–4)} \qquad V_{pure} = \eta_p V_{ch} = \eta_p C_{rel} V_{disp},$$

which with (4–3) gives

$$\text{bmep} = \frac{1050}{\lambda f r_{th}}\eta_p C_{rel}.$$

This can also be written as

$$\text{(4–5)} \qquad \text{bmep} = 180\,\frac{0.4}{f}\,\frac{14.5}{r_{th}}\,\frac{\eta_p}{\lambda}\,C_{rel}.$$

Since in a diesel engine the numerical values of $0.4/f$ and $14.5/r_{th}$ are both approximately equal to unity, the bmep is approximately $180\,C_{rel}\eta_p/\lambda$, which is a very simple relation indeed.

It is evident that there is nothing in the derivation of the final formula (4–5) which does not apply equally to two- and four-stroke cycle engines. The concepts: delivery ratio, trapping efficiency, relative charge, and purity are as useful in describing the charging process of a four-stroke as a two-stroke cycle engine. The only difference is that with a four-stroke cycle engine without valve overlap the short-circuited air is zero, and in Fig. 3–1 the area below the dash line is missing. To a supercharged four-stroke cycle engine the analysis applies in its present form.

By using the relation (3–24), equation (4–5) becomes

$$\text{(4–6)} \qquad \text{bmep} = 180\,\frac{0.4}{f}\,\frac{14.5}{r_{th}}\,\frac{\eta_{sc}}{1 + (\lambda - 1)\eta_{sc}}\,C_{rel}$$

which relation is graphically represented by Fig. 4–1.* It is useful in predicting the engine output from a knowledge of efficiency of the scavenging.†

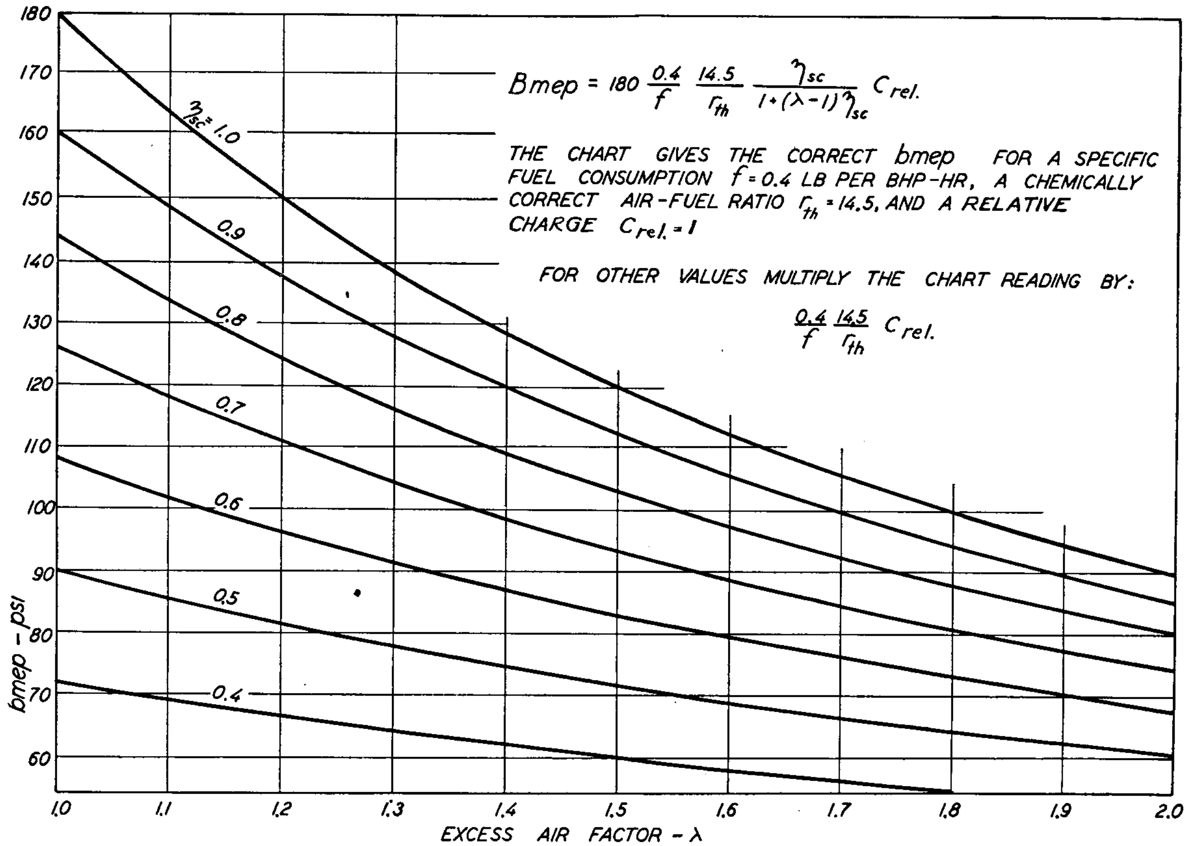

Fig. 4-1. Relation Between Excess Air Factor and Brake Mean Effective Pressure. This chart covers both four-stroke and two-stroke cycle engines at any load.

---

* Figure 4–1 is identical with Fig. 1 of Ref. Schweitzer, 1943, except that C_{rel} replaces η_{vol} which term has been abandoned.

† At this point it may be interesting to mention that by using the more or less conventional definition for volumetric efficiency: $\epsilon_{vol} = V_{ret}/V_{disp}$, equation (4–6) would lead to a paradox. From equations (3–3) and (3–5)

$$\epsilon_{vol} = \frac{V_{ret}}{V_{disp}} = \eta_{sc} C_{rel}$$

therefore C_{rel} in equation (4–6) would be replaced by ϵ_{vol}/η_{sc} and result in

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

It is seen that ϵ_{vol} being constant, the higher the scavenging efficiency η_{sc}, the lower is the bmep. For instance, assuming $0.4/f = 1$, $14.5/r_{th} = 1$ and $\epsilon_{vol} = 0.8$, with $\lambda = 1.5$ and $\eta_{sc} = 1$, bmep = 96 psi; with $\eta_{sc} = 0.8$, bmep = 103 psi, which is obviously absurd. Of course, it is not the algebra which is responsible for the absurd result but the improper definition of the volumetric efficiency, as explained in the preceding chapter.

Figure 4–2 is a crossplot of Fig. 4–1 and shows that the bmep increases slower than η_{sc} except for ($\lambda = 1$).

Another way to write equation (4–5) is by taking account of (3–24) and (3–25)

$$\text{(4–7)} \qquad \text{bmep} = 180 \frac{0.4}{f} \frac{14.5}{r_{th}} \frac{\eta_{ch}}{1 + \eta_{sc}(\lambda - 1)}$$

which shows that the power output increases directly with the air retained.

Any of the three equations (4–5), (4–6), or (4–7) may serve for predicting the engine output from its scavenging performance, and the one for which data are available will naturally be preferred. In case ($\lambda = 1$), any of them becomes

$$\text{(4–8)} \qquad \text{bmep} = 180 \frac{0.4}{f} \frac{14.5}{r_{th}} \eta_{ch},$$

which, in view of $\eta_{ch} = P$, could have been obtained from equation (4–3) as well.

This relation is shown in Fig. 4–3 for perfect scavenge and perfect mixing, which graph has been obtained by utilizing the relations between P and L as shown in Fig. 3–2 and 3–3.

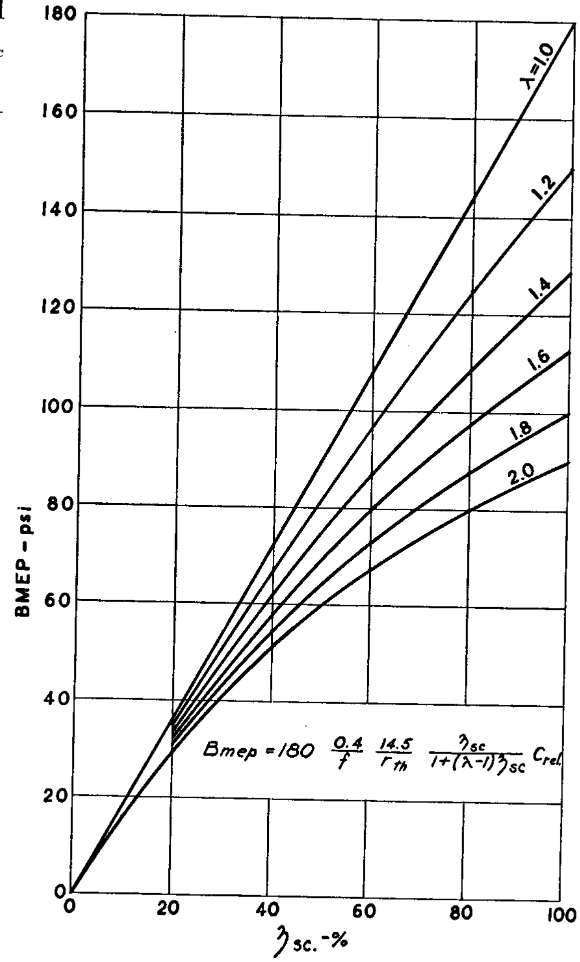

Fig. 4-2. BMEP vs Scavenging Efficiency. $f = 0.4$ lb per bhp-hr, $r_{th} = 14.5$, $C_{rel} = 1$.

4.2 Effect of the Relative Charge.

Both equations (4–5) and (4–6) contain the factor C_{rel} which represents the degree of filling of the cylinder. The relative charge is close to one in most separately scavenged engines but substantially less in crankcase-scavenged engines. It can be determined experimentally by measuring (with a sampling valve) the pressure and temperature of the charge at any instant after the ports closure. It can also be estimated fairly closely if the pressure and the temperature of the scavenge air are known.

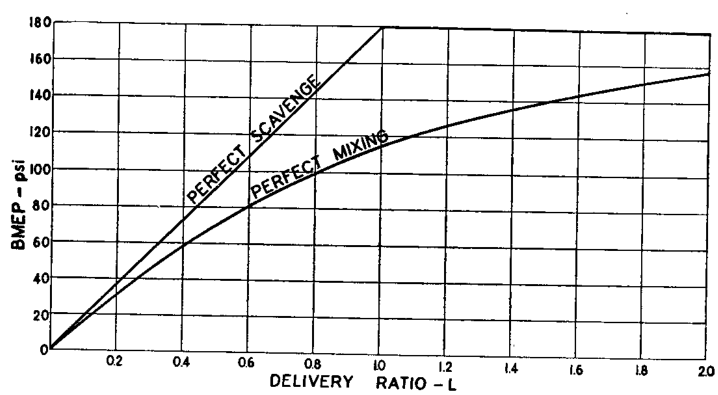

Fig. 4-3. BMEP vs Delivery Ratio at Zero Excess Air. $f = 0.4$ lb per bhp-hr, $r_{th} = 14.5$.

In symmetrically scavenged engines the pressure in the cylinder at exhaust port closure approximately equals the exhaust pipe pressure, which is little higher than atmospheric in atmospheric engines. Then, if the relative height of the exhaust ports is h_e

$$C_{rel} = (1 - h_e)\frac{T_0}{T_i} = (1 - h_e)\frac{520}{460 + t_{sc}}. \tag{4-9}$$

The actual relative charge may be somewhat higher than this if the cylinder pressure at exhaust port closure is higher than atmospheric, or somewhat lower if the cylinder pressure is subatmospheric at that time.

In nonsymmetrically scavenged engines the inlet ports close **after** the exhaust ports (or valves). After exhaust closure the cylinder pressure rises partly because of the piston motion and partly because of the continued flow of air from the inlet ports. In well-designed engines the cylinder pressure at inlet closure about equals the scavenge pressure. In that case

$$C_{rel} = (1 - h_e)\frac{p_i}{p_0}\frac{T_0}{T_i} = (1 - h_e)\left(1 + \frac{p_{sc}}{14.7}\right)\frac{520}{460 + t_{sc}} \tag{4-10}$$

The actual relative charge may be somewhat less in case the cylinder pressure does not quite equal the scavenge pressure at port closure.

Values of C_{rel} as calculated by equations (4–9) and (4–10) are listed for a number of engines in Table 18–I. Knowing C_{rel}, the engine bmep can be estimated on the basis of Fig. 5–8, 5–9, and 5–10. The estimated bmep may be compared with the actual or rated bmep listed in the same table. The deviations are due to the differences of the combustion efficiencies and of the scavenging efficiencies of similar types of engines.

CHAPTER 5

SCAVENGING ARRANGEMENTS

5.1 As was mentioned in Chapter 2, there are three general types of scavenging arrangements in use: cross scavenging, loop scavenging, and uniflow scavenging.

CROSS SCAVENGING

5.2 Cross scavenging is the type most commonly used. Exhaust ports are opposite the intake ports, either one occupying less than one-half of the cylinder circumference. To prevent short-circuiting, deflector pistons or upwardly directed intake ports are employed. The objective is to induce the scavenging air to move along the cylinder wall and mix with and/or sweep out the cylinder gases before reaching the exhaust ports.

Unsymmetrical deflector pistons such as those shown in Fig. 8–7 are no longer in favor except for very small engines because of the uneven heat stresses and the resulting piston distortions. Symmetrically shaped deflector pistons as shown in Fig. 9–1 are not always applicable to open combustion chambers and, therefore, in the latter case inclined intake ports are generally used as in Fig. 2–3 and 2–4.

Many disappointments have been experienced by assuming that upwardly inclined intake ports assure protection from short-circuiting. By making the airflow visible through magnesium powder fed with the scavenge air into a transparent cylinder model, Lindner has shown [Lindner, 1933] that the flow at cross scavenge assumes one of two fundamentally different characteristics. The air stream may adhere to the port and cylinder walls and travel up close to the cylinder head before being deflected. This is the desirable type of flow called by him *high scavenge.*

In other cases the air stream separates from the guiding surfaces and takes a short route across the cylinder to the exhaust. This *flat scavenge* is of course undesirable.

5.3 Duct Shapes.

The type of flow depends largely on the form of the intake ducts and on the piston travel or uncovered port height. Certain duct shapes favor flat scavenge, as the one shown in Fig. 5–1. Others are conducive to high scavenge as the one shown in Fig. 5–3. Finally there are duct shapes like the one shown in Fig. 5–2 that produce high scavenge and flat scavenge alternately. Generally at high piston position with small uncovered port height, high scavenge occurs, while with the piston approaching the bottom center position the flow pattern turns into flat scavenge.

The transition from high scavenge to flat scavenge and vice versa does not take place gradually but suddenly. Fig. 5–4 and 5–5 show model photographs with duct shape of Fig. 5–2, the piston being in the same position. While Fig. 5–4 shows pronounced flat scavenge with most of the air passing directly to the exhaust port, Fig. 5–5 shows perfect high scavenge with no visible short-circuiting.

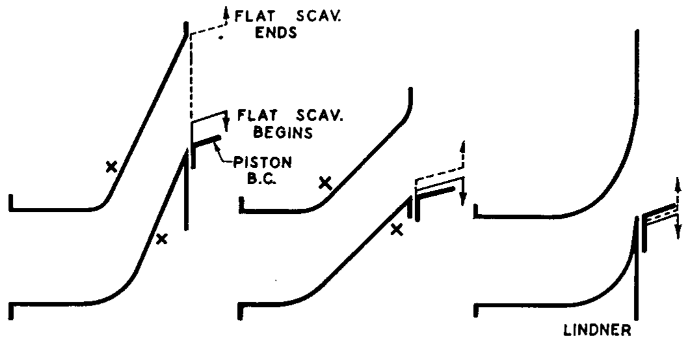

Fig. 5-1, 5-2, 5-3. Duct Shapes for Inlet Ports. Fig. 5-1 produces flat scavenge during almost the entire scavenging period, Fig. 5-2 only near the bottom of the piston stroke, while Fig. 5-3 is free from flat scavenge during the entire piston stroke. The crosses indicate locations of air stream separation.

It was found that in cases where a certain duct shape in a given piston position produces either high or flat scavenge, the character of the flow pattern depends on the side from which the piston approaches the given position. When the piston is going down, and traverses the overlapping region, high scavenge prevails; when the piston is going up, flat scavenge is observed at the **same piston position.**

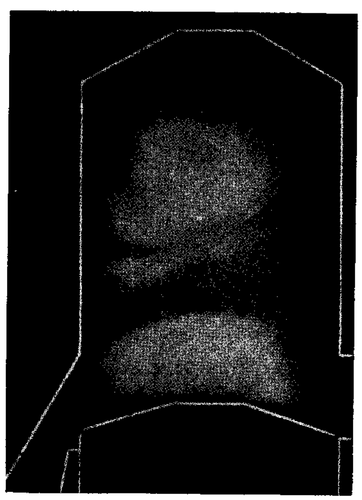

Fig. 5-4. Flat Scavenge with Most of the Air Passing Directly to the Exhaust Port (Right).

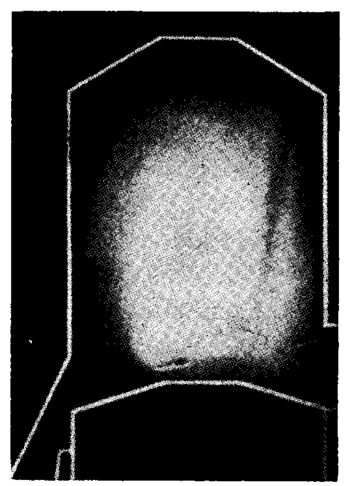

Fig. 5-5. High Scavenge with no Short-Circuiting Visible.

Fig. 5-4 and **5-5.** Model Photographs Taken with Magnesium Powder Mixed in the Intake Air. (Lindner.)

The region of overlap is sometimes surprisingly wide. In Fig. 5–1, 5–2, and 5–3, the piston positions corresponding to changes in the scavenging pattern are marked. The lower position always refers to down-going piston and indicates the piston position where high scavenge changes to flat scavenge. The higher position refers to up-going piston and indicates the change from flat scavenge to high scavenge. The bottom center position is also indicated on the figures.

With the duct form shown in Fig. 5–3 high scavenge prevails during the whole piston stroke. The indicated change-over to flat scavenge and vice versa are below bottom center. They can be observed on a model where the piston can be lowered below its bottom center but they cannot occur in an actual engine.

With the duct form of Fig. 5–1 high scavenge predominates during the down-stroke but it changes to flat scavenge at one-fourth of the port height. From then on flat scavenge persists during the remainder of the down- and the up-stroke of the piston. This is a very poor duct form.

With the duct form of Fig. 5–2, high scavenge prevails during all but the bottom quarter of the port height. Even then with down-going piston, high scavenge prevails down to 1/12 of the port height. In going up, flat scavenge remains until 21 per cent of the port height is reached.

The probable explanation for this irregular behavior in change-over from one type of scavenging flow to the other is found in the air swirls shown in Fig. 5–6 and 5–7.

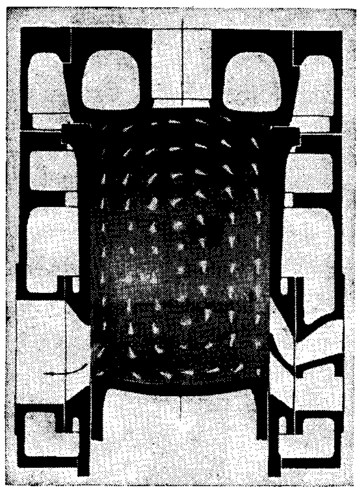

Fig. 5-6. High Scavenge with an Air Swirl in Counterclockwise Direction.

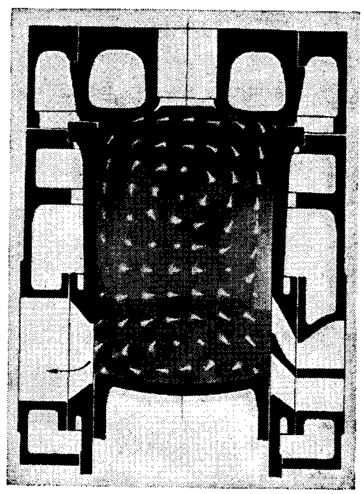

Fig. 5-7. Flat Scavenge with a Counterclockwise Swirl on the Bottom and a Clockwise Swirl Above It.

Fig. 5-6 and **5-7.** Air Current Direction Indicated by Silk Thread (Sulzer).

In Fig. 5–6 the air flow photographed by Sulzer [Sulzer, 1933] with the aid of small pieces of silk thread shows a single swirl of counterclockwise rotation. This is high scavenge. In Fig. 5–7 showing flat scavenge, two swirls are noted. One swirl on the bottom next to the piston is counterclockwise. Another large swirl above that is clockwise. With the main air stream blowing straight across, it is easy to see that the viscous friction of the air sets up a swirl of the opposite direction. The peculiar fact about these flow patterns is that they suddenly change from the one to the other, often without much provocation. Even that is less mysterious if we realize that a swirl once established has a momentum and that it tends to maintain its direction of rotation until it is forced to reverse it. After flat scavenge sets up a fast swirl, the latter does not change over to an opposite swirl as soon as the approaching air stream would favor high scavenge. Rather it opposes a change in rotation, and it takes considerable energy to destroy a swirl before a swirl of the opposite rotation is set up.

5.4 Stream Separation.

Further study [Lindner, 1933] revealed that eddies and stream separation in the duct are mainly responsible for flat scavenge. As long as the air stream follows the upper wall of the upperly inclined intake duct, the lower wall of the intake duct sends the stream up along the cylinder wall and a single swirl results which is characteristic of high scavenge. The nature of the boundary layer of the air next to the duct surface has a deciding influence upon the air stream. A turbulent boundary layer promotes stream separation. If the stream is forced to turn corners, a tendency to eddies and stream separation exists right after the curvature. In consequence, flat scavenge results with counter swirl.

One means to prevent stream separation used by the German Krupp firm [Ackeret, 1926] is by use of small aspirating holes in the cylinder wall above the intake ports, which, connected with the exhaust ports or with the suction side of the scavenging blower, suck away the turbulent boundary layer and direct the stream upward. The same objective can be achieved by suppressing the turbulent boundary layer and resultant stream separation by proper shaping of the intake ducts.

Lindner investigated a number of duct forms in models, by magnesium dust photography and hydraulic methods, and also in running engines. He found stream separation in the ducts Fig. 5–1 and 5–2 at the spots marked by crosses, while in the duct shown in Fig. 5–3 no stream separation was ever observed. He found that the stronger the decelerating forces that act upon the air stream, the sooner does a turbulent boundary layer cause stream separation. This is brought about by directional changes or increased cross-sectional area.

In order to avoid eddies and short-circuiting, in designing intake ducts, the following rules should be observed: The duct surface should always be curved convex in the same direction and the radius of curvature should increase or stay constant when approaching the cylinder. In the direction of the flow the cross-sectional area of the duct should preferably decrease slightly. Protrusions on the duct wall should be avoided, but moderate roughness characteristic of good cast iron surface was found harmless with regard to stream separation.

Generally the best port timing was found to be independent of the duct shape and the best duct shape independent of the port timing.

5.5 Scavenging Efficiency.

The scavenging efficiency of cross scavenge has been measured on models and on running engines. The results when plotted against delivery ratio (air delivered to the cylinder divided by the displacement volume) are fairly consistent for well-designed engines, except that the scavenging efficiency becomes lower with high stroke-bore ratio and also with high scavenge pressure. The curves shown on Fig. 5–8 were obtained experimentally by Hans List [List, 1932, 1933] a German engineer who made his experiments around 1930 in Woosung, China. The curves refer to engines with a stroke-bore ratio of 1.2 to 1.3. If the stroke is longer the scavenging begins to deteriorate at 1.5 stroke-bore ratio, and the scavenging efficiency figures must be reduced by 10 per cent.

Scavenging is also poorer with high scavenge pressure (for the same delivery ratio). The figure shows two experimental curves with heavy lines. The upper curve refers to low scavenge pressure of about 1.4 psig. The lower curve refers to high scavenge pressure 5 to 10 psig.

In the figure, for comparison, the lines of perfect scavenge and of perfect mixing are also shown. The actual amount of air retained by cross scavenging is seen to be somewhat less than by perfect mixing (complete dilution) if low scavenge pressure is used. With high scavenge pressure the air retained is considerably less.

Scavenging efficiency can be translated into brake mean effective pressure, because maximum engine output depends on the amount of air available for combustion.

Several assumptions have been made, such as:

1. The cylinder charge under NTP conditions is equal to the displacement volume ($C_{rel} = 1$).
2. The theoretical air-fuel ratio is 14.5 : 1 by weight. This corresponds to a carbon-hydrogen ratio of about 7 : 1, which is fairly typical of diesel fuels.
3. The excess air is 60 per cent, which means that the air actually available for combustion is 1.6 times the chemical ratio ($\lambda = 1.6$). The *actual* air-fuel ratio is, therefore, assumed to be $1.6 \times 14.5 = 23.2 : 1$; this is a good average figure for commercial engines. The nominal air-fuel ratio is higher because of short-circuiting.
4. The specific fuel consumption is 0.5 pound per bhp-hr. This is a fair average figure for two-stroke cycle engines although high-grade engines have better economy.

Under these conservative assumptions the maximum attainable bmep is shown on the right-hand scale of Fig. 5–8. With an air delivery of 60 per cent displacement volume, which represents average crankcase-scavenged engines, the scavenging efficiency reads 38 per cent and the attainable bmep is 45 psi. With 120 per cent air delivery, which represents average blower-scavenged engines, the scavenging efficiency reads 62 per cent and the attainable bmep becomes 65 psi. It must be borne in mind, however, that without supercharge the cylinder charge may be only 70 per cent of the displacement volume and the attainable bmep correspondingly 31.5 and 46 psi respectively. The relative charge depends on such factors as port height, scavenging air pressure, and temperature and compression ratio. These factors are not of concern now, in attempting to evaluate the various scavenging arrangements. Figure 5–8 represents good average conditions except as to relative charge, taken as 100 per cent, which can be attained (or exceeded) only by supercharging.

If the charge is 10 per cent less than the displacement volume ($C_{rel} = 0.9$), the bmep figure on the right-hand scale should be decreased by 10 per cent. If the fuel consumption is 10 per cent lower than 0.5 pound per bhp-hr, the bmep value should be increased 10 per cent. If the excess air factor is other than 60 per cent, equation (4–6) should be used.

These rules and Fig. 5–8 provide means for estimating the power obtainable from cross-scavenged engines. The conclusion is that cross scavenging is not very efficient. The attainable bmep is 31.5 and 46 pounds, respectively, for crankcase-scavenged and blower-scavenged engines, provided no supercharge is used. Furthermore, the benefit from supercharge is less than the amount of supercharge, because of the lower scavenging efficiency with high scavenging pressure. It must not be forgotten, however, that for many applications simplicity of construction is more important than high specific output.

LOOP SCAVENGING

5.6 Loop scavenging, also called *backflow scavenging*, is a more recent development and is gaining ground. The objective is so to direct the incoming air stream as to avoid short-circuiting to the exhaust ports. Sharply inclined intake ports in the conventional cross scavenging of a shape similar to that shown in Fig. 5–3 accomplish that purpose, but at the expense of reduced effective intake port area. Effective area is the narrowest cross sectional area perpendicular to the flow. With horizontal or slightly inclined intake ports, however, the trapping efficiency is low.

5.7 M.A.N. Scavenging.

In the M.A.N. version of loop scavenging (see Fig. 2–5*b*) the air is admitted in the transverse plane across the piston, and is forced to leave through exhaust ports placed above the inlet ports.

The ports are substantially horizontal; therefore their effective area is large. One drawback of M.A.N. scavenging is that the placement of the exhaust ports above the intake ports appreciably reduces the

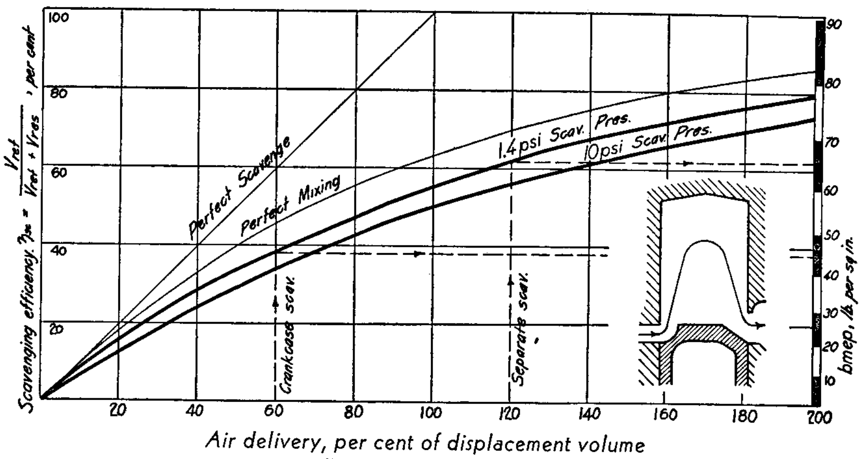

Fig. 5-8. Scavenging Efficiency of Cross Scavenging. Scale on right shows the bmep attainable with 60% excess air, 100% relative charge, and 0.5 lb per hp-hr fuel consumption. 70% of the bmep reading represents fair output for symmetrically scavenged engines and 100% for the supercharged engines.

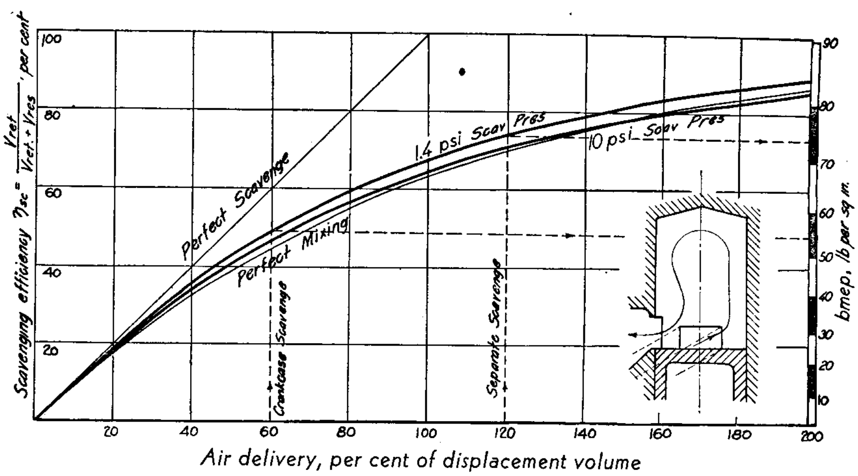

Fig. 5-9. Scavenging Efficiency of Loop Scavenging. Scale on the right shows bmep attainable with 60% excess air, 100% relative charge and 0.5 lb per hp-hr fuel consumption. 70% of the bmep reading represents fair output for symmetrically scavenged engines and 100% for supercharged engines.

effective piston stroke. This can be partly remedied by placing a mechanically actuated valve in the exhaust ports, which closes the exhaust much before the exhaust ports are covered. In this way the effective compression stroke is lengthened but the effective expansion stroke remains.

5.8 Schnuerle Scavenging.

In the Schnuerle version of loop scavenging (see Fig. 2–5*c*) the scavenge air is directed upward through slightly inclined intake ports on two sides and is forced to leave through exhaust ports placed between the intake ports. The obvious disadvantage of this arrangement is that only about one-half of the cylinder periphery can be utilized for accommodation of the ports, which restricts the air-taking capacity of the engine. The scavenging, however, is good and Schnuerle scavenging is successfully used in small engines.

5.9 Curtis Scavenging.

In the Curtis type of scavenging (see Fig. 2–4*d*) the side inlet ports are substantially horizontal while the center inlet ports are steeply inclined. The center ports are substantially radial while the side ports slope backward, away from the exhaust ports. The inlet ports occupy more than one-half of the cylinder periphery, leaving less than one-half to the exhaust ports.

This type of porting allows large effective inlet port areas in the side ports without promoting short-circuiting. According to model tests, the air streams from the side ports meet off the cylinder center and rise vertically together until deflected by the cylinder head. The steeply inclined center ports contribute some flow area but do not disturb the vertical flow of the air on the intake side of the cylinder. Between the horizontal side ports and the steep center ports moderately inclined ports are sometimes used.

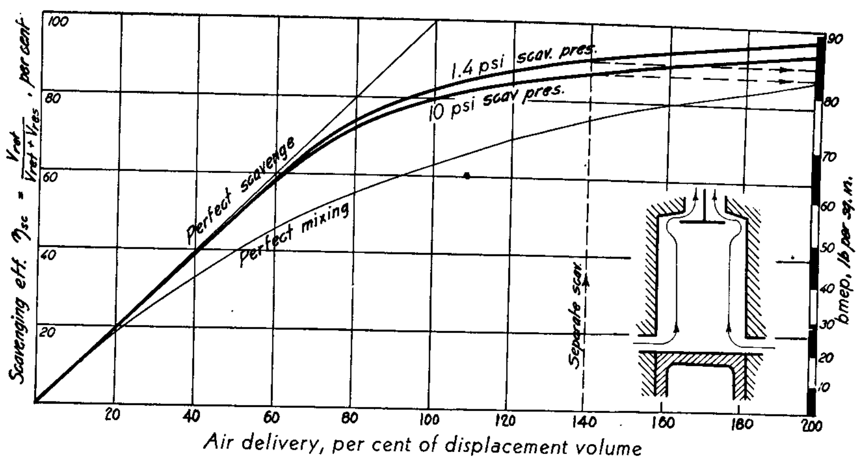

Fig. 5-10. Scavenging Efficiency of Uniflow Scavenging. Scale on the right shows bmep attainable with 60% excess air, 100% relative charge and 0.5 lb per hp-hr fuel consumption.

As with cross scavenging, the scavenging efficiency of loop scavenging depends a great deal on the skill of the designer. The characteristic curves obtained experimentally by List are shown in Fig. 5–9. For comparison the lines for perfect scavenge and perfect mixing are also shown. It is evident that the scavenging efficiency is considerably better than with cross scavenging. At high scavenge pressure it is about equal to that of perfect mixing and at low scavenge pressure it is higher. With 60 per cent air delivery, which represents average for crankcase-scavenged engines, the scavenging efficiency reads 48 per cent, compared with 38 per cent with cross scavenging. With 120 per cent air delivery, which represents the average with blower scavenge, the scavenging efficiency reads 74 per cent against 62 per cent with cross scavenging.

The corresponding bmep have been calculated under the same assumptions as before, and a bmep scale is shown on the right. The values for the selected cases are 54 and 73.5 psi, respectively, for the crankcase-scavenged and the separately scavenged engine. Again corrections need to be made for relative charge other than 100 per cent, for fuel consumption other than 0.5 pound per bhp-hr, and for excess air other than 60 per cent over the theoretical ratio. The scavenging suffers somewhat when high scavenge pressure is used (for the same delivery ratio) but not as much as with cross scavenging. Therefore the benefit from supercharge will be greater.

It should be noted, however, that the curves in Fig. 5–9 refer to an average loop-scavenged engine as of 15 years ago. Recently built loop-scavenged engines with Curtis-type scavenging show scavenging efficiencies 8 to 10 per cent higher than the values read from the chart. This means that with an economical delivery ratio of 1.2 the scavenging efficiency may reach $1.1 \times 74 = 81$ per cent, corresponding to 79 psi bmep with a cylinder charge that is equal to the displacement volume.

Loop scavenging has another advantage which does not appear on the charts. A longer stroke may be used. The figures on cross scavenging apply only with a stroke-diameter ratio not greater than 1.3. If the stroke is longer the scavenging deterioriates; in case of 1.5 stroke-diameter ratio the bmep figures must be reduced by 10 per cent. In the case of loop scavenging, the figures hold up to a stroke-diameter ratio of 1.6 with little or no correction.

UNIFLOW SCAVENGING

5.10 Uniflow or end-to-end scavenging unquestionably does the best job of purging the cylinder. The scavenging air goes straight through without change of direction and it acts like a piston in pushing out the burnt gases before the cylinder is filled with the fresh air charge. At least that is the ideal, even if the actual process falls short of it. Uniflow scavenging is perfect if the scavenging air has approximately the same axial velocity over the entire cylinder cross section. In practice that is not always the case. The scavenging air may pass through the center of the cylinder and leave the residual gases near the cylinder wall unscavenged or it may crowd the periphery and leave a core in the cylinder unscavenged. Both of these cases are to be avoided.

5.11 Turbulence.

A factor which complicates uniflow scavenging is air turbulence. Combustion in a diesel combustion chamber is greatly promoted by a motion of the air which helps to mix it with the fuel. In a high-speed engine, turbulence is indispensable no matter how perfect may be the spray dispersion and atomization. Turbulence is also desirable in a slow-speed engine. Cross scavenging and loop scavenging inherently furnish some sort of a turbulence, but with uniflow scavenging it must be provided specially.

The almost universal method for providing turbulence in a uniflow two-stroke cycle engine is through *tangential* inlet ports. Instead of being cut radially, the inlet ports in the cylinder wall are cut at an angle to the radial in the plane perpendicular to the cylinder axis. This produces a rotary air motion in the cylinder in addition to the axial flow through it. An air particle may be viewed as describing a helix while it goes through the cylinder. It has been proved [Sass, 1929; Lee, 1939] that the subsequent compression does not damp out the rotary swirl of the air; on the contrary it frequently becomes more pronounced at the time when the piston reaches top dead center, because of the squeeze between piston crown and cylinder head. Uniflow engines without air swirl invariably give poor performance. The angle of the inlet ports affects both the combustion and the scavenging. Combustion demands an intense turbulence which can be produced by more nearly tangentially directed inlet. That, however, may crowd the air to the periphery, leaving a poorly scavenged core in the center. Good scavenging requires a uniform dispersion of the inlet air over the cylinder cross section, which can be obtained by proper selection of the inlet port angle.

5.12 Types of Uniflow Scavenging.

It has been mentioned before that the three known types of uniflow scavenging are (1) poppet valves, (2) opposed piston, and (3) slide or sleeve valve, as shown schematically in Fig. 2–6. List's characteristic curves refer to a Junkers opposed-piston engine and are reproduced in Fig. 5–10. They can probably be used without change for the poppet valve and sleeve valve arrangements.

The scavenging efficiency of uniflow scavenging is notably higher than that of the other two sets. Below 80 per cent air delivery it approaches the line of perfect scavenge and in the practical range, which is around 140 per cent air delivery, the scavenging efficiency is 90 and 86 per cent at 1.4 and 10 psig scavenge pressure, respectively. The deterioration of scavenging at high scavenge pressure is moderate, which makes supercharging worth while.

The attainable bmep is about 84 psi for 1.4 and 82.5 psi for 10 pounds scavenge pressure at 140 per cent air delivery (assuming the same relative charge). The expected bmep were calculated in the same manner as formerly, although it may be justifiable to modify the assumptions inasmuch as uniflow engines average more nearly 0.4 pound of fuel consumption per bhp-hr. Better combustion and higher mechanical efficiency are responsible for the improved fuel economy. Consequently, the attainable bmep would be 105 and 103 psi. Nevertheless, to facilitate comparison with the previous charts, the same bmep scale is used.

The line for crankcase scavenging has been omitted from the chart as no uniflow crankcase-scavenged diesel engines are being built. Yet the scavenging performance at low air deliveries should not go unnoticed. Up to about 60 per cent air delivery uniflow scavenging is practically identical with that of perfect scavenge. *All* air delivered to the cylinder is retained by it. This means that short-circuited air is zero. The air truly acts as an imaginary piston, pushing out the gases without going out with them. This has significance for engines the delivery ratio of which drops at low load or slow speed. Reduction of the intake air delivered by the blower at part load operation would have such an effect. Uniflow scavenging would permit such engines to operate more efficiently by reducing the work of the blower.

Comparing Fig. 5–8, 5–9, and 5–10, it becomes clear that considerable differences exist between the scavenging performances of various methods. These differences affect the attainable engine output correspondingly.

CHAPTER 6

PORTING

6.1 The difference in the scavenging efficiencies of the various scavenging methods goes far to explain the differences in performance of two-stroke cycle engines. An equally important factor is the shaping and dimensioning of the ports. Proper dimensioning of the inlet and exhaust ports is not a simple task. The unsteady flow of air and hot gases through the variable openings cannot be represented by simple equations. Simplifying assumptions must be used. Even so, complete computations are rather laborious.

Laying out the ports requires judgment also, if best results are to be obtained, as all factors can never be included in the calculations. In addition to the factors ordinarily considered, such as engine dimensions, speed, and load, optimum porting depends on such circumstances as:

1. The characteristics of the scavenging blower
2. The effect of the intake systems including air cleaners, silencers, receivers, and ducts
3. The effect of the exhaust system, including exhaust ducts, header, muffler, and piping
4. The altitude
5. Whether the engine is to operate at a constant speed or variable speed most of the time
6. Whether the load is to be constant or vary much in service
7. Whether high power output or high fuel economy is desired most.

These factors do not figure in the formulas, although their combined effect may exceed the effect of some essential variable and may change the porting dimensions considerably. For this reason it is useless to strive for extreme accuracy in the calculations. There is no point in worrying about the fourth decimal in applying a formula when some of the disregarded factors play havoc with the third decimal.

In view of these uncertainties, and also of the known aversion of the practical designer to involved mathematics, a deliberate attempt has been made to simplify porting calculations to a greater extent than has been done heretofore. Even if the methods used to obtain the simple formulas have been anything but simple, the final conclusions are expressed in easy recipes, amply illustrated with examples.

6.2 Essence of Port Design.

The essence of port design is to provide sufficient port areas at the right time. The sequence of the engine cycle requires first removal of the exhaust gases from the cylinder, and then filling the cylinder with fresh air. Both of these events must take place rapidly while the piston is near bottom center. Consequently, optimum porting requires that at first a large exhaust area be uncovered while the inlet ports are closed, and then for a short overlap period that both ports be open for effective scavenging, and finally that the inlet ports be opened wide while the exhaust ports are closed or nearly closed. Unfortunately freedom in port design is rather limited by the kinematics of the engine, and the ideal can never be attained or even closely approached.

6.3 Scavenging Arrangements.

Figure 6–1 depicts schematically the port area diagrams of six scavenging arrangements. That at *a* shows ordinary piston-controlled ports. Since the piston motion is symmetrical to the bottom center, it is impossible to have both large exhaust openings before and large inlet openings after bottom center. The selection of the exhaust- and inlet-port opening times determines their closing times and the port heights.

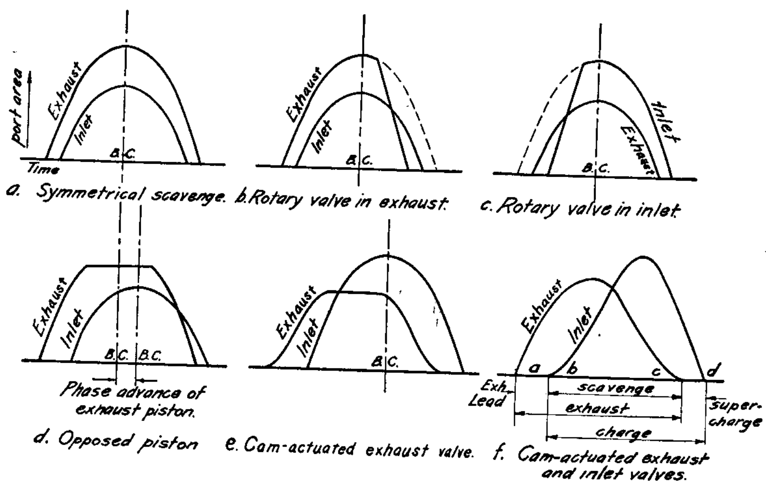

Fig. 6-1. Porting Diagrams of Various Scavenging Arrangements. Uncovered port area vs crank rotation.

One more degree of freedom can be had by placing a mechanically actuated or automatic valve in the inlet or exhaust connections, shown at *b* and *c*. In that way either the inlet or the exhaust can be unsymmetrical, which permits better porting and supercharge. A somewhat similar effect can be realized with the opposed piston by employing a phase difference between the exhaust and inlet pistons, as shown at *d*. Still more freedom is afforded by the use of cam-actuated exhaust valves (or inlet valves), as at *e*. Complete kinemàtic freedom is achieved by actuating both the inlet and the exhaust valves by means of cams (*f*) but this arrangement is mechanically unattractive and, if all valves are placed in the cylinder head, is conducive to short-circuiting.

The objective in each case is to trap as much fresh air as possible for combustion without sacrificing much of the effective stroke for the purpose. No part of the stroke would be sacrificed if the whole scavenging and charging process took place at bottom dead center, which, of course, is impossible. But an excellent job can be done in this respect with an arrangement such as in Fig. 6–1*f*. Referring specifically to the latter diagram, the cylinder pressure drops rapidly during the exhaust-lead period *a–b*, so that no exhaust gas can enter the inlet when the latter opens. During the scavenging period *b–c* the residual gases are blown out and replaced by fresh air. Near the end of the scavenging period *c* the inlet area greatly exceeds the exhaust area, which causes air pressure to build up. This becomes more pronounced during the supercharge period *c–d* when the exhaust is closed, while the inflow of air continues until the inlet closes.

The timing of *a*, *b*, *c*, and *d* are important. The period *a–b* must be long enough to permit sufficient blowdown of the cylinder content, *b–d* must be long enough to scavenge the cylinder with fresh air, c–d must be long enough to reach the desired supercharge, and d must not be so late that the compression in the cylinder has caused a reversed flow and loss of air.

6.4 Kinematic Limitations.

The more the kinematic limitations, the less can all these requirements be satisfied. The least attractive in this respect is symmetrical scavenge. With it supercharge is impossible. Sufficient exhaust lead severely diminishes the inlet time-area or reduces unduly the effective compression and expansion strokes. The cylinder **loses** air between the closure of the inlet and exhaust ports. The other cases are intermediate.

In cases other than piston-controlled ports, the porting layout should be preceded by a kinematic layout of the piston motion and control mechanism. A porting should be selected on that basis and the porting so determined should be checked by calculations. If unacceptable, another selection should be made until one is found which proves to be acceptable.

6.5 Inlet Ports.

In laying out the porting, it is customary to begin with laying out and calculating the inlet ports. The total width of the inlet port should be made as large as possible to permit large air flow with low ports. The bridges between the ports cannot be reduced beyond a certain minimum because of the necessary mechanical strength, the space required for cooling, and other considerations.

Inlet ports are seldom made radial; they are inclined in the axial or in the transverse plane, or in both. The object of the inclination is to direct the flow in a predetermined manner, to obtain good scavenging by minimizing short-circuiting in cross- or loop-scavenged engines. In uniflow engines the inlet ports are made to form an angle of 15 to 60 degrees with the radial to create a rotary swirl of the air, which favors combustion. To provide effective guidance to the air, the inlet ports must not be too wide, relative to their depths. Narrow ports were found to admit as much air per square inch of port area as did wide ports [Waldron, 1941], and to direct the air much more effectively. As to the number and spacing of ports, consideration should also be given to the action of piston rings. No rectangular ports should be much wider than 15 degrees if free-floating piston rings are used, or 30 degrees if the rings are pinned. Nonobservance of this rule may lead to breakage of rings. Both considerations favor narrow ports. Wide ports on the other hand are cheaper to make and do not clog up so easily.

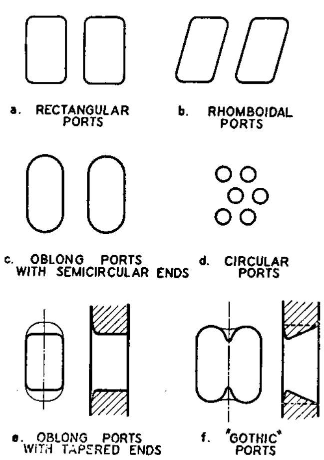

Fig. 6-2. Port Shapes.

6.6 Ring Breakage.

Ports are most frequently made rectangular with a slight rounding of the corners, as shown in Fig. 6–2a; this shape offers the greatest flow area with a given bridge width. A possible objection is that they are prone to cause ring scoring, especially if the rings are pinned. The rhomboidal ports shown in b avoid that. Both rectangular and rhomboidal ports are disadvantageous from the viewpoint of ring breakage caused by impingement of the ring on the port edge. This is particularly dangerous if an end of the floating ring happens to be in the port. To prevent ring breakage the upper and lower edges of the port are frequently rounded.

Another way to avoid ring breakage is to make the upper and lower edges semicircular as shown in c. Rows of circular ports are sometimes used for inlet but they are not recommended for exhaust ports because of their greater tendency to clog with carbon. Both *c* and *d* give less port area with a given bridge width than do rectangular or rhomboidal ports.

In order to protect rings from breakage without excessive rounding of the edges, which causes gas blow-by immediately before the ports open and after they close, ports shown in *e* have a slight taper relief. The taper relief gradually compresses the protruding part of the ring and thereby prevents breakage. Such taper relief requires laborious filing or grinding with hand tools.

6.7 Gothic Ports.

The author's favorite is the *Gothic* or lip ports, shown in *f*. They have taper-relieved lip in the center. Figure 6–3 shows some dimensions for a 2¾ in. wide exhaust port. The lip, which has a gradual relief from 1/32 inch at the nose to nothing at the root, does not break the ring, and it actually cuts the unsupported span of the ring in two. At the same time it reduces port area only slightly. Fig. 6–4 shows the photograph of a Gothic port.

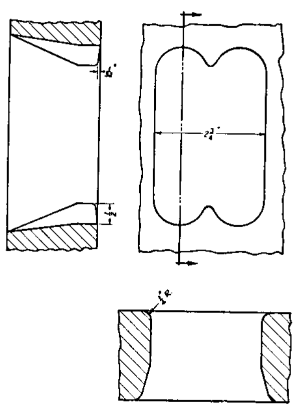

Fig. 6-3. Gothic Exhaust Port. Note relieved lip and rounding of side edges.

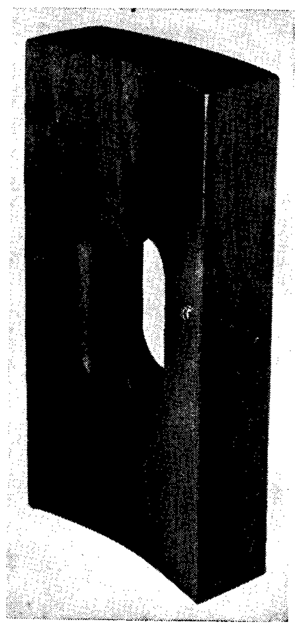

Fig. 6-4. Photograph of Gothic Exhaust Port with Rounded Side Edges.

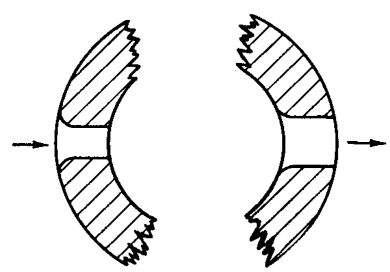

Fig. 6-5. Cross Sections of Intake and Exhaust Ports.

6.8 Rounding of Edges.

Figure 6–5 shows cross sections of inlet and exhaust ports. The inlet ports have sharp downstream edges to afford best guidance to the incoming air stream. The upstream edges should be flared to promote air flow. The exhaust ports on the other hand should have flared inside edges. The rounding of the upper edge has the disadvantage of permitting gas blow-by just before the exhaust port opens. The high-pressure, high-temperature exhaust gas flowing with a high velocity has detrimental scoring effect; therefore appreciable rounding of the upper edge is not recommended. There is no valid objection, however, to a liberal rounding of the side edges of the exhaust ports. This increases the flow, which is very beneficial especially during the blowdown period. According to tests made by the author on a 2¾ by 4½-inch exhaust port, rounding of the side edges to ¼-inch radius increased the flow by 22 per cent and is therefore equivalent to a 22 per cent increase in port width.

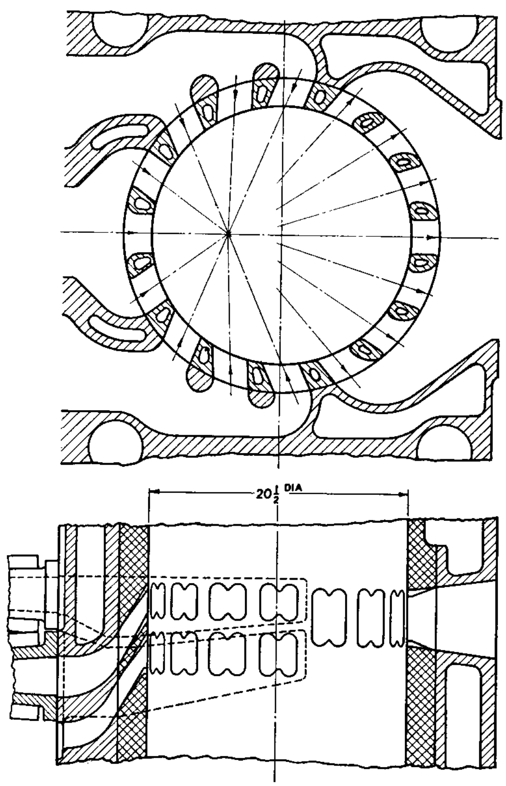

Fig. 6-6. Gothic Ports with Water-Cooled Bridges.

6.9 Bridges.

The greater the percentage of the periphery that is occupied by the ports, the greater the flow. Port widths are limited, however, by the necessary bridge width. A certain bridge width is necessary for strength and this usually must be increased for the sake of cooling. Overheating of the bridges causes them to bulge inward and seize the piston. Narrow exhaust port bridges are particularly sensitive to overheating. The most effective way of preventing overheating of the bridges is by water-cooling them through cast-in, drilled or milled holes. Figure 6–6 shows the cross section and longitudinal section of a liner with Gothic ports and water-cooled bridges.

If no water cooling is used the width of the bridges or spacing of the ports must be such as to prevent the damming up of the heat in the bridges. The cylinder is frequently relieved in the bridge region but too much relief is an undesirable expedient.

6.10 Shaping of Ports.

Ports are either cast or machined. Casting is cheaper but gives poorer results. For large engines cast ports are popular. Usually they are hand-dressed afterward. Figure 6–7 shows the milling of ports in a large cylinder liner.

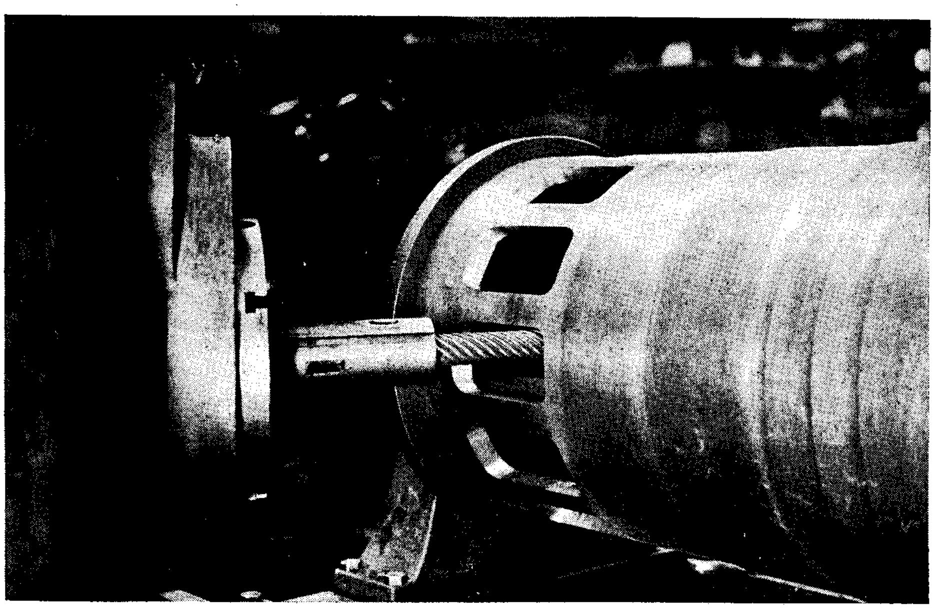

Fig. 6-7. Milling of Ports in Cylinder Liner. (By permission from F. Sass, *Kompressorlose Dieselmaschinen*. Copyright, 1929 by Julius Springer in Berlin.)

A port of a given peripheral area accommodates the greatest flow if both its side and end walls are perpendicular to the cylinder wall. Radial ports, however, are not always practical. For cross

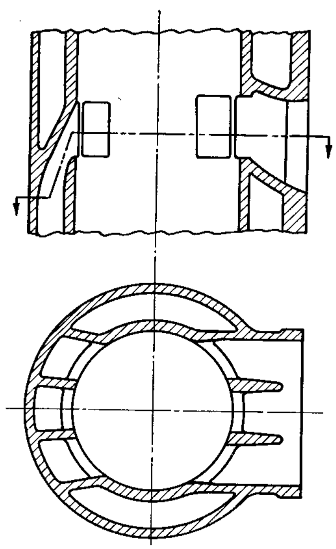

Fig. 6-8. Ports of Typical Cross-Scavenged Engine.

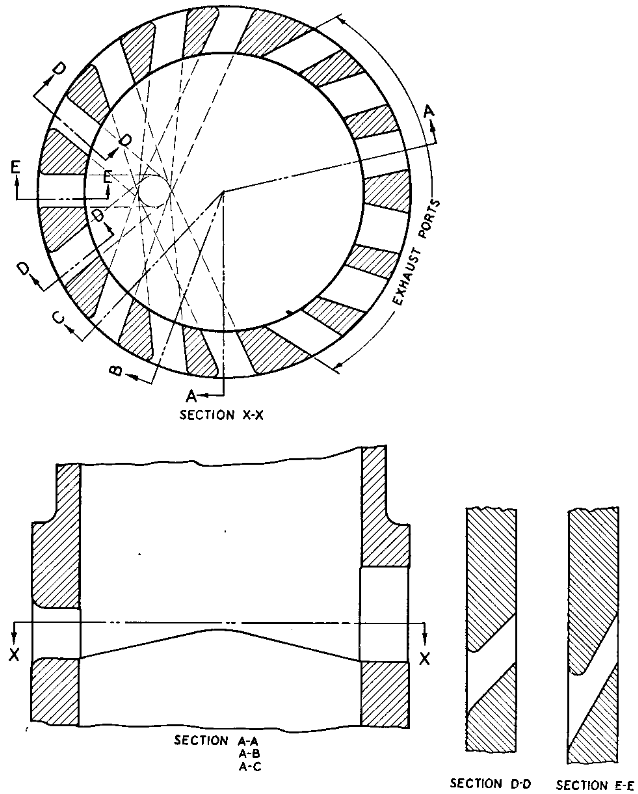

Fig. 6-9. Ports of Loop-Scavenged Engine with Curtis-Type Scavenging.

scavenging the inlet ports must be slanted upward from 45 to 60 degrees, unless deflector pistons are used, in order to avoid short-circuiting. In the transverse plane the ports are more or less radial to

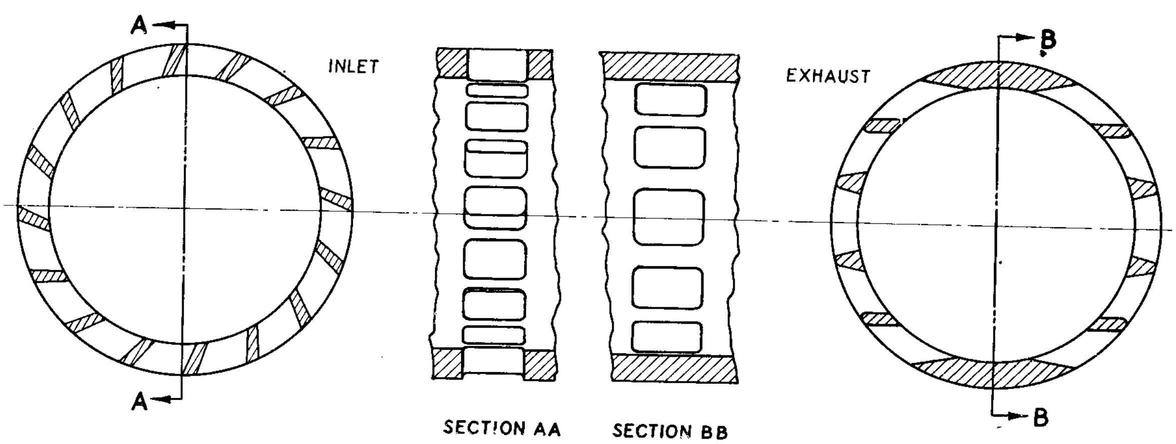

Fig. 6-10. Ports of a Typical Opposed-Piston Engine.

provide the greatest possible flow area. Figure 6–8 shows the ports of a typical cross-scavenged engine.

In loop scavenging the exhaust ports are radial or close to radial, but the inlet ports have angularities to force the inlet air to make a loop in the cylinder. Figure 6–9 shows the ports of a typical loop-scavenged engine with Curtis-type scavenging.

The center port of section E is radial in the transverse plane but is inclined 60 degrees upward in the vertical plane. Ports of sections A, B, C, and D are all directed toward a spot about halfway between the cylinder and periphery at the center port. In the vertical plane D has 45 degree inclination, while A, B, and C have zero degree inclination to the transverse plane. The exhaust ports are more or less radial.

In uniflow-scavenged engines the inlet ports usually have an angle to the radial in the transverse plane and zero or negligible inclination in the vertical plane. The exhaust ports are either radial or are directed toward the exhaust duct. Figure 6–10 shows the ports of a typical uniflow (opposed-piston) engine.

After the shape, width, and angularlity of the inlet and exhaust ports have been fixed, their heights are determined by calculation. The procedure is slightly different for various scavenge systems.

CHAPTER 7

INLET PORTS

7.1 Depending on the method of scavenging, the calculations of the inlet ports vary, but the principle of the calculations is always the same. It is necessary to provide sufficient time-area adequately to fill the cylinder with the available scavenging pressure head. It should be borne in mind, however, that the available pressure head is not a predetermined quantity, and the word "adequate" also admits of various interpretations.

Cylinder bore, stroke, and revolutions per minute being given and the scavenging method having been chosen, the first step ordinarily is to select the amount of air delivery.

If the underside of the power piston is employed for scavenging, as in crankcase-scavenged engines, the delivery ratio is naturally less than one, being equal to the volumetric efficiency of the air pump which has the same bore and stroke as the power piston. In view of the large clearance volume which contains the entire crank mechanism, the delivery ratio is seldom in excess of 70 per cent and frequently it is much less.

7.2 Choice of Delivery Ratio.

With step-piston or separately scavenged engines, it is to a certain extent a matter of choice how much air it is desired to deliver to the cylinder. The more air delivered, the more is retained by the cylinder and the more fuel the engine can burn and the more power it can develop. However, the amount of air retained does not increase in proportion to the air delivered. Assuming a relative charge of 100 per cent which approximately prevails except in superatmospheric engines (Chapter 15), it is seen from Fig. 5–8 that by doubling the delivery from 100 to 200 per cent of the displacement volume the air retained increases only from 55 to 80 per cent or from 50 to 74 per cent, which amounts to less than 50 per cent. The reason for this is that the short-circuited air increases faster than does the delivered air. On the other hand, the furnishing of scavenge air takes power, which puts a limit to the delivery ratio that can be economically employed. **Designers commonly make the mistake of furnishing *too much* air to the engine,** instead of making sure that the air furnished really gets into all of the cylinders and that as large a portion of air as possible is retained. Some well-designed engines have high power outputs with surprisingly small delivery ratios. They have ample air-box volume, adequate port areas, and the ports and ducts are so shaped as to discourage short-circuiting from inlet to exhaust. Neither from the standpoint of first cost nor of fuel economy is it practical to use a very large blower or pump in order to obtain a slight increase in power.

PROCEDURE IN DETERMINING INLET-PORT DIMENSIONS IN SEPARATELY SCAVENGED ENGINES

7.3 In selecting a delivery ratio for normal separately scavenged engines, 1.4 is a good figure to start with. If high output is of paramount importance and/or a cheap source of air is available from a high efficiency blower, a delivery of 1.5 times displacement volume may be feasible. If good fuel economy is important and/or in large and/or high-speed engines 1.3 V_{disp} or even less may be used.

7.4 Selection of Scavenge Pressure.

The next step is the selection of the scavenging pressure, which usually varies between 2 and 10 psig. List [List, 1937] maintains that the correct scavenging pressure is independent of the system (cross, loop, uniflow) and depends only on an index number which is a function of a few significant factors. Figure 7–1 gives the results of his calculations in English units. Besides the cylinder diameter and speed, only the relative width of the inlet ports has an influence on the selection of the scavenge pressure.

7.5 Width of Ports.

The total width of the inlet ports should be made as large as possible. With uniflow scavenging, when these ports extend all around the periphery, the total width may be as much as 75 per cent of the circumference, but ordinarily do not exceed 65 per cent. In the case of angular inlet ports this value must be multiplied by the cosine of the flow angle to the radial. With cross scavenging the total width of the inlet port usually is 30 to 35 per cent of the circumference. If the inlet and exhaust ports are confined to one side of the cylinder, the total width may be only 20 to 25 per cent of the circumference, whereas in the Curtis type of port arrangement it may reach the figures given for cross scavenging. In any case where the bridges between the ports are water-cooled, lower values must be selected.

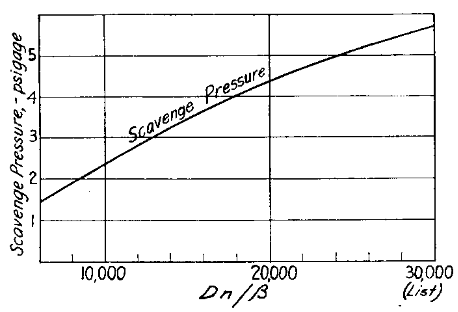

Fig. 7-1. Selection of Scavenge Pressure. D = Cylinder diameter in inches, n = Revolutions per minute, β = The ratio of the inlet port width (square to the flow) to the circumference = $b_i/D\pi$.

Although the scavenge pressure selected on the basis of Fig. 7–1 may prove to be somewhat low for high-output engines, it is a good figure with which to start. A method for determining the scavenge pressure that would give maximum horsepower is outlined later.

7.6 Mean Inlet Velocity.

After the scavenging pressure has been selected the next step is the determination of the mean inlet velocity. For this purpose the chart in Fig. 7–2 is used. The derivation of the respective formula together with the definition of the scavenge factor S is given in the appendix at the end of this chapter. According to the chart the mean inlet velocity depends on the ratio of scavenge pressure to external pressure, on the temperature of the scavenge air, and on the scavenge factor.

7.7 Scavenge Factor.

In using the chart of Fig. 7–2 for the determination of mean inlet velocities the value of the scavenge factor S must be known. The scavenge factor S must be selected on the basis of experience. In case no experimental data are available, Table 7–I is a guide for selecting the scavenge factor.

The figures refer to medium scavenge pressures. For high scavenge pressures somewhat higher values, for low scavenge pressures somewhat lower values, should be selected.

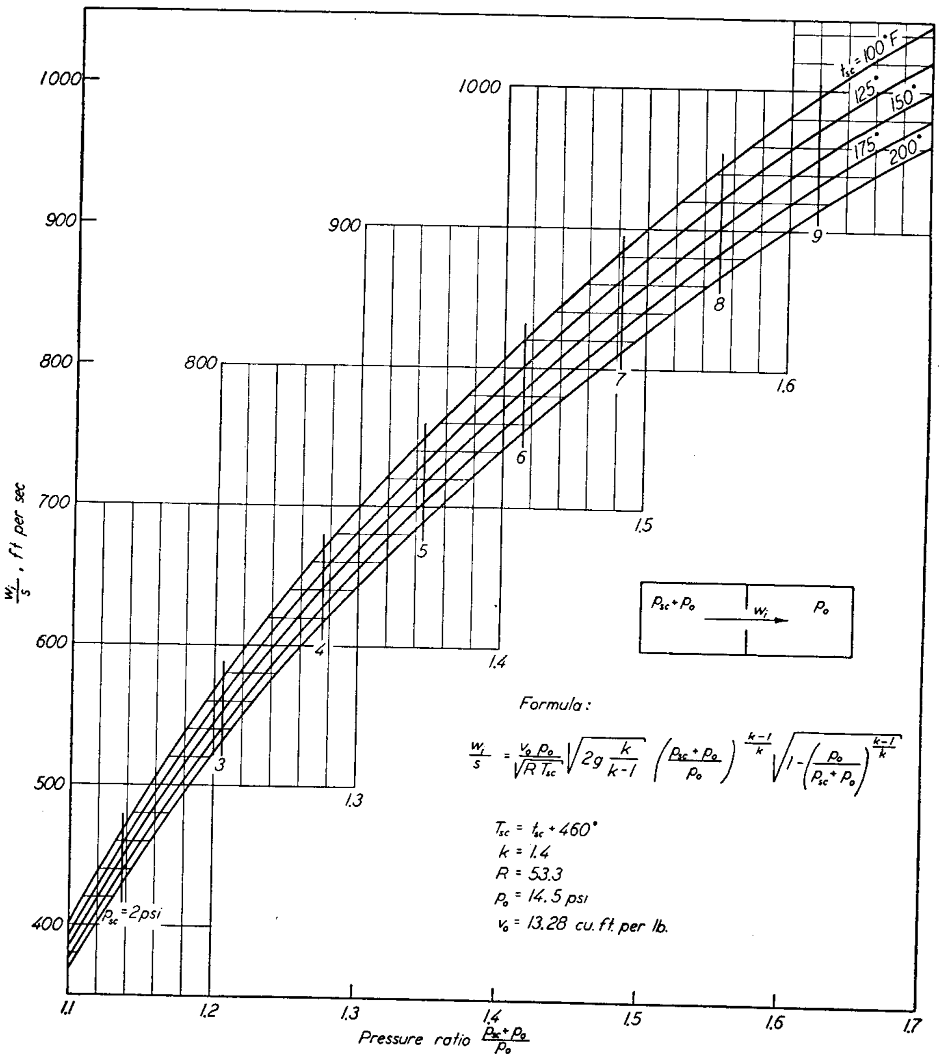

Fig. 7-2. Chart to Determine Mean Velocities. To obtain the mean velocity, multiply the reading on the vertical scale by the estimated scavenge factor which may be taken from Table 7-1.

It is noted that the scavenge factor is much higher in symmetrically scavenged engines than in engines with considerable supercharge. In the latter case, exhaust restriction materially reduces the air flow. Rough walls, sharp entry edges, and high exhaust pressures also reduce the scavenge factor, while high scavenge pressure increases it slightly.

The scavenge-air temperature can be calculated on the basis of the blower characteristics (see Section 11.17), but its effect is small and estimated scavenge-air temperatures will give fairly accurate results. Up to 3 psig scavenge pressure the scavenge-air temperature can be taken as 125 F. Between 3 and 5 psig 150 F may be used. In the case of high scavenge pressures, 6 psig or higher, the scavenge-air temperature should be calculated or measured for accurate porting calculations.

Table 7-I. For the Selection of Scavenge Factor S.

	TYPE OF SCAVENGE	WITH ROUNDED ENTRY EDGES AND SMOOTH PORTS	ROUGH PORTS	AVERAGE
Symmetrically scavenged engines	Cross	0.9	0.7	0.8
	Loop	0.65	0.45	0.5
Unsymmetrically scavenged engines, considerable supercharge	Cross	0.5	0.35	0.4
	Loop	0.35	0.2	0.25
	Uniflow	0.5	0.35	0.4
Unsymmetrically scavenged engines, slight supercharge	Cross	0.65	0.55	0.6
	Loop	0.4	0.3	0.35
	Uniflow	0.65	0.55	0.6

7.8 Time-Area.

The mean inlet velocity being known, the required inlet time-area is calculated by the following formula:

$$A_{im}\alpha_i = \frac{L}{2w_i} V_{disp}n \tag{7-1}$$

The derivation of this formula is found in the appendix at the end of this chapter.

In this formula A_{im} is the mean inlet-port area at right angles to the flow in square inches (see Fig. 7-3), α_i the inlet duration in degrees of crank angle, L the delivery ratio, V_{disp} the displacement volume, w_i the mean inlet velocity, and n the rpm.

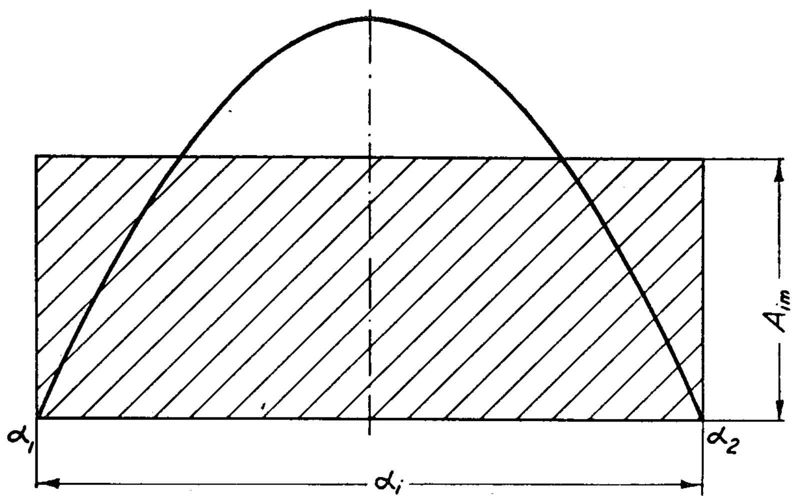

Fig. 7-3. Calculation of Inlet Port Dimensions. The formula is $A_{im}\alpha_i = \dfrac{L}{2w_i} V_{disp}n$; w_i is to be taken from Fig. 7-2.

Equation 7-1 tells how large a time-cross-sectional area is required to fill the cylinder by the available pressure drop. This area is in direct proportion to the cylinder displacement, delivery ratio, and rpm, and in inverse proportion to the mean inlet velocity. Of course, to get a certain time-cross-sectional area it is possible to use either large port openings A_i during a short time α_i, or small port openings during a long time. The former is clearly preferable as it permits longer effective strokes. Unfortunately in case of piston-controlled inlet ports, which are used almost exclusively, the port height and inlet duration cannot be chosen independently. The upper edge of the inlet ports is already determined by the inlet duration α_i. The relation is shown in Fig. 7–4, which is a plot of the piston travel against crank rotation for a 4.5: 1 connecting rod-crank ratio.

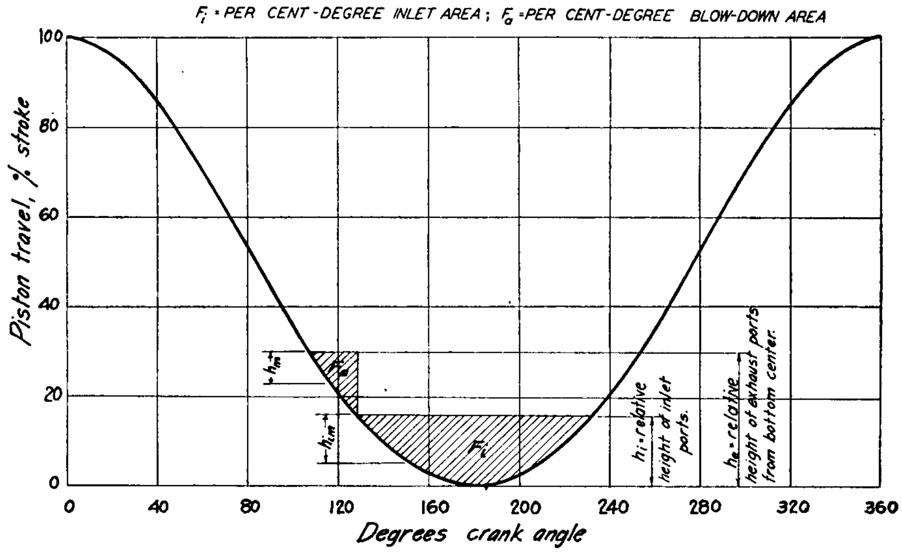

Fig. 7-4. Piston Travel vs Crank Rotation for Connecting Rod-Crank Ratio of 4.5:1. F_i = Per cent-degree inlet area; F_a = Per cent-degree blowdown area.

If the inlet ports are rectangular (or rhomboidal) with a total width of b_i, then

$$A_{im} = b_i h_{im} s \tag{7-2}$$

where h_{im} is the average uncovered height of the inlet port in relation to the stroke s during the inlet period. It is convenient to figure in terms of *per cent-degrees*. By denoting $h_{im}\alpha_i = F_i$, the inlet time-area $A_{im}\alpha_i$ can be expressed as

$$A_{im}\alpha_i = b_i s F_i \tag{7-3}$$

where F_i is the lower shaded area in Fig. 7–4.

7.9 Piston-Controlled Ports.

If the inlet ports are piston-controlled and extend, as they should, to the edge of the piston at the bottom center position, Fig. 7–4 makes it clear that the inlet duration α_i determines both h_i and F_i. Figure 7–5 which was constructed from the relationship represented in Fig. 7–4 is helpful in computations. If one of the quantities, α_i, h_i, h_{im}, or F_i is known, the others can be read from the figure. From equations (7–1) and (7–3) we have

$$F_i = \frac{L}{2w_i}\frac{V_{disp}}{b_i s}n \tag{7-4}$$

which permits the calculation of F_i from known quantities. From Fig. 7–5, F_i being known, α_i and h_i can be determined. If the inlet ports are not piston-controlled, the original equation (7–1) is used by determining A_{im} graphically.

7.10 Optimum Scavenge Pressure.

Because of the presence of w_i in equation (7–1), the result of the inlet-port calculation always depends on the selection of the scavenge pressure. The lower the scavenge pressure the longer must be the inlet duration. A long inlet duration reduces the effective compression stroke and therefore the output of the engine. From this standpoint a high scavenge pressure is preferable. But it takes more

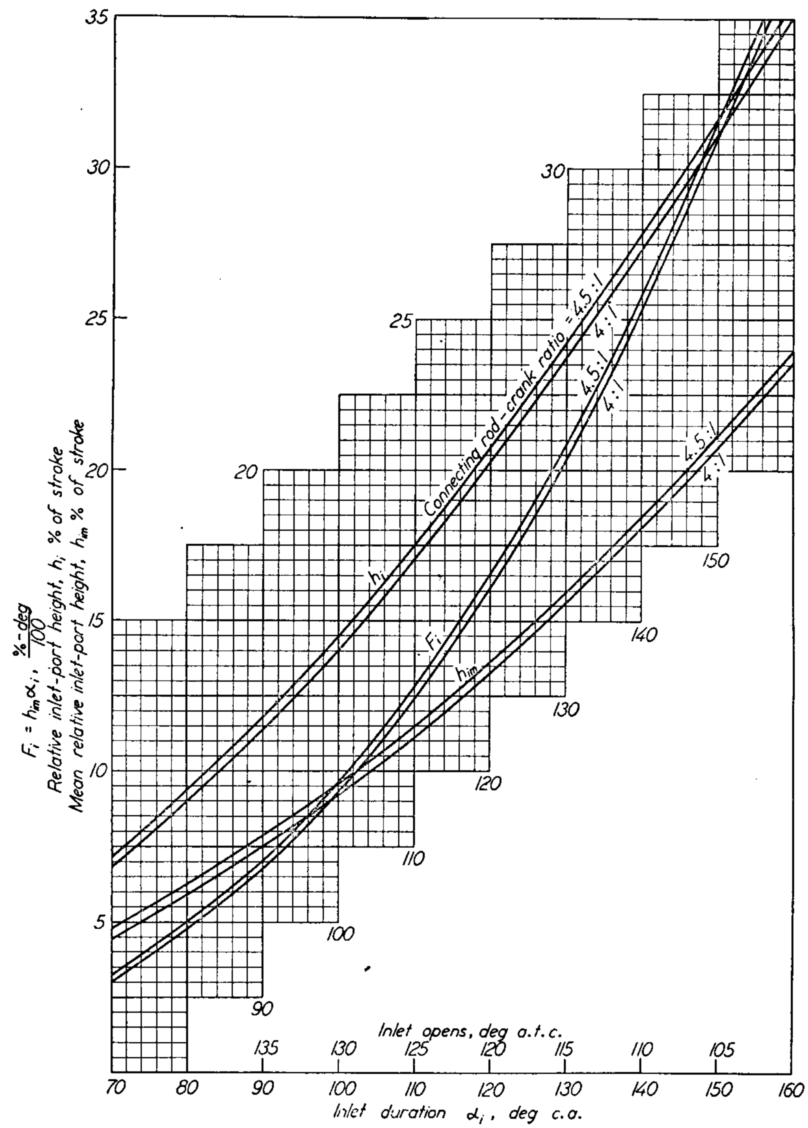

Fig. 7-5. Port Height and Per Cent-Degrees vs Inlet Duration for 4.5:1 and 4:1 Connecting Rod-Crank Ratios. As h_i and h_{im} are expressed in % stroke, the chart readings must be divided by 100 before they are used. F_i readings are 1/100 of the %-deg. values, therefore they are to be used unchanged. By substituting α_e for α_i, h_e for h_i and F_e for F_i, the curves are applicable to exhaust ports as well.

power to drive the blower to produce a high scavenge pressure. Beyond a certain point the power expended for driving the blower exceeds the gain in indicated horsepower due to the longer effective stroke. Naturally a certain scavenge pressure gives the maximum net power output.

The scavenge pressure that gives maximum net power can be found by a rather simple method shown in Fig. 7-6. The method essentially consists of determining the *net bmep* as a function of the variable scavenge pressure. The net bmep is the difference between the *gross bmep* and the *blower mep*, the latter being defined as that portion of the engine mep which is absorbed in driving the blower.

The first step is to plot, with scavenge pressure p_{sc} as abscissa, the inlet time-area $A_{im}\alpha_i$ from equation (7-1) with the aid of Fig. 7-2. Knowing the inlet time-area, the inlet duration and inlet-port heights can be determined by Fig. 7-5 and also plotted against p_{sc}.

The next step is to estimate the gross bmep of the engine for any one scavenge pressure. This may be done with the help of Fig. 5-8, 5-9 and 5-10. Even a substantial error in the estimate of the bmep is of small consequence in the final result, and therefore we need not be concerned over the accuracy of the basic bmep. It is important to know, however, how the bmep varies with the scavenge pressure. Since the gross bmep can be assumed to vary linearly with the effective compression stroke $s(1 - h_i)$, and h_i is already plotted against p_{sc}, the gross bmep can also be plotted against p_{sc}.

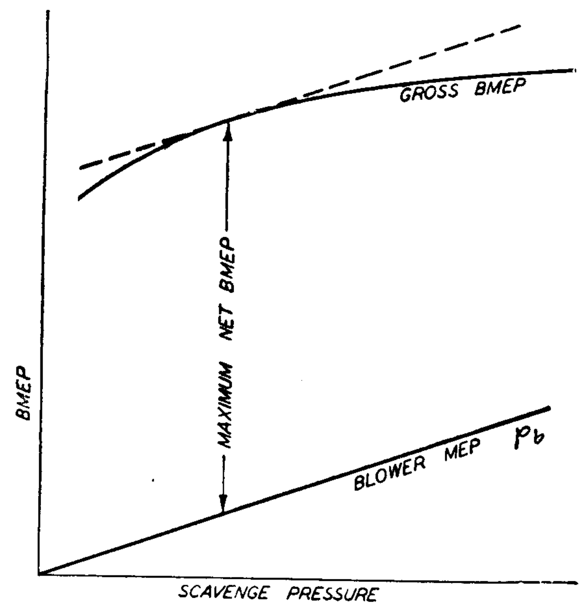

Fig. 7-6. Showing the Principle of Determining the Scavenge Pressure that Gives Maximum Net Bmep.

The third step is the plotting of the blower mep, p_b, against p_{sc}. If experimental data are not available it can be calculated by the following approximate formula:

$$p_b = p_{sc} \frac{L}{\eta_b}. \tag{7-5}$$

Since delivery ratio L is around 1.4 and the blower efficiency η_b around 0.7, equation (7-5) can be further simplified to

$$p_b = 2p_{sc} \tag{7-6}$$

which means that the blower mep is roughly double the scavenge pressure.

The ordinate between the gross bmep and the blower mep is the net bmep. That is a maximum where a tangent parallel to the p_b line touches the gross bmep line (see Fig. 7-6). The scavenge pressure corresponding to that point is the optimum scavenge pressure from the standpoint of net power output.

7.11 Example 1. Simple Piston-Cottrolled Engine.

The engine shown in Fig. 7-7 is an air-injection engine with 14.6-inch bore and 20.1-inch stroke, and is expected to deliver 100 bhp per cylinder at its normal speed of 200 rpm which corresponds to 58.5 psi bmep. What should be the timing and dimensions of the inlet ports?

The permissible width of the inlet ports is 30 per cent of the periphery, that is 13.8 inches. Selecting the scavenge pressure from Fig. 7-1 as 2.2 psig, to be produced by a reciprocating compressor, the pressure ratio figures out to be $(14.5 + 2.2)/14.5 = 1.15$. In view of the cast ports the scavenge factor S is estimated from Table 7–I as 0.7 and the scavenge-air temperature as 150 F. With these values, Fig. 7–2 gives a mean inlet velocity of $w_i = 0.7 \times 470 = 329$ feet per second.

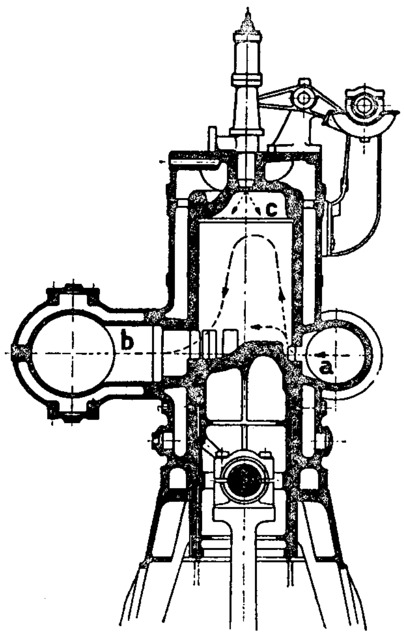

Fig. 7-7. Engine with Simple Port Scavenge. 14.6 in. bore, 20.1 in. stroke, 200 rpm. (Löffler-Riedler, *Oelmaschinen*. Copyright by Julius Springer, 1916.)

In view of the slow speed, the delivery ratio L may be made 1.5, in which case according to equation (7–1) the required inlet time-area is

$$A_{im}\alpha_i = \frac{1.5}{2 \times 329}\frac{14.6^2\pi}{4}20.1 \times 200 = 1540$$

square inch-degrees.

To get per cent-degrees, equation (7–3) gives

$$F_i = \frac{1540}{13.8 \times 20.1} = 5.55.$$

From Fig. 7–5 with a connecting rod-crank ratio of 4.5, $\alpha_i = 83$ degrees, and $h_i = 10.1$ per cent stroke, which corresponds to 2.03 inches inlet-port height.

7.12 Example 2. Opposed-Piston Engine.

An 8 by 10 + 10-inch opposed-piston engine, pictured in Fig. 7–8, designed to develop 150 bhp per cylinder at the rated speed of 720 rpm, is to have 16 machined rectangular inlet ports, each $\frac{7}{8}$ inch wide, at right angles to the flow. The air is supplied by a Roots blower. What must be the timing and height of the inlet ports?

With $\beta = (16 \times \frac{7}{8})/8\pi = 0.557$, Fig. 7–1 indicates a preferred scavenge pressure of 2.4 psig, corresponding to a pressure ratio of 1.165. The temperature of the scavenge air is taken as 150 F.

The scavenge factor S must be estimated low because of the intended supercharge, but not very low because the supercharge necessarily must be of short duration in an opposed-piston engine. On the basis of Table 7–I, $S = 0.64$ is selected.

With these values, Fig. 7–2 gives a mean inlet velocity of $0.64 \times 490 = 314$ feet per second. With an assumed delivery ratio $L = 1.4$, formula (7–1) gives for the required inlet time-area

$$A_{im}\alpha_i = \frac{1.4}{2 \times 314}\frac{8^2\pi}{4} 20 \times 720 = 1618 \text{ sq in.-deg.}$$

To get per cent-degrees, equation (7–3) gives

$$F_i = \frac{1618}{16 \times 7/8 \times 10} = 11.55.$$

With F_i being known, the following are read from Chart 7–5, for 4.5 connecting rod-crank ratio: $\alpha_i = 106.5$ degrees, and $h_i = 16.4$ per cent stroke, which corresponds to 1.64 inches.

In aiming at maximum output, the possibility of obtaining more power by the use of higher scavenge pressure must be explored. The higher the scavenge pressure the lower the inlet-port heights need be, which gives a longer effective stroke and greater bmep. On the other hand, the blower mep increases with increasing scavenge pressure. To determine the scavenge pressure which gives the maximum net bmep we apply the graphical calculation described in 7–10.

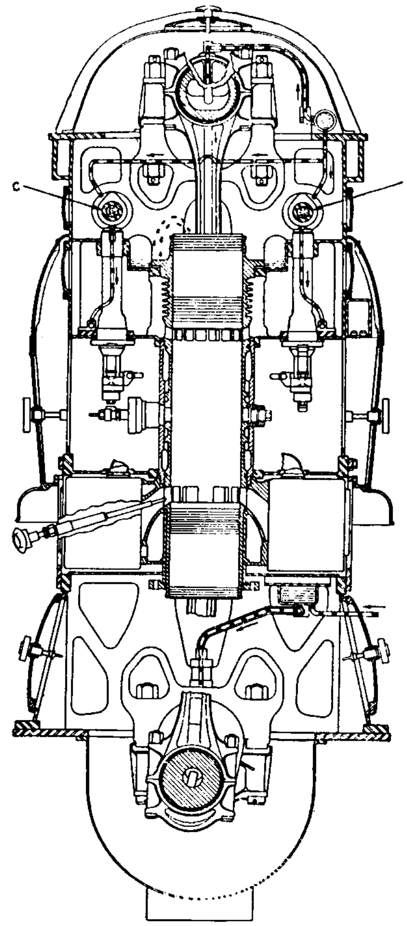

Fig. 7-8. Fairbanks-Morse Opposed Piston Engine. 8 in. bore, 10 + 10 in. stroke, 720 rpm. (By permission of Fairbanks-Morse & Co.)

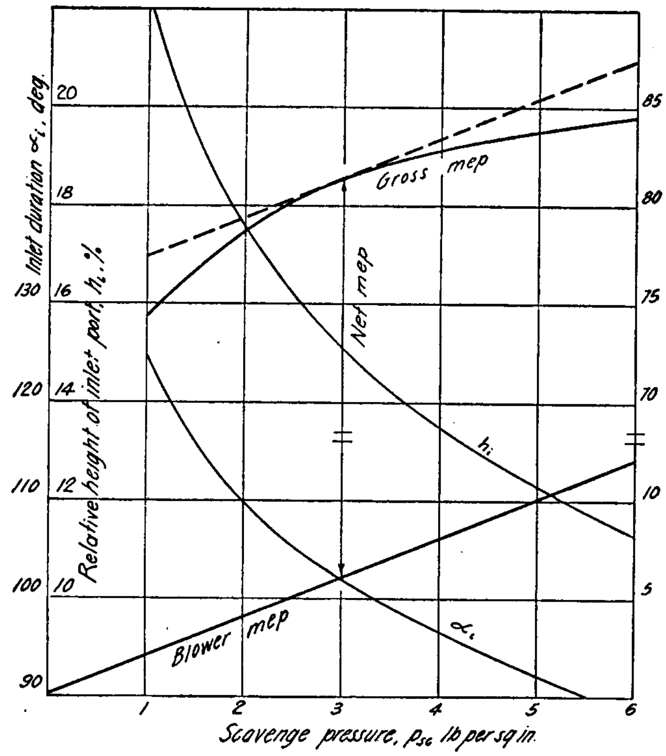

Fig. 7-9. Determining the Optimum Scavenge Pressure for Engine Shown in Fig. 7-8.

In Fig. 7–9, h_i is first plotted against the scavenge pressure by using the calculation shown in Table 7–II. Next the gross mep is assumed to be 80 psi with the scavenge pressure of 2.4 psig; 10 to 15 per cent error in this estimate hardly affects the final result. The gross mep for another scavenge pressure is then

$$80 \frac{100 - h_i}{100 - 16.4}.$$

To obtain the net mep the blower mep must be deducted. With $L = 1.4$ and $\eta_b = 0.7$, from equation (7–5)

$$p_b = 2p_{sc}.$$

If the efficiency of the blower is other than 70 per cent the volume of p_b is different, but this is sufficiently accurate for the purpose.

The blower mep can, therefore, be represented by a straight line, and the point of tangency on Fig. 7–9 represents the maximum output. In the present case this corresponds to 3 psig scavenge pressure. Correspondingly, the inlet duration is 102 degrees and the relative height of the inlet ports $h_i = 15.2$ per cent which corresponds to 1.52 inches.

Table 7-II. Calculation of Optimum Inlet Porting, for 8 by 10 + 10-inch Opposed-Piston Engine Shown in Fig. 7–8.

Scavenge pressure p_{sc} (psig)	2	3	4	5	6
Pressure ratio $(14.5 + p_{sc})/14.5$	1.137	1.206	1.276	1.344	1.414
Scavenge factor S (estimated)	0.630	0.65	0.67	0.69	0.71
Scavenge air temperature t_{sc} (F, estimated)	125	150	150	175	175
Mean inlet velocity[1] w_i (ft per sec)	288	358	429	484	545
Required mean inlet time-area[2] $A_{im}\alpha_i$ (sq in.-deg)	1760	1417	1182	1048	931
$F_i = \dfrac{A_{im}\alpha_i}{bs_i} = \dfrac{A_{im}\alpha_i}{14 \times 10}$ (per cent deg)/100	12.56	10.12	8.45	7.49	6.65
Inlet duration[3] α_i (deg)	$109\frac{1}{2}$	102	96	92	88
Relative height of inlet ports[3] h_i (per cent stroke)	17.5	15.2	13.50	12.25	11.25
Gross mep $= 80 \dfrac{100 - h_i}{100 - 16.4}$ (psi)	78.9	81.0	82.7	83.8	84.8
Blower mep $= Lp_{sc}/\eta_b = 2p_{sc}$ (psi)	4	6	8	10	12
Net mep (psi)	74.9	75.0	74.7	73.8	72.8

[1] From Fig. 7–2. [2] From Formula (7–1). [3] From Fig. 7–5.

It will be noted from Table 7–II that the net bmep is not sensitive to the scavenge pressure. One psi difference in scavenge pressure reduces the power by only 0.7 per cent.

7.13 Example 3. Poppet Exhaust-Valve Engine.

The engine shown in Fig. 7–10 is a popular $4\frac{1}{4}$ by 5-inch two-stroke cycle engine with inlet ports at the lower end of the cylinder and two exhaust valves in the head. The engine is rated 20.5 bhp at 2000 rpm. It is supplied with air by a Roots blower through two rows of circular inlet ports $\frac{5}{16}$ inch diameter, with 32 ports in each row. The ports are drilled 16 degrees from the radial in the transverse plane. The inlet ports open at 132 degrees and close at 228 degrees after top center. A delivery ratio of 1.3 or more is desired at a scavenge pressure of 6.5 psi or less at 2000 rpm.

In view of the considerable supercharge to be used a scavenge factor of 0.4 and a scavenge air temperature of 200 F are assumed. From Fig. 7–2 it can be seen that these values with a pressure ratio of $(14.5 + 6.5)/14.5 = 1.45$ give a mean inlet velocity of

$$w_i = 0.4 \times 785 = 314 \text{ ft per sec.}$$

From equation (7–1)

$$A_{im}\alpha_i = \frac{1.3}{2 \times 314} \frac{(4\frac{1}{4})^2\pi}{4} 5 \times 2000 = 294 \text{ sq in.-deg.}$$

That much inlet time-area is required. How much is available? Figure 7–11 shows the porting arrangement to scale. The lower line represents the piston travel during the scavenging period and shows the uncovered part of the ports at any crank position from 132 degrees after top center to bottom center. The square inches uncovered at each crank position are plotted above the axis and the inlet time-area is obtained by planimetration. At the end of the inlet period this amounts to 317.4 square inch-degrees, which is 8 per cent more than is required. The inlet porting, therefore, is satisfactory.

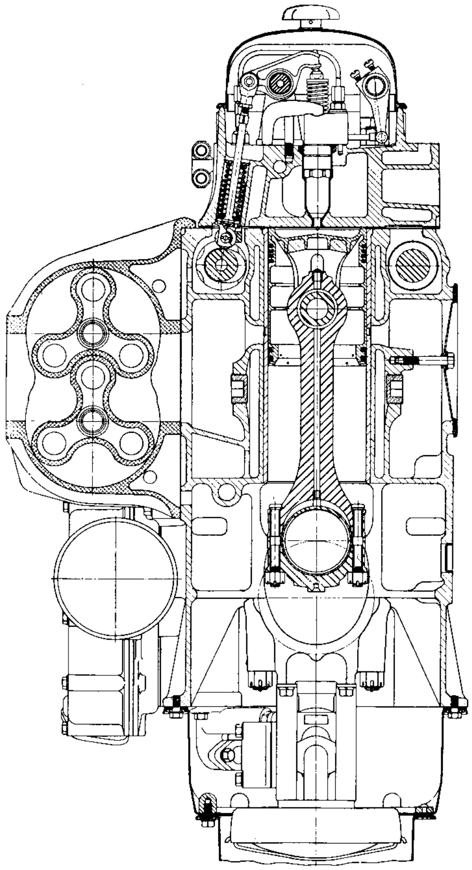

Fig. 7-10. General Motors Model 71 Engine. $4\frac{1}{4}$ in. bore, 5 in. stroke, 2000 rpm. (By permission of General Motors Corp.)

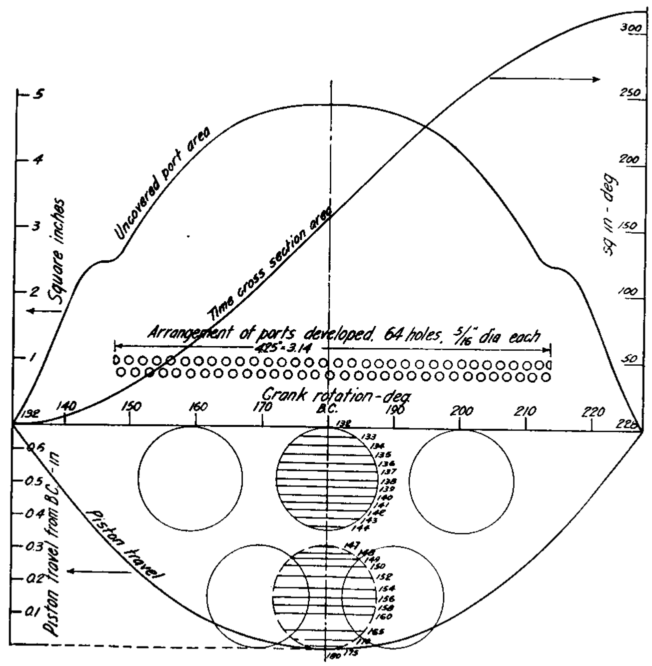

Fig. 7-11. Inlet Porting of Poppet Exhaust Valve Engine in Fig. 7-10.

INLET PORTS OF CRANKCASE-SCAVENGED ENGINES

7.14 The procedure in determining the dimensions of the inlet ports is the same in the case of crankcase-scavenged engines, except that the air delivery cannot be chosen freely because it is determined by the system. The air delivery depends on the clearance volume, air valves, and flow resistance. In typical crankcase-scavenged engines it varies from 0.5 to 0.8 V_{disp}.

The scavenge pressure varies during the scavenging process as the crankcase is emptied of its content. In well-designed engines the scavenge pressure is 4 to 5 psig when the inlet opens, dropping to almost atmospheric at inlet-port closing. The cylinder pressure during the same time drops from a couple of pounds to almost atmospheric and sometimes a little below. The effective pressure head is approximately 2.5 psi and the pressure ratio 1.175 in a good engine with *tuned* exhaust.

The scavenge factor may safely be taken as 0.5, assuming cast ports with fairly rough edges. On this basis, with 150 F scavenge-air temperature, the mean inlet velocity from Fig. 7–2 is 0.5×505, or in a round number 250 feet per second. This figure may be used for all types and sizes unless experimental data indicate otherwise. In case of loop scavenging it should be 20 to 30 per cent lower. Exact information is lacking.

The rest of the procedure is the same as with separate scavenging. The same formula

$$A_{im}\alpha_i = \frac{L}{2w_i} V_{disp} n$$

is applied, which gives the mean inlet time-area, from which the inlet duration in degrees of crank angle and the corresponding port height can be determined on the basis of Fig. 7–5.

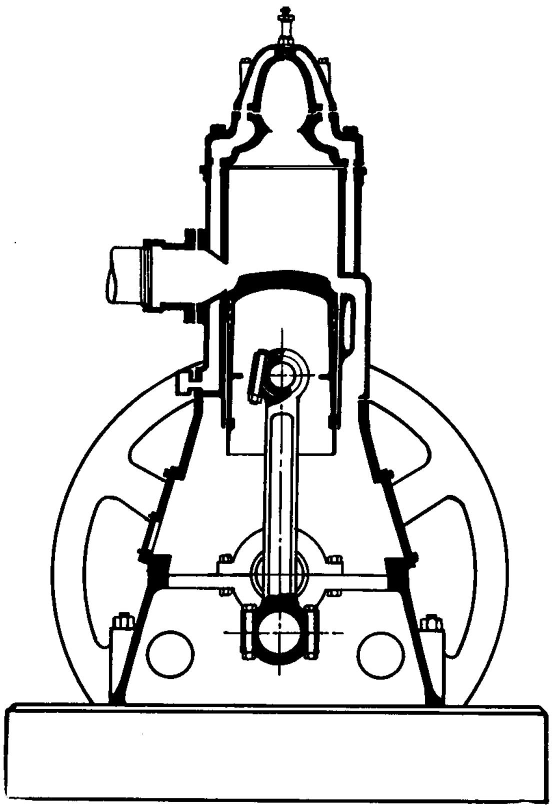

Fig. 7-12. Venn-Severin Crankcase-Scavenged Engine. $10\frac{1}{2}$ in. bore, 12 in. stroke, 425 rpm. (By permission of Venn-Severin Machine Co.)

7.15 Example. Crankcase-Scavenged Engine.

A $10\frac{1}{2}$ by 12 inch, 425 rpm crankcase-scavenged engine with cross scavenging shown in Fig. 7–12 has four upwardly inclined inlet ports that extend 21.9 per cent of the cylinder circumference. Since the ports are cast, it was thought inadvisable to go higher lest the bridges between the ports become too weak. The engine having a snug-fitting crankcase and feather-type inlet valves, the air delivery is assumed to be 0.7 V_{disp}, and the required inlet time-area becomes

$$A_{im}\alpha_i = \frac{0.7}{2 \times 250} \frac{(10.5)^2\pi}{4} 12 \times 425 = 620 \text{ sq in.-deg.}$$

With rectangular ports extending to the bottom of the piston stroke, from equation (7–3)

$$F_i = \frac{620}{0.219 \times 10.5 \times \pi \times 12} = 7.15.$$

From Fig. 7–5 the corresponding inlet duration and port height are $\alpha_i = 91$ degrees and $h_i = 12$ per cent which corresponds to 1.44 inches. The production engine is made with $1\frac{5}{8}$-inch high inlet ports, which increases the inlet period to 96 degrees.

APPENDIX TO CHAPTER 7

7.16 Calculation of Inlet Ports.

In the calculation of inlet ports it is convenient to use the concepts inlet time-area, mean inlet area, and mean inlet velocity. The first two are geometric concepts. If the uncovered inlet port area is plotted against the crank angle (see Fig. 7–3) the area under the curve is

$$\int_{\alpha_1}^{\alpha_2} A_i d\alpha$$

the integral of port area over the opening period, and is to be termed for brevity *inlet time-area.* It can be represented as

$$\int_{\alpha_1}^{\alpha_2} A_i d\alpha = A_{im}(\alpha_2 - \alpha_1) = A_{im}\alpha_i$$

where A_{im}, the mean height of the curve, is termed *mean inlet area*, and α_i the *inlet duration.*

Since all of the air V_{del} that is delivered to the engine per cycle passes the inlet ports, it is convenient to consider V_{del} as a product:

$$V_{del} = A_{im}\tau_i w_i$$

where A_{im} is the mean inlet area as defined above, τ_i is the duration of the inlet port opening during the cycle in seconds, and w_i is the *mean inlet velocity.* Since V_{del} is usually expressed in cubic inches, A_{im} in square inches, while w_i is in feet per second,

$$V_{del} = LV_{disp} = A_{im}\tau_i 12 w_i \tag{7–7}$$

which by expressing the inlet duration in crank degrees

$$\tau_i = \frac{60\alpha_i}{360n} = \frac{\alpha_i}{6n}$$

can also be written as

$$LV_{disp} = \frac{A_{im}\alpha_i 12 w_i}{6n},$$

from which is obtained the basic formula

$$A_{im}\alpha_i = \frac{L}{2w_i} V_{disp} n \tag{7–1}$$

used for inlet-port calculations.

It is, however, well to keep in mind that this equation expresses nothing but the definition of w_i. Nevertheless as soon as w_i is known, the inlet time-area $A_{im}\alpha_i$ can be calculated. To estimate the mean inlet velocity w_i, for the sake of porting calculations, the empirical method may be used. In an engine similar to the one to be calculated the delivery ratio $L = V_{del}/V_{disp}$ is determined by air consumption measurements, $A_{im}\alpha_i$ by planimetration of the port opening diagram, and equation (7–1) gives w_i.

Another way to estimate w_i is by thermodynamic calculations.

If air flows from a container of absolute pressure p_1 into a container of absolute pressure p_2 through a fixed opening, and on its way neither receives nor imparts energy from or to the surrounding bodies, the difference on the total energy at any two sections is equal to the net work performed:

$$\left(\frac{w_1^2}{2g} + E_1\right) - \left(\frac{w_2^2}{2g} + E_2\right) = p_2v_2 - p_1v_1.$$

Expressing the internal energy of the air at moderate temperature the specific heat may be assumed constant, and

$$E_1 = 778c_vT_1; \quad E_2 = 778c_vT_2.$$

Since

$$pv = RT, \ \frac{c_p}{c_v} = k \text{ and } (c_p - c_v) = \frac{R}{778}$$

$$E_1 = \frac{p_1v_1}{k-1} \text{ and } E_2 = \frac{p_2v_2}{k-1}$$

which makes the energy equation

$$\left(\frac{w_2^2}{2g} + \frac{p_2v_2}{k-1}\right) - \left(\frac{w_1^2}{2g} + \frac{p_1v_1}{k-1}\right) = p_1v_1 - p_2v_2.$$

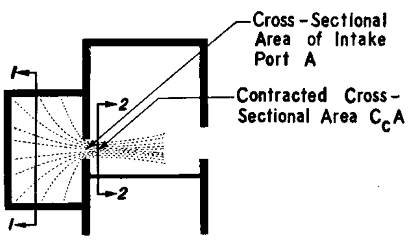

Fig. 7-1. Schematic Representation of Flow of Intake Air.

For section 1 a section through the air box as shown on Fig. 7-13 is selected, and for section 2 is selected that section of the air stream where its area is minimum (vena contracta) which may be in or slightly beyond the inlet port. This minimum cross section is equal C_cA, where C_c is the coefficient of contraction which varies from approximately 0.7 in case of a thin plate orifice to approximately 1.00 in case of a well-rounded orifice. The pressure, velocity, and specific volume in the contracted cross section are designated by p_m, w_m, and v_m, respectively; therefore the energy equation is written as

$$\left(\frac{w_m^2}{2g} + \frac{p_mv_m}{k-1}\right) - \left(\frac{w_1^2}{2g} + \frac{p_1v_1}{k-1}\right) = p_1v_1 - p_mv_m. \tag{7-8}$$

In view of the fact that the opening A is relatively small compared to the cross section of the air box, w_1^2 may be neglected relative to w_m^2 and solving equation (7–8) for w_m

$$w_m = \left[2g\frac{k}{k-1}(p_1v_1 - p_mv_m)\right]^{\frac{1}{2}}. \tag{7-9}$$

The actual flow velocity w_{ma} is always somewhat smaller than this theoretical value, therefore a factor C_v, the coefficient of velocity, is added.

$$w_{ma} = C_v\left[2g\frac{k}{k-1}(p_1v_1 - p_mv_m)\right]^{\frac{1}{2}}.$$

The rate of flow (lb per sec) is a product of velocity $\times$ cross section $\times$ density

$$G = w_{ma}C_cA\rho_m = \frac{C_cA}{v_m}C_v\left[2g\frac{k}{k-1}(p_1v_1 - p_mv_m)\right]^{\frac{1}{2}}.$$

By combining C_c and C_v into a discharge coefficient C

$$G = \frac{CA}{v_m}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}(p_1v_1)^{\frac{1}{2}}\left(1 - \frac{p_mv_m}{p_1v_1}\right)^{\frac{1}{2}}. \tag{7-10}$$

Dealing with ideal gas and isentropic flow

$$p_1v_1{}^k = p_mv_m{}^k$$

and equation (7–10) changes to

$$G = \frac{CA}{v_m}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}(p_1v_1)^{\frac{1}{2}}\left[1 - \left(\frac{p_m}{p_1}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}.$$

Under the existing conditions there is no appreciable difference between the pressure in the contracted contraction and the average pressure in container 2,

$$p_2v_m{}^k = p_1v_1{}^k$$

which makes the rate of flow equation

$$G = \frac{CA}{v_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{k}}(RT_1)^{\frac{1}{2}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}. \tag{7-11}$$

Before applying this equation to the scavenging problem it is well to realize that the scavenging process differs from a steady flow through a constant opening between two constant pressure containers. In the engine the port opens and closes intermittently, and while it is open the uncovered area varies. This involves repeated acceleration and deceleration of an air column, although this is not of considerable magnitude [Waldron, 1941]. Both the scavenge pressure and the cylinder pressure fluctuate during the process. The cylinder ordinarily is open during most of the scavenging process through the exhaust port, but this communication is frequently restricted and becomes completely closed during the supercharging period. Exhaust gas column vibrations and their resonance or lack of resonance with the engine speed cause the cylinder pressure to fluctuate between wide limits during the charging process.

In spite of these marked differences it is proposed to treat the two-stroke engine as a fixed orifice with regard to the scavenging air flow. Of course, all air delivered to the engine goes through the inlet ports, through the exhaust ports, and through the exhaust pipe to the atmosphere. The system beginning at the scavenging air receiver and ending at the exit of the exhaust pipe is considered as a single unit, and an attempt is made to replace it with a single equivalent orifice. The equivalent orifice with air of scavenging pressure and temperature on one side and ambient pressure on the other side must transmit the same amount of air per minute as the running engine. By knowing the size and coefficient of flow of that orifice, the air consumption of the engine can be calculated by equation (7–11), substituting $(p_{sc} + p_0)/p_0$ for p_1/p_2, where p_{sc} is the gage pressure of the inlet air.

As to the size of the equivalent orifice, the first idea that suggests itself is to set it equal to the mean inlet area. Such procedure would be fully justified if the air flow would always be controlled by the inlet ports. This, however, is not always the case. In a normal two-stroke engine the cylinder pressures fluctuate during the scavenging period between a couple of pounds above atmospheric to possibly a fraction of a pound below atmospheric. The positive pressure is caused by: (1) the pressure drop in the cylinder (this is negligible), (2) the pressure drop across the exhaust ports, (3) the pressure drop through the exhaust ducts, (4) the pressure drop through the exhaust pipe, (5) the positive pressure waves generated by the gas column oscillations in the exhaust pipes. The negative pressure is caused by: (1) the negative pressure waves generated by the gas column oscillations, (2) the momentum of the mass of gas discharged at a high velocity during the blowdown period.

For these reasons the air flow through the engine cannot be considered as being controlled solely by scavenge pressure and the inlet ports. Nevertheless under certain conditions this is permissible. If it is assumed that (1) the engine has no exhaust pipe but that the exhaust ports discharge directly to the atmosphere, (2) the exhaust-port openings are always considerably larger than the inlet-port openings during the charging period, then the pressure in the cylinder does not differ appreciably from atmospheric while the inlet ports remain open. A simple piston-scavenged engine (symmetrical scavenge) with separate scavenging, with a large receiver next to the inlet ports, and a large exhaust pot next to the exhaust ports comes very close to the above assumption.

In other cases, where an appreciable pressure exists in the cylinder during part or all of the charging period, like a supercharged engine or an engine exhausting into a long exhaust pipe, etc., the preceding assumptions are not valid and the actual pressure ratio is different from $(p_{sc} + p_0)/p_0$. Nevertheless it is possible to apply the simple equation (7–11) to these cases too by the simple expedient of introducing a scavenge factor S to take care of the uncertainty of the pressure ratio and other incalculable factors.

Proceeding on the basis of an equivalent orifice, set in equation (7–11) for A the mean inlet area

$$A_{im} = \frac{\int_{\alpha_1}^{\alpha_2} A_i d\alpha}{\alpha_2 - \alpha_1},$$

for p_1 the absolute scavenge pressure $(p_{sc} + p_0)$, for p_2 the normal atmospheric pressure p_0, for T_1 the temperature of the scavenge air T_{sc}, for discharge coefficient C the scavenge factor S,

$$\text{(7–12)} \qquad G = SA_{im}\frac{1}{v_{sc}}\left(\frac{p_0}{p_{sc} + p_0}\right)^{\frac{1}{k}}(RT_{sc})^{\frac{1}{2}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left[1 - \left(\frac{p_0}{p_{sc} + p_0}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}.$$

On the other hand the volumetric amount of **free air delivered in one cycle** is

$$V_{del} = G\tau_i v_0$$

where v_0 is the specific volume of the air under NTP conditions.

Introducing now the *mean inlet velocity* defined as

$$w_i = V_{del}/A_{im}\tau_i$$

and combining it with (7–12),

$$w_i = \frac{G\tau_i v_0}{\tau_i A_{im}} = S\frac{v_0}{v_{sc}}\left(\frac{p_0}{p_{sc} + p_0}\right)^{\frac{1}{k}}(RT_{sc})^{\frac{1}{2}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left[1 - \left(\frac{p_0}{p_{sc} + p_0}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}$$

$$\text{(7–13)} \qquad w_i = S\frac{v_0 p_0}{\sqrt{RT_{sc}}}\sqrt{2g\frac{k}{k-1}\left(\frac{p_{sc} + p_0}{p_0}\right)^{\frac{k-1}{k}}}\sqrt{1 - \left(\frac{p_0}{p_{sc} + p_0}\right)^{\frac{k-1}{k}}}.$$

From this equation w_i/S can conveniently be plotted against the pressure ratio $(p_0 + p_{sc})/p_0$ for any scavenging temperature. A chart representing equation (7–13) is presented in Fig. 7–2 with $p_0 = 14.5$ psi. Knowing the scavenge factor, the chart of Fig. 7–2 gives the mean inlet velocity.

For scavenge pressures in the range of 2 to 8 psi gage and atmospheric exhaust, equation (7–13) may be replaced with the simpler approximate formula

$$\text{(7–14)} \qquad \frac{w_i}{S} = 7900\sqrt{\frac{p_{sc}}{T_{sc}}}$$

which gives identical results with equation (7–13) or the chart of Fig. 7–2 within 1 per cent error.

No great error can result from inaccurately estimating the inlet-air temperature either. The range of its variations is probably from 100 to 200 F, and by assuming it to be 150 F the maximum error in w_i is not more than 4 per cent. However, the possible range of S is wide even in the case of simple port-scavenged engines, and still wider for other types of engines. An error of x per cent in estimating S causes an error of the same per cent in the calculated mean inlet velocity. Therefore, the selection of the scavenge factor S requires most careful attention.

It is to be kept in mind that S is not strictly a constant. Only if equation (7–13) had included all the factors that effect S, and all of them correctly, could S be expected to be a constant. Only experiments can decide whether the equation is serviceable and if so how the coefficient S is to be selected.

CHAPTER 8

EXHAUST PORTS

8.1 The exhaust period consists of two parts. The first is the blowdown period which lasts from the opening of the exhaust to the opening of the inlet ports. The second is the scavenging period which lasts from the opening of the inlet ports to the closing of the exhaust ports. Of the two the first is of paramount importance.

EXHAUST LEAD FOR NONSYMMETRICAL ENGINES

8.2 Blowdown Period.

The objective is to provide during the blowdown period sufficient time-area to permit the cylinder content to expand to approximately scavenge pressure, lest exhaust gases be blown into the inlet ports. The most important part of the exhaust-port layout consists, therefore, of determination of the exhaust lead, that is, of the angular difference between inlet opening and exhaust opening. What counts, of course, is not the exhaust lead itself but the uncovered exhaust-port area during the exhaust lead. In fact, the layout of the exhaust ports is frequently done with the single point in view of providing sufficient time-area during the blowdown period. If that is sufficient then the rest of the exhaust-port areas are as a rule more than sufficient. This is particularly true in symmetrically scavenged engines. But the provision of sufficient exhaust lead is always one of the hardest requirements to fill. Even elaborate formulas cannot be depended on to give optimum values. It is preferable therefore to start out with simple formulas, to check and improve the values as the layout proceeds, and to put the finishing improvements on when the engine is tested on the stand.

In the layout, the first concern of the designer should be to open the largest possible exhaust-port areas as rapidly as possible and to promote high exhaust velocities by unobstructed exhaust ducts and proper exhaust piping. This not only permits a quick drop of the cylinder pressure but produces a certain amount of scavenging by the inertia of the exhaust-gas column.

8.3 Approximate Equation.

In calculating the exhaust lead, for first approximation use the equation

$$A_m\alpha = 0.00033\,V_{disp}n \tag{8-1}$$

where $A_m = h_m s\, b_e$ is the average uncovered exhaust-port area during the exhaust blowdown period α in square inch-degrees (see Fig. 7–4), V_{disp} the displacement volume in cubic inches, and n the rpm.

This equation, the derivation of which is given in the appendix at the end of this chapter, gives the blowdown time-area that is necessary to make the cylinder pressure drop close to the scavenge pressure. The equation is strictly valid only under the following conditions: (1) The cylinder volume at the time the exhaust opens is 0.8 displacement volume; (2) The discharge coefficient of the exhaust ports (valves) is 0.442; (3) The temperature of the cylinder content at the beginning of the exhaust is 1250 F; (4) The ratio of the cylinder pressure (at the beginning of the exhaust) to the scavenge pressure is 3:1; (5) The pressure in the exhaust duct remains atmospheric; and (6) The polytropic exponent of expansion is 1.3.

These secondary variables seldom vary enough to affect seriously the preliminary calculation. The primary variables determining the required blowdown time-area are V_{disp} and n.

A_m and α are interrelated by the kinematics of the port control. In case of piston-controlled exhaust ports, Fig. 7–4 illustrates the conditions. In order to facilitate computations, the chart of Fig. 8–1 has been constructed. This gives the exhaust lead in degrees crank angle if the inlet opening angle and the per cent-degree blowdown area $F_a = h_m\alpha = A_m\alpha/b_e s$ are given, provided the total width of the exhaust ports is constant during the blowdown period. The chart is applicable primarily to symmetrically scavenged engines but it holds also for nonsymmetrical engines if the exhaust ports

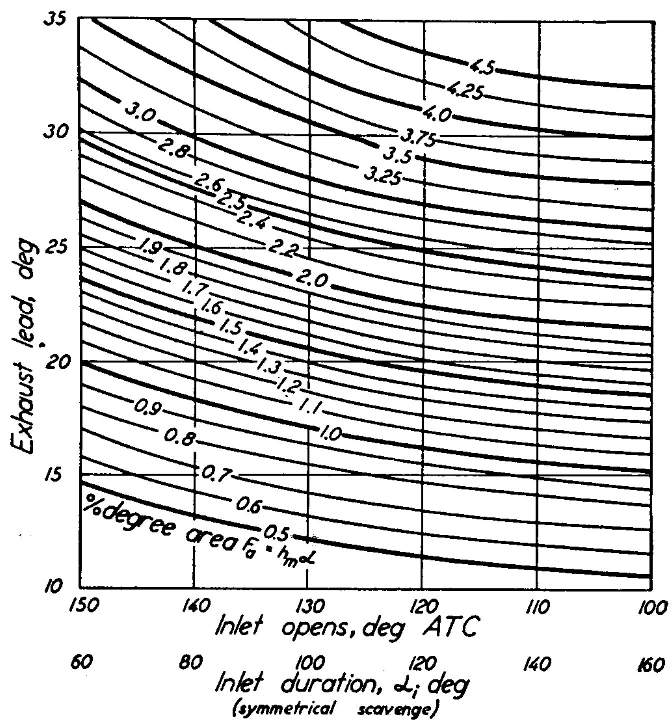

Fig. 8-1. Exhaust Lead with Piston Controlled Ports. Connecting rod-crank ratio 4.5:1. (The inlet duration scale is not valid for nonsymmetrical engines.)

are piston-controlled. However, the inlet duration then differs from the figures shown on the lower horizontal scale. If the exhaust ports are not simple piston-controlled, a graphical determination by the trial and error method is recommended.

The total width of the exhaust ports should be made as large as possible. In uniflow engines b_e is up to 70 per cent of the circumference (radial exhaust port). In cross-scavenged and loop-scavenged engines the figures are 26 to 33 and 20 to 33 per cent respectively. If poppet exhaust valves are used, a knowledge of valve dimensions and lift diagrams is required to calculate the time-area.

8.4 Precise Calculation.

After a preliminary figure for the exhaust lead has been obtained a more precise calculation should be made. For this purpose equation (8–2) can be used.

$$A_m\alpha = \frac{0.98}{\sqrt{RT_e}} V_e n y \tag{8–2}$$

where

$$y = \frac{1}{k-1}\left[\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{2k}} - 1\right].$$

The gas constant is $R = 53 \times 12 = 636$ (because everything is expressed in in.-lb-sec-F units). The temperature T_e may vary with the engine and the load, but its effect on the result of the calculation is not great. In the absence of experimental information the figure 1250 F = 1710 R may be used. V_e can be computed by the following formula.

$$V_e = V_{disp}\left(\text{Relative effective expansion stroke} + \frac{1}{\text{compression ratio} - 1}\right).$$

The value of y depends on the pressure ratio and the polytropic exponent. For the latter a value of 1.3 may be used. The pressure ratio depends on the expansion end pressure p_e and the blowdown pressure p_i which exists in the cylinder when the inlet port opens. The expansion end pressure is in linear relation with the mean indicated pressure as shown in Fig. 8–2 taken from List [List, 1937]. The blowdown pressure may be set equal to the scavenge pressure selected on the basis of Fig. 7–1, or on some other basis. For a given pressure ratio and exponent, y can be read from Fig. 8–3, and equation (8–2) can then be evaluated. The derivation of equation (8–2) is given in the appendix at the end of this chapter.

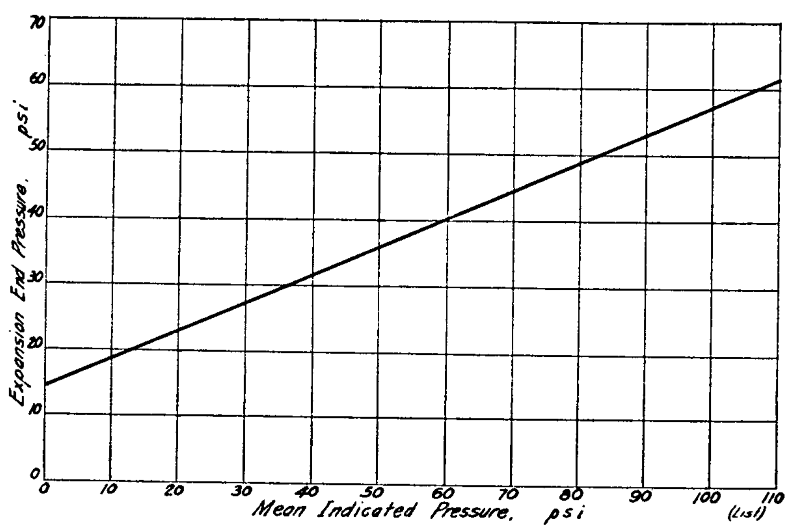

Fig. 8-2. Relation between Expansion End Pressure and Mean Indicated Pressure.

The result obtained from the exact equation (8–2) does not differ much from that of the simple equation (8–1) if the cylinder volume at the time of exhaust opening is approximately 0.8 displacement volume and the other variables have approximately the values mentioned. However, if V_e differs considerably from V_{disp}, the simple equation (8–1) can still be used by adding or deducting as many per cent to the blowdown time-area as V_e is greater or smaller than 0.8 V_{disp}.

While equation (8–2) represents the exhaust lead necessary to cause the cylinder contents to drop approximately to scavenge pressure, circumstances may force the designer to accept a lesser exhaust lead, as discussed in Chapter 10.

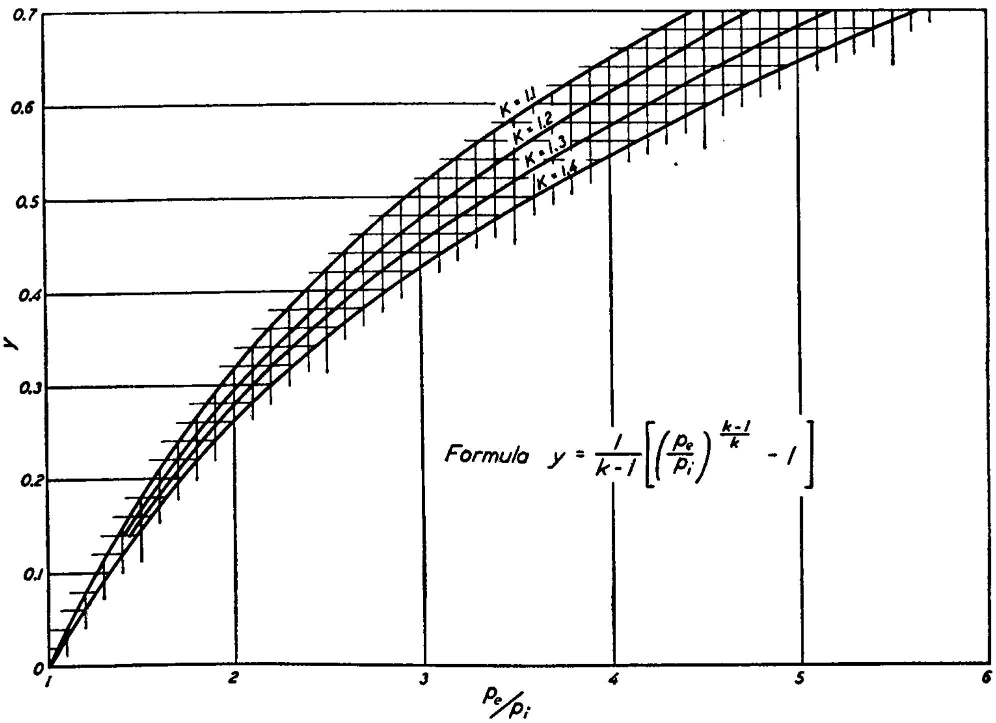

Fig. 8-3. Function y for Calculating Exhaust Lead.

8.5 Example 1. Opposed-Piston Engine.

The 8 by (10 + 10) inch, 720 rpm opposed-piston engine described in 7.12, was calculated for inlet porting that gave an inlet duration of $\alpha_i = 102$ degrees and inlet-port height of 1.52 inches. The layout (Fig. 8–4) permits exhaust ports with a total width of 12.7 inches (51 per cent of the circumference). What must be the exhaust lead?

In the preliminary calculation equation (8–1) is used:

$$A_m\alpha = 0.00033\ V_{disp}n$$
$$= 0.00033(8^2\pi/4) \times 20 \times 720$$
$$= 238 \text{ sq in.-deg.}$$

The per cent-degree blowdown area therefore is

$$F_a = \frac{238}{12.7 \times 10} = 1.87 \times 100 \text{ per cent-deg.}$$

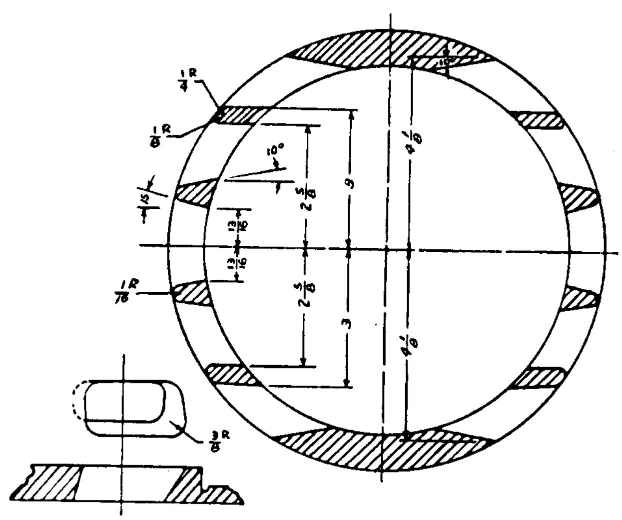

Fig. 8-4. Section through Exhaust Ports of Fairbanks-Morse Opposed-Piston Diesel Engine. (By permission of Fairbanks-Morse Company.)

With this and $\alpha_i = 102$ degrees, an exhaust lead of $\alpha = 23$ degrees is obtained from Fig. 8–1.

In the case of symmetrical scavenging this would give an exhaust duration of $102 + (2 \times 23) = 148$ degrees, and an exhaust closure at $180 + (148/2) = 254$ degrees after top center, which is very late.

The first remedy is to make the porting unsymmetrical by giving the exhaust piston a 12-degree crank-advance lead (see Fig. 8–5). Relative to the exhaust piston the inlet would open instead of $102/2 = 51$ degrees before bottom center, at $51 - 12 = 39$ degrees before bottom center or 141 degrees after top center. In order to get the required 238 square inch-degree blowdown time-area or 1.87 per cent-degree area, $\alpha = 24$ degrees is now read from Fig. 8–1.

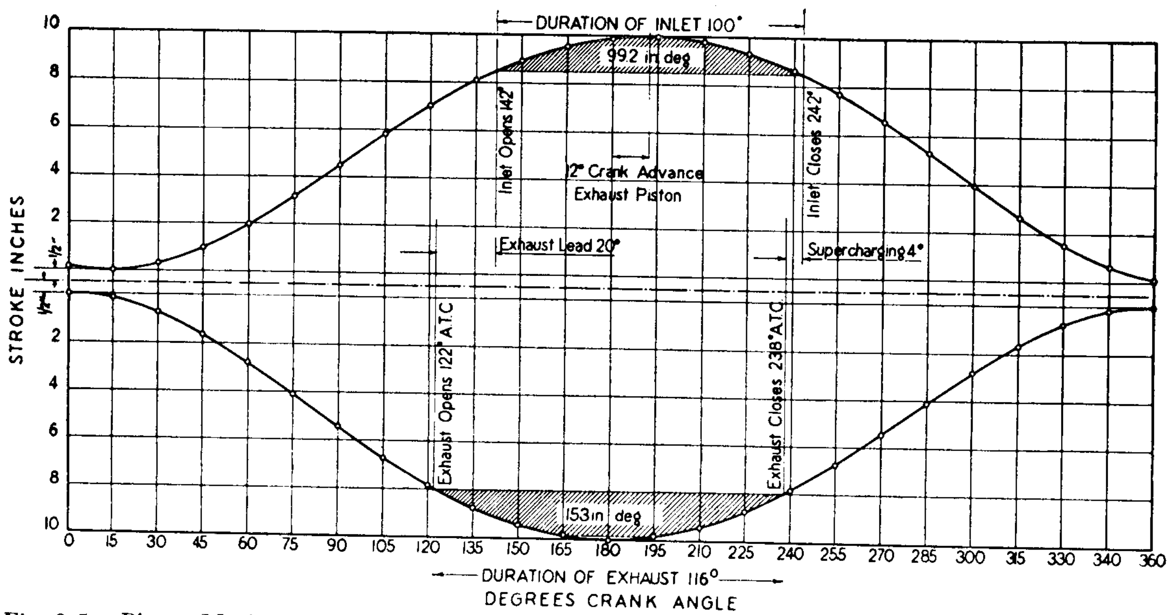

Fig. 8-5. Piston Motion of Fairbanks-Morse Opposed-Piston Engine. This represents the final porting adopted after compromise described in Section 10.10.

This was a preliminary calculation. Now the more precise equation (8–2) is applied, using the values of $R = 12 \times 53 = 636$, and $T_e = 1710$ Rankine.

Corresponding to an exhaust opening of $141 - 24 = 117$ degrees, the cylinder volume when scaled from Fig. 8–5 is found to be

$$V_e = 15.5 \times \frac{\pi(8)^2}{4} = 780 \text{ cu in.}$$

From Fig. 8–2 it is found that with an estimated mip of 83 psi, $p_e = 50$ psia.

On the basis of 3 psig scavenge pressure,

$$\frac{p_e}{p_i} = \frac{50}{14.5 + 3} = 2.85$$

and from Fig. 8–3 with $k = 1.3$, $y = 0.428$.
With these values

$$A_m\alpha = \frac{0.98}{\sqrt{636 \times 1710}} \times 780 \times 720 \times 0.428 = 225 \text{ sq in.-deg.}$$

The corresponding per cent-degree area figures out to be

$$F_a = \frac{225}{12.7 \times 10} = 1.77 \times 100 \text{ per cent-deg.}$$

From Fig. 8–1 with inlet opening at 141 degrees this gives a required exhaust lead of $23\frac{1}{2}$ degrees. Consequently, relative to the crank of the exhaust piston, the timing is as follows: inlet opens 141 degrees, inlet closes 243 degrees, exhaust opens $141 - 23\frac{1}{2} = 117\frac{1}{2}$ degrees, exhaust closes $(180 + 180) - 117\frac{1}{2} = 242\frac{1}{2}$ degrees after top center.

This gives a supercharging period of only $\frac{1}{2}$ degree. In order to get more it is necessary to cut down on the exhaust lead as is shown in Chapter 9.

8.6 Example 2. Poppet Exhaust-Valve Engine.

The $4\frac{1}{4}$ by 5 inch 2000 rpm engine shown on Fig. 7–10 and described in 7.13 has two exhaust valves in the head, of 1.56-inch effective diameter, 0.375-inch lift, and 45-degree seat angle. The exhaust valve, lifted by the cam shown on Fig. 8–6, begins to open 94.5 degrees after top center. Is the exhaust lead adequate?

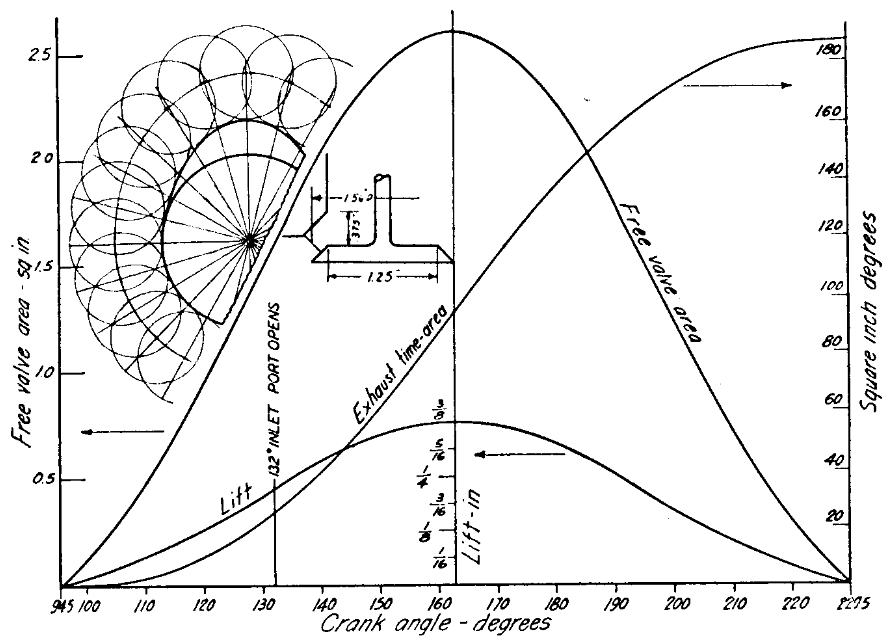

Fig. 8-6. Exhaust Valve Lift, Valve, and Time-Area in General Motors Model 71 Engine. Valve area computed square to the flow.

The required exhaust lead may be calculated by means of equation (8–2). The temperature T_e is not known but may be assumed to be 1250 F. R is $53 \times 12 = 636$.

An exhaust opening at 94.5 degrees after top center corresponds to an effective expansion stroke of 60 per cent. With this and an assumed compression ratio of 16 to 1,

$$V_e = \frac{(4.25)^2\pi}{4} \times 5\left(0.6 + \frac{1}{16 - 1}\right) = 47.3 \text{ cu in.}$$

Since the engine is rated 20.5 bhp at 2000 rpm and its mechanical efficiency is assumed to be 70 per cent, the mean indicated pressure is

$$\text{mip} = \frac{20.5 \times 33000}{71/12 \times 2000 \times 0.7} = 81.5 \text{ psi.}$$

The corresponding probable expansion end pressure, from Fig. 8–2, is $p_e = 48$ psi. The scavenge pressure developed by the Roots blower at 2000 rpm engine speed is 6.5 psi. Therefore, the pressure ratio is

$$\frac{p_e}{p_i} = \frac{48}{14.5 + 6.5} = 2.28.$$

From Fig. 8–3 with $k = 1.3$, $y = 0.326$, and therefore, from equation (8–2), a blowdown time-area

$$A_m\alpha = \frac{0.98}{\sqrt{636 \times 1710}} \times 47.3 \times 2000 \times 0.326 = 29.09 \text{ sq in.-deg}$$

is required; how much is available?

From the lift curve the flow area can be obtained and by integration the time-cross-sectional-area curve as shown in Fig. 8–6.

The opening angle of the inlet ports was given (see 7.13) as 132 degrees after top center. The exhaust time-area curve rises at that point to 26.2 square inch-degrees, which is about 10 per cent less than the value required. However, this pertains to the maximum speed of a variable speed engine. For speeds up to 1800 rpm the exhaust blowdown is adequate.

EXHAUST LEAD IN SYMMETRICALLY SCAVENGED ENGINES

8.7 Calculation of the exhaust lead in symmetrically scavenged engines, which include practically all of the crankcase-scavenged types, is in principle the same as that in nonsymmetrically scavenged engines. Equation (8–2) still holds, although there is difficulty in its application. If the engine is given sufficient inlet-port area to fill the cylinder and sufficient exhaust lead to permit adequate blowdown, the effective expansion and compression strokes become so short that they seriously reduce the engine output.

A somewhat shorter exhaust lead is permitted in crankcase-scavenged engines because: (1) The expansion-end pressure is low, and (2) The scavenge pressure at the point of inlet opening is maximum and usually over 4 psig.

In consideration of these circumstances and to obtain a favorable compromise, the equation

$$A_m\alpha = 0.0002\, V_{disp}n \tag{8–3}$$

is recommended for crankcase-scavenged engines. This permits a decompression from about 27 psi cylinder-release pressure to a little above 4 psig.

For separately scavenged engines with symmetrical porting the recommended equation for rough calculation is

$$A_m\alpha = 0.00023\, V_{disp}n \tag{8–4}$$

but whenever possible equation (8–2) should be employed with the provision that in the case of high-speed engines (p_{sc} + 2 psi) may be used for p_i. This permits a decompression to a pressure 2 psi above scavenge pressure, which causes a slight amount of backflow of exhaust gases into the inlet ports.

Existing engines frequently have exhaust leads still shorter than those corresponding to equations (8–3) and (8–4), thus showing the extent of compromise the designer has been forced to make for the sake of an acceptable effective stroke length, as discussed later.

The opening of the exhaust, of course, depends as much on the inlet opening as on the exhaust lead; both vary with the scavenge pressure. To obtain maximum output, the scavenging pressure must be so chosen that the *net bmep* becomes maximum. In an unsymmetrically scavenged engine the height of the inlet port largely determines the *gross mep* because that controls the effective compression stroke. In symmetrically scavenged engines the exhaust port closes later, and therefore the calculation for the selection of the optimum scavenge pressure must be based on the height of the exhaust ports.

8.8 Example. Crankcase-Scavenged Engine.

The $10\frac{1}{2}$ by 12-inch 425 rpm crankcase-scavenged engine, shown in Fig. 7–1, has exhaust ports that occupy 19.7 per cent of the cylinder circumference. The inlet ports are open through 96 degrees, from 132 to 228 degrees after top center. When should the exhaust open?

From equation (8–3)

$$A_m\alpha = 0.0002(10\tfrac{1}{2})^2\pi/4 \times 12 \times 425 = 88.3 \text{ sq in.-deg.}$$

The total width of the rectangular exhaust ports

$$b_e = 0.197 \times 10.5 \times \pi = 6.48 \text{ in.}$$

Therefore.

$$F_a = \frac{88.3}{6.48 \times 12} = 1.136 \times 100 \text{ per cent deg.}$$

From Fig. 8–1, with 96-degree inlet duration, the corresponding exhaust lead is found to be 18.5 degrees. Therefore the exhaust should open at 113.5 degrees after top center. According to Fig. 7–5, this is equivalent to 25.2 per cent port height which corresponds to 3.01 inches. Figure 8–7 shows the porting diagram of the production engine. The blow-down time-area is about one-third smaller than the calculated requirement, for reasons discussed in Chapter 10.

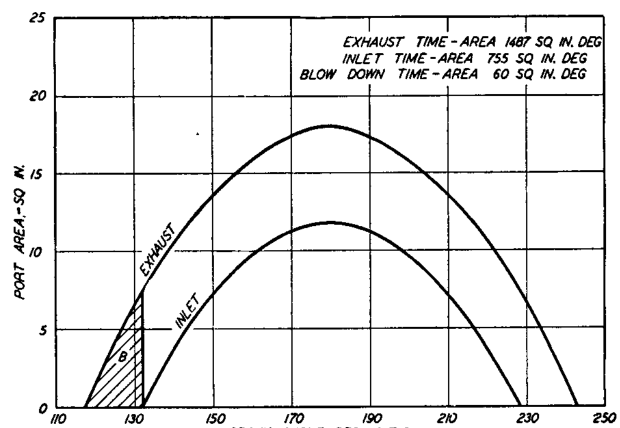

Fig. 8-7. Porting Diagram of Venn-Severin Engine. 10.5 in. bore; 12 in. stroke; 425 rpm.

The 14.6 by 20.1-inch 200 rpm engine shown in Fig. 7–7 and described in 7.11 was found to require an inlet period of 83 degrees duration. The permissible width of the radial exhaust ports is 16.5 inches, or 36 per cent of the circumference. How much is the necessary exhaust lead?

Using equation (8–3)

$$A_m\alpha = 0.00023(14.6)^2\pi/4 \times 20.1 \times 200 = 155 \text{ sq in.-deg.}$$

Converting this into per cent-degree area

$$F_a = \frac{155}{20.1 \times 16.5} = 0.47 \times 100 \text{ per cent deg.}$$

With $\alpha_i = 83$, the corresponding exhaust lead is found from Fig. 8–1 to be $\alpha = 13$ degrees. Consequently, the duration of the exhaust period will be $83 + (2 \times 13) = 109$ degrees. From Fig. 7–5 it is found that this gives an exhaust port length of 17 per cent or 3.41 inches.

APPENDIX TO CHAPTER 8

CALCULATION OF EXHAUST LEAD

8.10 The flow of the gas through the exhaust ports can be considered as a high-pressure discharge from a fixed-volume container through an orifice. The rate of flow through the port at any instant may again be represented by equation (7–11),

$$G = CA\frac{1}{v_m}\sqrt{2g\frac{k}{k-1}p_1v_1}\sqrt{1-\left(\frac{p_m}{p_1}\right)^{\frac{k-1}{k}}} \tag{8–5}$$

where A is the cross-sectional area of the exhaust port and the other symbols are identical with those above. If the pressure drop is greater than the critical, p_m becomes higher than the counter pressure p_2. Assuming adiabatic change,

$$v_m = v_1\left(\frac{p_1}{p_m}\right)^{\frac{1}{k}} \tag{8–6}$$

$$G = CA\sqrt{2g\frac{k}{k-1}\frac{p_1}{v_1}}\sqrt{\left(\frac{p_m}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_m}{p_1}\right)^{\frac{k+1}{k}}}. \tag{8–7}$$

If now

$$C\sqrt{\frac{k}{k-1}\left[\left(\frac{p_m}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_m}{p_1}\right)^{\frac{k+1}{k}}\right]} = \psi \tag{8–8}$$

equation (8–7) can be written

$$G = \psi A\sqrt{2g\frac{p_1}{v_1}} \tag{8–9}$$

which has the form of the equation for the efflux of liquids.

In the foregoing, ψ and k are functions of the pressure ratio p_1/p_2. ψ includes the contraction coefficient, which in case of sharp-edged orifices is considerably smaller than unity, and which also is a function of the pressure ratio. Nusselt [Nusselt, 1932] made a very authoritative investigation of the flow of gases through orifices, and obtained as a result Table 8–I. ψ is called the *Nusselt coefficient.*

For a pressure ratio $p_2/p_1 = 0.245$ the contraction is zero, and the Nusselt coefficient becomes a maximum. In reality, for any pressure ratio p_2/p_1 between 0.245 and 0 the Nusselt coefficient remains constant, instead of following the theoretical values listed in the table. With this correction, experiments very closely support Nusselt's theory.

In Fig. 8–8, ψ is plotted against the pressure ratio p_1/p_2 for sharp-edged orifices. The upper line represents the equation which applies to rounded orifices, in which there is no contraction of the jet or stream.

$$\psi = \sqrt{\frac{k}{k-1}\left[\left(\frac{p_m}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_m}{p_1}\right)^{\frac{k+1}{k}}\right]}$$

Up to the critical pressure ratio

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

p_m is equal to the outside pressure p_2, and ψ is represented by the ascending line. Beyond the critical pressure, ψ remains 0.484.

Table 8-I. Contraction Coefficient α, Nusselt Coefficient ψ, and Controlling Pressure Ratio p_m/p_1 in Function of the Pressure Ratio.

p_2/p_1	p_1/p_2	p_m/p_1	α	ψ
1.0	1.00	1.0	0.597	0
0.9	1.11	0.918	0.617	0.185
0.8	1.25	0.897	0.641	0.282
0.7	1.43	0.846	0.670	0.304
0.6	1.67	0.798	0.706	0.338
0.4	2.50	0.716	0.820	0.383
0.3	3.33	0.692	0.920	0.393
0.245	4.07	0.689	1.000	0.395
0.2	5.00	0.692	1.096	0.393
0.1	10.00	0.725	1.519	0.382
0.0	∞	0.874		0.280

The lower line represents ψ for sharp-edged orifices, where considerable contraction takes place. The maximum is now 0.395 and the critical pressure ratio $p_2/p_1 = 0.245$. The maximum flow is 21 per cent greater through a rounded orifice than through a sharp-edged orifice of equal size, and at lesser pressure drops the difference is still greater. This shows the advantage of rounding the entry edge of any orifice if large flow is desired.

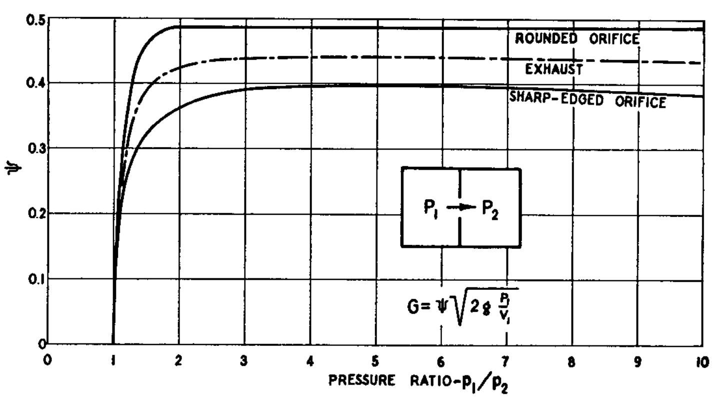

Fig. 8-8. Flow of Gas through an Orifice. Exhaust ports are considered midway between sharp-edged and rounded orifices.

Exhaust ports may be considered as midway between sharp-edged and rounded orifices, because the edges in the cylinder wall usually are sharp (although they need not be), but the piston provides a guide to the lower edge. Therefore, the dash-dotted line may be used in calculating the flow.

The weight of gas passing through the exhaust in dt seconds will be

$$dG_e = \psi_e A \sqrt{2g\frac{p}{v}}\, dt \tag{8-10}$$

where p and v are the instantaneous pressure and specific volume of the gas in the cylinder, respectively.

Since the cylinder volume can be considered constant during the exhaust blowdown period ($V = V_c$), the reduction of the cylinder content is

$$dG_c = -d\left(\frac{V_e}{v}\right). \tag{8-11}$$

Equating (8–10) and (8–11)

$$\psi_e A\sqrt{2g\frac{p}{v}}\,dt = -d\left(\frac{V_e}{v}\right). \tag{8-12}$$

If it is assumed that the cylinder charge expands adiabatically during the blowdown period and the index e refers to the condition that exists when the exhaust port begins to open

$$pv^k = p_e v_e^k, \tag{8-13}$$

which permits the following transformation of equation (8–12). Since

$$\frac{p}{v} = \frac{p_e}{v_e}\left(\frac{p}{p_e}\right)^{\frac{1}{k}+1}$$

the left-hand side of equation (8–12) can be written in the form

$$A \cdot \psi_e\sqrt{2g\frac{p_e}{v_e}\left(\frac{p}{p_e}\right)^{\frac{1}{2}+\frac{1}{2k}}}\cdot dt,$$

and since from equation (8–13)

$$d\left(\frac{1}{v}\right) = \frac{1}{v_e}\,d\left(\frac{p}{p_e}\right)^{\frac{1}{k}} = \frac{1}{v_e}\frac{1}{k}\left(\frac{p}{p_e}\right)^{\frac{1}{k}-1}\cdot d\left(\frac{p}{p_e}\right)$$

the right-hand side of equation (8–12) can be written as

$$-\frac{V_e}{v_e}\frac{1}{k}\left(\frac{p}{p_e}\right)^{\frac{1}{k}-1} d\left(\frac{p}{p_e}\right)$$

therefore,

$$A\psi_e\sqrt{2g}\sqrt{\frac{p_e}{v_e}}\left(\frac{p}{p_e}\right)^{\frac{1}{2}+\frac{1}{2k}} dt = -\frac{V_e}{v_e}\cdot\frac{1}{k}\cdot\left(\frac{p}{p_e}\right)^{\frac{1}{k}-1}\cdot d\left(\frac{p}{p_e}\right)$$

or

$$A\,dt = -\frac{V_e}{\psi_e\sqrt{2g}}\frac{1}{\sqrt{p_e v_e}}\frac{1}{k}\left(\frac{p}{p_e}\right)^{\frac{1}{2k}-\frac{3}{2}}\cdot d\left(\frac{p}{p_e}\right) \tag{8-14}$$

Integrating between the time limits when the exhaust port opens and the inlet port opens,

$$\int_{t_1}^{t_2} A\,dt = -\frac{1}{\psi_e}\frac{1}{\sqrt{2g}}\frac{V_e}{\sqrt{p_e v_e}}\frac{1}{k}\left[\frac{1}{\frac{1}{2k}-\frac{1}{2}}\left(\frac{p}{p_e}\right)^{\frac{1}{2k}-\frac{1}{2}}\right]_{p=pe}^{p=p_i},$$

$$\int_{t_1}^{t_2} A\,dt = \frac{1}{\psi_e}\frac{1}{\sqrt{2g}}\frac{V_e}{\sqrt{p_e v_e}}\frac{2}{k-1}\left[\left(\frac{p_i}{p_e}\right)^{\frac{1-k}{2k}} - 1\right].$$

Considering that $p_e v_e = RT_e$

$$\int_{t_1}^{t_2} A\,dt = \frac{1}{\psi_e}\frac{1}{\sqrt{2g}}\frac{V_e}{\sqrt{RT_e}}\frac{2}{k-1}\left[\left(\frac{p_i}{p_e}\right)^{\frac{1-k}{2k}} - 1\right]. \tag{8-15}$$

This gives the time integral in square inch-seconds, if V_e is inserted in cubic-inches, R in inch per degree F and T_e in degrees Rankine. If calculation in crank degrees is preferred,

$$t = \frac{\alpha}{360\frac{n}{60}} = \frac{\alpha}{6n}$$

and

$$\int_{\alpha_1}^{\alpha_2} A\, d\alpha = nV_e \frac{12}{\psi_e\sqrt{2g}\sqrt{RT_e}} \frac{1}{k-1}\left[\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{2k}} - 1\right].$$

If the average height of the exhaust port opening during the blowdown period is h_m, the total width of the exhaust ports b, and the duration of the blowdown period α degree crank angle, then

$$\int_{\alpha_1}^{\alpha_2} A \cdot d\alpha = h_m b\alpha.$$

The blowdown usually begins with a pressure ratio far above the critical. Near the end the discharge becomes subcritical. Figure 8–8 shows the coefficient ψ for both periods. Since the blowdown ordinarily takes place from 5 atmospheres to a little above 1 atmosphere, a constant value of $\psi = 0.44$ has been selected for calculations.

With this value and $g = 385$ in. per sec², equation (8–15) becomes

$$(8\text{–}2) \qquad A_m\alpha = \frac{0.98}{\sqrt{RT_e}} V_e n \frac{1}{k-1}\left[\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{2k}} - 1\right].$$

The gas constant of the cylinder content is $12 \times 53 = 636$, the temperature of the gas at the beginning of the exhaust blowdown period may vary from 1000 to 1500 F at full load, depending on the engine. If an average temperature of 1250 F = 1710 Rankine is assumed an error of as much as 7 per cent one way or the other may result; the probable error is considerably less. With these values equation (8–16) becomes

$$(8\text{–}16) \qquad A_m\alpha = 0.00094\, V_e n \frac{1}{k-1}\left[\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{2k}} - 1\right].$$

In this equation, V_e and p_e, the cylinder volume and pressure at the time the exhaust port opens depend on the port timing, the very thing which is to be calculated. The practical procedure is first to estimate these quantities and get an approximate answer, which in turn permits calculation of these quantities and making a more accurate determination of the exhaust port opening.

For the preliminary calculation it is permissible to make $V_e = 0.8\ V_{disp}$ and $p_{ie}/p = 3$. The exponent k is taken as 1.3, which makes

$$\frac{1}{k-1}\left[\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{2k}} - 1\right] = 0.4533$$

and equation (8–16) becomes

$$A_m\alpha = 0.00094 \times 0.8 \times 0.4533\ V_{disp}n$$

or

$$(8\text{–}1) \qquad A_m\alpha = 0.00033\ V_{disp}n.$$

This is the equation recommended for preliminary calculations.

CHAPTER 9

SUPERCHARGE

9.1 Even before scavenging is completed, the charging of the cylinder with fresh air begins. The objective is to trap as much air in the cylinder as possible. Success in doing this depends largely on the timing of the closure of the exhaust ports.

The preceding calculations served to determine the proper opening time of the exhaust port or valve, but they gave no clue as to the proper closing time or the proper time-area after the blowdown period.

9.2 Symmetrical Scavenge.

In a symmetrically scavenged engine, as soon as the opening is fixed, the closing of the exhaust port is beyond control. Some control is possible over the port areas. They may be made full-length rectangular, extending down to the piston edge at its bottom dead-center position, shorter or narrower on the bottom, or both.

In practice, exhaust ports of symmetrically scavenged engines are generally made rectangular (or rhomboidal) and full length. It is true that the exhaust port has finished its most important task by the end of the blowdown period, and there is generally an abundance of exhaust-port area to permit unobstructed scavenging afterward. By restricting exhaust-port area after the blowdown period, it might be possible to build up more pressure in the cylinder. Whatever pressure might have been built up near bottom center would be lost, however, during the period between inlet-port closure and exhaust-port closure. This period is exactly equal to the exhaust lead and sufficiently long to bring the cylinder pressure down to practically atmospheric. For this reason no pressure trapping or supercharge of any appreciable amount is possible in symmetrically scavenged engines, and the exhaust ports ordinarily are of full length and full width throughout, although occasionally they are made shorter for other good reasons.

9.3 Nonsymmetrical Scavenge.

In nonsymmetrical engines, however, the designer tries to trap as much air as possible by skillful design of the exhaust ports. In the first place, by closing the exhaust before the inlet, he uses a supercharging period to build up pressures. This period is comparatively short, however, unless it encroaches too much on the effective compression stroke, which is to be avoided. The pressure build-up must therefore be started earlier by restricting the exhaust-port cross section during the later part of the scavenging period. The greater the surplus of inlet-port area over the exhaust-port area, the more fresh air is trapped in the cylinder. The exact closing time of the exhaust is really less important than the throttling effect produced by a restricted port area after bottom center, especially during the last 20 to 30 degrees of the charging period.

9.4 Pressure Build-Up.

The pressure build-up in the cylinder after bottom center can be calculated by the step-by-step method, but it is a laborious calculation and is seldom done. The theoretical requirement is to

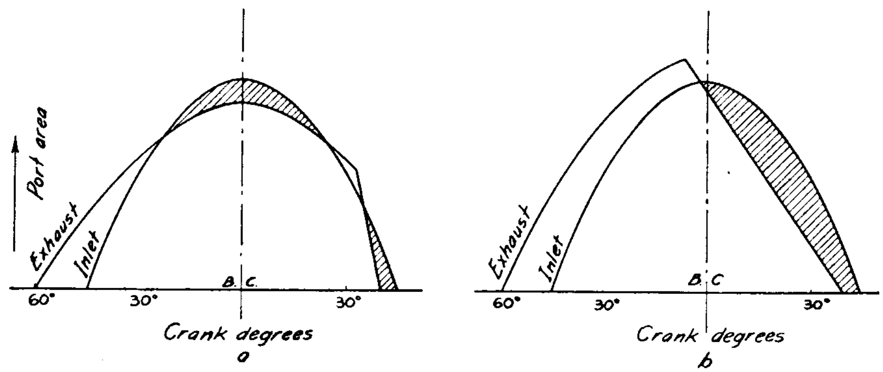

Fig. 9-1. Porting Diagrams with Supercharge. Diagrams *a* and *b* have equal timings and time-areas, yet *b* shows better supercharge.

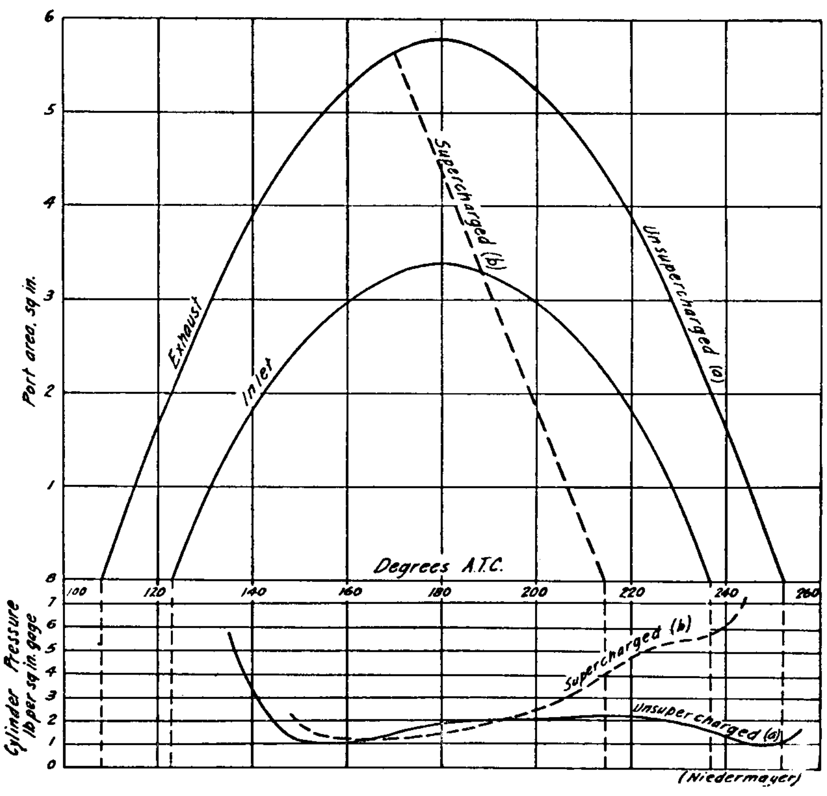

Fig. 9-2. Porting Diagrams and Measured Cylinder Pressures during the Charging Process with (*a*) Symmetrical and (*b*) Nonsymmetrical Porting. 4.7 in. bore, 5.9 in. stroke, 700 rpm loop-scavenged engine developing (*a*) 67.5 and (*b*) 86 psi bmep respectively. The supercharge was produced by a rotary valve in the exhaust port.

provide for a pressure build-up such that the pressure just about equals the scavenging pressure when the inlet closes. In some cases the pressure build-up may overshoot this mark. With liberal inlet areas and restricted exhaust areas, the compression of the piston may raise the cylinder pressure above the scavenging pressure before the inlet closes. Then, naturally, a reverse flow sets in from the cylinder through the inlet ports to the scavenging blower. The engine loses air during that period and this occurrence should be avoided. This case is not very rare, and when it happens a late inlet closure can usually be blamed. Therefore, while an increase of the exhaust cross sections near the inlet closure would eliminate this reverse flow, it is not the proper remedy for it.

The problem frequently is the opposite: to throttle the exhaust sufficiently near the completion of the charging period, in order to build up the cylinder pressure to the scavenge pressure by the time the inlet closes. It has been stated that a short supercharging period alone does not always accomplish this, because the air requires time to pass through the ports. This is illustrated in Fig. 9–1. Port-area diagram *a* has the same supercharging period as port-area diagram *b*, and yet it represents a much poorer supercharge. On the other hand, excellent supercharging effect has been obtained with engines with a slightly negative supercharging period, in which, during the latter part of the charging period, the inlet-port area considerably exceeded the corresponding exhaust-port area (see Fig. 9–9).

9.5 Supercharge Valve in the Exhaust.

In principle the simplest way to convert a symmetrically scavenged engine into a supercharged engine is by placing a rotary valve in the exhaust duct. Figure 9–2, top, shows the change in the port-area diagrams brought about by a rotary valve in the exhaust. The exhaust closure is advanced from 252 to 215 degrees after top center and thus a considerable surplus inlet area is created. Weak-spring diagrams gave the pressure curves shown below the port-area diagram for symmetrical and supercharged scavenging respectively. The gain in charging pressure is considerable, 5.7 psig as compared with 1 psig unsupercharged. In addition, the effective compression stroke has been lengthened. The measured increase in power output was 12.7 per cent. A Hamilton–M.A.N. Double Acting Diesel engine using a rotary valve in the exhaust duct is shown in Fig. 9–3.

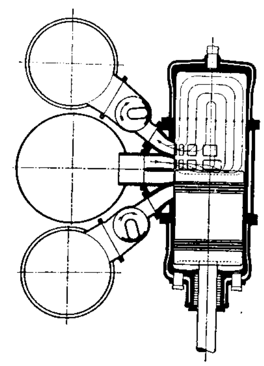

Fig. 9-3. Rotary Valve in Exhaust Hamilton-M.A.N. double acting engine. (By permission of Lima-Hamilton Corp.)

A simplified version of the exhaust rotary valve was recently developed by the Cooper-Bessemer Corporation. As is shown in Fig. 9–4, the rotary valve consists of a rotating blade or butterfly

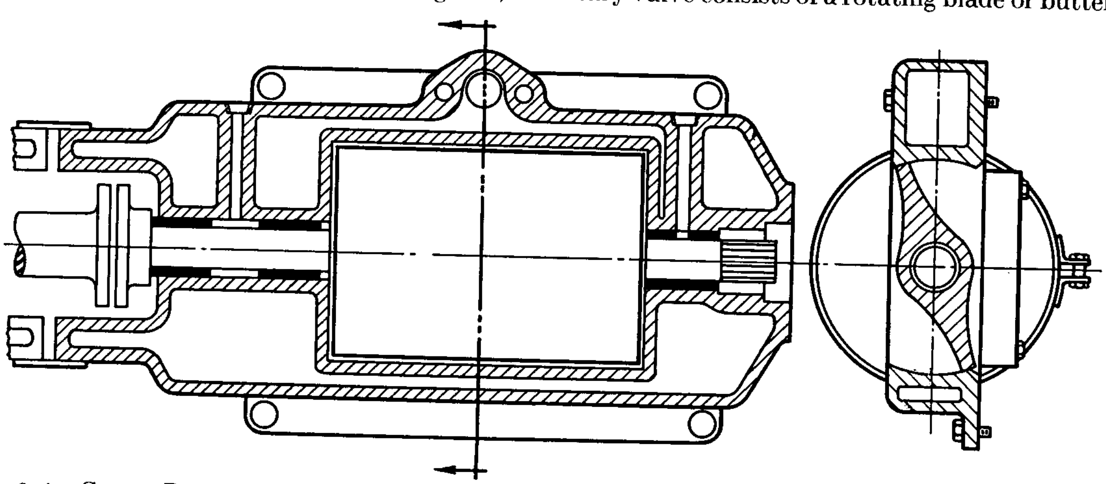

Fig. 9-4. Cooper-Bessemer Supercharge Valve in the Exhaust. (By permission of Cooper-Bessemer Corp.)

valve mounted with a rather loose fit between the blades and the cylindrical segment, and driven at crankshaft speed. This type of valve is less likely to clog up than is a regular rotary valve, yet it produces a satisfactory supercharge period. Figure 9–5 shows the port timing of an engine employing Cooper-Bessemer supercharge valve and the geometry of the valve which produces that timing.

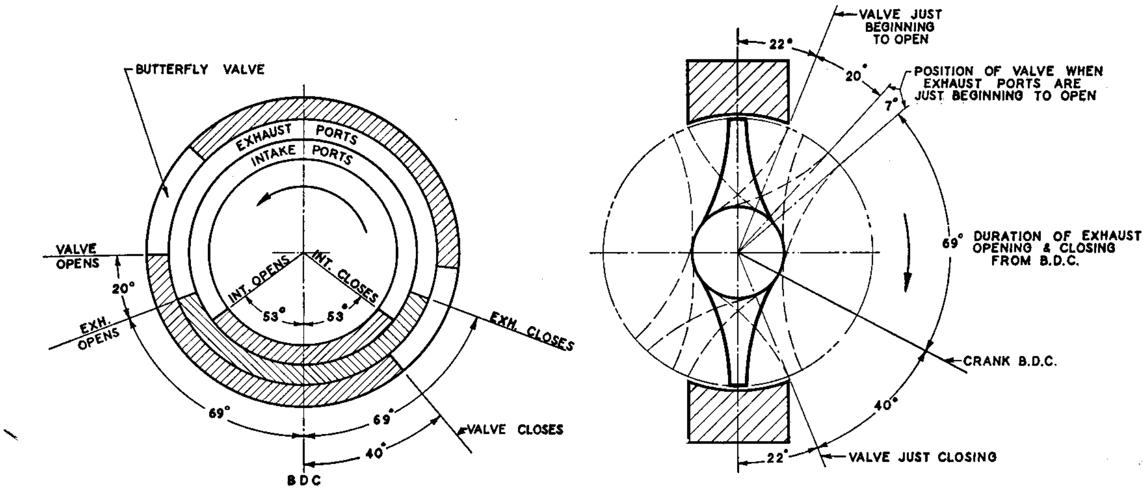

Fig. 9-5. Port Timing with Cooper-Bessemer Supercharge Valve. (By permission of Cooper-Bessemer Corp.)

9.6 Supercharge Valve in the Inlet.

More frequently the supercharge valve is placed in the inlet duct. This arrangement, pioneered by Sulzer in 1910, is more attractive mechanically than valving the hot exhaust gases. However, from the standpoint of charging, it does not quite equal the performance represented by Fig. 9–2. The port area diagram with a rotary supercharge valve in the inlet would look like Fig. 9–6. The unfavorable aspect of that diagram is that the nonsymmetry is produced by cutting out some of the inlet time-area, which is precious. (By placing the supercharge valve in the exhaust, only the later part of the exhaust time-area is cut out, which is practically worthless.) The effect is, that in order to get adequate inlet time-area the closing time of the inlet port must be later in Fig. 9–6 than in Fig. 9–2, with the result that the effective compression stroke becomes shorter.

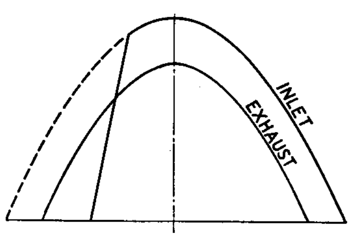

Fig. 9-6. Port Area Diagram with a Rotary Supercharge Valve in the Inlet.

Nevertheless supercharge valves in inlet became rather popular because of their mechanical simplicity. As a rule not all of the inlet air flows through the supercharge valve, but only a portion of it. Two rows of inlet ports are used. The lower row is placed lower than the exhaust ports so that when the piston uncovers the upper edge of the lower ports the blowdown of the cylinder has been completed at least to the level of the scavenging pressure. During that time the upper row of inlet ports is closed. On the up-stroke of the piston the upper ports remain open until they are covered by the piston. Since the upper edge of the upper ports is higher than the upper edge of the exhaust ports, during the time period the piston travels that distance a supercharging at the cylinder takes place.

Supercharge valves in the inlet may be mechanically actuated, as shown in Fig. 9–7 or automatic as shown in Fig. 9–8.

It is a matter of opinion whether supercharge valves located in the inlet have sufficient merit to be worth the complication involved, as almost the same output can be obtained with symmetrical scavenging. On the other hand, supercharge valves of simple construction placed next to the exhaust ports may add at least 10 psi to the bmep obtainable.

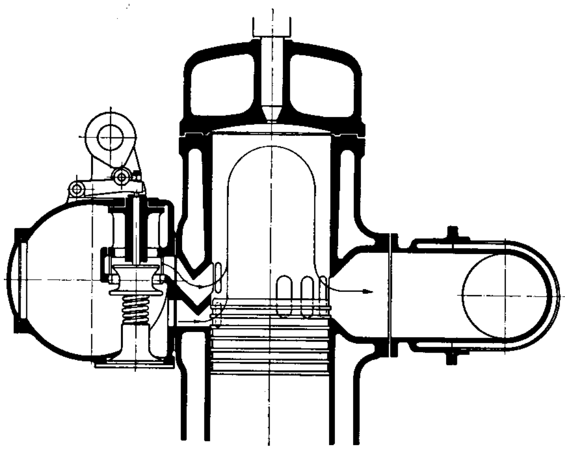

Fig. 9-7. Mechanically Actuated Supercharge Valve in Inlet Duct. (By permission of Sulzer Bros.)

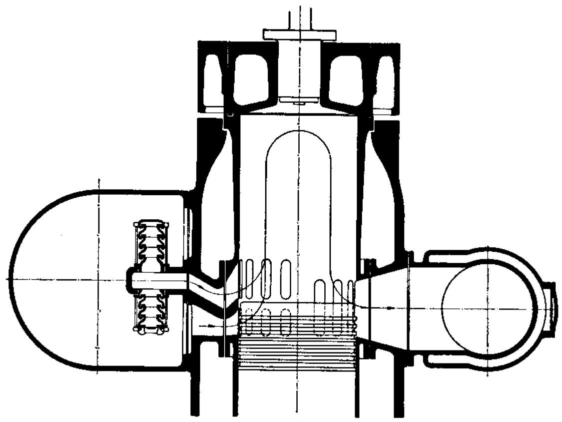

Fig. 9-8. Automatic Supercharge Valve in Inlet Duct. (By permission of Sulzer Bros.)

A good supercharge can also be realized with cam-controlled exhaust valves. Figure 9-9 shows the port areas of the poppet-exhaust engine described in 7.13 and 8.6 and shown in Fig. 7-10. It is

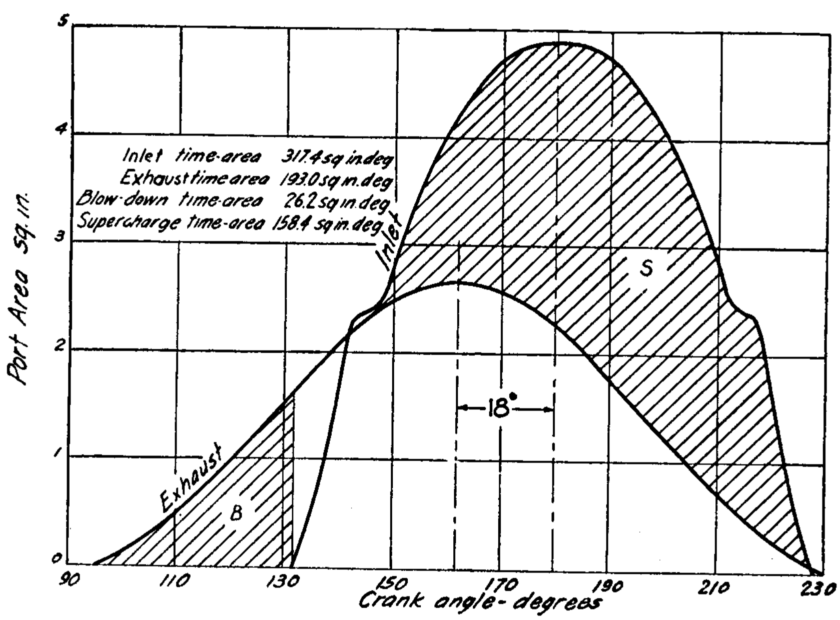

Fig. 9-9. Porting Diagram, General Motors Poppet Exhaust Valve Engine.

evident that although the exhaust actually closes $1\frac{1}{2}$ degrees after the inlet, there is an enormous surplus inlet area, marked *S*, before the inlet closes.

The McCullum-Burt sleeve-valve drive also permits a favorable surplus of inlet over exhaust as illustrated in Fig. 9-10, which represents Ricardo's experimental $5\frac{1}{2}$ by 7-inch sleeve-valve engine [Pye, 1934].

Less favorable in this respect is the opposed-piston engine shown on Fig. 9–11, which corresponds to the 8 by 10 + 10-inch engine described in 7.12 and 8.5 and shown in Fig. 7–8. Since the

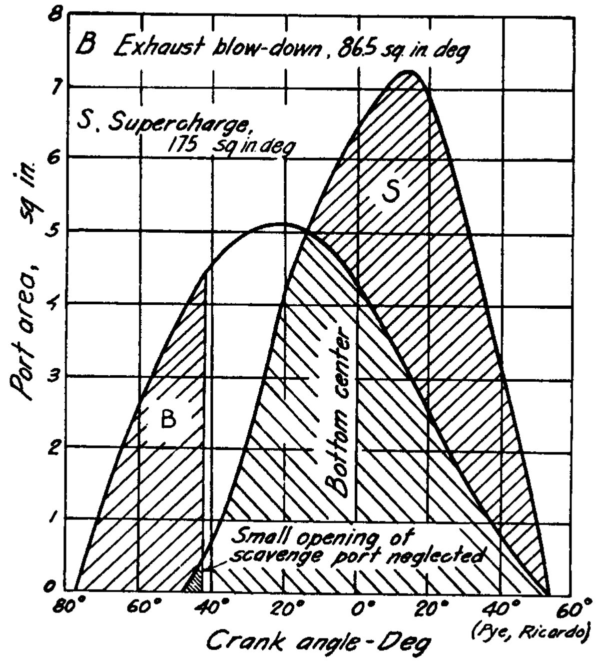

Fig. 9-10. Porting Diagram, Ricardo Sleeve Valve Diesel Engine. (D. R. Pye, *The Internal Combustion Engine*, Oxford at the Clarendon Press, 1934.)

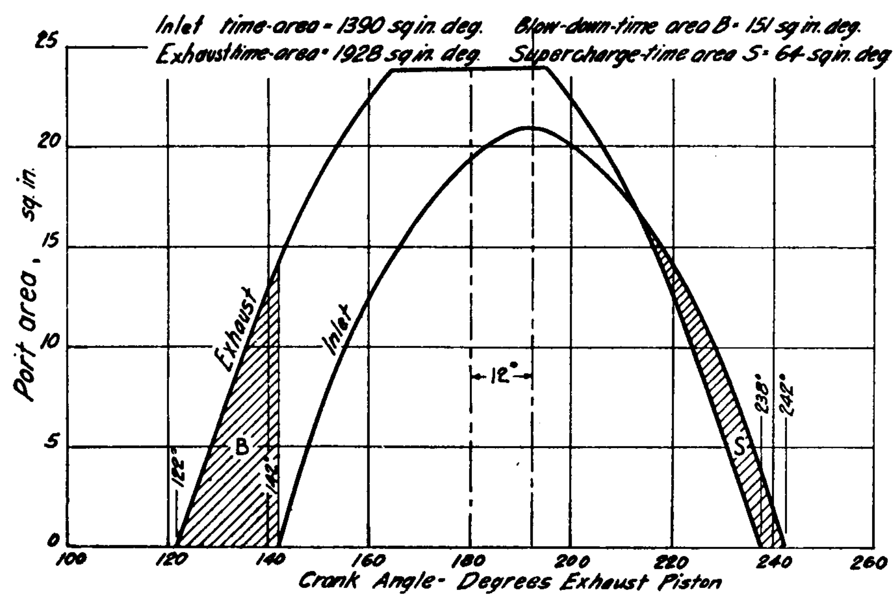

Fig. 9-11. Porting Diagram, Fairbanks-Morse Opposed-Piston Engine.

opening and closing of the exhaust are symmetrical to each other and controlled by the fast-moving piston, a slow-closing exhaust is impossible.

9.7 Phase Advance.

A common expedient for obtaining a supercharge is to give a phase advance of 10- to 15-degree crank angle to the crank which controls the exhaust piston. The diagram shown in Fig. 9-11 is based on a phase advance of 12 degrees which produces a supercharging period of 4 degrees.

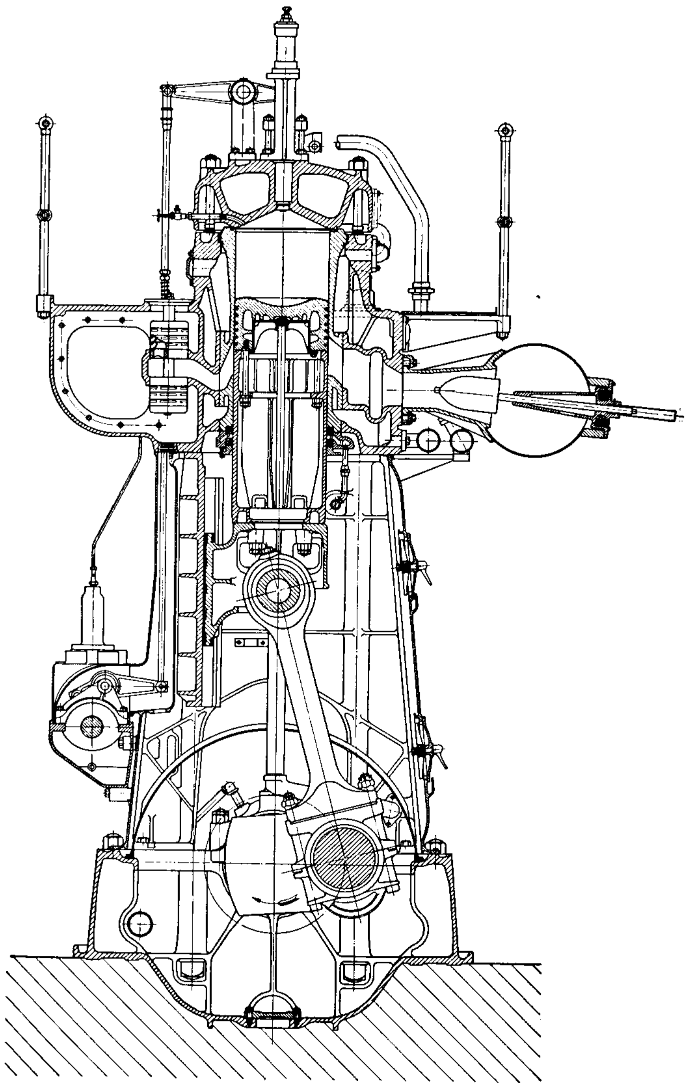

Fig. 9-12. Nordberg Reciprocating Blower Scavenged Engine with Sulzer Type Supercharge Valve (Crosshead Type). 21 in. bore 31 in. stroke, 225 rpm.

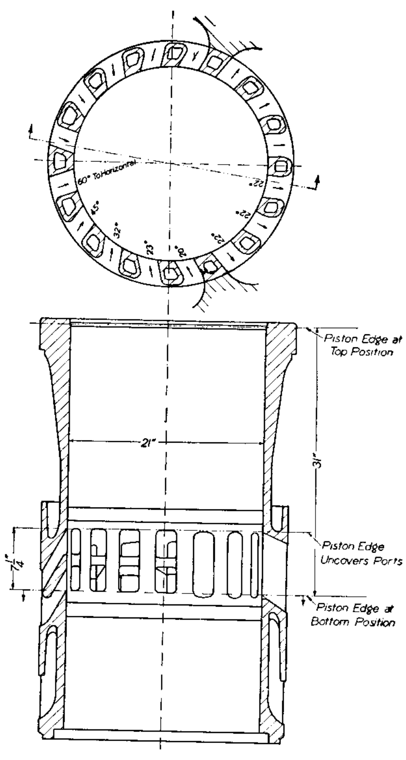

Fig. 9-13. Port Arrangement of Nordberg 21 in. × 31 in. Engine. (By permission of Nordberg Manufacturing Company.)

Another expedient for pressure trapping is to restrict the exhaust-port areas by making them shorter or narrower or both. If instead of extending the exhaust ports down to the end of the stroke, they are terminated some distance above that point, the exhaust-port-area diagram becomes truncated as in Fig. 9–11. Naturally, it is desirable to have all the port width possible during the blowdown period. The lower part of the exhaust ports, however, can be made narrower, trapezoidal, triangular, or semicircular in shape. The gain, however, is of no great consequence. Some throttling of the exhaust is achieved near bottom center, but none near the exhaust-closing point, where it is most desirable.

Even with these expedients, the exhaust ports of opposed-piston engines invariably come out too large for effective pressure build-up.

9.8 Example.

Figure 9–12 shows the section of a large piston-controlled two-stroke cycle engine, supercharged with a Sulzer-type automatic supercharge valve located in the intake header. Curtis-type loop scavenging has been used and the port arrangement is shown on Fig. 9–13. Scavenge air in excess of 32.5 per cent over the piston displacement is supplied by a reciprocating pump at 2.35 psig pressure. The automatic supercharge valves are assumed to open as soon as the cylinder pressure drops below the scavenge pressure and to remain open as long as the scavenge pressure exceeds the cylinder pressure, unless the piston closes the inlet ports in the meantime.

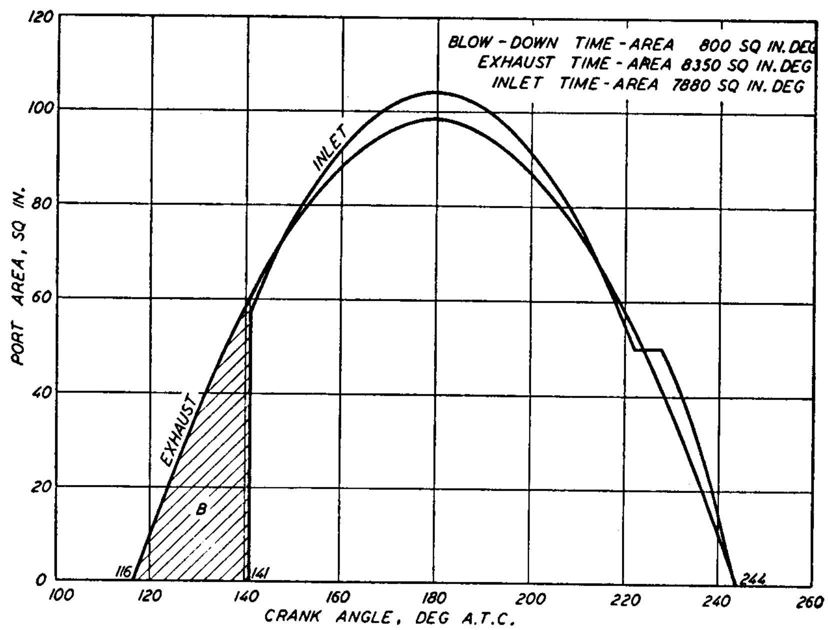

Fig. 9-14. Porting Diagram of Nordberg Engine.

The task ahead is to check the porting by calculation. As a first step the port-area diagram is constructed on the basis of piston travel and uncovered port areas. This diagram is shown in Fig. 9–14. Knowing the length of the stroke (31 in.) and that of the connecting rod (61 in.), the port areas can be laid out from the port dimensions shown in Fig. 9–14. The only data missing is the time at which the supercharge valve opens. Weak-spring indicator diagrams taken on an engine of similar type indicate that at full load the cylinder pressure drops to 2.35 psig (scavenge pressure) at 141 degrees after top center; therefore that point is taken for opening of the supercharge valve.

The rest is routine calculation. The horizontal threshold in the inlet curve (Fig. 9–14) is caused by the vane in the inlet ports (Fig. 9–13). While the piston passes the vane, the inlet port area remains constant.

Table 9-I. Nordberg 21 by 31-inch Engine, with Curtis-Type Loop Scavenging.

Bore (in.)	21
Stroke (in.)	31
Rpm	225
Displacement Volume (cu in.)	10,750
Rated hp per Cylinder	429
Rated bmep (psi)	70.5
Length of Connecting Rod (in.)	61.0
Air-Delivery Ratio	1.325
Scavenging Pressure (psi)	2.35
Scavenging Air Temperature	100 F
Exhaust	
Beginning	116
End (deg A.T.C.)	244
Inlet	
Beginning	141
End (deg A.T.C.)	244
Exhaust Lead	25°
Exhaust Port Height	
Inches	7.25
Per cent of Stroke	23.4
Exhaust Port Width	
Inches	14.625
Per cent of Circumference	22.2
Inlet Port Height	
Inches	7.25
Per cent of Stroke	23.4
Inlet Port Width	
Inches	21.5
Per cent of Circumference	32.6
Effective Expansion Stroke (per cent)	76.6
Effective Compression Stroke (per cent)	76.6
Maximum Port Area — Exhaust (sq in.)	98.3
Maximum Port Area — Inlet (sq in.)	104.2
Mean Port Area — Exhaust (sq in.)	65.2
Mean Port Area — Inlet (sq in.)	76.5
Mean Exhaust Time Area (sq in.-deg)	8350
Mean Inlet Time Area (sq in.-deg)	7880
Mean Port Velocity — Exhaust (ft per sec)	192.5
Mean Port Velocity — Inlet (ft per sec)	204

The important porting data are listed in Table 9–I, and the required inlet time-area is calculated by equation (7–1)

$$A_{im}\alpha_i = \frac{L}{2w_i} V_{disp}n = \frac{1.325}{2w_i} \times 10750 \times 225$$

With a pressure ratio of $(14.5 + 2.35)/14.5 = 1.16$, and 100 F temperature, Fig. 7–2 gives $w_1/S =$ 510 feet per second. The scavenge factor of the nonsymmetrical loop-scavenged engine with rounded entry edges and smooth ports is estimated according to Table 7–I as $S = 0.4$. With these values

$$A_{im}\alpha_i = \frac{1.325}{2 \times 0.4 \times 510} \times 10750 \times 225 = 7850 \text{ sq in.-deg.}$$

That much inlet time-area is required. Planimetration of the inlet port-area curve in Fig. 9–14 gives the available inlet-time area as 7880 square inch-degrees. The agreement is closer than one-half of one per cent.

In calculating the required exhaust lead let us first use the simple equation (8–1)

$$A_m\alpha = 0.00033\ V_{disp}n = 0.00033 \times 10750 \times 225$$
$$= 800 \text{ sq in.-deg.}$$

This just happens to be exactly equal to the available blowdown area from Fig. 9–14. The more precise formula (8–2)

$$A_m\alpha = \frac{0.98}{\sqrt{RT_e}}\, V_e n y$$

requires the knowledge of T_e, V_e and p_e. V_e can be estimated as 1710 Rankine. Since the effective expansion stroke is 76.6 per cent and the compression ratio is approximately 13.1,

$$V_e = 10750\left(0.766 + \frac{1}{13 - 1}\right) = 9150 \text{ cu in.}$$

From Fig. 8–2 with an estimated mip of 83 psi, $p_e = 50$ psi, and from Fig. 8–3 with $p_e/p_i = 50/16.85 = 2.97$, and $k = 1.35$, $y = 0.43$. Consequently,

$$A_m\alpha = \frac{0.98}{\sqrt{636 \times 1710}} \times 9150 \times 225 \times 0.43 = 825 \text{ sq in.-deg.}$$

The precise calculation gives about 3 per cent more required blowdown area than is available. The agreement is remarkably close.

REDUCED TIME-AREA METHOD

9.9 The method of calculating the amount of air flowing through the inlet ports as developed in Chapter 7 was based on integrating the inlet-port time-area and disregarding the exhaust-port areas. The method is justified in the case of symmetrically scavenged engines because there the inlet ports constitute practically the entire resistance to the air flow. The open portions of the exhaust ports are always considerably larger than those of the inlet ports as is seen for instance in Fig. 8–7. The additional resistance caused by the exhaust ports is small.

In nonsymmetrically scavenged (supercharged) engines during the later part of the inlet period the inlet port areas are larger than the exhaust port areas, and by ignoring the resistance offered to the flow by the exhaust ports a considerable error is committed. The actual air flow is appreciably smaller than that calculated on the basis of the inlet port time-areas. In order to prevent such an overestimation of the air flow the recommended scavenge factors S in Table 7–I were selected very low for unsymmetrically scavenged engines; in case of a loop-scavenged engine with considerable supercharge, as low as 0.2.

A better method of calculating the air flow through nonsymmetrically scavenged engines was proposed by Hold [Hold, 1941]. In his method of calculation it is recognized that two flow resistances are in series: the inlet ports and the exhaust ports. Ignoring variations of compression of the air inside of the cylinder, the amount of air entering through the inlet port at any time is equal to the amount of air leaving through the exhaust ports at the same time. If density variations of the air while passing the cylinder are ignored,

$$G = \mu A_i \gamma \sqrt{2g \frac{p_i - p_c}{\gamma}} = \mu A_e \gamma \sqrt{2g \frac{p_c - p_e}{\gamma}}. \tag{9–1}$$

9.10 Equivalent Area.

If the inlet and the exhaust areas are replaced by a single equivalent area A_{red} (reduced area), which offers the same resistance to the air flow as the inlet and exhaust areas in series, then

$$G = \mu A_{red} \gamma \sqrt{2g \frac{p_i - p_e}{\gamma}}. \tag{9–2}$$

From equations (9–1) and (9–2)

$$A_i^2 = \frac{G^2}{\mu^2 \gamma 2g(p_i - p_c)}$$

$$A_e^2 = \frac{G^2}{\mu^2 \gamma 2g(p_c - p_e)}$$

$$A_{red}^2 = \frac{G^2}{\mu^2 \gamma 2g(p_i - p_e)}$$

Setting

$$\frac{1}{\mu^2 \gamma 2g} = C$$

$$p_i - p_c = \frac{CG^2}{A_i^2} \tag{9–3}$$

$$p_c - p_e = \frac{CG^2}{A_e^2} \tag{9–4}$$

$$p_i - p_e = \frac{CG^2}{A_{red}^2} \tag{9–5}$$

By adding equations (9–3) and (9–4),

$$p_i - p_e = CG^2 \left(\frac{1}{A_i^2} + \frac{1}{A_e^2} \right)$$

which compared with equation (9–5) gives

$$\frac{1}{A_{red}^2} = \frac{1}{A_i^2} + \frac{1}{A_e^2}. \tag{9–6}$$

Therefore for openings A_i and A_c in series a single opening can be substituted A_{red}, which has the relation to A_i and A_e expressed by equation (9–6) or (9–7)

$$A_{red} = \frac{1}{\sqrt{\frac{1}{A_i^2} + \frac{1}{A_e^2}}}. \tag{9–7}$$

This *reduced area* permits the same amount of air flow as the inlet and exhaust port areas combined.

On this basis A_{red} can be determined for any crank-angle. Figure (9–15) shows the relations for a piston-controlled engine with a rotary valve in the exhaust.

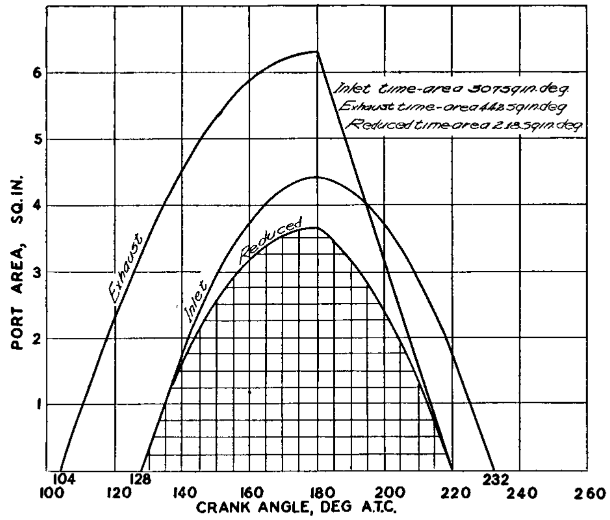

Fig. 9-15. Porting Diagram. 4.5 in. bore, 6 in. stroke, 1800 rpm.

Planimetration of this reduced port area gives the reduced time area

$$(A\alpha)_{red} = \int_1^2 A_{red}\, d\alpha$$

which in turn gives the mean inlet velocity (see equation (8–1)).

$$w_{ir} = \frac{L \cdot V_{disp} \cdot n}{2(A\alpha)_{red}}.$$

Since $(A\alpha)_{red}$ is always smaller than $A_{im}\alpha_i$, w_{ir} is always greater than w_i obtained in the conventional manner. Therefore, the scavenge factor, which is the ratio between the actual mean inlet velocity and the thermodynamically calculated (Fig. 8–2) inlet velocity, turns out to be greater if the reduced area method is employed. In addition, the scavenge factors determined by this method proved to be more constant, which supports the soundness of this method of calculation.

9.11 Example 1. Opposed-Piston Engine.

The porting diagram of the Fairbanks-Morse $8\frac{1}{2}$ by (10 + 10) opposed-piston engine is shown in Fig. 8–5. From this by the construction explained Fig. 9–16 has been obtained.

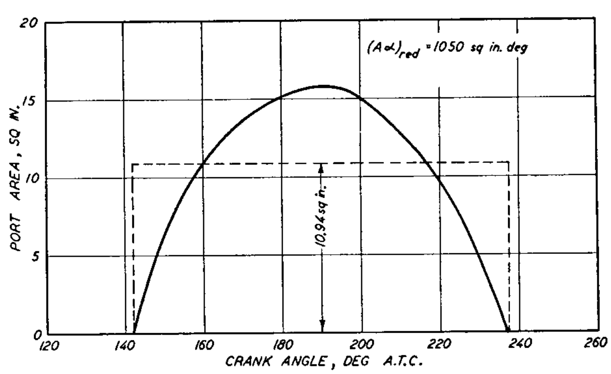

Fig. 9-16. Reduced Time-Area Diagram of Fairbanks-Morse Opposed-Piston Engine. 8 in. bore, 10 + 10 in. stroke.

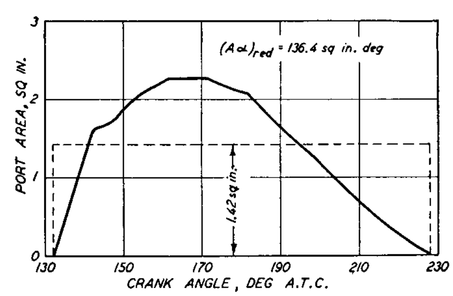

Fig. 9-17. Reduced Time-Area Diagram of General Motors Poppet Exhaust Valve Engine. $4\frac{1}{4}$ in. bore, 5 in. stroke.

The original inlet time-area was 1390 square inch degrees. The reduced time-area is 1050 square inch degrees. The scavenge factor was $S = 0.64$. The new scavenge factor is

$$S_{red} = 0.815.$$

9.12 Example 2. Poppet Exhaust-Valve Engine.

The porting diagram of the General Motors $4\frac{1}{4}$ by 5 poppet exhaust-valve engine is shown in Fig. 8–6. From this by the construction explained above Fig. 9–17 has been obtained.

In Fig. 8–6 the inlet time-area has been 317.4 square inch degrees. The reduced time-area in Fig. 9–17 is 136.4 square inch degrees or less than half of its former value. The scavenge factor has been $S = 0.37$. The new scavenge factor is

$$S_{red} = 0.86$$

We see that in the former case when the supercharge area was relatively small the scavenge factors calculated by the conventional method and by the new method differ by only 0.175 while in the latter case where the supercharge area is considerable, they differ by 0.49.

CHAPTER 10

COMPROMISE

10-1 Calculation of the inlet and exhaust ports by means of the proper formulas occasionally leads to results which make the effective compression and expansion strokes so short that the engine output is seriously affected. This is liable to happen if:

1. The engine has symmetrical scavenge,
2. The ports were made relatively narrow for mechanical reasons, or
3. The speed is high for the bore.

A combination of these factors makes the porting problem extremely difficult. Every effort should be made to have no more than one of these adverse conditions to cope with, otherwise a decent porting is almost impossible.

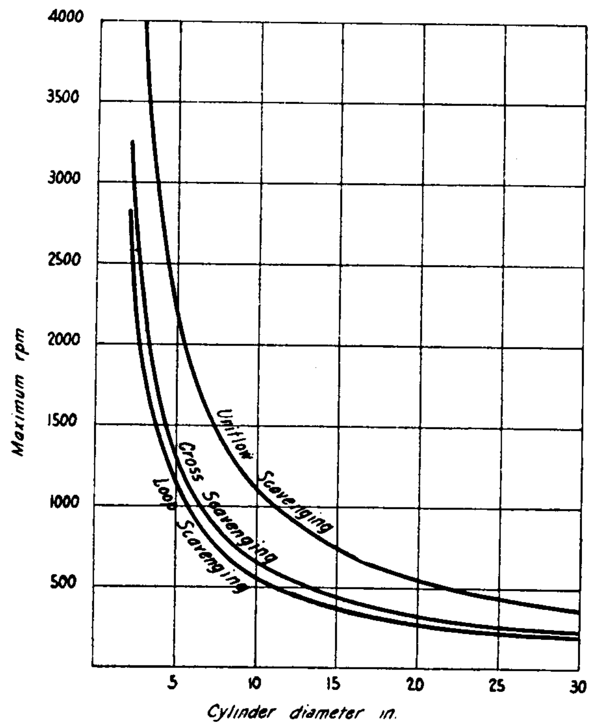

Fig. 10-1. Practical Speed Limits for Two-Stroke Cycle Engines.

Alternatives to symmetrical scavenge have been discussed earlier. The modern trend is toward nonsymmetrical or supercharged engines.

When the ports are narrow it may be possible to widen them by machining them instead of casting. Proper rounding of the entry edges alone may be equivalent to a 20 per cent increase of the port area.

In engines of relatively high speed, it is particularly important that the other factors be well taken care of and that the porting design be close to optimum. Even so, there is a speed limit which it is inadvisable to exceed. With increasing speed, the relative port heights and also the necessary scavenge pressure increase, so that a point is reached beyond which the power output must decrease.

The value of Dn/β* is the controlling factor. Experience indicates that 20,000 is about the practical maximum for Dn/β. Figure 10-1 shows the limiting speeds for engines of various bores, the average values of β being

* D is the cylinder diameter, n the revolutions per minute, and β the total width of the intake ports divided by the circumference.

assumed to be 0.55 for uniflow scavenging, 0.325 for cross scavenging, and 0.275 for loop scavenging. Of course, the curves do not give any assurance that the speeds indicated are mechanically feasible.

Even if the engine speed is below the value indicated by Fig. 10–1, the calculation sometimes results in ports so high that the effective stroke becomes intolerably short.

10.2 Effective Stroke.

The power output is affected by both the effective compression stroke and the effective expansion stroke. If the effective compression stroke is shortened, the relative charge is reduced in about the same proportion. The mean effective pressure is in direct proportion to the relative charge. It may, therefore, be said that one per cent decrease of the effective compression stroke decreases the power output approximately one per cent.

One per cent of the stroke corresponds to about 10 degrees of crank rotation near bottom center, to 3 degrees at 25 degrees after bottom center, and 2 degrees at 45 degrees after bottom center, but at 90 degrees after bottom center it is equivalent to only about one degree of crank rotation. Therefore, the effective compression stroke is little affected by the inlet duration as long as the latter is not too great. However, the effective compression stroke diminishes rapidly with delay of the port closure in the region of 240 to 270 degrees after top center. Such late inlet closures should be avoided if possible.

Late inlet closure is sometimes made necessary by circumstances, but it should be realized that close to the middle of the stroke every degree of delay in closing the ports cuts the power output by about one per cent. Frequently one per cent of the power output added to that consumed by the blower would result in a scavenge pressure so much higher that an appreciable shortening of the inlet period would be feasible.

The power output is affected also by the effective expansion stroke. By the exhaust blowdown through the exhaust ports, most of the energy contained in the cylinder charge at the point of exhaust opening is lost. From Fig. 10–2, by comparing diagram *a*, which is a typical four-stroke cycle card, the blowdown loss due to the shortened effective expansion stroke is made apparent.

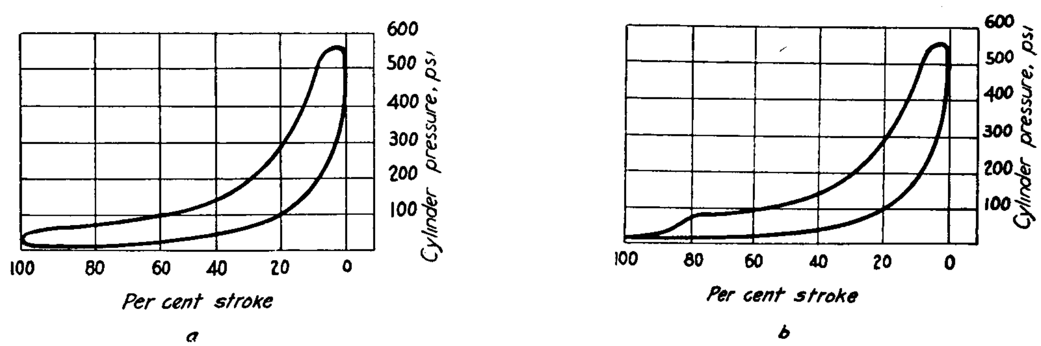

Fig. 10-2. Typical Indicator Diagrams (a) Four-Stroke Cycle, and (b) Two-Stroke Cycle Diesel Engines.

10.3 Power Loss.

The power loss due to the shortened expansion stroke is frequently overemphasized. The loss may be estimated on the basis of the theoretical constant-pressure cycle diagram, Fig. 10–3, based on a compression ratio of 15:1. The total work area is divided into 10 parts corresponding to 10, 20, 30, 40, 50, 60, 70, 80, 90, and 100 per cent of the effective stroke. The work area is shown by the number in the shaded area as per cent of the total indicated area, and the added amounts are represented by the curve running up to 100. The diagram shows that if the effective expansion stroke is 80 per cent, the total work will be 94 per cent instead of 100. This would mean that the 20 per cent loss in effec-

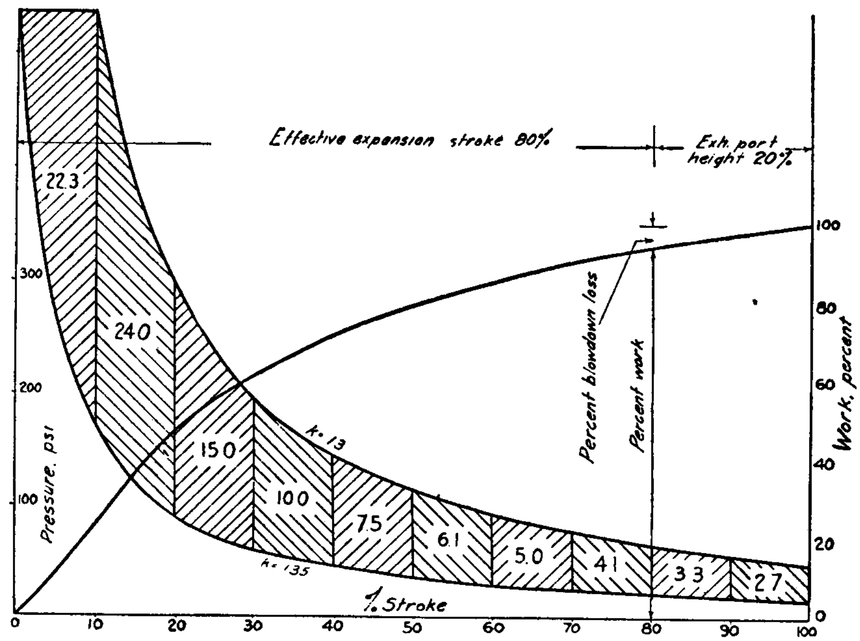

Fig. 10-3. Work-Area of Theoretical Indicator Diagram Reduced by Blowdown Loss. Compression ratio 15 : 1.

tive expansion stroke causes 6 per cent loss in power. The actual loss may be a little more, due to after-burning which makes the tail of the indicator diagram fatter. On the other hand, the pressure drop at the opening of the exhaust is never perpendicular, but sloping, which makes the blowdown less than that theoretically calculated.

10.4 Late Port Closure.

It is substantially true that a one per cent reduction of the expansion stroke reduces the power output less than one-half per cent, while one per cent reduction of the compression stroke reduces it by one per cent. Therefore, a long effective compression stroke is more important than a long effective expansion stroke. This fact is not generally recognized. It is believed that a shortness of effective compression stroke can always be made up by supercharge, that is, that the cylinder pressure can be brought up to and beyond what it would have been without any shortening of the effective compression stroke. A little calculation is sufficient to prove that this is a fallacy.

If the cylinder is filled with atmospheric air of 14.5 psia pressure at bottom dead center, and the compression is polytropic with an exponent of 1.35, and a compression ratio of 15:1 is used, the pressures attained at various stages of the compression stroke are approximately as follows:

Per cent stroke from top center	100	90	80	70	60
Pressure (psig)	0	2.0	4.2	8.0	12.2

Since the full scavenge pressure rarely exceeds 8 psig, supercharge cannot make up for a 30 per cent loss in effective compression stroke. Supercharge is very helpful in increasing the cylinder charge

if it takes place at the early part of the compression stroke, but if the port closure is very late, the cylinder actually loses air during the supercharging period.

In case one or more of the factors mentioned at the beginning of this chapter are present, it is very hard to avoid excessive shortening of the effective expansion stroke or the effective compression stroke, or both. A certain inlet duration is necessary to fill the cylinder. This amounts to 100 degrees or more in a high-speed engine, and/or in an engine the inlet ports of which extend only halfway around the cylinder. Then a certain exhaust lead is necessary which may be 30 degrees or more in a high-speed engine, and/or in an engine the exhaust ports of which extend only halfway around the cylinder.

In a symmetrically scavenged engine, 110-degree inlet duration and 35-degree exhaust lead mean that both the expansion and the compression stroke are 56 per cent of the full stroke. This amounts to sacrificing almost half the power output, in which case the attraction of the two-stroke cycle disappears.

10.5 Shortening of Ports.

In cases where the calculation results in a very short effective stroke, it is sometimes possible to obtain better performance by making the ports smaller than the values obtained by equations (7–1), (8–1), (8–2), (8–3), and (8–4). If the possibilities of widening and rounding the ports and boosting the scavenge pressure are exhausted, it may be necessary to compromise by reducing the inlet-port height or shortening the exhaust lead, or both.

In an unsymmetrically scavenged engine with a positive supercharging period, the effective compression stroke is lengthened by reducing the inlet-port height, which is desirable. On the other hand, the inlet-port area is reduced and with it the charge in the cylinder, which is undesirable. There is, of course, an optimum inlet-port height which gives maximum power. An analytical method for the determination of this optimum inlet-port height was described in 7.10. If the inlet-port height is reduced below the optimum, the power output diminishes.

By reducing the exhaust port height (or exhaust duration), the effective expansion stroke is lengthened in nonsymmetrically scavenged engines; and both the effective expansion stroke and the effective compression stroke are lengthened in symmetrically scavenged engines. By reducing the exhaust duration with unchanged inlet-port timing, the exhaust lead is reduced. A reduction of the exhaust lead may so interfere with the scavenging process that the power output is adversely affected.

In setting the exhaust lead, the aim usually is to make it such as to permit the cylinder pressure to drop to approximately scavenge pressure. A reduction of the exhaust lead somewhat **below** this point frequently results in an increase in power. The imperfect pressure release in the cylinder is sometimes more than compensated by the lengthening of the effective expansion stroke. Nevertheless such a reduction must be made cautiously. An exhaust lead which does not permit the cylinder pressure to drop to the scavenge pressure during the blowdown period causes a flow of exhaust gases into the inlet ports. This contaminates and heats up the inlet air and, for a while, prevents it from entering the cylinder. Exhaust gases are blown back into the cylinder, instead of clean cool air. Not only is the fresh air charge thereby reduced, but on account of the elevated temperature of the charge, piston overheating may result. Lubricating oil is blown into and clogs the intake ports. **In extreme cases exhaust blowback causes explosions in the air box or receivers.** If the exhaust lead permits a pressure drop to within a couple of pounds of the scavenge pressure, the exhaust blowback is of short duration and of small consequence. However, a considerable excess of cylinder pressure over scavenge pressure is normally injurious to the engine. In designing an engine, it may be found advisable to compromise on the blowdown time-area up to 30 per cent, provided there is no danger of piston overheating. In any case the compromise should be checked experimentally, as described later.

10.6 Speed and Load.

In engines which operate under variable speed or variable load or both, a compromise is frequently resorted to in order to accommodate the speed and load variations.

The preceding calculations were based on full load at maximum speed. Formulas used for porting calculations include engine speed. The necessary exhaust blowdown is dependent on the exhaust release pressure which in turn depends on the mean indicated pressure or load.

It is assumed first that the load is constant and the question is how a reduction of speed affects the porting. According to equation (7–1), the required inlet-time area is proportional to the engine speed divided by the mean inlet velocity

$$A_{im}\alpha_i = \tfrac{1}{2} L V_{disp} \frac{n}{w_i}.$$

If the scavenging blower is driven by the engine and its volumetric efficiency is assumed to be constant, as is approximately the case with positive-displacement Roots blowers and reciprocating piston blowers, the scavenge pressure varies approximately as the square of the engine speed, because on the per minute basis

$$\text{Blower delivery} = \text{constant} \times n$$

$$\text{Engine air consumption} = \text{constant} \times \sqrt{p_{sc}}$$

The two being equal

$$p_{sc} = \text{const} \times n^2.$$

On the other hand the delivery ratio is, from equation (7–1),

$$L = \frac{2A_{im}\alpha_i}{V_{disp}\dfrac{n}{w_i}} = \text{const}\,\frac{w_i}{n}.$$

The mean inlet velocity varies as the square root of the scavenge pressure or as the first power of the engine speed; therefore

$$L = \text{const}\,\frac{\sqrt{n^2}}{n} = \text{const}$$

or the engine receives the same amount of air every revolution irrespective of the engine speed, which is a gratifying result, indeed. However, this is true only if the scavenging blower is of constant volumetric efficiency, and if the effect of the exhaust system on the charging process is negligible.

10.7 Blowers.

Ordinarily the volumetric efficiency of Roots-type blowers increases while that of the reciprocating type decreases with speed. Accordingly, an engine with a Roots-type blower is somewhat short of air at slow speed, while one with a reciprocating blower has more than enough air at low speed. Since, however, the volumetric efficiency of the blower seldom changes more than 10 per cent in the operating range, and separately scavenged engines are designed with a delivery ratio between 1.3 and 1.4, the effect of speed change on the filling of the cylinder is of little consequence.

In the case of centrifugal blowers, volumetric efficiency has no definite meaning, nevertheless the air delivery varies directly in proportion to the rotor speed. Therefore an engine driven centrifugal blower has no ill effect upon the charging of the engine at lower than normal speeds. If the blower is electric motor driven, the motor speed should vary as the engine speed does, otherwise the engine receives too much air at low speed, which is a waste of power at best, and may cause overcooling and misfiring at worst.

Reduced speed shortens the required exhaust lead that is necessary to allow the cylinder pressure blowdown to the level of the scavenge pressure. No harm results from a too-long exhaust lead except that a shorter lead would permit a longer inlet period and higher power output. Therefore engines which run at less than maximum speed most of the time are given a shorter exhaust lead than the maximum speed would require. This results in some back-blow into the inlet ports at high speed.

The effect of speed manifests itself mostly during the supercharge period. In a well-designed engine, the supercharge period (from exhaust closure to inlet closure) is just long enough to permit equalization between scavenge pressure and cylinder pressure. During that period the cylinder pressure rises, both because of compression and because of filling up with delivered air. During the same time the scavenge pressure drops, although with a large receiver volume the drop is insignificant. If at the point of inlet-closure the cylinder pressure just equals the scavenge pressure, ideal supercharging conditions exist. If the supercharging period is too long the pressure in the cylinder rises so high that before the inlet ports close, the flow is reversed; air flows from the cylinder to the air box. This is likely to happen at low speed when filling the cylinder up to scavenge pressure requires fewer crank degrees than at high speed. After the cylinder pressure has reached scavenge pressure, the compression of the piston tends to raise it still further and if the inlet ports are still open, air flows out of the cylinder. This represents a waste of air (power) on one hand, and is likely to cause lubricating oil deposits in the inlet ports on the other hand, both of which are undesirable. Therefore for variable speed engines a compromise is frequently used in employing a shorter supercharge period than the rated speed would call for. The supercharge period is sometimes completely omitted.

In reducing the phase angle difference between the exhaust piston (or cam) and inlet piston (without changing the lifts and time-areas) both the exhaust blowdown and the supercharge periods are shortened. This is the expedient sometimes employed in variable-speed engines.

10.8 Effect of Load.

Change of load has no direct effect of charging. The indirect effect is such that in a well-designed engine the charging improves somewhat at high load and is poorest under motoring and at resonant speeds. This is discussed in later chapters. The difference in the charge between full load and motoring may be of the order of 5 per cent, and between full load and part load much less. Therefore, the effect of the load on charging is of no consequence.

Reduction of load affects the exhaust blowdown favorably. Equation (8–2) with Fig. 8–3 indicates that the required blowdown area decreases if the ratio of exhaust release pressure to inlet pressure decreases. Figure 8–2 shows that the exhaust release or expansion end-pressure increases with the load. Therefore, if at full load the exhaust blowdown period is adequate, at part load it is more than ample.

10.9 Propeller Drive.

In many applications, like propeller drive, speed and load change together. Both reduced speed and reduced load permit reduction of the exhaust lead, the requirements for which at full load and speed are hard to satisfy in any event. This is the reason that a compromise by cutting the exhaust lead is so frequently used. Such a compromise is tolerable or even desirable in engines that run at part load and reduced speed most of the time, but it should be avoided in engines designed for constant speed operation with relatively small load variations as is the case in power generation.

10.10 Example 1. Opposed-Piston Engine.

For the 8 by (10 + 10) inch, 720 rpm, opposed-piston engine described and port-calculated in 7.12 and 8.5, the calculation gave the following results:

Exhaust opens 117½ degrees after top center (exhaust piston)
Exhaust closes 242½ degrees after top center
Exhaust duration, 125 degrees
Inlet opens 141 degrees after top center (exhaust piston)
Inlet closes 243 degrees after top center
Inlet duration, 102 degrees

It may be objected to that this port timing gives an extremely short (½ degree) supercharging period.

To overcome this difficulty, first the scavenge pressure was raised to 3.4 psi which requires only 100 degrees duration (see Fig. 7–9). As a result, the blowdown pressure ratio was reduced to $50/(14.5 + 3.4) = 2.79$. It can be seen from Fig. 8–3 that for $k = 1.3$, this gives $y = 0.42$. The required blowdown time-area is thereby reduced to 213 square inch-degrees.

In order to further reduce the exhaust lead, this blowdown time-area was reduced by 30 per cent, to 149 square inch-degrees, and the corresponding per cent-degree area to

$$F_a = \frac{149}{12.7 \times 10} = 1.174 \times 100 \text{ per cent deg.}$$

Figure 8–1 then gives an exhaust lead of 20 degrees. As a result, the compromise porting is as follows:

Exhaust opens 122 degrees after top center (exhaust piston)
Exhaust closes 238 degrees after top center
Exhaust duration, 116 degrees
Inlet opens 142 degrees after top center (exhaust piston)
Inlet closes 242 degrees after top center
Inlet duration, 100 degrees

This porting is shown in Fig. 8–5 and 9–8.

10.11 Example 2. Crankcase-Scavenged Engine.

The 10½ by 12 inch 425 rpm crankcase-scavenged engine shown in Fig. 9–1 was calculated to require a blowdown time-area of 88.3 square inch-degrees (section 8.8), which corresponds to an exhaust lead of 18.5 degrees. In order to shorten the exhaust lead, the blowdown time-area is reduced by 30 per cent to 61.8 square inch-degrees. This gives a per cent-degree area of

$$F_a = \frac{61.8}{12 \times 0.21 \times 10.5 \times \pi} = 0.74 \times 100 \text{ per cent deg.}$$

From Fig. 10–1 it is evident that an inlet duration of 96 degrees corresponds to an exhaust lead of 15 degrees.

The exhaust period is $96 + 2 \times 15 = 126$ degrees, which permits an exhaust-port height of 22.9 per cent stroke or 2.75 inches. This is the height of the exhaust port of the production engine.

CHAPTER 11

INTAKE SYSTEM

11.1 In the calculations presented in the previous chapters the roles of the blower, piping, and receivers were ignored. It was assumed that air of constant scavenge pressure is always present at the inlet ports and that atmospheric pressure exists outside the exhaust ports. It is in order now to direct attention to the intake and exhaust systems.

The intake system consists of a blower, a receiver, intake pipes, and ducts; an air filter, silencer, and cooler are frequently included. Unless all these items are designed properly the engine performance suffers. Furthermore, because of the interrelation of all parts, their selection influences the porting calculations.

BLOWERS

11.2 With the exception of some engines built using the Kadenacy system (Chapter 13), all two-stroke cycle engines have blowers. Crankcase-scavenged engines have no separate blowers but the underside of the piston reciprocating in the closed crankcase acts as a blower. Therefore, the crankcase-scavenged engine includes a single acting reciprocating blower possessing a rather large (ordinarily 300 to 400 per cent) clearance volume and its volumetric efficiency is correspondingly low.

If in order to furnish more air, the compressor side of the piston has an enlarged area, the engine is called a *step-piston* engine. The step piston may open into the crankcase but more often it is separated from it by a wall or diaphragm in order to reduce the clearance volume. Then in order to be able to seal the lower side of the compressor cylinder a crosshead must be employed. The reciprocating blower is sometimes located at an angle to the power cylinders, receiving the drive from the main crankshaft through separate connecting rods, or in line with the power cylinders, similarly driven. Rotary types of blowers are usually geared to the crankshaft.

The blower is required to furnish an adequate amount of air to the engine under all operating conditions without using up a disproportionate amount of power for the purpose. From the performance standpoint, its important characteristics are volumetric efficiency, adiabatic efficiency, and temperature rise of the air across the blower.

11.3 Volumetric Efficiency.

The higher its volumetric efficiency, the smaller may the blower be to deliver the required amount of air. No less important is the condition that the volumetric efficiency curve shall be flat, meaning that the delivery shall be little affected by variations in speed and back pressure, so that the engine shall receive about the same amount of air per cycle.

11.4 Adiabatic Efficiency.

The mechanical efficiency of the blower is usually expressed in terms of adiabatic efficiency, which means a ratio

$$\eta_{ad} = \frac{\text{work of adiabatic compression}}{\text{actual work of compression}}$$

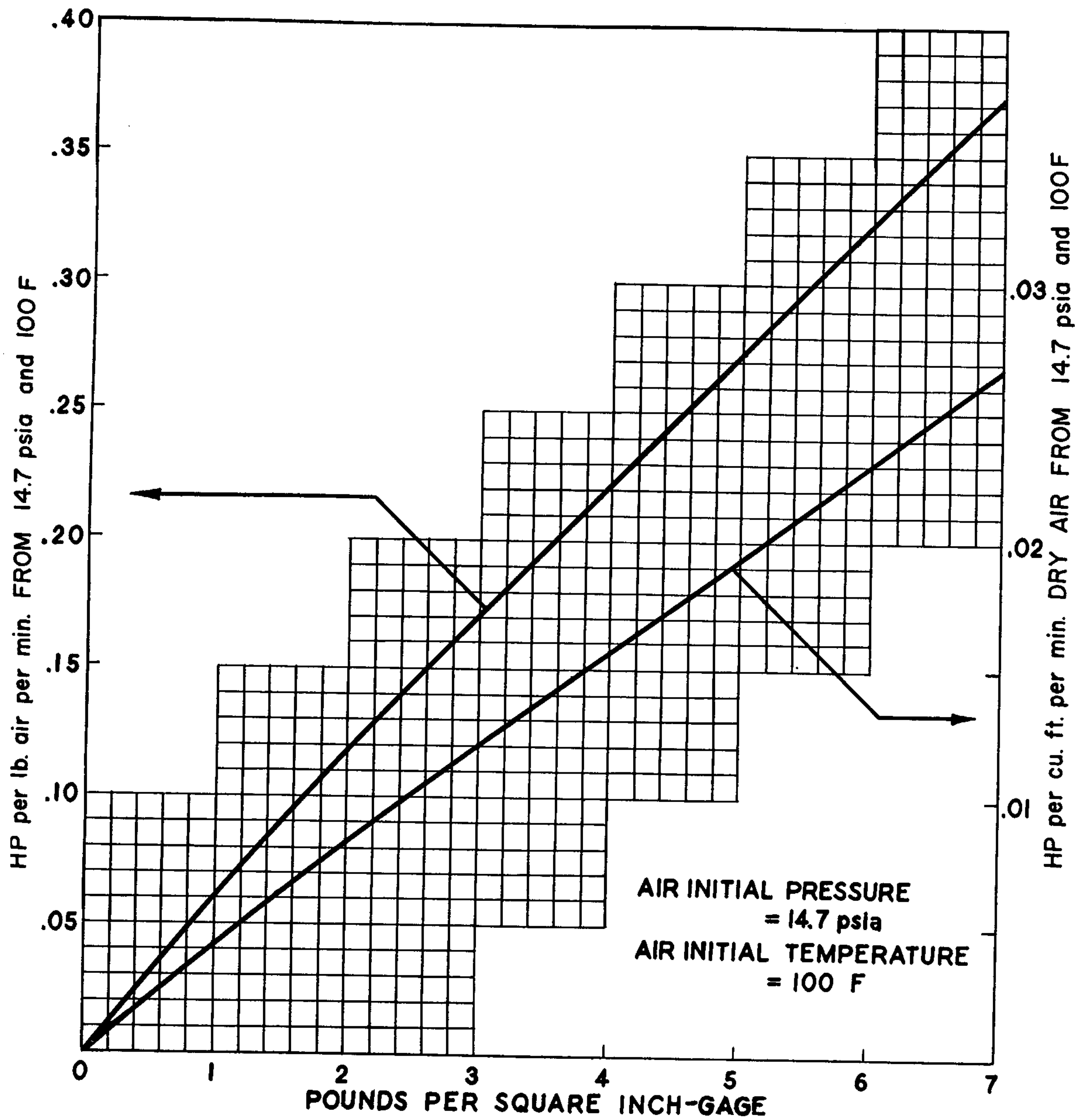

Fig. 11-1. Theoretical Horsepower Requirements for Adiabatic Compression.

The power required to compress 1 pound (and 1 cubic foot) of air per minute is shown in Fig. 11-1 which has been plotted for dry air of 14.7 psia pressure and 100 F temperature from thermodynamic tables [Keenan and Kaye, 1945]. In principle, the actual work of compression may even be less than the work of adiabatic compression because the compression may be close to isothermal, which consumes considerably less work. In practice adiabatic efficiency is used almost exclusively for expressing the over-all mechanical efficiency of the blower, and its value varies according to type and operating conditions usually between 50 and 80 per cent, but is occasionally lower.

Low adiabatic efficiency is undesirable in a blower for two reasons. First, because the power required to drive the unit reduces the engine output. Second, because the air delivered has a higher temperature (friction work converts into heat) and a correspondingly lower density, which reduces the cylinder charge and the maximum amount of fuel that can be burned in the cylinder. High intake temperature is also undesirable because it aggravates the cooling of cylinder and piston and may cause lubrication troubles.

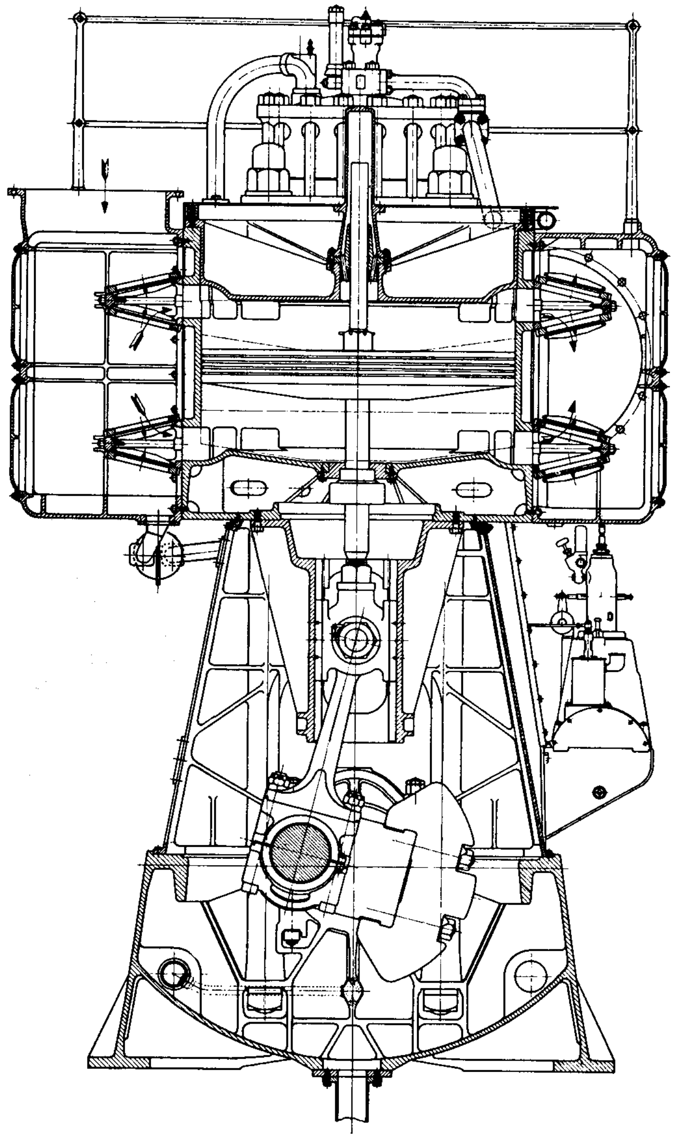

Fig. 11-2. Large Reciprocating Blower. (By permission of Nordberg Manufacturing Company.)

11.5 Reciprocating Blowers.

Three types of blowers are used for two-stroke cycle engines: the reciprocating blower, the positive-displacement rotary blower, and the centrifugal blower.

Reciprocating piston blowers are frequently used on slow-speed engines, but rarely on high-speed engines. They have good efficiency and desirable delivery characteristics, but occupy considerable space. Figure 11–2 shows a large reciprocating blower used on Nordberg engines.

If each engine cylinder has its own reciprocating blower its delivery can be so timed in relation to the power piston as to make charging take place under optimum conditions. The best that is obtainable from rotary type compressors is a constant scavenge pressure, by using a very large receiver volume close to the inlet ports. Ordinarily when the inlet ports open, the scavenge pressure drops, and starts to rise again when the exhaust closes. The scavenge pressure usually reaches its lowest value at the end of the inlet period when supercharging is to take place. This is unfavorable because the trapped charge adjusts itself to the low pressure occurring at inlet closure. When individual reciprocating blowers are used it is possible so to phase the motion of the pump and power piston that the scavenge pressure is low during the greater part of the cycle, fairly high when the inlet ports open, after that taking a dip due to the transfer of air to the power cylinder, but soon making it up and reaching its maximum when the inlet ports are about to close, insuring maximum trapped charge to the

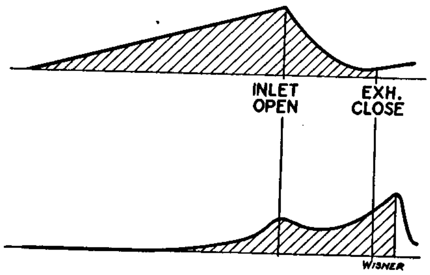

Fig. 11-3. Evolution of the Scavenge Pressure. Top: Rotary blower. Bottom: Reciprocating blower. (R. Wisner, *High Speed Two-Stroke Compression-Ignition Engine*. The Institute of Automobile Engineers, June, 1944.)

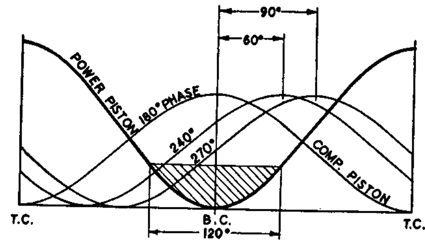

Fig. 11-4. Motion of Power Piston and Compressor Piston. Three phase angles are shown, 180°, 240°, and 270° between the power and compressor pistons.

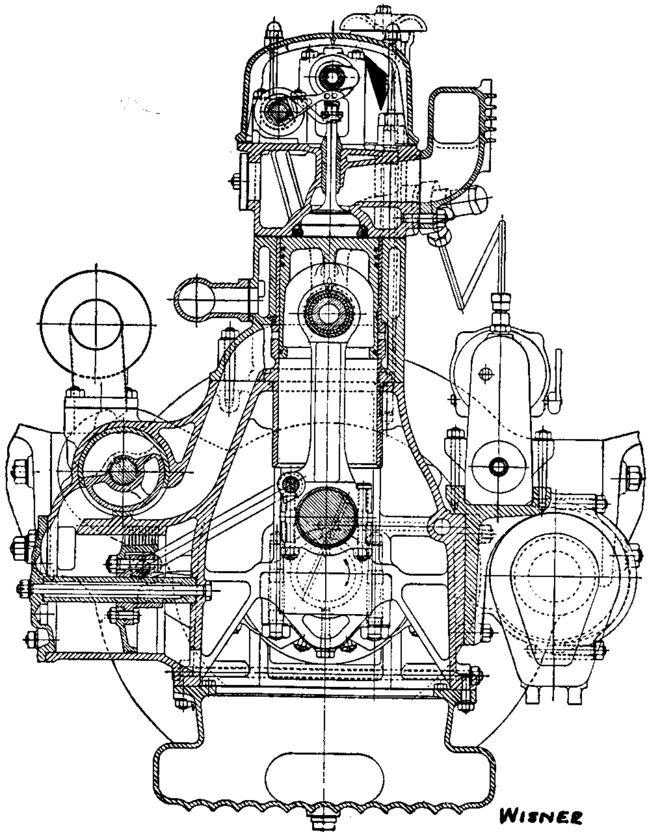

Fig. 11-5. Ricardo's 63/3 Experimental Engine. (R. Wisner, *High Speed Two-Stroke Compression-Ignition Engine*. The Institute of Automobile Engineers, June, 1944.)

cylinder. Figure 11–3 illustrates the difference between a rotary blower and a well-phased reciprocating blower.

The phasing of the reciprocating compressor is important. Figure 11–4 shows the motion of the power piston and of a compressor piston with a phase angle of 180, 240 and 270 degrees between them. Assuming a single-acting compressor and an inlet period of 120 degrees, the piston motion diagram shows that the compressor piston reaches its top dead center position at bottom dead center of the power piston or at 60 or 90 degrees after that, respectively. The engine shown in Fig. 2–2 represents 180 degrees, that of Fig. 2–3 approximately 240 degrees, and that shown in Fig. 11–5, 270 degrees phase advance. With 240 degrees phase advance maximum compressor pressure is shown to occur exactly when the inlet ports of the diesel cylinder close, which is ideal.

11.6 Volumetric Efficiency.

The volumetric efficiency of a reciprocating air compressor is determined primarily by its clearance volume and its discharge pressure. Intake surges, valve resistance, and temperature effects caused by the valves are ignored at present. Referring to Fig. 11–6, from the polytropic expansion line

$$p_i(V_{disp} + CL - V)^k = p_d CL^k, \tag{11-1}$$

where p_i and p_d are absolute inlet and discharge pressure,

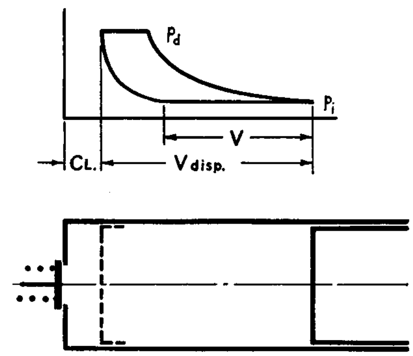

Fig. 11-6. Reciprocating Compressor.

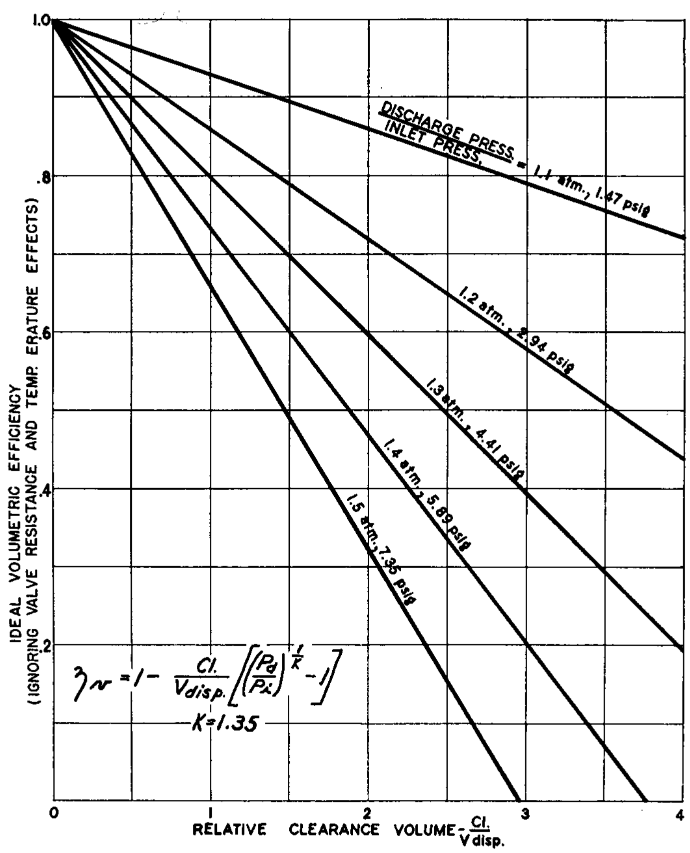

Fig. 11-7. Volumetric Efficiency of Reciprocating Air Compressor vs. Clearance Volume. Deduct 5–10% slippage from ideal volumetric efficiency.

V the volume of air at inlet conditions, CL the clearance volume and k the polytropic exponent. From (11–1)

$$\eta_v = \frac{V}{V_{disp}} = 1 - \frac{CL}{V_{disp}}\left[\left(\frac{p_d}{p_i}\right)^{\frac{1}{k}} - 1\right]. \tag{11–2}$$

Figure 11–7 shows volumetric efficiency and relative clearance volume for various discharge pressures with 14.7 psig inlet pressure and $k = 1.35$. The effect of the clearance volume upon the volumetric efficiency at elevated discharge pressures is very great.

Volumetric efficiency is also greatly affected by surges in the intake pipe which in turn depends on the length of the pipe or its *tuning*. A single-cylinder crankcase-scavenged engine acting as an air pump was shown [List, 1932] to vary its volumetric efficiency from 70 to 130 per cent depending on the length of the intake pipe. Voiszel [Voiszel, 1911], List [List, 1932], and Dennison [Dennison, 1932] developed analytical methods for the selection of favorable intake pipe lengths.

In multicylinder reciprocating blowers the effect of inlet-pipe tuning is less pronounced [Kluesener, 1932] unless individual intake pipes are used. Ordinarily the optimum pipe lengths are longer than are convenient. Even a limited tuning may, however, improve the performance of the engine. Such limited tuning can best be effected experimentally by the method described in section 16.26.

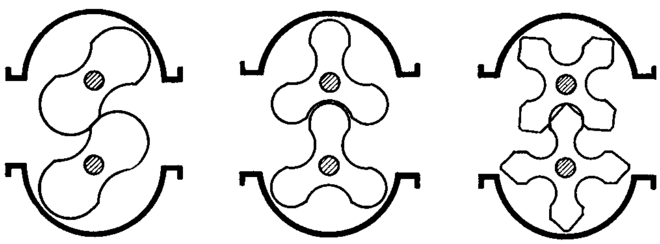

Fig. 11-8. Roots Blowers; Two-Lobe, Three-Lobe and Four-Lobe.

11.7 Roots Blower.

The most popular type of rotary blower is the Roots blower. It is made with 2, 3, and 4 lobes as shown in Fig. 11–8. A typical 3-lobe Roots blower is shown in Fig. 11–9.

Roots blowers are of the positive-displacement type and transport the air from the inlet side to the discharge side

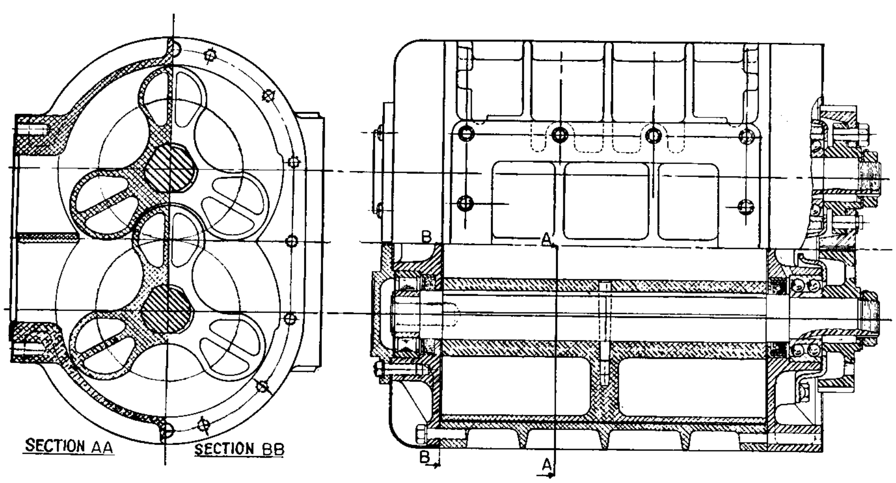

Fig. 11-9. Typical Three-Lobe Blower, Made by B–W Superchargers, Inc. (By permission of B–W Superchargers, Inc.)

of the blower without compression, substantially at intake pressure. The compression takes place suddenly when the lobe space opens into the discharge pressure and the previously compressed air flows back into it.

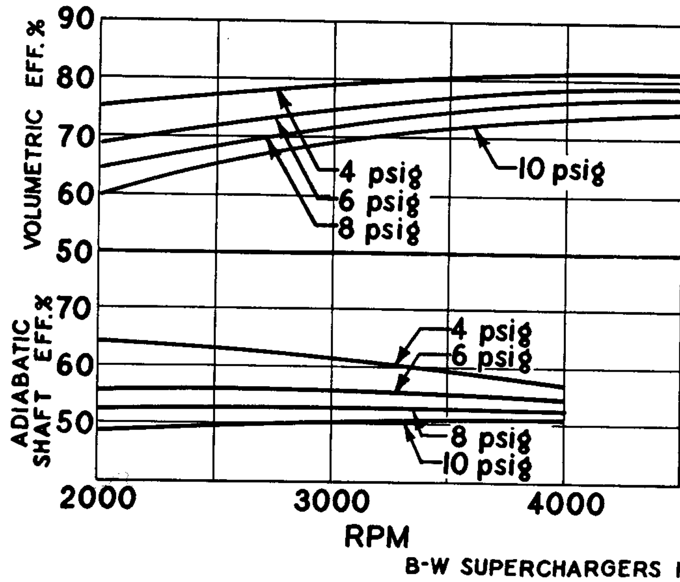

Fig. 11-10. Effect of Speed and Discharge Pressure on the Performance of a Roots-Type Blower. (By permission of B–W Superchargers, Inc.)

Figure 11–10 shows the volumetric efficiency and the adiabatic efficiency of a typical Roots-type blower. A blower without slip and clearance would always deliver the theoretical amount irrespective of speed or back pressure. Actually, the volumetric efficiency increases with the speed as the slip (leakage) decreases, and decreases with the back pressure as both slip and back delivery through the clearance volume increase.

11.8 Prediction of Performance.

The prediction of the performance of a Roots blower from design data is not easy. An attempt to analyze the power and delivery losses of positive displacement blowers was made by Pigott [Pigott, 1947] who obtained fair agreements between experimental results obtained by National Advisory Committee for Aeronautics [Ware and Wilson, 1928] and his theoretically derived but prohibitively complicated formulas. The blower designer may profitably go into a thorough theoretical analysis but the engine designer would prefer experimental data on the blower he is to use. Unfortunately complete delivery and horsepower data on Roots blowers are frequently either unavailable or unreliable. They may be available for one impeller speed or for one discharge pressure but not for various combinations of impeller speeds and discharge pressures. Since in porting calculations such information is desirable, the designer frequently must take recourse to interpolations or extrapolations from available data.

If delivery and horsepower input data are available for one speed but various discharge pressures, it is safe to set the air delivery at another impeller speed proportional to the speed. There is also a straight-line relationship between the theoretical horsepower input and the impeller speed if the discharge pressure (or rather pressure difference between intake and discharge pressure) is constant. The friction horsepower on the other hand increases as the square of the impeller speed. The brake or shaft horsepower, which is assumed to be the total of the adiabatic and friction horsepower, can then be computed.

Extrapolations to other than tested blower discharge pressures are more difficult. For low discharge pressures the input horsepower may be set proportional to the pressure difference (discharge–intake). The change in delivery or the change in *slip* for a given change in pressure difference is hard to predict with any accuracy. For anything but rough calculations it is better to depend on experimental data.

11.9 Vane-Type Blower.

Another type of positive displacement blowers are the vane types, shown in Fig. 11–11 and 11–12. In the single-vane type a cylindrical eccentric rotor and a reciprocating vane separate the suction space from the discharge space. The displacement volume is approximately the difference between the housing cylinder and the rotor cylinder. This volume is delivered once every revolution into the air receiver. The vane is either automatic spring-loaded or mechanically actuated. The delivery and discharge pressure fluctuates similarly to that of a reciprocating pump. Therefore if used individually, one for each cylinder, it could be phased to obtain an effect similar to that described under the heading of reciprocating blowers. Since the volumes on the suction and discharge side increase and decrease alternately, it is necessary to employ nonreturn valves at least on the discharge side in order to avoid backflow of compressed air into the blower.

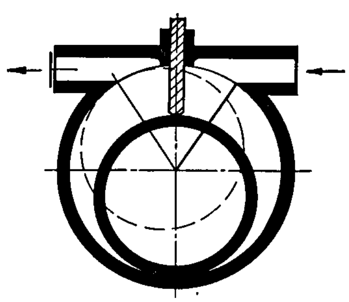

Fig. 11-11. Single Vane Blower.

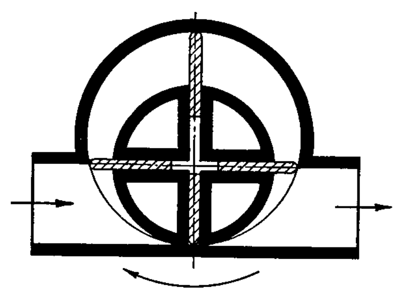

Fig. 11-12. Multivane Blower.

In the multivane-type blowers which are built with 2, 3, 4 and more vanes, the compression that the air undergoes while it is trapped between the vanes is insignificant, and the device operates more like a Roots blower with a fairly uniform delivery and a backflow of compressed air upon uncovering the vane space. The vanes frequently seal by action of centrifugal force.

The volumetric efficiency of vane-type blowers with good seal varies between 75 and 95 per cent, and their adiabatic efficiency between 35 and 65 per cent. Lubrication difficulties in conjunction with maintaining good seal prevented their more general acceptance.

11.10 Screw-Type Blowers.

More recent developments of lobe-type blowers promise to eliminate some disadvantages of the Roots blowers without sacrificing much of their simplicity and favorable operating characteristics. One of these, the Elliott-Lysholm blower [Lysholm et al., 1943] is shown in cross section in Fig. 11–13 and a pair of rotors of another, the Whitfield blower [Whitfield, 1943] is shown in Fig. 11–14. The main improvement of these screw-type machines over the Roots blower is in the rotor design. Not only do these rotors carry the air from inlet to discharge, but they effect simultaneously an

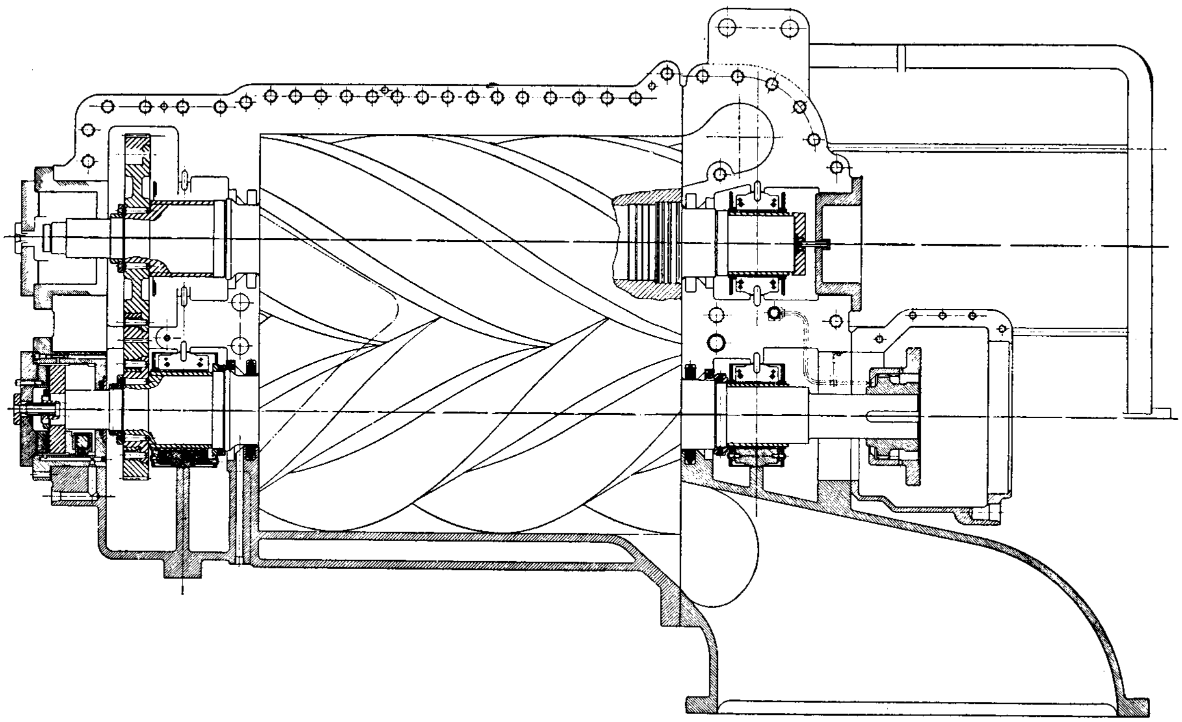

Fig. 11-13. Elliott-Lysholm Blower. (By permission of the Elliott Company from W. A. Wilson and J. Crocker, "Fundamentals of the Elliott-Lysholm Compressor," *ASME Paper 45-A-45*, 1945.)

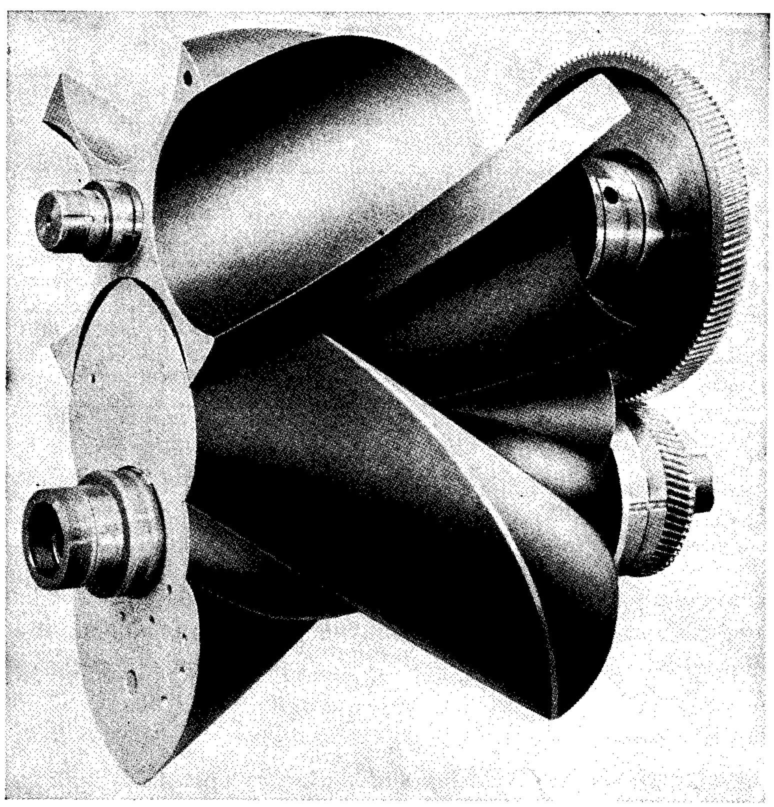

Fig. 11-14. Whitfield Blower with the Rotors in Mesh and Housing Removed. (By permission of J. E. Whitfield Company.)

adiabatic compression. As described by Lysholm, "A charge is initially enclosed in the space bounded by the tooth flanks, casing bore and end walls. The rotor helices are so chosen that a particular thread space is completely filled and sealed off from the suction just as it is entered at its opposite end by a coacting lobe on the other rotor. Further rotation establishes an axial seal, which separates this charge from the charge in the succeeding grooves. As rotation proceeds, this seal moves axially affecting a reduction in volume of the charge and a substantially adiabatic compression. When the leading lobes of the grooves pass the boundaries of the discharge port, the compressed medium is brought into communication with the discharge. The location of the port thus determines the 'built-in' compression ratio." The blower can operate, however, at a pressure ratio different from that built-in, but its efficiency is somewhat lower. If the compression is adiabatic throughout at a pressure ratio of 1.5, the compressor work is 12 per cent less that of the constant volume compression which takes place in a Roots-type blower. This means that instead of a possible 60 per cent efficiency, 67 per cent can be obtained. If, however, the built-in ratio is other than the pressure ratio at which the blower operates, its efficiency is somewhere in between.

Figure 11–15 shows [Wilson and Crocker, 1945] the performance curves of an Elliott-Lysholm blower of 10,000 cfm capacity with a built-in pressure ratio of 1.85. It is evident that under favorable operating conditions the adiabatic efficiency reaches 82 per cent and the volumetric efficiency is better than 90 per cent. With Whitfield blowers of smaller size almost as good results have been obtained.

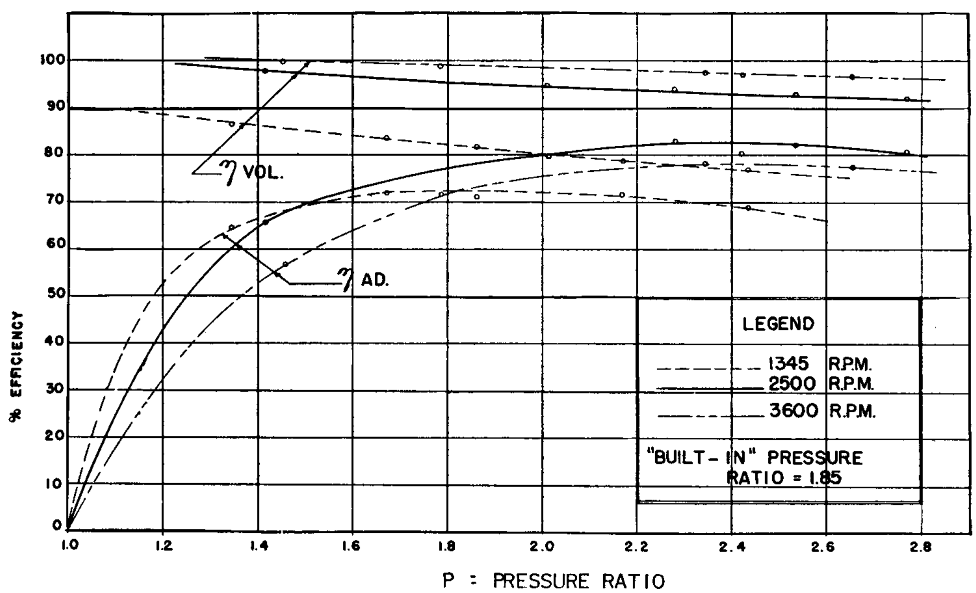

Fig. 11-15. Performance Curves of a 10,000 cfm Elliott-Lysholm Blower. (By permission of the Elliott Company from W. A. Wilson and J. Crocker, "Fundamentals of the Elliott-Lysholm Compressor," *ASME Paper 45-A-45*, 1945.)

Another advantage of screw-type blowers is that they eliminate the sudden backflow of the previously compressed air into the lobe space, which is believed responsible for much of the noise of the Roots-type machines.

11.11 Centrifugal Blowers.

Centrifugal blowers are light and compact because they operate at high speeds and without rubbing surfaces. Normal speeds vary from 3000 rpm to 20,000 rpm. The impellers (rotors) are open, closed, or semiclosed; the vanes are radial, straight, angular, or curved. The diffusers in the casing are one piece volute or multivane-type. Figure 11–16 shows the cross section of a single-stage centrifugal blower with gearing.

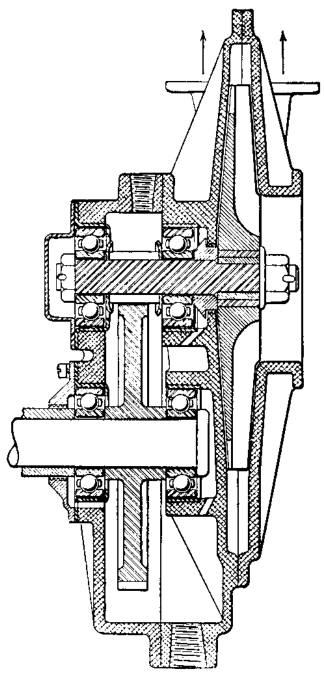

Fig. 11-16. Section through Single-Stage Centrifugal Blower.

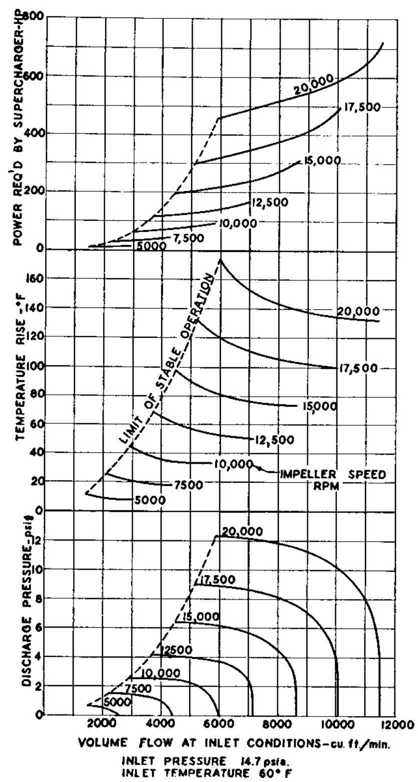

Fig. 11-17. Characteristic Curves of a Centrifugal Blower.

Single-stage centrifugal blowers are well suited for 2 to 6 psig pressure head, which may be required for scavenging two-stroke cycle diesel engines. Figure 11–17 shows the characteristic curves

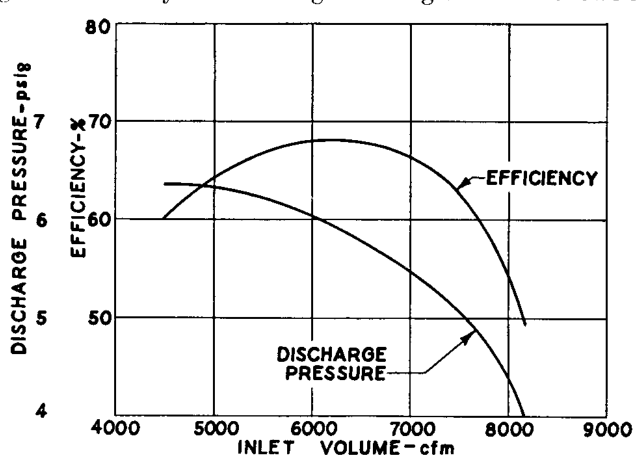

Fig. 11-18. Adiabatic Efficiency of Centrifugal Blower.

of a centrifugal blower designed for scavenging a 3000 hp high-speed two-stroke cycle diesel engine. The adiabatic efficiencies of the same blower are shown in Fig. 11–18.

11.12 Axial Blowers.

Axial blowers have characteristics somewhat similar and perhaps superior to those of centrifugal blowers but they have not been used extensively for charging two-stroke cycle diesel engines.

11.13 Motor Drive.

A centrifugal blower may have an independent electric motor drive or may be geared to the engine. Electric motor drives are frequently used for blowers supplying air to large diesel engines of ships or electric power plants. Variable speed motors are sometimes employed so that the blower speed may be adjusted to the engine speed. If the engine is to operate at a constant speed as is the case in electric power plants, constant speed motors for the blowers are adequate. Constant speed motors are, however, also used with variable-speed engines, as in ships. Sometimes two motor-driven blowers furnish the air to one engine, and when the engine runs slowly as in maneuvering one blower is cut out and the engine is run on one blower only. Sometimes, however, the engine is slowed down without any adjustment of the air supply so that it receives an oversupply of air; this is not a good practice because it not only wastes power but it overcools the engine, which may cause incomplete combustion, exhaust odor, and in extreme cases misfiring. An automatic control which changes the blower speed approximately in proportion to the engine speed is more desirable.

Internal combustion engines, gasoline or diesel, could be advantageously used for driving centrifugal blowers for large diesel engines, by having a simple control that would regulate the speed of the blower engine according to the speed of the main engine.

Centrifugal scavenging blowers are occasionally geared to the engine or driven by an exhaust turbine. The exhaust turboblower in its present form is unsuitable for two-stroke cycle engines because at starting there is no exhaust gas available to drive the blower, and therefore the engine would not start. Geared centrifugal blowers are also popular only with four-stroke cycle engines, but are seldom used on two-stroke cycle engines.

11.14 Geared Centrifugal.

It is a fairly widely held belief that geared centrifugal blowers are unsuitable for two-stroke cycle engines because at low speed they furnish an insufficient amount of air to the engine. This belief has no solid foundation in fact. The delivery of a centrifugal blower is approximately proportional to the impeller speed and the discharge pressure to the square of the impeller speed. This is just what the engine requires. In order to maintain a fixed delivery ratio (air delivered to the engine divided by piston displacement), delivery should be proportional to engine speed. In order to push the same amount of air through the engine, which behaves in this respect as a fixed orifice (as explained in Chapter 7), during each cycle, the scavenge pressure must increase approximately as the square of the engine speed. If the centrifugal blower is geared to the engine, it fulfills both requirements.

Figure 11–19 shows a diagram derived from Fig. 11–17. If that blower is connected to the two-stroke cycle engine for which it was designed, the volume flow varies with engine speed in accordance with the line through the circles marked "engine receives." In order to obtain a fixed delivery ratio of 1.4 the volume flow would have to vary as the dash-dot line. The two lines are close together except below 400 rpm which is less than the lowest operating speed of the engine.

11.15 Instability Region.

However, a particular feature of centrifugal blowers must always be given proper attention.

They all have a region of instability. If the flow is reduced to about one-half of what is maximum at that particular impeller speed, the operation becomes unstable. The reason is that a given impeller speed can produce only a given pressure head and if the counter pressure is more than that, the flow becomes reversed. The reverse flow lasts until the discharge pressure drops sufficiently for the blower to begin delivery again. Therefore, delivery is intermittent, accompanied by a loud shrieking noise. This happens always when the air delivery (due to too much throttling by clogged ports for instance) is less than a certain minimum. This minimum, the limit of stable operation, changes somewhat with the blade design, but it is around one-half of normal or maximum delivery.

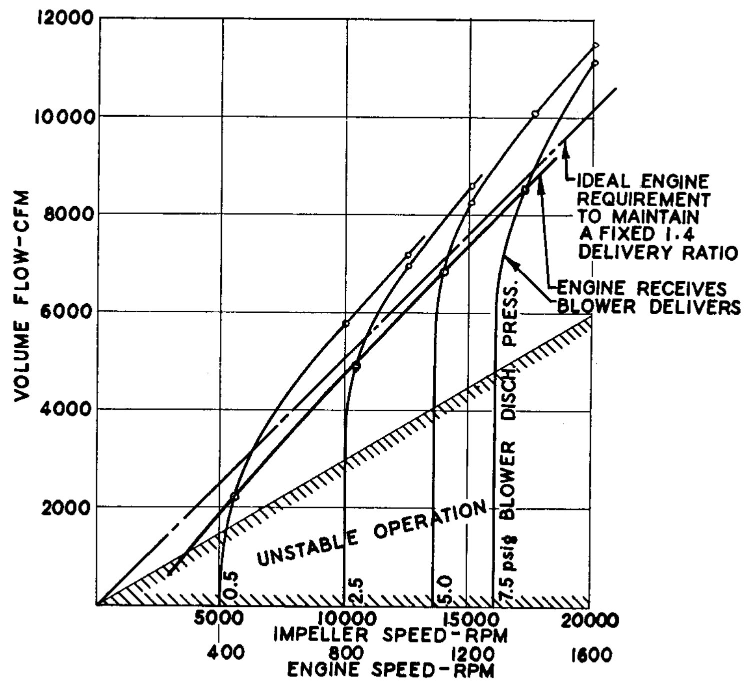

Fig. 11-19. Co-action of Engine and Centrifugal Blower.

The regions of unstable operation are marked on the graphs of Fig. 11–17. In practical operation those regions must be avoided. Referring to Fig. 11–19, the only time that unstable operation would be encountered is at about 250 engine rpm, which is less than the lowest operating or cranking speed of the engine.

Properly designed centrifugal blowers are found suitable for scavenging either slow- or high-speed two-stroke cycle engines. If a rigid gear drive is considered undesirable because of the high impeller speeds which represent a considerable inertia force, a hydraulic drive or an overrunning clutch, or simply a vee belt may be interposed between the engine crankshaft and the blower shaft, so that the momentum of the impeller wheel is unable to do any damage.

11.16 Power Requirements of Blowers.

Irrespective of the type of blower used, the engine porting is influenced by the power absorbed and the temperature rise of the air in the blower.

The power absorbed may be calculated on the basis of known or estimated adiabatic efficiency. It is most convenient to express the power absorbed by the blower in terms of engine mep and to designate it as *blower mep*. If the adiabatic efficiency of the blower is η_b, the blower mep is *approximately*

$$\text{(11-1)} \qquad \text{mep}_b = \frac{Lp_{sc}}{\eta_b}$$

Precise calculation has shown that the blower mep is somewhat lower than indicated by the simplified equation (11–1). The exact equation and a comparison between the approximate and the exact equation are shown in Fig. 11–20. In using the simplified equation, the error is on the side of safety

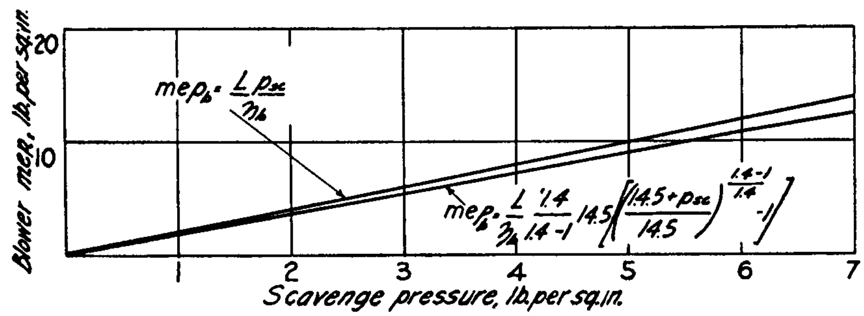

Fig. 11-20. Blower mep for a Delivery Ratio of $L = 1.4$ and a Blown Efficiency of $\eta_b = 0.7$. Lower curve is based on formula for adiabatic compression, upper curve on simplified formula.

because the power absorbed by the blower is somewhat overestimated. If η_b is taken as 0.7, the efficiency of the blower is probably also overestimated, which compensates for using the approximate equation. Finally if the delivery ratio is taken to be 1.4, as is customary,

$$\text{mep}_b = \frac{1.4p_{sc}}{0.7} = 2p_{sc}$$

which is a very simple rule indeed.

11.17 Temperature Rise.

The temperature rise of the air in the blower can also be calculated fairly well if the efficiency of the blower is known. The temperature rise is composed of two parts, the first of which is the heat produced by the compression, and the second of which comes from heat produced by friction.

The heat of compression may be calculated from the adiabatic change by the equation

$$\text{(11-2)} \qquad \frac{T_d}{T_a} = \left(\frac{p_d}{p_a}\right)^{\frac{k-1}{k}}$$

Taking for ambient temperature 460 + 70 F and ambient pressure 14.5 psia, the temperature rise $t_{comp} = T_d - T_a$, is listed in Table 11–I.

Table 11-I. Temperature Rise in Blower Due to Adiabatic Compression.

Discharge pressure p_d (psig)	1	2	3	4	5	6
Temperature rise t_{comp} (F)	10	19.6	29	39	46.4	54.6

The temperature rise originating from friction in the blower can be calculated as

$$\Delta t_{fr} = \frac{Q}{\dfrac{V_b}{v} c_p} \tag{11–3}$$

where Q is the heat quantity produced by friction absorbed by the air, V_b and v are the delivery in cubic feet per second and specific volume of the blower air at the suction side respectively, and c_p is its specific heat. Heat lost by radiation and conduction is disregarded.

The power output corresponding to Q is

$$N = \frac{Q}{550} 778 = 1.41Q,$$

if N is expressed in horsepower and Q in Btu.

The mechanical efficiency of the blower is (approximately)

$$\eta_m = \frac{\text{Useful work}}{\text{Useful work} + \text{friction loss}} = \frac{V_b(p_d - p_a)144/550}{V_b(p_d - p_a)144/550 + \dfrac{V_b}{v} \times \dfrac{c_p \Delta t_{fr}}{550} \times 778}$$

where p_d and p_a are, respectively, discharge pressure and atmospheric pressure in psia, and V_b is the blower delivery in cubic feet per second. The specific volume of the atmospheric air (14.5 psia and 70 F) is 13.8 cubic feet per pound, and its specific heat is 0.243; therefore,

$$\eta_m = \frac{1}{1 + \dfrac{778}{144} \times \dfrac{0.243 \Delta t_{fr}}{13.8(p_d - p_a)}} = \frac{1}{1 + 0.095 \times \dfrac{\Delta t_{fr}}{(p_d - p_a)}}.$$

From this

$$\Delta t_{fr} = \left(\frac{1}{\eta_m} - 1\right) \frac{p_d - p_a}{0.095} \tag{11–4}$$

The temperature rise due to mechanical friction for various discharge pressures (psig) is given in Table 11–II.

Table 11-II. Temperature Rise of Air in Blower Due to Mechanical Friction.

η_m (%) p_d (psig)	1	2	3	4	5	6
40	15.8	31.6	47.5	63.3	79	95
50	10.5	21	31.5	42	52.5	63
60	7	14	21	28	35	42
70	4.5	9	13.5	18	22.5	27
80	2.63	5.26	7.9	10.5	13.1	15.8

The total temperature rise may be taken as the total of the compression temperature rise and friction temperature rise and is shown in Fig. 11–21. It is seen that with low blower efficiency and high scavenge pressure the temperature rise becomes considerable. For instance, with 6 psig blower pressure and 60 per cent blower efficiency, the total temperature rise of the intake air would be 96.6 F, which would entail an appreciable power loss.

If it is desired to estimate the effect of blower efficiency upon engine performance, the per cent loss in power output due to the power absorption of the blower and due to the loss in charge density which results from the temperature rise of the air across the blower must be known. Figure 11–22 shows the per cent power loss for 4 psig and 6 psig scavenge pressure. It is evident that low blower efficiency increases the power loss materially and that the loss is much greater with 6 psig scavenge pressure than with 4 psig scavenge pressure. It amounts to approximately 24 per cent with 60 per cent blower efficiency.

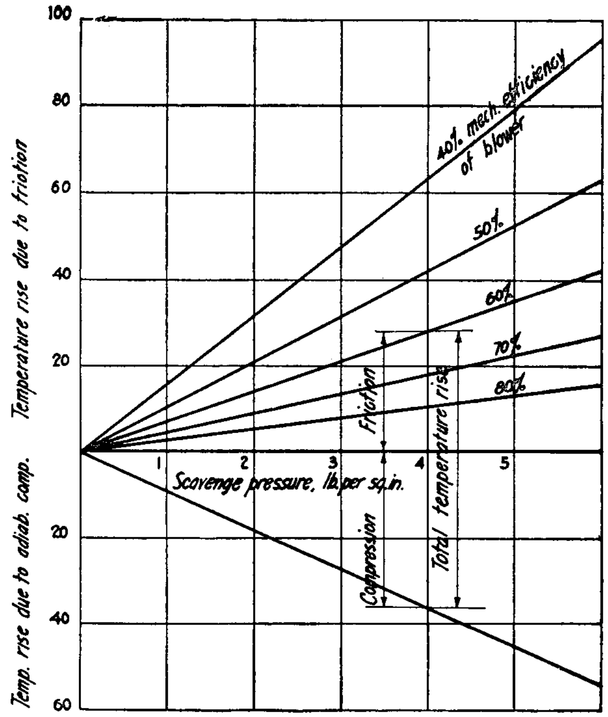

Fig. 11-21. Air Temperature Rise in Blower or Compressor. Example shows that for 4 psig scavenge pressure at 60 per cent blower efficiency the total temperature rise is 67 F.

SCAVENGING RECEIVER

11.18 In crankcase-scavenged engines the crankcase acts as a receiver. For the sake of high volumetric efficiency the crankcase should be made as small as possible, but the limit is somewhere around four times displacement volume for structural reasons.

In a blower-scavenged engine it is customary to use a separate receiver, although the crankcase is sometimes employed. The reason the latter practice is not more widespread is that it complicates lubrication.

To be more effective in damping out pressure fluctuations in the air supply to the inlet ports, the receiver should be as close to the ports as possible. Its size depends on the tolerable pressure fluctuation, on the type of blower, and on the number of cylinders to be supplied.

11.19 Receiver Volume.

In order to calculate the necessary receiver volume, it is well to start by drawing a graph comprising two lines: the delivery of the air by the blower to the receiver, and the drawing of air by the cylinders from the receiver. These two superimposed lines plotted on a crank-angle basis with the proper phase difference (if any) between them show the surplus or shortage of air existing in the receiver at any particular moment.

The first step is the drawing of the air delivery as in Fig. 11–23. The scale is immaterial as the blower delivery is made equal to the cylinder feed. The Roots blower or centrifugal blower having a continuous delivery is represented by a straight line. For a reciprocating blower the dash-dot line may be used if it is single acting, or the dash line if it is double acting.

The next step is to plot the delivery from the receiver. This is approximately proportional to the inlet opening. The inlet-opening area is proportional to the piston travel in the case of piston-controlled quadrangular inlet ports, see Fig. 7–4. By planimetering the area in increments from inlet opening to inlet closing the total volume of air drawn from the receiver in the period between the opening of the inlet ports and any crank degree is obtained. Figure 11–24 presents a series of air-delivery curves for various inlet durations. If the receiver supplies two or more cylinders, two or more of these banks must be added in proper phase relations.

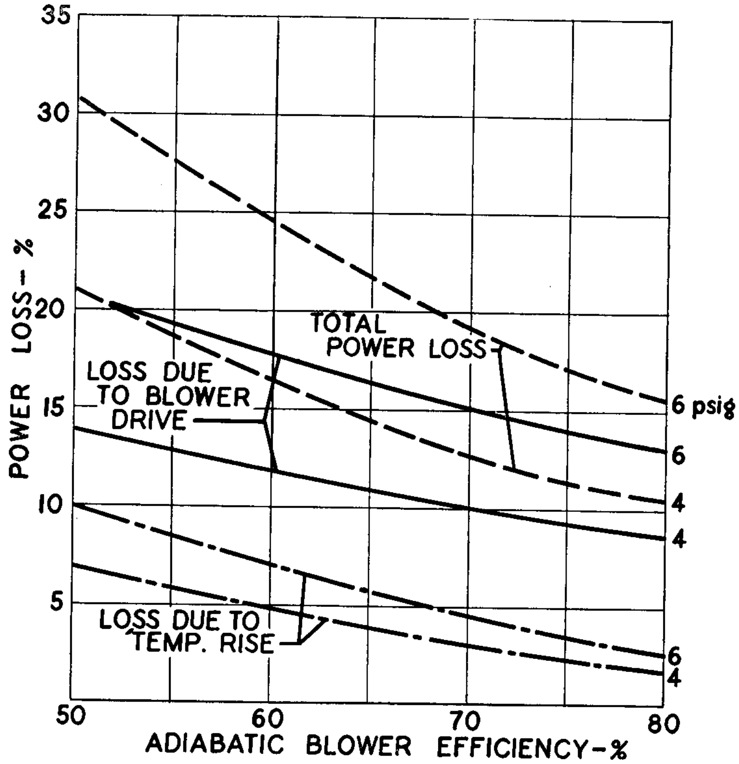

Fig. 11-22. Effect of Blower Efficiency on Engine Power with 4 and 6 psig Scavenge Pressure. Engine gross bmep is assumed to be 80 psi.

By superimposing the blower-delivery line and the receiver-delivery line, the surplus air or shortage at any instant can be scaled off. The scale is immaterial as long as both lines end at the same (100 per cent) point. The blower delivery is, of course, equal to the receiver delivery.

Figure 11–25 shows such a graph for a double-cylinder engine with 180-degree cranks, fed by a double-acting piston compressor. This has been obtained by selecting the proper curves from Fig. 11–23 and 11–24 and assembling them over 360 degrees. Because of the piston rod, the lower piston area is 18 per cent smaller than the upper piston area. The deliveries are in the same proportion. The total delivery is made 100 per cent for the blower and the receiver.

The areas marked (+) correspond to periods of excess blower delivery. During those periods the pressure in the receivers rises. The areas marked (−) correspond to

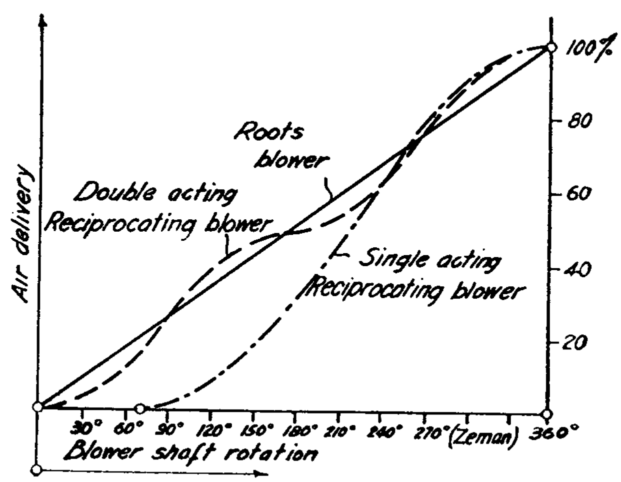

Fig. 11-23. Air Delivery of Roots Blower and Single- and Double-Acting Reciprocating Blowers. (Zeman, *Zweitakt-Dieselmaschine*, Springer, Vienna. Copyright, 1935, by Julius Springer.)

periods of deficient blower delivery. In those periods the pressure in the receiver drops. In Fig.

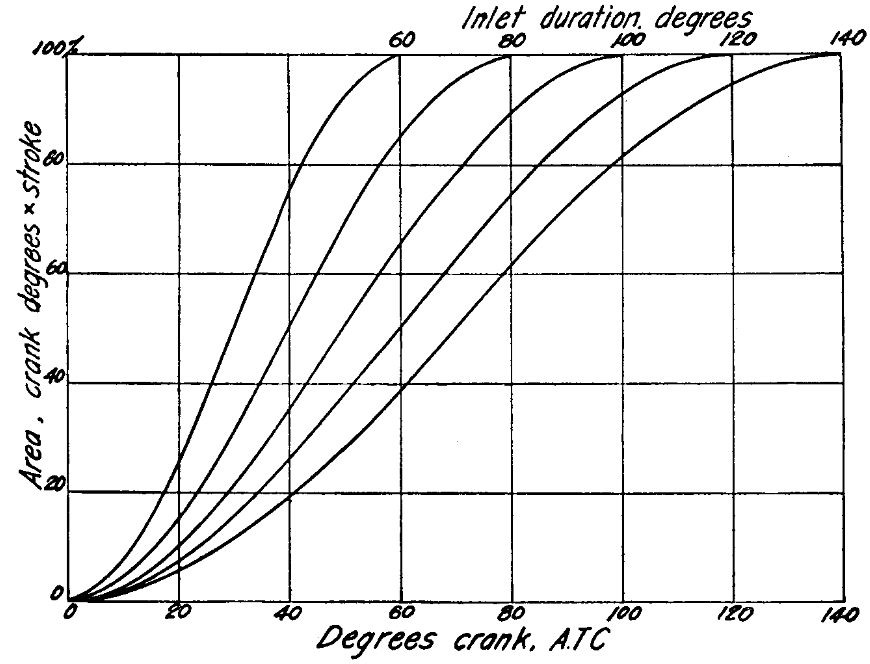

Fig. 11-24. Air Delivery from Receiver into Cylinder with Various Inlet Durations. Piston controlled quadrangular ports and connecting rod-crank ratio of 4.5 : 1.

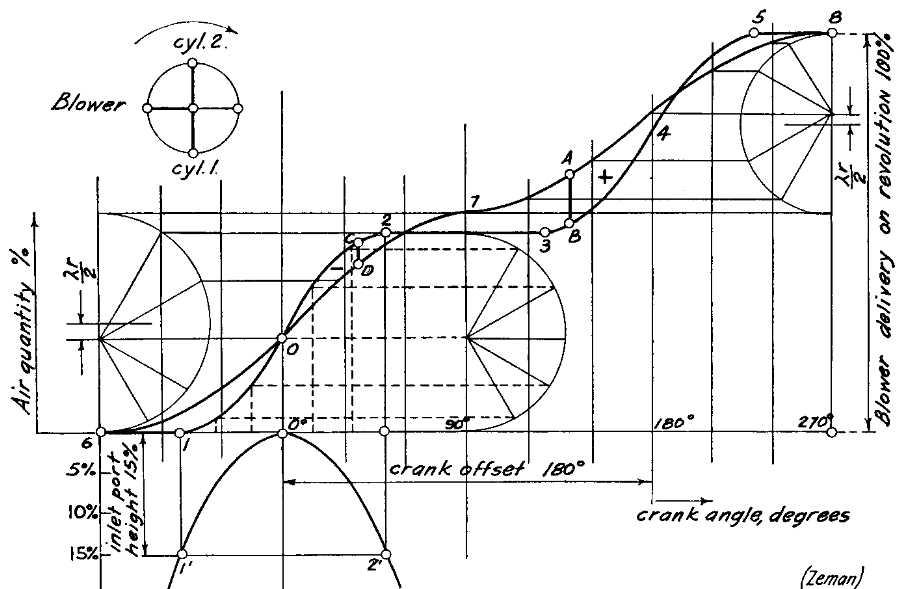

Fig. 11-25. Quantity of Air Delivered into (Line 6-7-8), and Drawn from (Line 1-2-3-4-5-8), the Receiver of a Two-Cylinder Engine with Double-Acting Compressor. (Zeman, *Zweitakt-Dieselmaschine*, Springer, Vienna. Copyright, 1935, by Julius Springer.)

11–25 the maximum air surplus is 13.5 per cent and the maximum shortage is 6.5 per cent of the total delivery during one revolution.

11.20 Fluctuations.

If the delivery fluctuations are known, the necessary receiver volume can be calculated. The calculation is given in the appendix following this chapter. Assuming a tolerance of 10 per cent in pressure fluctuations $(p_{max} - p_{min})/(p_{av} = 0.1)$, and an air consumption of 1.4 times displacement volume, the required receiver volume is

$$V_r = 19.6 \text{ (maximum excess + maximum shortage) } V_{disp}. \tag{11-5}$$

Applying this equation to the example above,

$$V_r = 19.6(0.135 + 0.065)\ V_{disp} = 3.9\ V_{disp}.$$

11.21 Volume Requirements.

In the case of Roots or other uniform-delivery blowers, the maximum delivery excess or shortage can be determined from the maximum deviation of the air consumption from the straight line. Table 11-III gives the necessary receiver volume in units of displacement volume. This is on

Table 11-III. Necessary Receiver Volume to Keep Pressure Fluctuations Within ± 10 Per Cent. Values are expressed in terms of piston displacement.

INLET DURATION, DEGREES	NUMBER OF CYLINDER 1	2	3	4
60	16.2	6.6	3.4	2.0
80	15.2	5.5	2.6	1.3
100	14.4	4.9	1.9	0.6
120	13.6	4.1	1.3	0.3
140	12.2	3.3	0.9	0.1

the assumption that the permissible pressure fluctuation is 10 per cent. If only 5 per cent can be tolerated the values must be doubled, and if 20 per cent is permitted the values are to be halved. It is natural that multicylinder engines and engines with longer inlet periods require less receiver volume.

Equation (11–5) and Table 11–III express the approximate requirements of receiver volume. If the air receiver consists of an air chest surrounding the individual cylinder as is normally the case with multicylinder engines, the values given refer mathematically to the total volume surrounding all cylinders. However, in view of the fact that an individual cylinder can depend on drawing only from the volume adjacent to it, the safe thing is to provide *at least* that much air-box volume to *each* cylinder. It is a good practice to provide two to three times that much to reduce the pressure fluctuations in the air chest and to counteract possible obstacles to the air flow as well as pressure fluctuations caused by the exhaust. Much too often inadequate air-chest volume is responsible for inferior engine performance.

Figure 11–26 shows an example of liberally sized air chest.

If any pipe or duct is used between the receiver and inlet ports, it should have as large a cross section and be as short as possible. In this respect placement of the blowers on the side of the engine is better than an end location. With reciprocal and centrifugal blowers the latter is frequently unavoidable, which is one more reason for making the air chest sufficiently large. So many two-stroke

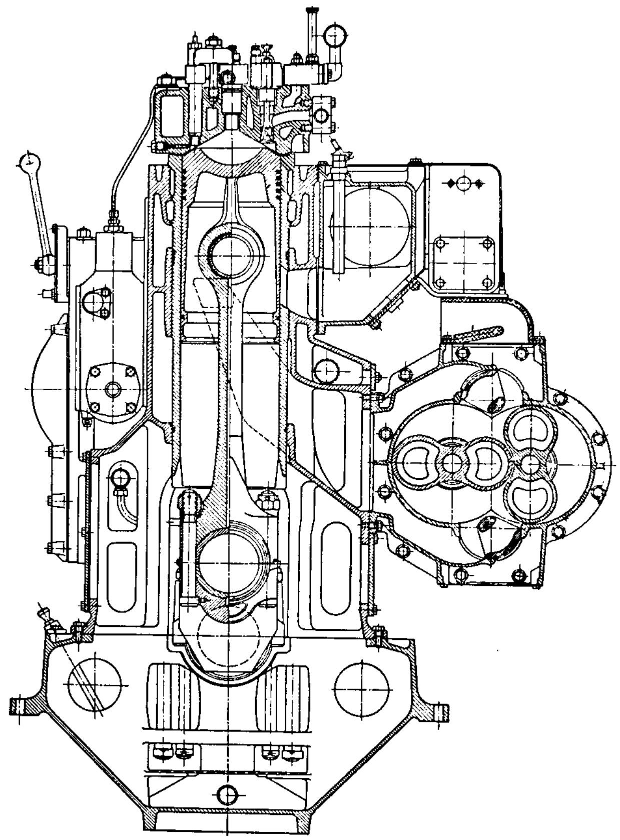

Fig. 11-26. Engine with Liberal Air Chest. Deutz 9.5 in. bore, 13.4 in. stroke, 500 rpm.

cycle engines suffer from inadequate air-chest volume that it is worth emphasizing that it usually pays to make the air-chest volume larger than would seem necessary.

APPENDIX TO CHAPTER 11

CALCULATION OF REQUIRED RECEIVER VOLUME

11.22 Calculation of the receiver volume is ordinarily based on the arbitrary rule that the pressure fluctuation in the receiver must not exceed a certain percentage, such as 10 per cent:

$$\frac{p_{max} - p_{min}}{p_{av}} = 0.1$$

V_r denotes the volume of the receiver, G_r the weight of air in it, G_{del} the weight of air delivered by the blower during *one* engine revolution, and p_{max}, p_{min}, and p_{av} denote, respectively, the maximum, minimum, and average pressure in the receiver. The amount of air contained in the receiver fluctuates during the cycle. There is an excess amount G_{exc} over the normal amount, when the pressure is a maximum, and there is a shortage G_{sho} when the pressure is a minimum.

Assuming the change to be adiabatic, which is justified in view of the low temperature differences,

$$p_{max}\left(\frac{V_r}{G_r + G_{exc}}\right)^{1.4} = p_{av}\left(\frac{V_r}{G_r}\right)^{1.4}$$

and

$$p_{min}\left(\frac{V_r}{G_r - G_{sho}}\right)^{1.4} = p_{av}\left(\frac{V_r}{G_r}\right)^{1.4}$$

from which

$$\frac{p_{max} - p_{min}}{p_{av}} = \left(\frac{V_r}{G_r}\right)^{1.4}\left(\frac{G_r + G_{exc}}{V_r}\right)^{1.4} - \left(\frac{V_r}{G_r}\right)^{1.4}\left(\frac{G_r - G_{sho}}{V_r}\right)^{1.4} = \left(1 + \frac{G_{exc}}{G_r}\right)^{1.4} - \left(1 - \frac{G_{sho}}{G_r}\right)^{1.4}.$$

By relating G_r, G_{exc}, and G_{sho} to the displacement volume ($G_r = r\ V_{disp}/v$, $G_{exc} = exc\,L\ V_{disp}/v$ and $G_{sho} = sho\,L\ V_{disp}/v$),

$$\frac{p_{max} - p_{min}}{p_{av}} = \left(1 + \frac{exc\,L}{r}\right)^{1.4} - \left(1 - \frac{sho\,L}{r}\right)^{1.4} \tag{11–6}$$

is obtained. Each member of the right side of the equation can be developed into a binominal series of the form:

$$(1 + x)^n = 1 + \binom{n}{1} x + \binom{n}{2} x^2 + \cdots$$

which gives for equation 11–6

$$\frac{p_{max} - p_{min}}{p_{av}} = 1.4\,\frac{L}{r}\,(exc + sho) + \frac{1.4 \times 0.4}{2}\left(\frac{L}{r}\right)^2 (exc^2 + sho^2) + \cdots$$

Since L/r is less than $\frac{1}{5}$, the second member of the series is already negligible and

$$\frac{p_{max} - p_{min}}{p_{av}} = 1.4 \frac{L}{r} (exc + sho), \tag{11-7}$$

hence the required receiver volume ratio

$$r = 1.4\, L\, (exc + sho) \frac{p_{av}}{p_{max} - p_{min}} \tag{11-8}$$

If it is decided arbitrarily that the pressure fluctuation shall not exceed 10 per cent and assume L to be 1.4,

$$r = 1.4 \times 1.4\, (exc + sho)\, 10 = 19.6 \times (exc + sho). \tag{11-9}$$

In the case of rotary or centrifugal blowers, $exc = sho$; therefore,

$$r = 39.2\, exc \sim 40\, exc,$$

which means that if the maximum delivery excess is 15 per cent, the receiver volume required to keep the pressure fluctuations within 10 per cent is $40 \times 0.15 = 6$ times the total displacement volume.

CHAPTER 12

EXHAUST SYSTEM

12.1 The primary function of the exhaust system is to discharge the combustion products into the atmosphere. It frequently includes means for silencing the exhaust. Silencers, as such, are not dealt with here. They are usually made by specialty manufacturers who offer engineering assistance for their installation.

The design of the exhaust system has a pronounced effect on the performance of a two-stroke cycle engine. The formulas offered in the foregoing chapters for calculating inlet and exhaust ports are valid only if the exhaust back pressure is negligible. Such is the case only if the engine exhausts directly to the atmosphere or into a large expansion chamber which connects to the atmosphere with an exhaust pipe of ample diameter. Empirical rules are mostly used for the dimensioning of exhaust pipes. To avoid throttling of the exhaust, the gas velocity in the pipe or duct must be lower than that in the exhaust ports, preferably at least a third less.

12.2 Empirical Rules.

In multicylinder engines the gas velocity in the common header or exhaust pipe should be still lower. Burgess-Manning Company recommends 50 fps for crankcase-scavenged engines; from 65 to 115 fps for low-speed (up to 350 rpm) separately scavenged engines; from 100 to 150 fps for medium-speed (350 to 1200 rpm); and from 135 to 165 fps for high-speed (above 1200 rpm) two-stroke cycle engines. In calculating the conduit size from the permissible gas velocities, it must be taken into account that the volume of the exhaust gas is greater than the volume of the intake air, in the ratio of absolute temperatures.

Another way of figuring required exhaust-pipe dimensions is on the basis of permissible exhaust back pressure which in a two-stroke cycle engine is not to exceed one inch of mercury, and it is preferable to have it less than 5 inches of water. According to a common formula, the volume of gas flowing in a pipe is

$$Q = 72\sqrt{\frac{\Delta p d^5}{\rho L}}$$

where Q is cubic feet per minute, Δp the pressure drop in psi, d the pipe diameter in inches, ρ the specific weight of the gas in pounds per cubic foot, and L the length of the pipe in feet. For a two-stroke cycle engine the average temperature of the exhaust gas in the pipe does not exceed 500 F, which would make $\rho = 0.0765\ (460 + 60)/(460 + 500) = 0.0423$. With this the formula can be written as

$$\Delta p = \frac{Q^2L}{122{,}500 \times d^5} \text{ psi, or}$$

$$h = 27.7 \times \frac{Q^2L}{122{,}500 \times d^5} = \frac{Q^2L}{4430 \times d^5} \text{ in. of water.}$$

The exhaust back pressure so calculated should be less than 5 inches of water.

Example: A 16-cylinder $8\frac{1}{2}$ by 10-inch 800 rpm blower engine has an air-delivery ratio of 1.325 and a gas temperature (in the exhaust pipe) of 500 F at full load. What is the required size for the exhaust pipe?

The exhaust gas volume is

$$1.325 \times 16 \times \frac{8.5^2\pi}{4} \times 10 \times \frac{800}{1728} \times \frac{460 + 500}{460 + 60} = 10200 \text{ cfm.}$$

Allowing an average gas velocity of 150 feet per second, the cross section of the common exhaust pipe must be

$$A = \frac{10200 \times 1728}{60 \times 150 \times 12} = 162 \text{ sq in.,}$$

so that the inside diameter of the conduit is to be 14.4 inches. The next standard welded steel pipe size is 16 inches with $15\frac{1}{4}$-inch inside diameter.

The pressure drop in a 100-foot length of such pipe is

$$h = \frac{10200^2 \times 100}{4450 \times 15.25^5} = 2.85 \text{ in. of water}$$

which is less than the permissible exhaust back pressure.

12.3 Pressure Pulsations.

The value of the above rules and calculations is limited by the fact that they ignore the existence of pressure fluctuations. The exhaust of a two-stroke cycle engine is not a steady flow but a pulsating flow and it is much affected by the number and order of exhaust impulses, etc.

Because of the pressure waves, the two-stroke cycle engine is also very sensitive to the length of the exhaust pipes or ducts. A well-tuned exhaust pipe enables the engine to give even more power than it gives with no exhaust pipe at all. Paradoxically, widening and/or shortening of the exhaust pipe sometimes reduces the horsepower output in spite of lower back-pressure readings on the manometer.

Exhaust system design with regard to pressure waves is now in a status similar to that of crankshaft design 20 years ago with regard to torsional vibrations. Considerable information has been obtained, but it has not spread widely enough and is generally ignored by practical designers.

12.4 Oversized Exhaust Pipes.

To be on the "safe" side, exhaust pipes are frequently oversized. But an oversize exhaust pipe is no better guarantee against synchronism than is an oversize crankshaft. Synchronism in torsional vibrations results in crankshaft breakage. Synchronism in exhaust pressure wave results in poor scavenging, reduced air charge, low power output, high fuel consumption, and high exhaust temperatures. By avoiding the *criticals* the crankshaft is safe from torsional failure. By avoiding synchronism of pressure waves with engine speed, poor filling and resultant power loss are avoided. Further improvement is obtained by harnessing the pressure waves of appropriate frequency and thereby gaining powerful assistance in scavenging and charging the cylinder. The result is higher output, lower fuel consumption, lower exhaust temperatures, lower piston and cylinder-head temperatures, reduced maintenance, and longer life.

The effect of tuning is particularly pronounced on single-cylinder engines, or on multicylinder engines with an individual exhaust pipe for each cylinder. The difference between well tuned and poorly tuned exhaust systems is then sometimes 30 per cent in engine output. In multicylinder engines with a common exhaust header another difficulty called *interference* is encountered. The exhaust of one cylinder interferes with the exhaust and scavenging of an adjacent cylinder. Such *bucking* regularly handicaps certain cylinders of a multicylinder engine.

The exhaust system is frequently evaluated on the basis of the exhaust back pressure. The exhaust back pressure as read on a manometer or pressure gage attached to a certain point of the exhaust pipe is not significant, however, because the actual exhaust pressure at any one point fluctuates during the cycle and the reading is some kind of an average. It is not this average which controls the charging and scavenging and with it the engine performance, but the actual pressures during the scavenging period. **The engine which has low (still better, subatmospheric) exhaust pressures during most of the scavenging period gives a better performance than does one with high exhaust pressures, although the manometer readings of the exhaust pressures may be the same.**

12.5 Tuning.

If no exhaust pipe were used, evidently the pressure at the exhaust opening would constantly be atmospheric. If a pipe is used, the sharp impulse initiated when the high pressure gases are put in communication with the exhaust pipe sets up pressure waves which travel back and forth in the pipe with the velocity of sound. The pressure next to the exhaust opening rises and falls in accordance with the natural frequency of the system but with declining amplitudes until a new impulse from the subsequent exhaust puff is superimposed on the existing pressure waves. The effect of these pressure fluctuations on the scavenging and charging process may be favorable or unfavorable, depending on the timing of the waves, which in turn depends on the geometry of the exhaust system.

12.6 Bucking.

It is easy to see that if the natural frequency of the exhaust column is exactly equal to the number of engine revolutions per second, the pressure wave so adjusts itself that the rise regularly coincides with the exciting impulse. This is undesirable, as it makes the pressure peaks in the exhaust duct coincide with the opening of the exhaust. The pressure wave bucks the exhaust and also the subsequent intake. The same happens if the natural frequency of the exhaust is twice the revolutions per second.

12.7 Favorable Conditions.

In order to secure favorable exhaust conditions the period of natural oscillations should approximately equal the scavenging period. A partial vacuum then exists during the latter part of the scavenging period, which is very helpful in drawing fresh air into the cylinder through the intake port. The depression should taper out when approaching the exhaust closure, so that the scavenged cylinder may then fill up with fresh air rather than have this air sucked out again by the exhaust vacuum. Figure 12–1 shows 3 pairs of weak-spring indicator diagrams taken near the exhaust ports in the crankcase of a crankcase-scavenged engine. The first pair shows unsatisfactory scavenging. Since the pressure never drops below 0.4 psig in the cylinder during the whole scavenging period, the scavening air cannot expand lower than to this value. Only after the closing of the ports does the pressure in the crankcase drop sufficiently to draw in fresh air from the outside. The pressure drop between the crankcase and exhaust port is small or negative during the second half of the charging period, which results in poor filling.

The third (bottom) pair of diagrams show very good scavenging. The vacuum in the exhaust near port closure causes a corresponding vacuum in the crankcase. The effect is that fresh air from the outside is drawn vigorously through the engine. The absolute exhaust pressure begins to rise when the piston is approaching port closure. That helps to fill up the cylinder by the ramming effect of the air. The resulting good filling raised the thermal efficiency from 0.222 in the top diagram to 0.252, an increase of 13.5 per cent. In the middle pair of diagrams conditions are in between. The only difference in the engine conditions represented by the diagrams was in tuning of the exhaust system.

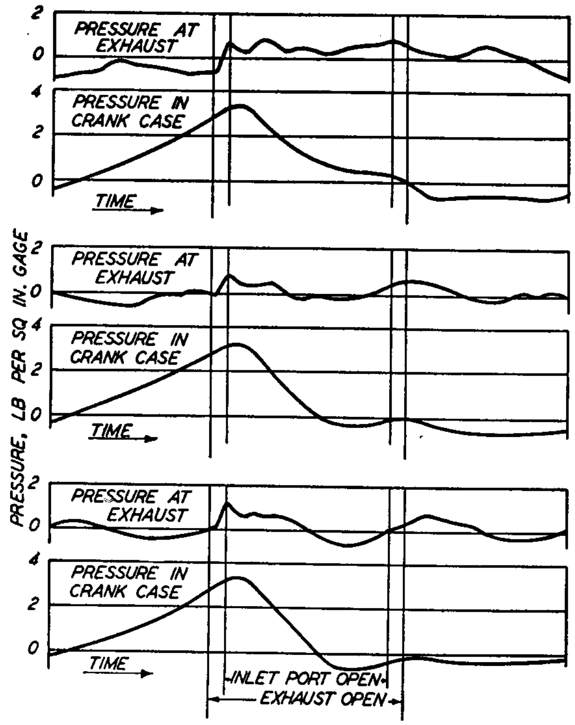

Fig. 12-1. Effect of Tuning on Pressure Fluctuations at Exhaust Ports and in Crankcase of Crankcase-Scavenged Engine during one Cycle. Top diagram shows poor tuning; thermal efficiency was 22.2 per cent. Middle diagram is slightly better, with thermal efficiency of 24.5 per cent. Bottom diagram shows good tuning; thermal efficiency rose to 25.2 per cent. (Th. Schmidt, "Schwingungen in Auspuffleitungen von Verbrennungsmotoren," *VDI Dieselmaschinen* **VI,** 79, 1936.)

12.8 Geometry.

It may be in order to point out here that the tuning of the exhaust system depends primarily on its geometry and the engine's rotative speed. *Geometry* refers to the length and diameter of the exhaust pipes and the volume of the various containers attached to or interposed in the exhaust system. If an exhaust system is tuned it stays tuned irrespective of load and operating conditions provided the speed does not change. An engine can be tuned for one speed only, and therefore tuning has the greatest significance for constant speed engines. Variable-speed engines should be tuned to the speed at which optimum performance is desired. This may be the most common operating speed or the speed of maximum power output.

Tuning of the exhaust system can be accomplished in the less complicated cases by calculation and design. In the more complicated cases experimental methods must be used such as those described in sections 16.24–16.26, which can serve also for checking the results of the mathematical analysis.

ANALYTICAL METHOD

12.9 Taking advantage of such work as was done by Schmidt [Schmidt, 1936] and Zeman [Zeman, 1936] in Germany, Farmer [Farmer, 1938] and Mucklow [Mucklow, 1942] in England, and Belilove [Belilove, 1943], Schweitzer [Schweitzer, 1944] and Warming [Warming, 1946] in this country, the problem of exhaust tuning can be made accessible to analytical or graphical calculations. In this manner the exhaust system can be tuned before it is built, saving much time and expense. In the following, methods of calculating exhaust systems are presented in a simplified manner, ignoring the derivations of the formulas and preceding theoretical work.

12.10 Frequency of Oscillations.

The frequency of the gas column oscillation in the exhaust system is determined primarily by the length of the pipe and secondarily by its diameter and the volumes interposed in the system. It is reasonable to start the calculations by selecting the diameter of the main exhaust pipe. If the cross section of the exhaust pipe is inadequate the exhaust is throttled irrespective of its tuning. If it is too large, the amplitude of the pressure waves is small and the effect of tuning is thereby minimized. That is an advantage if the tuning is incorrect but a disadvantage if the tuning is correct. The past tendency to use oversize exhaust pipes for two-stroke cycle engines presumably was due to the uncertainty caused by the supposedly incalculable pressure waves.

The rules given in the beginning of this chapter may ordinarily be used for the selection of the exhaust pipe diameter. However, in case of careful tuning on constant-speed engines somewhat smaller diameters are preferable.

After the size of the exhaust pipe has been selected, its tuning is effected by adjusting its length and the volumes interposed in the system. The controlling factor in the calculations is the natural frequency of the generated pressure waves.

12.11 Pressure Waves.

Pressure waves in exhaust pipes are similar to sound waves in organ pipes and are controlled by identical laws. In a plain pipe, closed at both ends, the period of the pressure wave is $2L/a$ where L is the length of the pipe and a is the velocity of sound in the gas. The value of a is identical with the velocity of sound and varies with the gas temperature according to the equation

$$a = C\sqrt{kpv} = C\sqrt{kRT} \tag{12–1}$$

where k is the ratio of specific heats of the gas, p, v and T its mean pressure, specific volume, and absolute temperature, respectively, and C a constant which decreases with the diameter of the pipe.

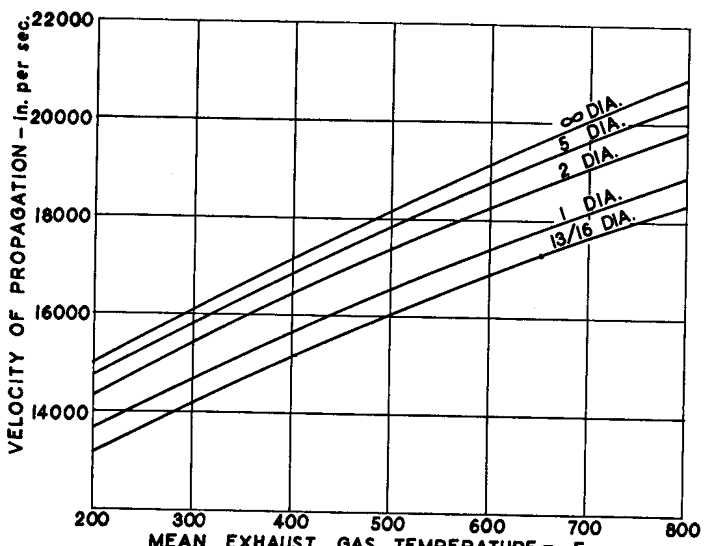

Fig. 12-2. Variation of Propagation Velocity of Pressure Waves in Pipes of Various Diameters.

According to recent determinations [Belilove, 1943], Fig. 12–2 represents the velocity of pressure propagation in pipes with average composition of exhaust gas for various pipe diameters plotted against the mean temperature of the exhaust gas.

For an exhaust system consisting of a single pipe of uniform cross section attached at one end of the exhaust ports, the other end being open to the atmosphere, the period of gas column vibration is $4L/a$. The reason for the double period is that the pressure wave is reflected at the open end of the pipe with sign reversed. The period of this negative wave is also $2L/a$, and therefore the total time of the complete cycle is $4L/a$.

12.12 Equivalent Pipe Length.

In calculation of pressure waves, it is customary to replace the oscillating system with a pipe of length L_e of uniform cross section, closed at both ends, which has the same frequency or period as the oscillating system. This is called the **equivalent pipe length.** For a plain exhaust pipe of uniform cross section throughout and open to the atmosphere, the equivalent pipe length is

$$L_e = 2(L + C_R) \qquad (12\text{–}2)$$

where L is the actual pipe length and C_R the so-called Rayleigh correction. Since the reflection does not take place exactly at the open end of the pipe, an additional length roughly equal 0.4 times the inside diameter is added to the actual pipe length. Except with very short pipes the Rayleigh correction is negligible; therefore

$$L_e = 2L. \qquad (12\text{–}3)$$

Using the convenient concept of equivalent pipe length, the complete natural period of the vibration of the gas column always is

$$t = \frac{2L_e}{a}. \qquad (12\text{–}4)$$

Consequently, if the exhaust port is connected through a plain pipe L to the atmosphere, the period of its exhaust column vibration is $t = 4L/a$.

If the period of the exhaust column vibration is equal to the period of the exhaust impulses, conditions are the most **unfavorable**, as represented by

$$t_w = \frac{60}{n} \text{ sec.} \qquad (12\text{–}5)$$

If the period of the exhaust column vibration is equal to the period of the port opening, conditions are the most favorable, as represented by

$$t_b = \frac{\alpha_e}{360}\,\frac{60}{n} = \frac{\alpha_e}{6n} \text{ sec,} \qquad (12\text{–}6)$$

where α_e is the period of the exhaust opening in crank degrees.

12.13 Example.

A 7.8 by 11.8-inch single-cylinder crankcase-scavenged engine operating at 370 rpm has an exhaust opening period of 136 degrees crank angle. The exhaust ports connect directly into the exhaust pipe of 5.35 inches inside diameter. What is the worst exhaust pipe length and what is the best?

The worst gas column frequently is the one which is equal to the engine frequency, the period of which is

$$t_w = \frac{60}{370} = \frac{1}{6.15} \text{ sec.}$$

The corresponding equivalent pipe length is, from equation (12–4),

$$L_{ew} = \frac{a}{2 \times 6.15}.$$

With an estimated mean exhaust temperature of 210 F, the propagation velocity of the pressure waves, from Fig. 12–2, is 14900 inches per second, and therefore

$$L_{ew} = \frac{14900}{2 \times 6.15} = 1210 \text{ in.}$$

The actual pipe length, from equation (12–3), is

$$L_w = \frac{L_{ew}}{2} = 605 \text{ in.}$$

This would be the worst pipe length.

The best gas column period, from equation (12–6), is

$$t_b = \frac{136}{6 \times 370} = \frac{1}{16.33} \text{ sec},$$

and the corresponding equivalent pipe length, from equation (12–4),

$$L_{eb} = \frac{14900}{2 \times 16.33} = 456 \text{ in.}$$

The actual pipe length is given from equation (12–3):

$$L_b = \frac{L_{eb}}{2} = 228 \text{ in.}$$

as the best pipe length for this engine.

It can be seen that the optimum exhaust-pipe length changes in inverse ratio to the engine speed. If, for general purposes, α_e is assumed to be 120 degrees of crank angle and a to be 15200 inches per second, the values of L for various speeds are:

When	$n = 100$	200	400	800	1600	3200 rpm
then	$L = 760$	380	180	90	45	22.5 in.

Frequently the location of the engine is such that the desirable pipe length is not sufficient to reach the outside atmosphere. In such case a large exhaust pit or expansion chamber can be used in place of the atmosphere at the end of the exhaust pipe, and the expansion chamber can be connected to the atmosphere by a tail pipe. The length of the latter is immaterial and has no effect on the pressure waves in the primary exhaust pipe, provided that its cross section is large enough to keep the pressure in the expansion chamber substantially atmospheric. A rule is given later in the text for the size of the expansion chamber.

12.14 Volume near Cylinder.

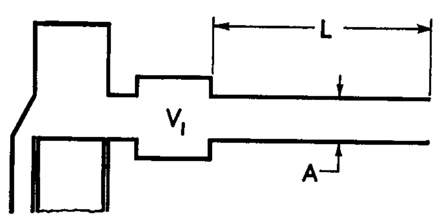

Fig. 12-3. Arrangement of Exhaust System.

Ordinarily the exhaust pipe does not connect to the exhaust ports directly, but a duct, sleeve, or chamber of a certain volume is placed between the exhaust ports and the exhaust pipe. The arrangement is shown in Fig. 12–3. The volume V_1 next to the exhaust ports affects the exhaust tuning and in this text is called the *exhaust pot* even if it consists only of a small enlargement over the exhaust-pipe cross section. The sizes of an expansion chamber and of a connected tail pipe do not affect the tuning if both are large enough to keep the pressure in the expansion chamber substantially atmospheric.

The equivalent pipe length of such a system has been calculated by Thomas Schmidt [Schmidt, 1936], and can be expressed by the following formula:

$$\tan \frac{\pi L}{L_e} = \frac{A L_e}{\pi V_1} \tag{12–7}$$

where L is the actual pipe length, L_e the equivalent pipe length, A is the cross-sectional area of the exhaust pipe, and V_1 the volume of the exhaust pot close to the engine. Figure 12–4 shows the relation in graphical form.

It is evident that equivalent pipe length (and therefore, the natural frequency of the exhaust system) is determined not by the pipe length alone, but also by its cross section and the volume between the engine and the exhaust pipe. Even a small volume V_1 increases the equivalent pipe length considerably.

In using equation (12–7) or the chart of Fig. 12–4, the exhaust pot volume V_1 should include any enlargement found beyond the exhaust port such as ducts, sleeves, etc., but only the volume in excess of the corresponding exhaust pipe should be counted. The actual exhaust-pipe length should be counted from the cylinder to the atmosphere or the large expansion chamber, and to that the Rayleigh correction of 0.4 × inside diameter may be added.

12.15 Example.

A 7.9 by 11.8-inch, one-cylinder, 18-hp crankcase-scavenged engine operating normally at 370 rpm has an exhaust opening period of 136 degrees crank angle. The exhaust ports connect directly into an exhaust pot of 2000 cubic inches volume. From this an exhaust pipe of 5.35 inches inside diameter leads to the atmosphere. What is the worst exhaust-pipe length, and what is the best?

The worst frequency is that equal to the engine frequency which corresponds to a period of

$$t_w = \frac{60}{370} = \frac{1}{6.15} \text{ sec.}$$

The corresponding equivalent pipe length is, from equation 12–4,

$$L_e = \frac{at_w}{2}$$

With an estimated mean gas temperature of 210 F in the pipe, the propagation velocity of the pressure waves from Fig. 12–2 is 14,900 inches per second, and therefore

$$L_e = \frac{14900}{2 \times 6.15} = 1210 \text{ in.}$$

The ratio $V_1/A = 2000/22.5 = 89$ inches, which gives, from Fig. 12–4, an actual pipe length of 515 inches, less $(0.4 \times 5.35) = 2.14$ inches Rayleigh correction; therefore

$$L_w = 513 \text{ in.}$$

This would be the worst pipe length.

The best frequency is that which corresponds to the period of exhaust duration, which is

$$t_b = \frac{136}{6 \times 370} = \frac{1}{16.33} \text{ sec}$$

and the corresponding equivalent pipe length

$$L_e = \frac{at_b}{2} = \frac{14900}{2 \times 16.33} = 456 \text{ in.}$$

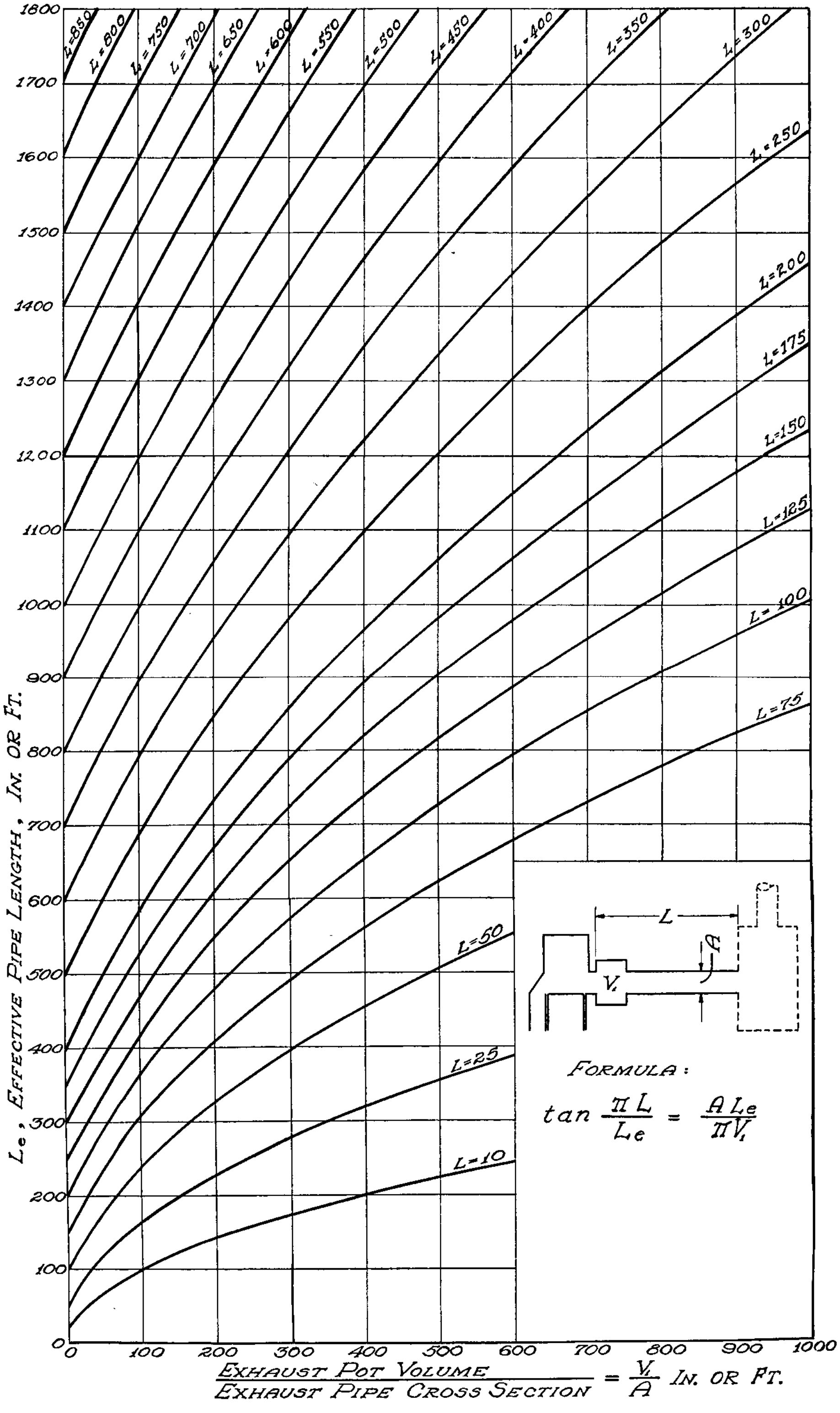

Fig. 12-4. Chart Showing the Equivalent Pipe Lengths for a Frequently Used Exhaust System.

Corresponding to the same value of 89 inches for the ratio V_1/A, Fig. 12–4 shows 150 inches; applying again the Rayleigh correction,

$$L_b = 148 \text{ in.}$$

as the best pipe length.

It so happens that Schmidt [Schmidt, 1936] has tried out a number of pipe lengths with the above engine and accurate records were taken. He found a 493-inch pipe length very bad and a 132-inch length very good. The corresponding pressure diagrams are reproduced in Fig. 12–5.

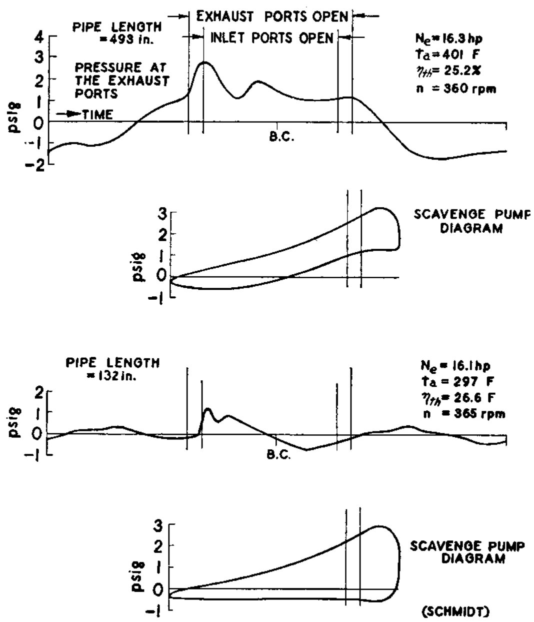

Fig. 12-5. Pressures at Exhaust Ports and in Crankcase. Top diagram shows pressures at ports with exhaust pipe 493 in. long; second diagram shows corresponding crankcase pressures; third diagram shows pressures at ports with exhaust pipe 132 in. long; bottom diagram shows corresponding crankcase pressures. (Th. Schmidt, "Schwingungen in Auspuffleitungen von Verbrennungsmotoren," *VDI Dieselmaschinen* **VI**, 79, 1936.)

The top diagrams were taken with the 493-inch pipe. The crankcase pressure diagram shows poor air delivery. The maximum power with this pipe was 19 hp and the thermal efficiency was 25.1 per cent.

With the 132-inch pipe the crankcase diagram shows a favorable delivery; the maximum power was 24.5 hp and the thermal efficiency rose to 29.5 per cent. It is evident that the pressure at the exhaust ports shows a complete oscillation during the exhaust period.

12.16 Large Expansion Chamber.

The above calculation is correct for an exhaust system such as that shown schematically in Fig. 12–3, where the main exhaust pipe connects to the atmosphere or to an expansion chamber sufficiently large to maintain substantially atmospheric pressure at the end of the exhaust pipe. This expansion chamber ordinarily connects through a tail pipe to the atmosphere and this tail pipe must be, of course, of such size as to prevent throttling of the exhaust gases. The following rules are used to select the size of the tail pipe and of the expansion chamber.

The manometric back pressure in the expansion chamber should not exceed 5 inches of water. The pressure drop in the pipe can be expressed as

$$h = \frac{Q^2L}{4430d^5} \text{ in. of water}$$

where Q is the volume of the exhaust gas in cubic feet per minute, d the diameter of the tail pipe in inches, and L its length in feet.

For the size of the expansion chamber the arbitrary requirement is set that it shall not change the natural period of the exhaust system by more than 5 per cent. Conforming to this requirement Fig. 12–6 shows the minimum ratios V_2/V_1 for various pipe lengths and V_1/A ratios. Accordingly the secondary exhaust pot or expansion chamber must be at least about 10 times as large as the primary exhaust pot in order to be equivalent to the atmosphere and permit the use of formula and chart shown in Fig. 12–4.

12.17 Small Expansion Chamber.

When the secondary exhaust pot or expansion chamber is relatively small (see Fig. 12-7), it affects the natural frequency of the oscillating system. Schmidt [Schmidt, 1936] developed the calculation for this case also and the resulting equation is as follows:

$$\tan \frac{\pi L}{L_e} = \frac{\dfrac{A}{V_1} + \dfrac{A}{V_2}}{\dfrac{\pi}{L_e} - \dfrac{L_e}{\pi} \dfrac{A^2}{V_1 V_2}}. \tag{12-8}$$

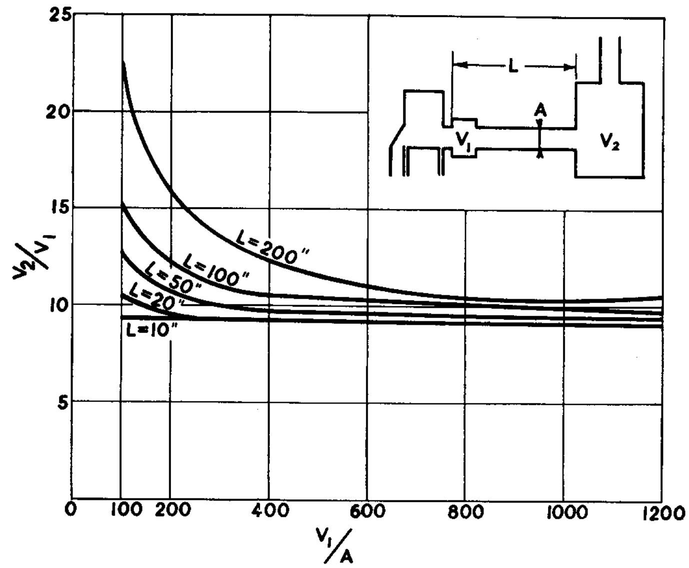

Fig. 12-6. Chart Showing Selection of Size of Expansion Chambers. For given value of V_1/A and L, ordinate shows minimum value of V_2/V_1 so that period differs no more than 5 per cent from the value obtained with $V_2/A = \infty$.

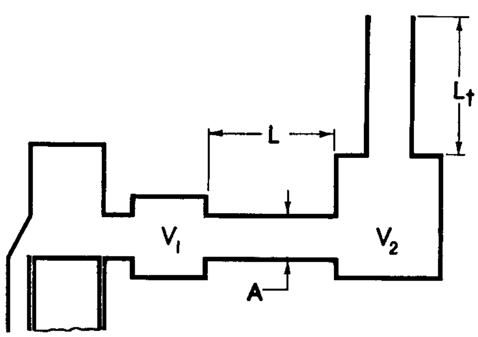

Fig. 12-7. Exhaust System Including a Relatively Small Expansion Chamber.

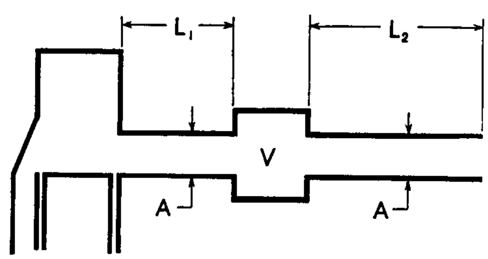

Fig. 12-8. Exhaust System With an Exhaust Pot in the Middle of the Exhaust Pipe.

Another system is shown in Fig. 12-8 with a single exhaust pot in the middle of the exhaust pipe, for which the following equation is valid:

$$\cot \frac{\pi L_2}{L_e} - \tan \frac{\pi L_1}{L_e} = \frac{\pi V}{A L_e}. \tag{12-9}$$

12.18 Example.

In Fig. 12–9, four pressure diagrams of the exhaust pipe are shown, presenting successively deteriorating tuning. The only variable in the setup was the size of the expansion chamber V_2. In order to equal the performance with no expansion chamber, Schmidt found it necessary to supply an

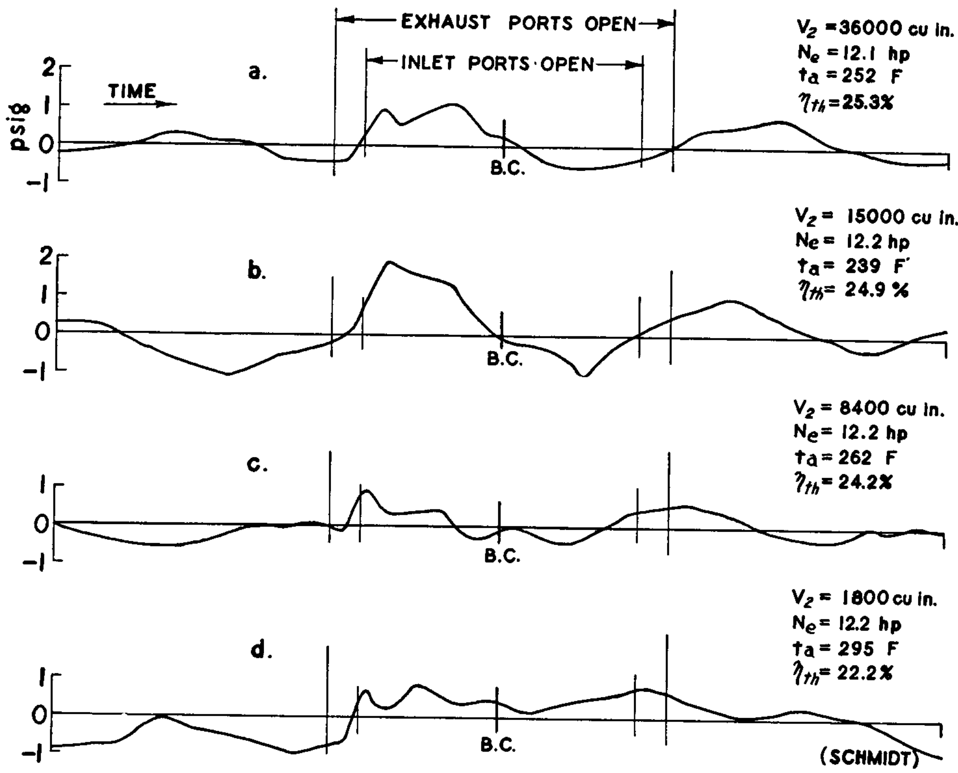

Fig. 12-9. Indicator Diagrams Showing the Effect of Size of Expansion Chamber. Pressure fluctuations in the exhaust near the exhaust ports.

expansion chamber of 0.6 cubic meter = 36,000 cubic inches. According to Fig. 12–5 the volume of the expansion chamber must be 17 times that of the primary exhaust pot or 12 × 2000 = 34,000 cubic inches, which checks closely with the experimental value. By reducing the secondary volume to 13,300, 8600, and 1840 cubic inches, respectively, diagrams *b*, *c*, and *d* were obtained and performance deteriorated, as shown by the indicated figures on thermal efficiency and exhaust temperature. The power output was kept substantially constant during the series.

12.19 Warming's Polar Diagram.

Systems more complicated than those referred to cannot be conveniently handled analytically. If the pipe contains several enlargements or if the system is forked or branched, the equations for calculating the natural frequency of the system or its equivalent pipe length become hopelessly complicated. Warming [Warming, 1946], however, has developed an ingenious graphical method for determination of the natural frequency of such exhaust systems. In a polar diagram he represents a pipe closed at one end and open at the other end with an arc of 90 degrees, and a partial length of pipe with a proportionally smaller arc. In the polar diagram the abscissa is pressure and the ordinate the amplitude of the pressure wave propagating in the pipe. At any point where the cross section of the exhaust system changes, the amplitude of the pressure wave increases in inverse proportion to the change in cross-sectional area. The pressure, on the other hand, remains the same. If a polar diagram is constructed on this basis and the assumed natural frequency of the system is the correct one, the last point corresponding to the atmospheric pressure at the open end must fall on the ordinate axis.

Warming's method may be applied to a case of the type represented by Fig. 12–8, the algebraic handling of which is complicated. The volume V represents a muffler interposed in the exhaust pipe, which is a rather common arrangement.

12.20 Example.

A single-cylinder $10\frac{1}{2}$ by 12-inch 425 rpm crankcase-scavenged engine, shown in Fig. 7–12, has an exhaust period of 126 degrees crank angle. The arrangement of its exhaust system is shown in Fig. 12–10. A 5-inch inside diameter exhaust pipe is selected and a 16 by 54-inch snubber is mounted 96 inches distant from the engine. How long should the tail pipe be for good tuning?

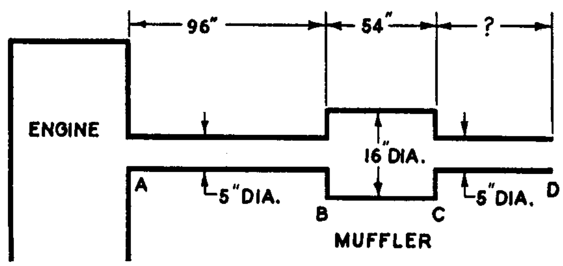

Fig. 12-10. Arrangement of Exhaust System.

Assuming a mean gas temperature of 450 F in the exhaust system, from Fig. 12–2 the propagation velocity of the pressure waves is 17400 inches per second. The duration of the exhaust is 126 degrees or

$$t = \frac{126}{6 \times 425} = \frac{1}{20.22} \text{ seconds.}$$

The best system frequency is the one which has a cycle period equal to the exhaust period, and is equal to

$$N = 60 \times 20.22 = 1213 \text{ cycles per minute.}$$

A plain pipe of

$$L = \frac{at}{4} = \frac{17400}{4 \times 20.22} = 215 \text{ in.}$$

would have the correct natural frequency.

In the polar diagram Fig. 12–11, 215 inches corresponds to 90 degrees, and a length AB corresponds to an angle

$$\text{arc } AB = \frac{L_{AB}}{215} 90 = 0.418\, L_{AB} \text{ deg.}$$

The first section of pipe is 96 inches long, which is represented in the polar diagram by the arc A_1B_1

$$\text{arc } A_1B_1 = 0.418 \times 96 = 40.1 \text{ deg.}$$

Therefore, 40.1 degrees are measured on an arc at *any* point of the abscissa axis and arrive at point B_1. At B the cross section changes from the pipe cross section to the snubber cross section; therefore, the area increases in the ratio of

$$\frac{16^2\pi/4}{5^2\pi/4} = \frac{10.23}{1}$$

The amplitude decreases at that point in the ratio of 10.23 to 1. Therefore, by dividing the ordinate of B_1 by 10.23 point B_2 is obtained.

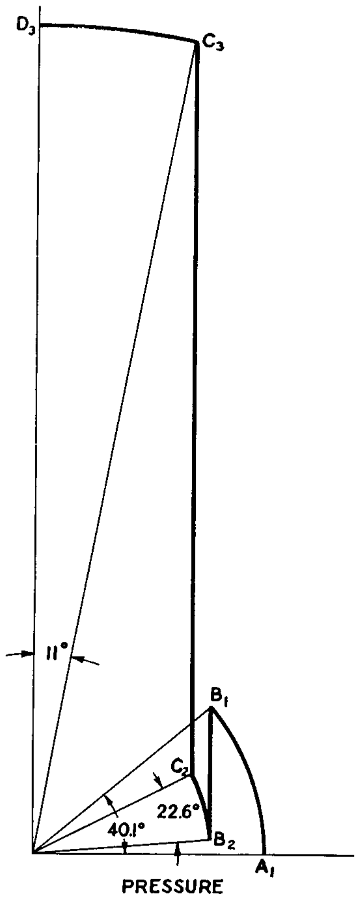

Fig. 12-11. Warming's Polar Diagram for Exhaust System Shown in Fig. 12-10. (T. Warming, "Polar Diagram for Tuning Exhaust Pipes," *Trans. A.S.M.E.* **68**, 31–3, 1946.)

The length of the snubber is 54 inches, which corresponds to an arc

$$\text{arc } B_2C_2 = 0.418 \times 54 = 22.6 \text{ deg.}$$

Scaling this angle from B_2 determines C_2. At this point the cross-sectional area contracts again in the ratio of 10.23 to 1, which increases the ordinate from C_2 to C_3. The arc from C_3 to complete the rectangle scales as

$$\text{arc } C_3D_3 = 11 \text{ deg,}$$

which would correspond to a pipe length of

$$L_{CD} = \frac{11}{0.418} = 26.3 \text{ in.}$$

By making the tail pipe $26\frac{1}{4}$ inches long the exhaust system is tuned for the engine.

Branched exhaust systems can also be treated by Warming's polar diagram method, the particulars of which are described in the paper to which reference was made.

12.21 Doubling the Period.

If the optimum exhaust pipe length is too short for structural reasons, even when used in conjunction with an expansion chamber, favorable results can also be obtained by multiplying the equivalent pipe length by two.

Example: A 12 by 15 367-rpm crankcase-scavenged engine had an 8-inch inside diameter exhaust pipe discharging into the open. It had a $78\frac{1}{2}$-inch long muffler, close to the engine, with a volume of 8550 cubic inches. The average exhaust gas temperature was estimated at 464 F and the exhaust port opening period was 135 degrees crank angle.

The exhaust-pipe cross section was $8^2\pi/4 = 50.3$ square inches. Since the muffler was in the line, the replaced pipe volume $50.3 \times 78.5 = 3950$ cubic inches must be deducted, giving an effective exhaust pot volume of $V_1 = 8550 - 3950 = 4600$ cubic inches, and $V_1/A = 4600/50.3 = 91.5$ inches.

The worst frequency is the one equal to the engine frequency, which had a period of $t_w = 60/367 = 1/6.125$ second. With a sound velocity of 17,800 inches per second, the equivalent pipe length is

$$L_e = \frac{at_w}{2} = \frac{17{,}800}{2 \times 6.125} = 1452 \text{ in.}$$

From equation 12–7 the actual pipe length corresponding to this, with $V_1/A = 91.5$ inches, is

$$L_w = 640 \text{ in.}$$

This would be the worst pipe length. For best pipe length the required period is

$$t_b = \frac{135}{6 \times 367} = \frac{1}{16.25} \text{ sec.}$$

which would give an equivalent pipe length of

$$L_e = \frac{at_b}{2} = \frac{17{,}800}{2 \times 16.25} = 550 \text{ in.}$$

or an actual pipe length of 190 inches, which was too short to reach from the building to the open; therefore, the equivalent pipe length is doubled, making 1100 inches. With this the actual pipe length turns out to be 460 inches as most favorable.

According to tests made by K. C. Whitefield [Whitefield, 1933] on this engine, the worst pipe length was 600 inches and the best 456 inches which is very good agreement indeed.

12.22 Multicylinder Engines.

The above equations are strictly applicable only to either single-cylinder engines or multicylinder engines with individual exhaust pipes connected to the atmosphere or to a large expansion chamber. The last arrangement is quite common with an exhaust pit under the floor into which the individual exhaust pipes connect. If the volume of the exhaust pit is larger than the size determined by Fig. 12–6, and if it connects with a large enough tail pipe to the open, the multicylinder engine can be treated as so many single-cylinder engines as far as exhaust tuning is concerned.

12.23 Helixhaust Manifold.

Similar effect can be obtained by the *Helixhaust* manifold [Diesel Progress, 1948] shown in Fig. 12–12, which is equivalent to a number of exhaust pipes laid parallel to each other. It is fabricated

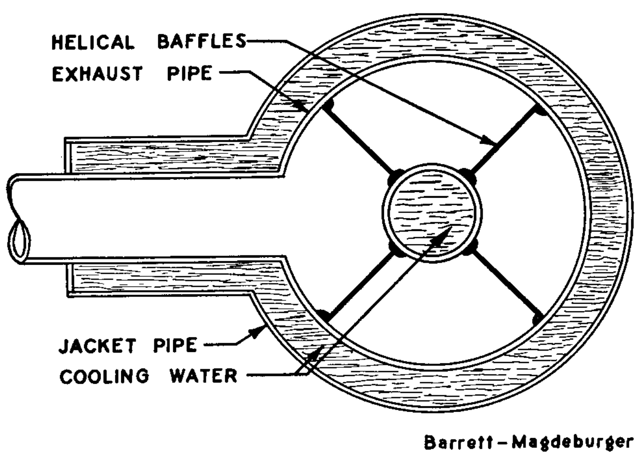

Fig. 12-12. Helixhaust Manifold. The Exhaust Manifold is divided by helically wound baffle plates into a number of sectors, each connecting either to a single exhaust port or to two or three ports from noninterfering cylinders. (Courtesy, The Stewart Iron Works Co., Cincinnati, Ohio.)

from a single pipe which is divided into a number of sectors by welded in baffle plates. The baffle plates are helically wound so in order that different sectors may pass next to the different cylinders.

However, with engines of eight or more cylinders, two or three cylinders are connected to the same sector. The cylinders discharging into the same sector must be so selected that their exhaust periods do not overlap each other.

It is a good practice not to allow more than two cylinders to exhaust into the same manifold. The firing distance of the joined cylinders must be equal (180 degrees crank angle). It is usually preferable to use a short manifold of adequate cross section and connect that to an expansion chamber or muffler rather than to use the manifold as an expansion chamber and let the cylinders exhaust immediately into it.

12.24 Untuned Exhaust System.

Most multicylinder two-stroke cycle engines have neither individual nor divided exhaust pipes but operate with a single exhaust header or manifold into which all cylinders are connected. No tuning of consequence can be effected under such circumstances if the number of cylinders exceeds three. Nevertheless, certain lengths of exhaust pipes prove to give slightly better performance than others. The favorable exhaust pipe length can be ascertained by the methods described in sections 16.25, 16.26 and 16.27.

12.25 Interference.

A more serious problem with multicylinder engines especially of 8, 9, 10 or more cylinders is *interference.* The exhaust of one cylinder interferes with the scavenging of an adjacent cylinder. If the firing order is such that when in cylinder A the exhaust blowdown is completed and the scavenging

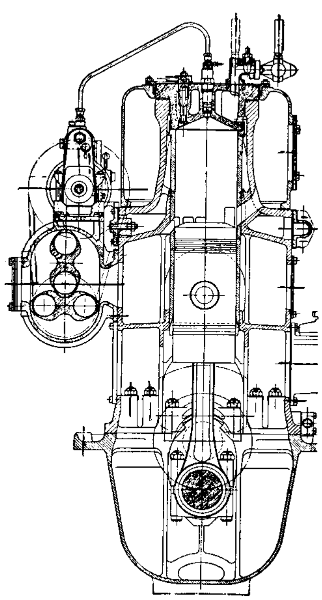

Fig. 12-13. M.A.N. 6.3 in. × 10.2 in., 600 Rpm Engine with Expansion Chamber Surrounding the Cylinders.

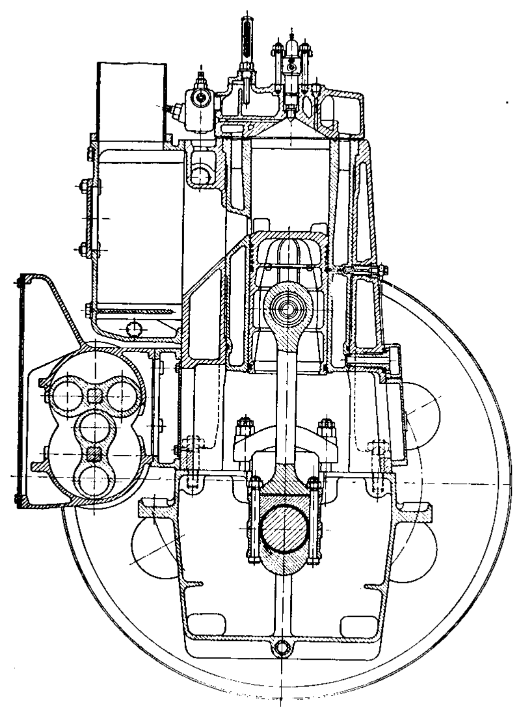

Fig. 12-14. Buckan 6.9 in. × 8.6 in., 600 Rpm Engine with Large Expansion Chamber.

process is in progress with both the inlet and exhaust ports open, then in an adjacent cylinder B (or in one a little distance away) the exhaust just opens, the exhaust puff naturally finds its way into cylinder A and opposes the scavenging of that cylinder. However, such a firing order is unavoidable with a certain number of cylinders and the best that can be done is to mitigate the effect of such interference.

Some designers use good-sized exhaust pots or an expansion chamber next to the cylinders, which prevent development of any appreciable pressure rise even momentarily in the chamber. Such an arrangement is shown in Fig. 12–13 where the expansion chamber for the exhaust is made of cast iron and surrounds the cylinders. Another good-sized expansion chamber is shown in Fig. 12–14.

With large engines the cost of a large cast iron expansion chamber is usually prohibitive, and more often welded steel exhaust headers are used like the one shown in Fig. 12–15. The pipe is uncooled but lagged with insulating material for the comfort of the operators.

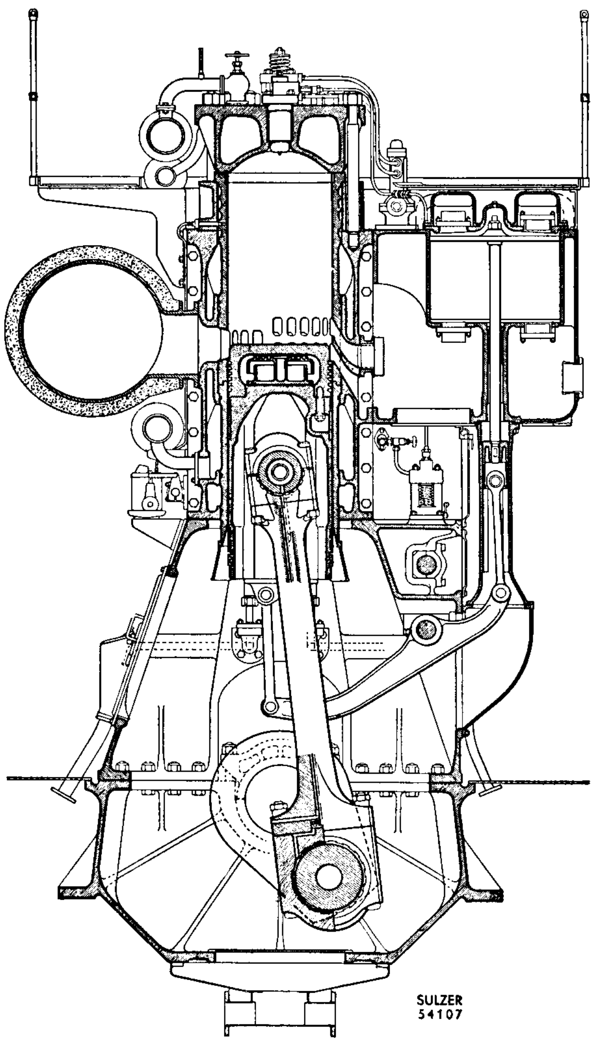

Fig. 12-15. Sulzer 22 × 39.4 in., 165 Rpm Engine with Welded Steel Exhaust Header.

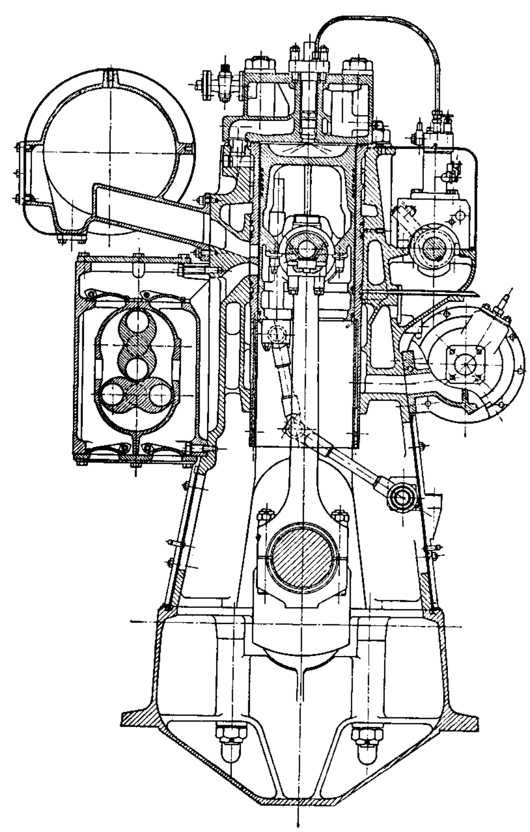

Fig. 12-16. M.A.N. 11.8 in. × 16.5 in., 400 Rpm Engine with Tangential Exhaust Branches.

Various geometrical arrangements are being tried by some designers for neutralizing exhaust interference, preventing the bucking of one cylinder by another cylinder, without conclusive information as to their merits. Pear-shaped plugs have been placed in the end of the exhaust ducts to discourage reverse flow; tangential connections with the header, angular ducts, elbows, and combinations of these have been used to make "one way streets" of the exhaust ducts, an example of which is shown in Fig. 12–16, others in Fig. 13–12, 13–13 and 13–14. It is a fairly general practice to make the exhaust ducts slightly divergent for the beneficial venturi effect. No methodical tests or conclusive results have been reported on these expedients so far.

12.26 Exhaust Pulse Supercharging.

Well-timed exhaust pressure pulses can also be used for supercharging a symmetrically scavenged engine. It will be recalled that in a symmetrically scavenged engine the exhaust ports close after the intake ports, and, consequently, with normal atmospheric exhaust it is difficult to charge the cylinder to a pressure much higher than atmospheric, because the extra air simply escapes through the exhaust ports. This is a deplorable circumstance, since the valveless loop-scavenged engine is an attractive type of engine, because of the simplicity of its construction. It has, however, been considered essentially a low output engine with maximum bmep of approximately 65 psi. The reason for the relatively low output has been that no supercharging can be applied to it in the manner in which it is applied to the nonsymmetrically scavenged engines described in Chapter 9.

One way to supercharge a symmetrically scavenged engine is by throttling the exhaust and building up an exhaust back pressure. This, however, is not a practical way because ordinarily the power output is decreased rather than increased by such a procedure. A continuously high exhaust back pressure obstructs scavenging and imposes an extra burden upon the scavenging blower. The consequent losses are ordinarily greater than the power gain, unless the superatmospheric exhaust pressure is utilized in a gas turbine instead of allowing it to expand idly to the ambient pressure. This latter alternative is discussed in Chapter 15.

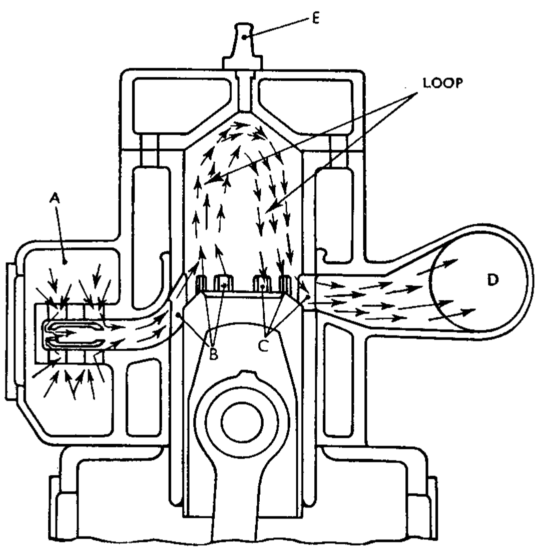

Fig. 12-17. Crossley-Carter Loop Scavenged Engine. A. Air under pressure in air manifold. B. Intake Ports. C. Exhaust Ports. D. Exhaust Manifold. E. Fuel Injector. (H. D. Carter, "The Loop-Scavenged Diesel Engine," *Proc. Inst. M. E.*, **154**, No. 4, 1946.)

It may be assumed, however, that the average pressure in the exhaust manifold is quite low, but that just at the time when the exhaust port of cylinder x is about to close, a pressure wave passes in front of the exhaust duct which stops the outflow of the air from the cylinder and even sends back some air that came out of the cylinder shortly before. Such a pressure wave may be a reflected wave from the end of the exhaust pipe, or in a multicylinder engine the exhaust impulse (puff) of an adjacent cylinder. Thus, a plug of gas is used instead of mechanical means to prevent escape of the cylinder charge through the exhaust ports early in the compression stroke, and the gas plug may even ram the charge to higher density at the same time that it is being forced in by the blower from the opposite intake ports. The average exhaust back pressure need not be high with such an arrangement, as only a pressure impulse of relatively short duration is needed to perform the supercharging, provided the pressure pulse coincides with the last 15 or 20 degrees of the exhaust port opening period. The exhaust back pressure being low (even subatmospheric) during the greater part of the scavenging period, the scavenging takes place against low resistance, and the intake pressure need not be high, and the blower would absorb little power from the engine. In this manner, exhaust pressure fluctuations can render powerful assistance to the scavenging and charging process.

This is the method that has been applied recently by H. Desmond Carter [Carter, 1946] to a Crossley two-stroke cycle engine with spectacular results. Figure 12-17 shows the vertical cross section of the engine and Fig. 12-18 the performance curves obtained with a $10\frac{1}{2}$-inch by $13\frac{1}{2}$-inch three-cylinder model normally running at 340 rpm.

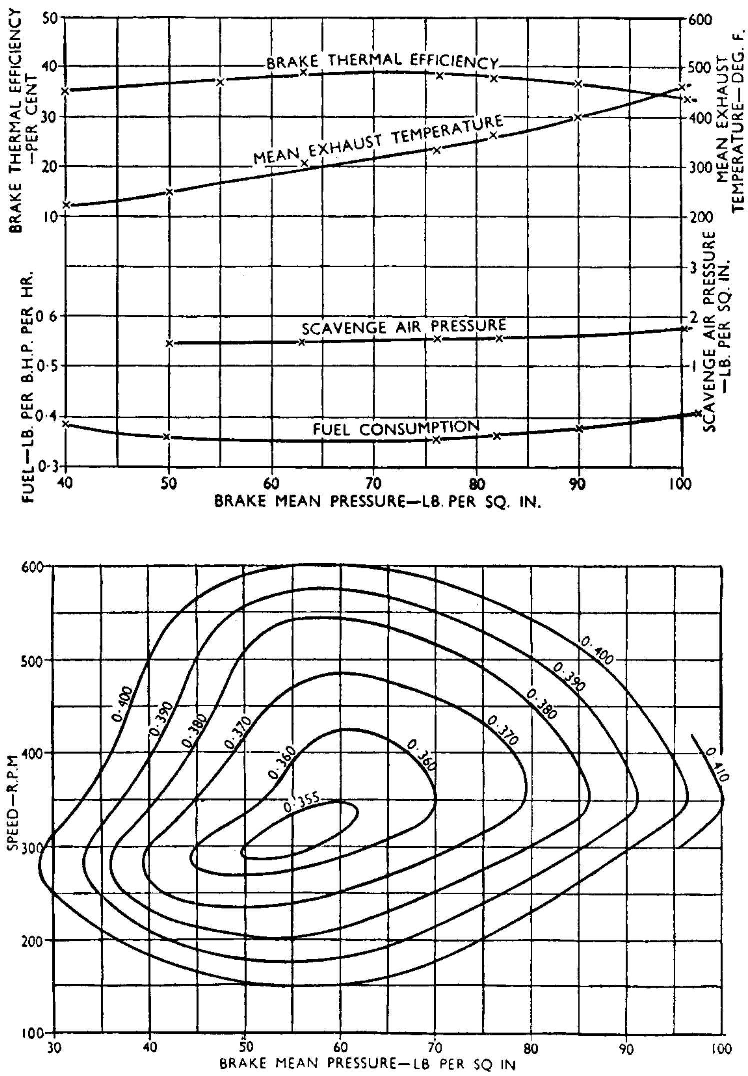

Fig. 12-18. Performance of Crossley-Carter Engine. $10\frac{1}{2}$ in. bore, $13\frac{1}{2}$ in. stroke, oil cooled pistons. (H. D. Carter, "The Loop Scavenged Diesel Engine," *Proc. Inst. M. E.*, **154**, No. 4, 1946.)

The 95 psi bmep was measured with a specific fuel consumption of 0.4 lb per bhp-hr. Between 45 and 80 psi bmep the fuel consumption was 0.37 lb per bhp-hr and at 55 psi bmep was as low as 0.355 lb per bhp-hr at the rated speed. Even at 600 rpm and 60 psi bmep the fuel consumption did not exceed 0.4 lb per bhp-hr. The scavenge pressure at 340 rpm was between $1\frac{1}{2}$ and 2 psig, which is exceptionally low for such high outputs. The exhaust temperature was also moderate.

This result was accomplished by proper timing and by skillful design of the exhaust system. To understand better the characteristics of exhaust pulse pressure charging, the sequence of events on weak-spring pressure diagrams is followed.

Figure 12–19(a), borrowed from Carter's paper, shows a diagram where exhaust pulse pressure charging is not used and is representative of that obtained from a four- or five-cylinder engine with a common exhaust manifold. Inlet and exhaust ports are shown opening at line O and closing at line C. The heavy line shows the cylinder pressures and the dash line the pressures in the exhaust duct.

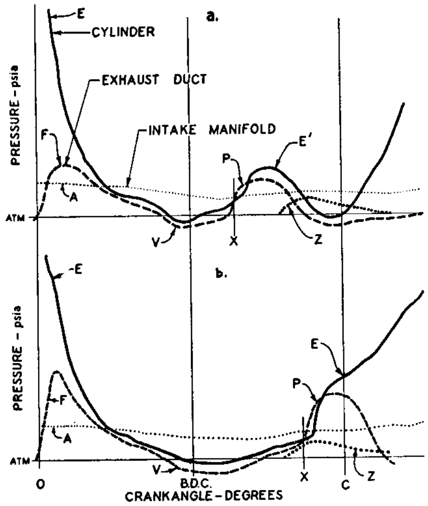

Fig. 12-19. Exhaust Pulse Supercharging Weak-Spring Diagrams. (H. D. Carter, "The Loop Scavenged Diesel Engine," *Proc. Inst. M. E.*, **154**, No. 4, 1946.)

After the exhaust opens, the pressure drops rapidly in the cylinder, and simultaneously a pressure pulse appears in the exhaust duct, the crest of which is reached shortly after exhaust port opening, after which it decreases and may even go below atmospheric as noted at V. At P another abrupt rise in the exhaust pressure appears which is obviously caused by the blowdown in the next cylinder, and the pulse finds its way into the exhaust branch of the subject cylinder. Naturally, while the exhaust ports are fully open, the cylinder pressures follow closely the pressures in the exhaust duct, which involves a rise in cylinder pressure also. This pressure rise is, however, of short duration and is followed by a dip which may go below atmospheric pressure as indicated in the diagram. Thus, when ports close, in this case, the cylinder pressure is subatmospheric and the cylinder charge is naturally very small. In a less extreme case the cylinder pressure might be above atmosphere, but hardly as high as the scavenge pressure indicated by the light line.

Figure 12–19(b) shows how, by a suitable combination of port timing, length, and cross-sectional area of exhaust ducts, manifold, and pipe, and the engine speed, the exhaust pulse of the succeeding cylinder in sequence has been so rephased that it reaches its crest just before the exhaust port closure, and the cylinder pressure at port closure is substantially above atmospheric and is even well above the scavenge pressure.

In a multicylinder engine with a common exhaust pipe, the pressure wave reflected from the end of the pipe is fairly well dissipated when it reaches the exhaust ducts and even with best timing cannot be expected to supply much pressure boost. The pressure impulse originating from the exhaust blowdown of the next firing cylinder is the chief source of the pressure boost. If the exhaust pressure charging pulse from the succeeding cylinder were absent, the pressure in the exhaust duct might be expected to follow the dotted line z with a tuned exhaust pipe [as shown in Fig. 12–19(a) and 12–19(b)]. While there is a positive pressure rise due to the second reflection of pulse F from the end of the pipe, it is, as already stated, effective only to a small extent as compared with the utilization pulse P from the next firing cylinder as a pressure charging means.

To make the exhaust pressure impulse most effective for supercharging, it appears that it should fulfill the following requirements. First, the exhaust blowdown must be rapid to get a good-sized puff. In this respect, exhaust ports are better than valves, and ports with square upper edges better than those with rounded upper edges.

Secondly, the impulse must arrive at the exhaust duct of the subject cylinder at just the right time, some 15 to 20 degrees before exhaust closure. If the opening period of the exhaust ports is 120 degrees, and the spacing of firing of a four-cylinder engine is 90 degrees, in the order of 1—3—4—2, and the travel time for the pressure wave from one duct to the other is 10 degrees crank angle, the pressure impulse originating from the exhaust blowdown of cylinder 3 would arrive in the exhaust duct of the adjacent cylinder 4, 100 degrees after the exhaust opening of cylinder 3, or 20 degrees before the exhaust closing of cylinder 4. Similar conditions obtain for cylinder 2 which is succeeded in firing by cylinder 3. Conditions would be somewhat less favorable to cylinders 1 and 4 because of their greater distance from the succeeding cylinder in firing order.

Conditions are also favorable in a 3-cylinder engine if the duration of the exhaust port opening is of the order of 140 degrees crank angle.

Conditions would not be so favorable with other cylinder numbers unless the exhaust pipes were divided. This is indeed being done in 6- and 8-cylinder engines with good results.

The arrival of the pressure wave at any one exhaust duct depends not only on the timing of the next exhaust blowdown, but also on the travel time of the wave from its source of origin. The pressure waves travel at sound velocity, which is of the order of 1400 feet per second in hot exhaust gas. This means that it takes $6n/1400$ crank degrees for the pressure impulse to travel one foot or 1.5 degrees per foot in a 350 rpm engine. From this it follows that the exhaust branches must also be tuned or the length of each must be selected according to the distance from the exhaust duct of the cylinder which is next in firing order. It also follows that exhaust pulse supercharging is good for only one engine speed.

Conditions are not quite so critical because the pressure pulses have a considerable spread, 20 to 45 degrees crank angle according to Carter, and there is a margin in the time of their arrival. Therefore a reasonable variation in engine speed or deviation from the optimum branch pipe length is not detrimental to the engine performance.

As an example of what can be accomplished by careful tuning, Fig. 12–20 is reproduced from Carter's paper, representing a cathode ray oscillogram taken from the above mentioned $10\frac{1}{2}$ by $13\frac{1}{2}$-inch engine. Attention is directed to the fact that at port closing the cylinder pressure of the trapped air charge was 12 psig, arising from a pressure charging pulse which in the exhaust branch has an amplitude of 8.5 psig, both in association with an air manifold pressure of only 2 psig. This represents an almost incredible degree of supercharging, possible only under exceptionally favorable circumstances. If the cylinder pressure registered at the time of ports closure prevailed over the entire cylinder volume, the volume of the trapped charge would permit twice as high a bmep as is indicated. In view of the transient nature of the charging process, it should not be expected that the same pressure would exist at every point of the cylinder. But even so, a diagram like Fig. 12–20 is very impressive and Carter's performance results equally so.

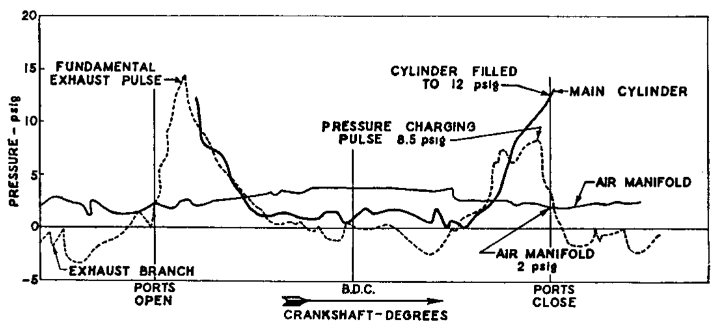

Fig. 12-20. Optimum Exhaust Pulse Supercharging. $10\frac{1}{2}$ in. bore, $13\frac{1}{2}$ in. stroke, three-cylinder engine, 330 bhp. Cathode ray oscillogram taken from cylinder No. 1. 76 psi bmep at 500 rpm. (H. D. Carter, "The Loop Scavenged Diesel Engine," *Proc. Inst. M. E.*, **154**, No. 4, 1946.)

APPENDIX TO CHAPTER 12

ENERGY IN THE EXHAUST

12.27 Utilization of the Exhaust.

Whenever the engine exhaust is put to work, be it for boosting the intake air as in turbocharging and in the Kadenacy type of scavenging (Chapter 13), or for a direct increase of the shaft power output as in a compound engine (Chapter 15), the energy contained in the exhaust gas is used to perform work. Either the pressure energy which cylinder gas possesses at the time of exhaust release is converted into mechanical work as it is in a steady-flow turbine or in a low-pressure reciprocating engine, or the kinetic energy of the exhaust gas is converted into work as it is in an impulse-type or blowdown turbine, or in inertia supercharging, or both.

In every case, when the exhaust is utilized to perform a certain useful function, there is a definite limit to what the exhaust can do. It is useful to know that limit.

12.28 Convertible Blowdown Energy.

One pound of gas of a certain pressure and temperature enclosed in a stationary container has the ability to do a certain amount of work. The heat value of that work is naturally less than its internal energy, because all heat energy can never be converted into mechanical work. It is also less than its *available energy* because it is not feasible to expand a gas to lower than atmospheric pressure (as is done in a steam condenser). The maximum work that can theoretically be obtained from the gas is called **convertible blowdown energy.**

The convertible blowdown energy can be converted into rotary motion either in an engine or in a turbine. It is shown below that a given amount of gas of certain pressure and temperature produces the same amount of work in an ideal engine as in an ideal turbine.

An **ideal engine** has no heat and friction loss and its expansion is long enough to permit the gas to reach atmospheric pressure. The work gained by complete expansion, Fig. 12–21, was shown [Schweitzer, 1925] to be equal to the tail triangle in the *pv* diagram, and it can be expressed either as

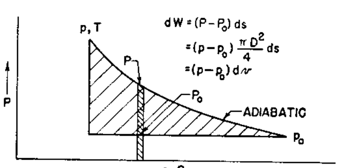

Fig. 12-21. Complete Expansion.

$$W_e = (u - u_0) - Ap_0(v_0 - v)$$

or as

$$W_e = (h - h_0) - A(p - p_0)v$$

From the latter with $pv = RT$ and $p_0 = 14.7$ psia

$$W_e = (h - h_0) - ART\left(1 - \frac{14.7}{p}\right) \tag{12–10}$$

$$= (h - h_0) - \frac{T}{14.62}\left(1 - \frac{14.7}{p}\right).$$

In Fig. 12–22, the convertible blowdown energy has been plotted as a function of the initial gas pressure and temperature. It is seen that the convertible blowdown energy increases almost in a straight line with the initial gas temperature, and it increases only slowly with initial gas pressure when the pressure is high.

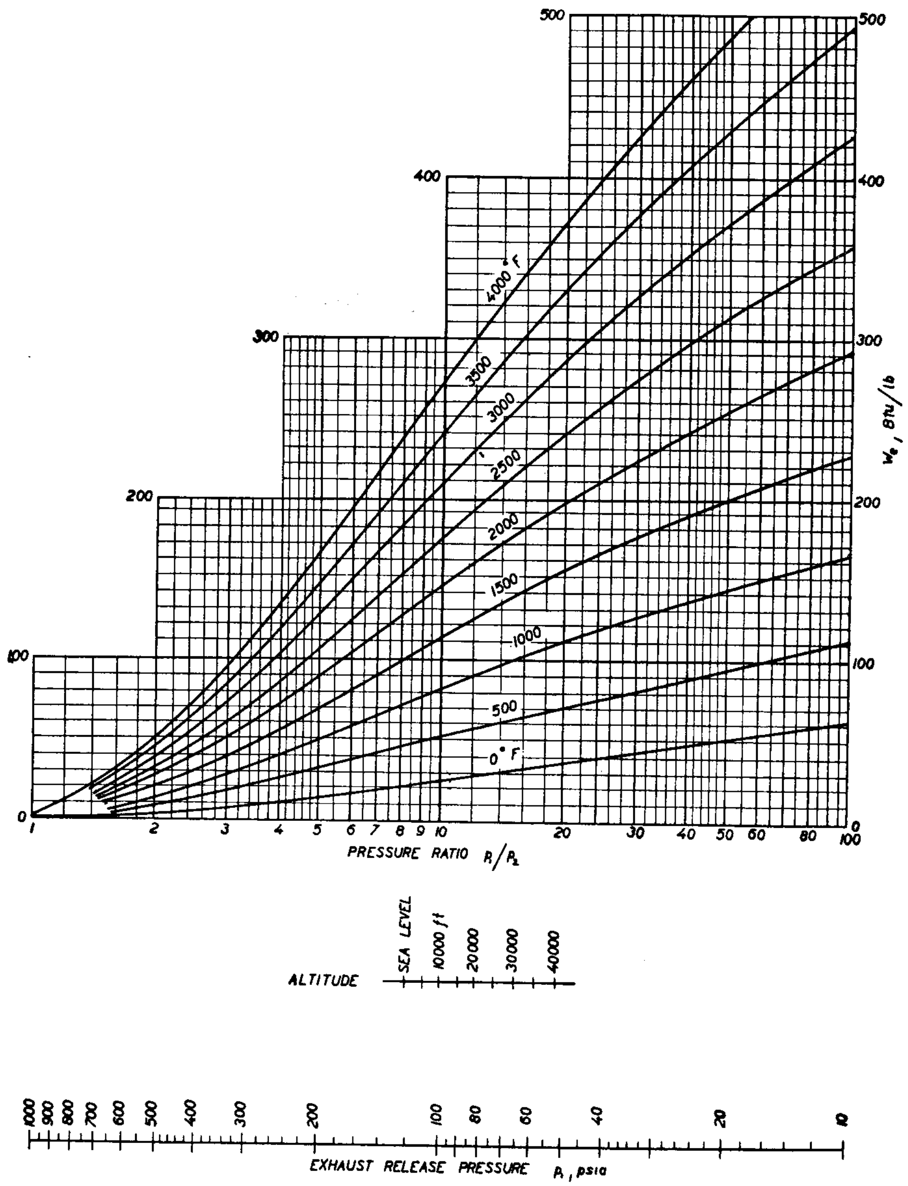

Fig. 12-22. Chart for Determining the Convertible Blowdown Energy.
Example: At 30,000 ft altitude (ambient pressure = 4.36 psia). Exhaust release temperature = 2000° F. Exhaust release pressure = 50 psia. Pressure ratio = 11.5. Convertible exhaust energy = 156 Btu/lb of cylinder gas.

As an example it is assumed that at the instant of exhaust opening, one pound of gas of 428 sia = 29.1 atmospheres pressure and 3470 R temperature is in the cylinder. The corresponding nvertible blowdown energy, read from Fig. 12–22, is

$$W_e = 323 \text{ Btu} = 252{,}000 \text{ ft-lb.}$$

With one pound of air used *per minute* the output of the ideal engine is

$$\frac{323}{42.42} = 7.63 \text{ hp}$$

12.29 Ideal Turbine.

An ideal turbine utilizes 100 per cent of the enthalpy drop across the turbine. That enthalpy naturally varies during the process. In an explosion turbine such as this, the jet velocity at the start is high and gradually, as the container discharges, tapers down to zero. At every instant the theoretical jet velocity v_x is such that

$$\frac{v_x^2}{2g} = 778(h_x - h_0)$$

where h_x is the instantaneous enthalpy of the gas in the container and h_0 the enthalpy corresponding to the temperature which obtains after an isentropic decompression of the gas to atmospheric pressure. It is also assumed that the turbine has an efficiency of 100 per cent irrespective of the jet velocity.

Under these circumstances, the total turbine work in Btu is:

$$W_t = \int (h_x - h_0)\, dG$$

The integration limits are 1 (envisaging 1 pound of gas) and the final amount of gas G_0 which is left in the cylinder when the blowout is completed.

The above integral can be evaluated graphically. Taking the values of the previous example, the pressure and temperature in the cylinder at the point of the exhaust releases are

$$p = 428 \text{ psia}; \; T = 3470 \text{ R.}$$

In the graph of Fig. 12–23, for various initial pressures and 14.7 psia final pressure, with isentropic

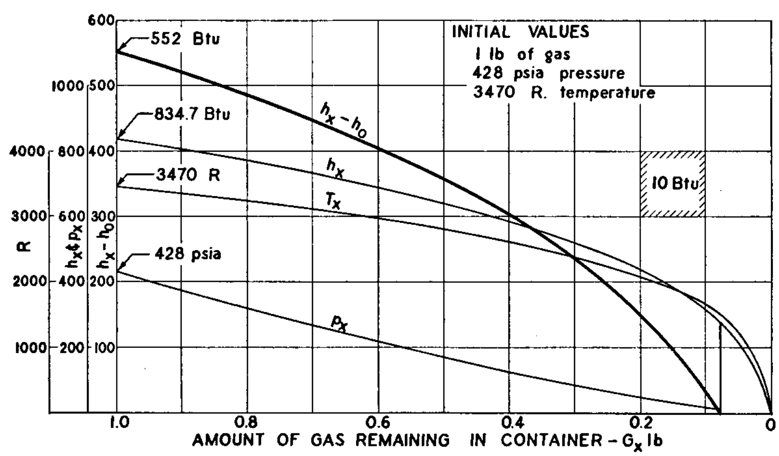

Fig. 12-23. Convertible Exhaust Energy in a Turbine.

expansion, T_x and h_x have been calculated from thermodynamic tables [Keenan and Kaye, 1945], and from them for each p_x

$$G_x = \frac{p_x V}{R T_x}$$

where $$V = \frac{1 \times 53.3 \times 3470}{428 \times 144} = 300 \text{ cu ft.}$$

In plotting p_x, T_x, and h_x against G_x, it is found that p_x becomes 14.7 psia at $G_x = .0778$ pound, and at that point $h_x = h_0 = 282.64$ Btu. Plotting $(h_x - h_0)$ and integrating it (by planimetration) between $G_x = 1$ and 0.0778 the total energy is obtained as

$$W_t = 323 \text{ Btu} = 252{,}000 \text{ ft-lb.}$$

This is the maximum amount of energy that can be obtained from an ideal turbine. Assuming that the container is filled up once every minute (air consumption 1 pound per minute), the energy corresponds to

$$\frac{323}{42.42} = 7.63 \text{ hp}$$

which is the same amount that was obtained in 12.28.

It is seen that **a given amount of gas delivers the same power in an ideal turbine as in an ideal (complete expansion) engine.**

In any mechanism or process which involves a mechanical utilization of the engine exhaust, the convertible blowdown energy represents the **maximum** work that can be recovered from the exhaust if the exhaust cannot expand lower than to atmospheric pressure. Figure 12–22 shows the values of the convertible blowdown energy available. Naturally, in most cases, the actual return is only a fraction of the theoretical limit.

CHAPTER 13

THE KADENACY SYSTEM

13.1 The conventional picture of the scavenging process in a two-stroke cycle engine consists of first releasing the exhaust gases so that the cylinder pressure is reduced to below scavenge air pressure, then delivering the fresh air charge which is caused to sweep out most of the remaining inert gases and fill the cylinder with clean air. In section 2.3 it was stated that "In ideal scavenging the scavenge air acts like a piston in pushing the burnt gases out of the cylinder without actually mixing with them." In a certain respect an actual engine can exceed the above stated ideal. In engines employing the *Kadenacy System* the fresh air charge is not called upon to **push out** any of the exhaust gases. On the contrary, it is **drawn in** by the outrushing mass of exhaust gases or rather by the depression created in its wake.

By making use of the inertia of gas columns, a two-stroke cycle engine can even be operated without any blower, as was demonstrated by Michel Kadenacy. The first commercial blowerless engine was built in 1932 by Petter [Petter, 1934] in England, and the same company is still building Kadenacy-type engines of improved design but with a blower attached.

13.2 Penn State Tests.

The essence of the Kadenacy system is the depression created in the cylinder by a well-timed rapid opening of the exhaust. In the tests conducted by Schweitzer, Van Overbeke and Manson [Schweitzer et al., 1946] at the Pennsylvania State College, the sudden evacuation of a cylinder filled with compressed air of 80 psig pressure was followed by a vacuum of approximately 10 inches of mercury. In these experiments a vessel of 6-inch diameter and $15\frac{1}{2}$-inch length was filled with cold air of a certain pressure. Then the lid of the vessel was suddenly removed by hitting a latch as shown in Fig. 13–1 and the resulting vacuum measured by the response of a diaphragm. Kadenacy utilizes this vacuum to suck fresh air into the engine cylinder and so effect the scavenging and charging, which operations are ordinarily performed by a blower. Well-tuned exhaust (and inlet) pipes are helpful but are not the essence of the Kadenacy system. As reported by Davies [Davies, 1937] a Junkers opposed-piston engine converted to the Kadenacy system was successfully operated with various lengths of exhaust pipes from 59 to 98 inches and in a speed range from 540 to 2000 rpm. The conversion resulted in a substantial increase in power. The brake mean effective pressure was increased from 85 to 142 psi, the normal rotative speed from 1200 to 1700 rpm, and the power output from 11 to 25 hp. This is claimed to have been obtained without any alteration of the combustion chamber or fuel-injection equipment, solely by eliminating the scavenge pump and changing the characteristics of the inlet and exhaust ports and passages, in accordance with the Kadenacy patents.

13.3 Geyer's Theory.

Various explanations have been offered for the Kadenacy effect. Kadenacy himself ascribes it to a *ballistic speed* of the exhaust, said to be about four times as high as sound velocity [U. S. Patent No. 2,131,957]. Giffen [Giffen, 1940] calculated the pressure waves generated by the sudden evacuation of a cylindrical vessel and obtained depressions of the order that have been observed. Geyer's [Geyer, 1941] rather simple theory attributes the effect to the kinetic energy of the gases rushing out of the cylinder. The potential energy of the compressed gas is, upon opening a passage, transformed into kinetic energy. The gas leaves the cylinder with a velocity that corresponds to this energy. At the time the pressure inside of the cylinder has dropped to atmospheric, most of the gas has left the cylinder, but the gas remaining in the cylinder still has some kinetic energy, which can perform work against the atmospheric pressure. This work consists of displacing a certain volume of air against ambient pressure. The equivalent volume of gas to replace that volume must come from the cylinder, and so a depression is created by the discharge of that amount of gas.

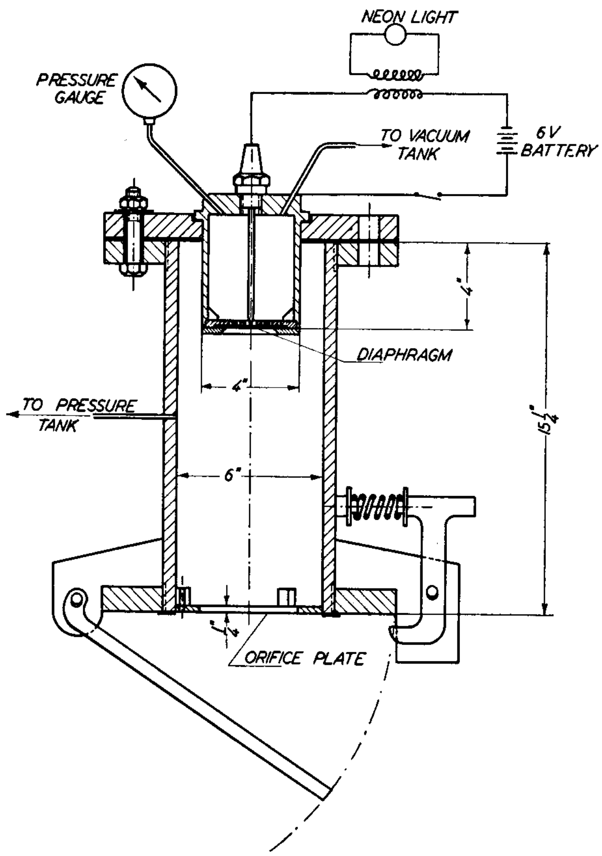

Fig. 13-1. Penn State Apparatus for Testing the Kadenacy Effect.

Geyer's simple theory can have no pretension of describing accurately a complex transient phenomenon as the sudden evacuation of a cylinder. Yet it not only gives an acceptable mental picture of the mechanism of the Kadenacy effect, but even gives tolerable agreement with observed results. Geyer has applied his calculations to Davies' experiments; Fig. 13–2 shows the comparison with the Penn State experiments, which in case of a full opening is remarkably close.

13.4 Slow Opening.

The upper two curves in Fig. 13–2 refer to the case when the entire end of the cylinder is suddenly removed. If the uncovered opening is smaller, both theory and experiment gave considerably smaller depressions. Figure 13–3 shows the resulting depressions when a 6-inch diameter cylinder containing air of 5 atmospheres pressure is evacuated by the instantaneous opening of an orifice to the atmosphere. It will be noticed that if the orifice diameter is one-half that of the cylinder (that means the discharge area one-quarter of the cylinder cross-sectional area), the resulting depression is less than 1 psi. This refers to *instantaneous* opening. Naturally, if the opening is less sudden, the resulting depression is still less. The Kadenacy effect practically vanishes if a relatively small discharge opening is uncovered relatively slowly. The inference is that the **full exploitation of the Kadenacy effect is obviated by our inability to open the exhaust ports or valves rapidly enough.**

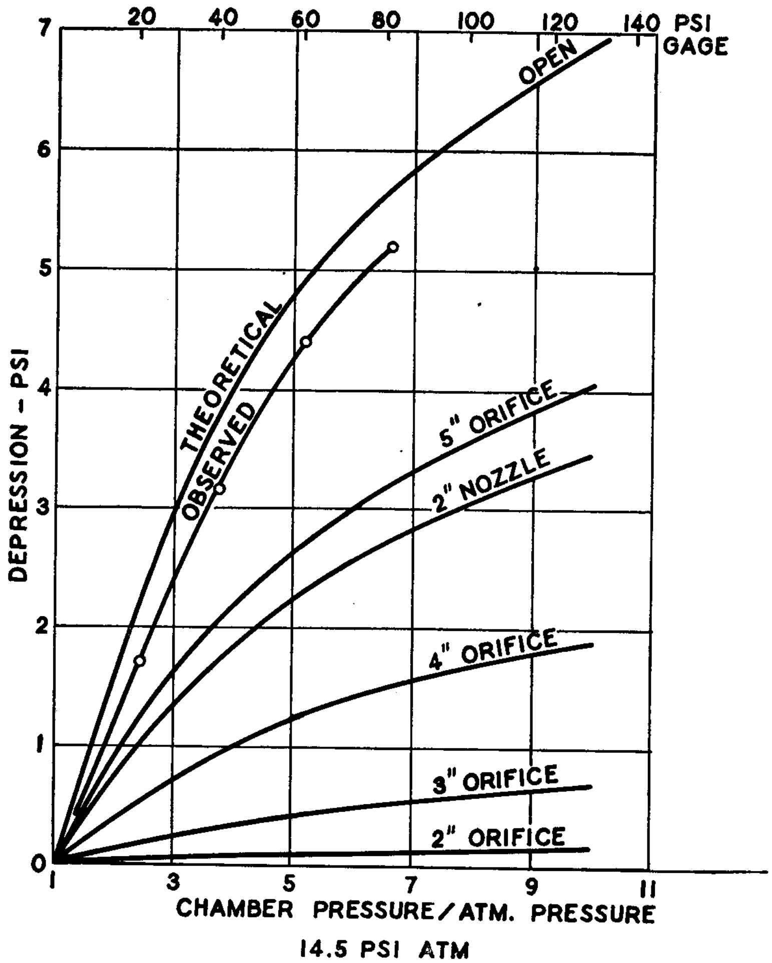

Fig. 13-2. Geyer's Theory Applied to Penn State Experiments.

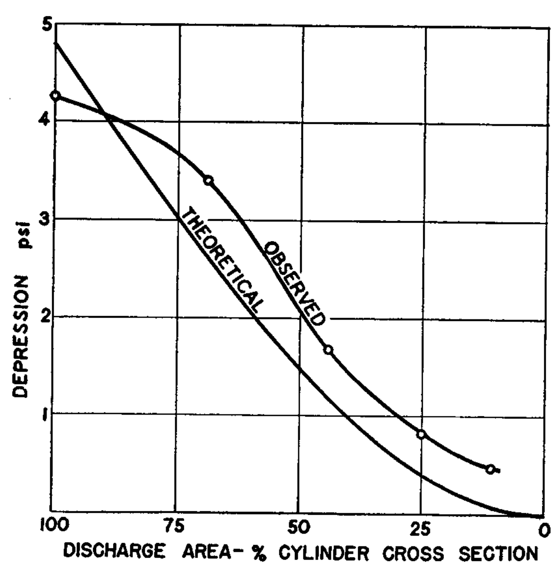

Fig. 13-3. Effect of Size of Opening on Kadenacy Effect. Chamber pressure 5 atmospheres.

13.5 Tuned Exhaust.

The Kadenacy effect can, however, be assisted powerfully by a properly tuned exhaust pipe. The kinetic energy of the gas contained in the exhaust pipe far exceeds the kinetic energy contained in the gas contained by the cylinder at the time of pressure equalization.

By a combination of the Kadenacy effect and a tuned exhaust pipe, the scavenging of any two-stroke cycle engine can be materially improved. Under favorable circumstances an engine can be made to run without any blower, but all engines *presently* manufactured under Kadenacy patents are built with blowers. A multicylinder engine which is to operate at variable speed does not lend itself well to blowerless operation because *with an exhaust header* the effect of tuning is less pronounced and at the wrong engine speed it is adverse. Even if a blower is used, the Kadenacy system offers decided advantages. The engine takes more air with a lower scavenge pressure and significantly the scavenge pressure decreases as the load increases. This is explained by the fact that as the load becomes greater the energy in the exhaust also becomes greater and by its aspiraling effect reduces the resistance to the delivery of the air from the blower. This benefits the engine in two ways: less power is used to drive the blower, and the increased air charge permits injection of more fuel with clear exhaust. In this way both the power output and the fuel economy are improved. Ordinarily, lower exhaust temperature also results due to the increased amount of short circuited air, and to the elimination of the temperature rise to which the air is subjected in a blower.

Although the Kadenacy effect, in the strict sense, and pipe tuning are in principle two different phenomena, they are related in nature—both are based on gas inertia—and their effects are almost indistinguishable from each other. A good way to determine whether an engine has or has not the benefit of the Kadenacy effect is to measure and plot the scavenge pressure against the engine speed while motoring and at various loads (fixed fuel rack settings). If the air-box pressure is generally less at load than at motoring and less at heavy load than at light load, Kadenacy effect is responsible. If the Kadenacy effect is less pronounced and/or the exhaust back pressure is considerable (small and/or long exhaust pipe), the air-box pressure will begin to increase at an intermediate load. If the Kadenacy effect is very pronounced, the air-box pressure will be minimum when the load is maximum. These tests are described in more detail in Chapter 16.

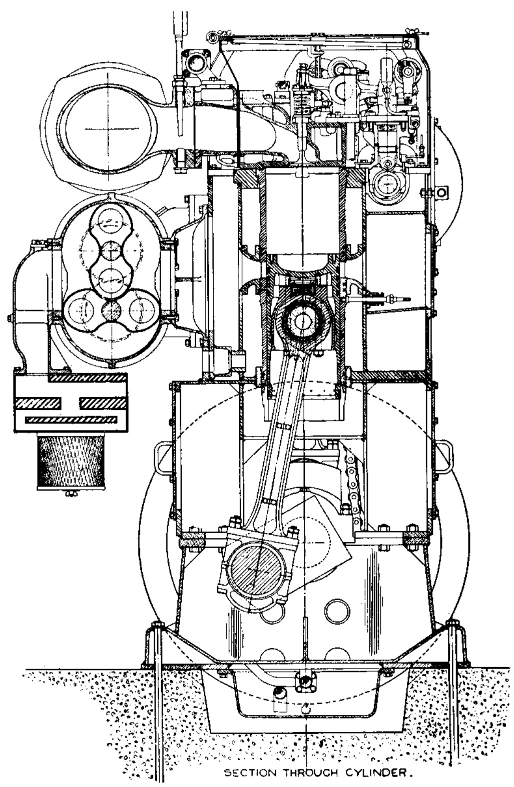

Fig. 13-4. Petter Superscavenge Engine. (By permission of Petters Ltd.)

CONSTRUCTIONAL FEATURES OF KADENACY ENGINES

13.6 Petter Engine.

While results obtained with the Kadenacy system are fully discussed in the literature [Dale, 1945], few of the constructional details have been disclosed. Figure 13-4 shows the longitudinal cross section of a modern Kadenacy type engine, the Petter Superscavenge Engine. It has $8\frac{1}{2}$-inch bore, 13-inch stroke, and normally operates at 600 rpm. It is built with 2, 3, 4, and 6 cylinders. It is rated at 85.88 psi bmep at which load the exhaust is said to be absolutely invisible. All engines carry the British Standard 10 per cent overload for one hour, and also a peak torque corresponding to 110 psi bmep or 30 per cent overload for considerable periods. The specific fuel consumption at rated load and speed is 0.367 pounds per bhp-hr with fuel of 19300 Btu heating value and the lubricating oil consumption less than $1\frac{1}{2}$ per cent of the fuel. The blower pressure used is only 2.8 psig at 600 rpm.

An examination of the section shown in Fig. 13–4 reveals a more or less conventional engine. Neither do the drawings and photographs of the components show very unusual features. Both the

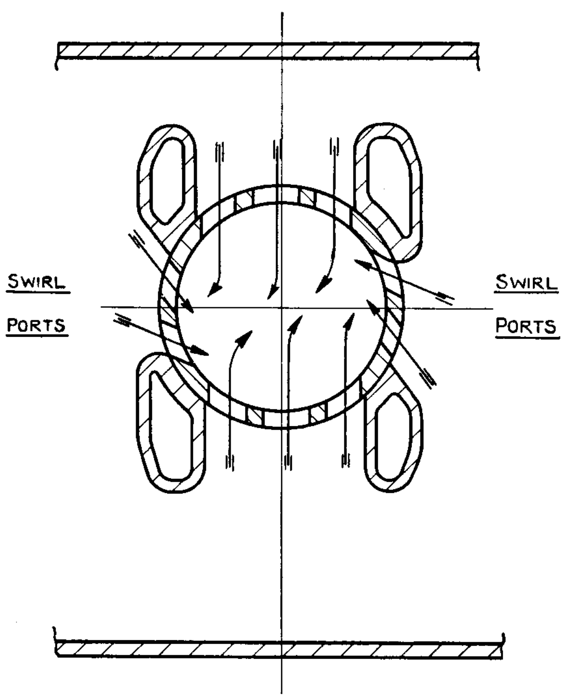

Fig. 13-5. Inlet Ports of Petter Superscavenge Engine. In this section through cylinder and liner arrows indicate the believed air flow causing rotation of charge.

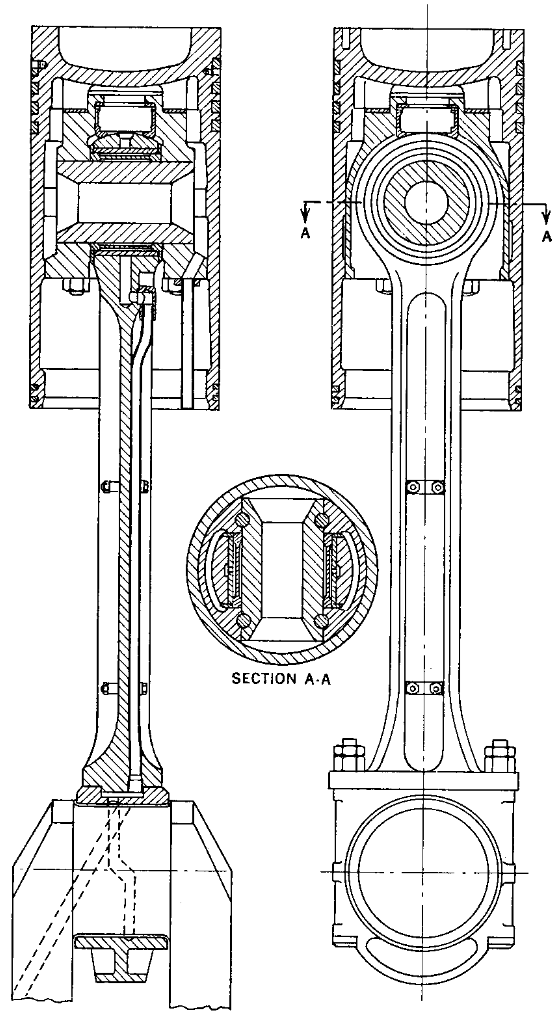

Fig. 13-6. Assembly of Piston and Connecting Rod Petter Superscavenge Engine.

inlet and exhaust ducts are carefully streamlined. The exhaust duct is divergent up to the exhaust header. Roots blowers are mounted on the side of the engine, each serving two cylinders. The inlet ports are made with various angles as shown in Fig. 13–5 which is claimed to cause the entire cylinder charge to rotate without leaving a stationary core in the center.

The exhaust is released through two valves in the head of rather large diameter and high lift but otherwise conventional.

The piston shown in detail in Fig. 13–6 is oil-cooled, of aluminum alloy, with fixed wrist pin clamped in an aluminum insert bolted to the piston body which is without the usual holes. Oil is fed

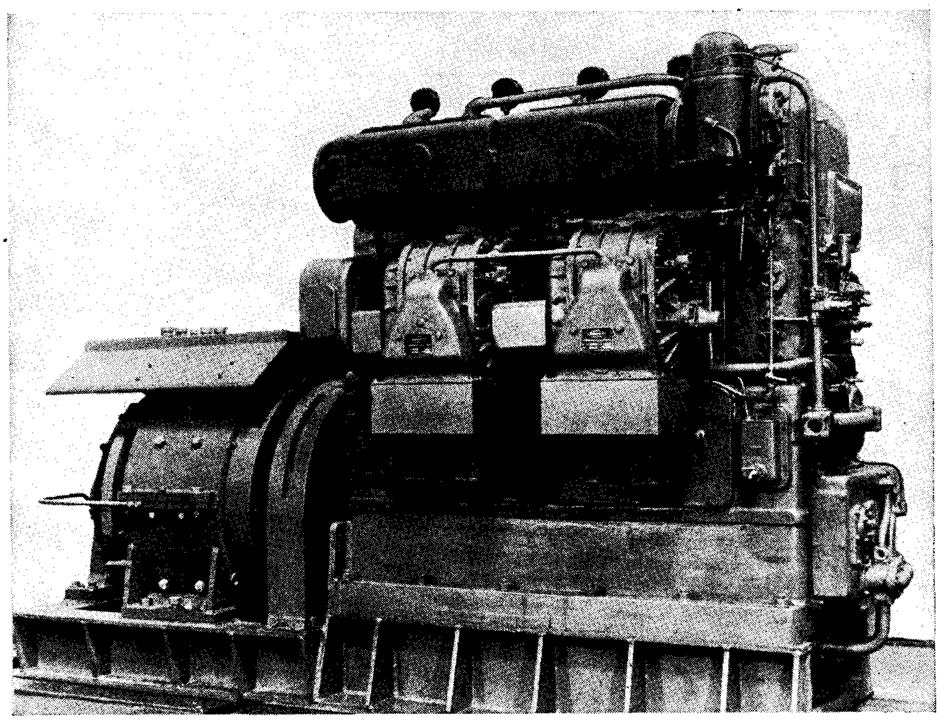

Fig. 13-7. Back View of a Four-Cylinder Petter Superscavenge Engine.

from the rod by a pipe from the crank pin bearing to the bushing of plastic bronze which is free to turn between the wrist pin and the carbon-chromium steel bushing fitted to the eye of the connecting rod. The piston-cooling oil passes from the channels in the bushing and the rod to the crown through a sealing piston with a spherical seat at the top of the rod, to which it is spring-loaded.

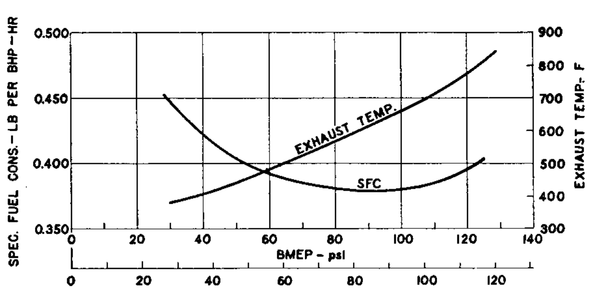

Fig. 13-8. Specific Fuel Consumption and Exhaust Temperatures of Petter Superscavenge Engine.

A back view of a four-cylinder auxiliary generator set is shown in Fig. 13–7 and Fig. 13–8 shows the specific fuel consumption and exhaust temperatures of this engine plotted against load.

Figure 13–9 shows a weak-spring indicator diagram of the engine, showing the slight dip below the atmospheric line after the completion of the blowdown period, which is characteristic of the Kadenacy system. This dip is much more pronounced in the diagrams of blowerless types of Kadenacy engines some of which are shown by Davies.

The inlet duration is 100 degrees and the exhaust lead 21 degrees.

DESIGNING FOR KADENACY EFFECT

13.6a It has been fairly well established that the Kadenacy effect cannot be dismissed as fictitious as it is based upon a demonstrable phenomenon: the rarefaction that follows sudden discharge of compressed gas from a closed vessel. Both thermodynamic theory and engines built according to the Kadenacy system prove the soundness of this principle.

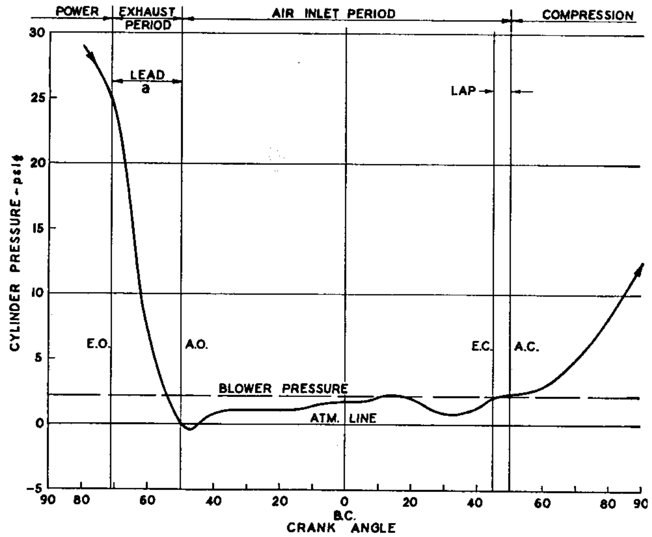

Fig. 13-9. Weak-Spring Indicator Diagram of Petter Superscavenge Engine. (B. M. Dale, "Recent Development in Medium-Speed Two-Cycle Oil Engines," *Motor Ship*, **25,** No. 300, Jan., 1945.)

The Kadenacy effect is not due to the inertia of the gas column in the exhaust pipe but to the inertia of the gas column in the cylinder, and arises therefore with any length of exhaust pipe if conditions are favorable. It is true, however, that gas inertia in the exhaust pipe can give powerful assistance to the Kadenacy effect if the pipe is properly tuned and destroy it if it is not.

All well-designed two-stroke cycle engines are benefited more or less by the Kadenacy effect, but in order to take full advantage of it, the following five points deserve particular consideration: (1) proper exhaust lead, (2) rapid exhaust opening, (3) streamlining of flow passages, (4) exhaust tuning, (5) elimination of reverse flow.

13.7 Proper Exhaust Lead.

The exhaust lead must be long enough to permit the cylinder gas to discharge from the cylinder and create a depression in its wake before much inlet air is admitted, but not so long that a return wave from the exhaust pipe would reach the exhaust orifice while it is still open and readmit the burnt gases to the cylinder.

For the calculation of the exhaust lead Kadenacy gives the following formula [U. S. Patent No. 2,144,065]

$$\frac{W}{100K\,Av} = \frac{a}{360N} \qquad (13\text{–}1)$$

where W is the cylinder volume in cubic centimeters, A the area of exhaust lead in square centimeters, v a *hypothetical* mean velocity of mass exit of the burnt gases of the order of 450 meters per second, K a constant depending upon the form of the exhaust orifice and the area opened per unit movement of the piston or crankshaft, a the angle of exhaust lead in degrees, and N the speed of the engine in revolutions per second. With the symbols used in this book

$$KA = A_m$$

$$a = \alpha$$

$$N = \frac{n}{60}$$

and

$$W = V_e$$

equation (13–1) can be written as

$$A_m\alpha = \frac{6n}{100}\frac{V_e}{v} \text{ cm}^2 \text{ deg.} \tag{13–2}$$

For the *hypothetical mean velocity of mass exit* Kadenacy recommends 450 meters per second but also mentions that it was found in practice that it should be in the neighborhood of 500 meters per second. Taking $v = 475$ meters per second equation (13–2) becomes

$$A_m\alpha = \frac{V_e n}{7900} \text{ sq cm deg}$$

or in English units

$$A_m\alpha = \frac{(2.54)^3}{(2.54)^2}\frac{V_e n}{7900} = 0.00032\ V_e n \text{ sq in. deg.} \tag{13–3}$$

This is practically identical with equation (8–1) which has been derived in the appendix of Chapter 8, without any assumption of *ballistic* velocities.

The proper exhaust lead for Kadenacy effect can therefore be calculated by the method described in 8.2.

According to either equation the required blowdown time area is proportional to the engine speed; therefore the *minimum* exhaust lead required to insure evacuation of the cylinder increases with the speed.

The time required for the return wave to reach the cylinder is approximately constant in terms of seconds but in terms of crank degrees it also increases directly with the engine speed.

It is seen that both the minimum and the maximum exhaust lead increase with the engine speed. In a variable-speed engine optimum exhaust lead can be designed for only one given speed. If the engine speed much exceeds the design speed, the blowdown will be incomplete and the Kadenacy effect is largely lost. If the engine speed is much lower than the design speed, the return waves from the exhaust pipe find the exhaust orifice still open and fill the cylinder with exhaust gases before fresh air has an opportunity to get in.

Kadenacy's U. S. Patent No. 2,113,480, pictured in Fig. 13–10, shows a mechanical arrangement for varying the exhaust lead to suit the engine speed. No such engine is known to have been built.

It may be concluded that the Kadenacy effect can best be applied to a constant-speed engine. In a variable-speed engine the exhaust lead must be designed for the preferred engine speed, and the calculation then is identical with the one described in Chapter 8. It requires, however, skill in design to insure adequate exhaust lead in a high-speed engine without unduly crowding the inlet period.

13.8 Rapid Exhaust Opening.

In all but very slow-speed engines rapid opening of the exhaust is desirable in order to provide adequate blowdown area without unduly reducing the duration of the inlet. With the Kadenacy system rapid opening of the exhaust is not only desirable but essential because the Kadenacy effect vanishes if the discharge opening is not fast enough.

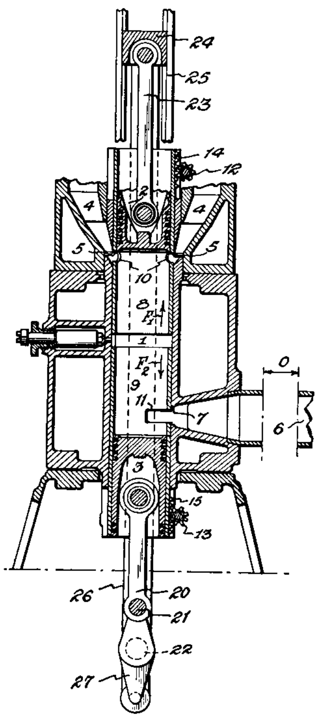

Fig. 13-10. U. S. Patent No. 2,113,480, M. Kadenacy. By pinions 12 and 13, and racks 14 and 15 the exhaust lead is so varied that the desired timing is obtained for varying engine speeds.

For the speed of opening the exhaust Kadenacy fails to give definite rules, but indicates (U. S. Patents No. 2,144,065 and 2,123,569) that the *critical exhaust lead* should be less than 1/300 second and preferably as low as 1/500 second. The critical exhaust area is given as approximately one-quarter of the piston area and should be uncovered in less time than that required for the blowdown period. The Penn State tests on the other hand have shown that the larger the suddenly opened exhaust orifice the greater is the ensuing depression. By reducing the exhaust opening from 100 per cent to 45 per cent piston area, the depression at 5 atmospheres chamber pressure dropped from 4.3 to 1.7 psig — that means to 40 per cent—and at 25 per cent piston area the depression was only 0.85 psig, or only 20 per cent of the maximum. It is obvious that a rapid opening is essential for the Kadenacy effect and the more rapid the opening the better the effect.

13.9 Mechanical Difficulties.

The mechanical difficulties involved in producing a very sudden exhaust opening are, however, considerable. Instantaneous opening of a large area in an engine cylinder cannot be realized or even closely approached by practical means. The uncovering of as much as one-half of the piston area is out of the question and even the opening of a smaller area involves appreciable time.

To satisfy Kadenacy's rather limited objective, one-quarter of the piston area must be uncovered during the exhaust lead. Piston-controlled ports are considered first. In an opposed-piston engine the rectangular exhaust ports may occupy as a maximum 75 per cent of the circumference. If the height of that portion of the exhaust ports which is uncovered before the inlet ports open is designated h_b,

$$0.75 D \pi h_b \doteq 0.25 D^2 \frac{\pi}{4}.$$

Assuming a stroke length of

$$s = 1.2D$$

$$D = \frac{s}{1.2}$$

$$h_b = \frac{D}{12} = \frac{s}{12 \times 1.2} = \frac{s}{14.4} = 0.07s.$$

An inlet duration of 100 degrees crank angle is assumed. From Fig. 7–5 this corresponds to 14 per cent stroke port height. Adding to this for h_b, 7 per cent stroke, 21 per cent stroke is shown. This would make the exhaust ports open at 119 degrees after top center, which is acceptable.

The exhaust lead consisting of $180 - \frac{100}{2} - 119 = 11$ degrees must be covered in less than, approximately, 1/300 second. From the relation

$$11 = 6n \times \frac{1}{300}$$

the engine speed must be higher than 550 rpm.

Considering next a cross- or loop-scavenged engine, the exhaust ports cannot occupy more than 30 per cent of the circumference and h_b must satisfy the following requirement:

$$0.3D\pi h_b = 0.25D^2\frac{\pi}{4}$$

which, with $s = 1.2D$,

$$h_b = \frac{s}{(0.3) \times (16) \times (1.2)} = 0.173s$$

or 17.3 per cent of the stroke. With 100 degrees inlet duration the upper edge of the exhaust port would be at 31.3 per cent from the bottom of the stroke, which makes the exhaust ports open at 105 degrees after top center, which is somewhat too early for a port-scavenged engine, but still acceptable.

The exhaust lead now is $130 - 105 = 25$ degrees, and this must be covered in not more than 1/300 second, which means that the engine speed must be higher than 1250 rpm. This would indicate that the Kadenacy system is not suited for cross- or loop-scavenged engines.

The final example is a cylinder with 2 poppet exhaust valves, the diameters of which are 40 per cent of the piston diameter. In order to provide a valve opening equal to one-quarter of the piston area

$$2 \times 0.4\,D\pi \times l_c = \frac{1}{4}\frac{D^2\pi}{4}$$

from which the *critical* valve lift is

$$l_c = \frac{D}{16}\frac{1}{0.8} = 0.078D$$

Although this is a fairly high lift, it is not the total lift required, but only that portion of it which corresponds to the exhaust lead period. In order to provide the necessary deceleration, the entire valve lift may have to be more than this.

The critical valve lift must be completed in less than 1/300 second. With a 6-inch bore cylinder the valves must travel 0.48 inch in 0.0025 second. Starting from standstill and assuming uniform acceleration

$$0.48 = \tfrac{1}{2}a(0.0025)^2$$

from which

$$a = 154{,}000 \text{ in./sec}^2 = 396g$$

which is very high indeed.

In ordinary valve gears, accelerations do not reach more than a fraction of this figure and even in engines built after the Kadenacy principle the valve accelerations and consequently the critical valve lifts are considerably below the minimum required by Kadenacy's patents. Instead of 1/500 to 1/300 second, the periods of critical exhaust opening vary between 1/100 and 1/200 second and even these figures require extraordinary valve gear design.

13.10 Masked Valve.

Various expedients are being used to avoid the enormous valve accelerations, and correspondingly enormous spring loads. One of them is the use of *masked* valves.

If the velocity of the valve is zero at the beginning, the opening is very small for the first 20 degrees. By recesssing the valve seat in such a manner that the outer diameter of the valve constitutes an easy fit in the recess, with a diametral clearance of approximately 0.030 inch, the valve acts as a piston valve and it is possible to start the motion earlier. Such a valve is shown in Fig. 13–11. The effective exhaust begins only when the valve head clears the recess. In this way the effective opening of the exhaust may remain the same but the time available is increased. In this way the acceleration can be greatly reduced.

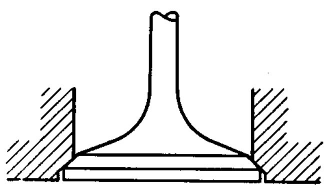

Fig. 13-11. Masked Valve.

A disadvantage of the masked valve is that the total valve lift is still greater. Another disadvantage is that the hot gases rushing through the small annular area between the cylindrical part of the valve and the recess are pretty hard on the seat.

Other expedients, such as piston valves and combination valves, have also been proposed for keeping the accelerations within bounds. If high accelerations are unavoidable, recourse is taken to unconventional springs such as multiple-coil, hairpin, and torsion-rod types to provide the very high spring load required for handling the high accelerations.

It is evident from the foregoing that the task of the designer to provide exhaust openings large enough and fast enough for the effective utilization of the Kadenacy effect is a difficult one. It may even be said that this difficulty constitutes the greatest obstacle to full exploitation of the Kadenacy effect. However, even a partial utilization of the Kadenacy effect is worth while, especially with blower-scavenged engines. Therefore, the above-cited figures should not be interpreted as a deterrent to any effort directed to the utilization of the Kadenacy effect.

13.11 Streamlining of Flow Passages.

It is natural that when the flow of air is controlled by low pressure differences, as is the case with the Kadenacy system, conservation of the kinetic energy in the stream is very important. Both the exhaust and inlet passages must be *streamlined* by providing ample cross-sectional area, avoiding sharp turns, protrusions, and other eddy-producing shapes. For inlet ducts and ports slightly convergent shapes are chosen to provide guidance for the air in the cylinder. For exhaust ducts slightly divergent shapes give good results because of the aspirator effect of the divergent tubes. Figure 13–4 shows an example of good streamlined duct forms.

For exhaust, tapered pipes have also been used, with an included angle of a little over 1 degree.

13.12 Exhaust Tuning.

Successful application of the Kadenacy system is greatly promoted by tuning the exhaust pipe. This is particularly true of the blowerless type of Kadenacy engine, where exhaust tuning can be of decisive importance. In multicylinder engines, with more than three cylinders, divided exhaust lines are generally used to avoid interference among the cylinders and to control better the pressure pulsations in the exhaust ducts.

For proper tuning the analytical methods described in Chapter 12 may be used and should be supplemented by the experimental methods described in Chapter 16. Tuning of a Kadenacy type engine does not differ from that of a conventional engine, except that with the former, exhaust tuning is more essential than with engines that depend on relatively high scavenge pressures for charging.

The necessity of proper exhaust tuning again tends to make the Kadenacy engine a single-speed machine. Naturally the design speed for port timing must agree with that of the piping. Operation at other speeds may be possible but less good performance must be reckoned with.

13.13 Elimination of Reverse Flow.

It has been pointed out that the Kadenacy effect can easily be destroyed by a reverse flow generated by a reflected pressure wave from the end of the exhaust pipe or from a receiver in the exhaust line. In such a case the charging of the cylinder with fresh air is spoiled by the return of burnt gases which mix with the fresh air and/or prevent its entry into the cylinder.

This is likely to happen if the exhaust pipe is so short that the return wave finds the exhaust orifice still open. To eliminate this contingency the relation of the natural frequency of the pressure wave and the engine speed must be such that the period of the exhaust pressure wave shall be longer than the duration of the exhaust opening. Methods for accomplishing this are described in Chapter 12.

Neither is it desirable to have the depression created by the exhaust column persist so long that it sucks out of the cylinder whatever fresh air has been admitted. Such effect would result from too long an exhaust pipe.

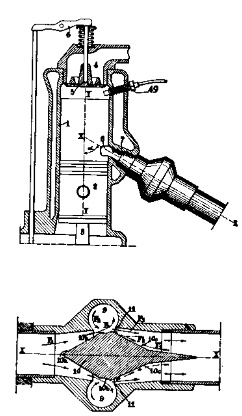

Fig. 13-12. U. S. Patent No. 2,110,986, M. Kadenacy. Outward flow at the exhaust gases obstructed, flow stopped by a "plug" of whirling gases in toroidal cavity 9.

Good results are obtained if the exhaust pipe is so tuned that the period of one complete oscillation approximately equals the duration of the exhaust, as was explained. But this is possible only for one speed. The question is how to make the Kadenacy system operable at variable speeds without contending with the obstruction or contamination caused by return of the burnt gases at certain engine speeds.

Solutions have been offered toward converting the exhaust duct into a one-way thoroughfare. The outwardly tapered exhaust pipe mentioned above may be considered such a device inasmuch as it discourages flow in the converging direction.

A more positive action could be expected from a nonreturn valve which would allow free passage to the outward flow of the exhaust gases but would prevent by its automatic closure any return flow. The objections to such valves so far proposed are of mechanical nature. The speed of operation, the high temperature of the exhaust gases and the smoke and soot which they contain make very difficult conditions for a valve to operate under. Therefore exhaust devices involving moving parts met with little success so far.

Kadenacy, in his numerous patents, proposes various stationary devices to accomplish such purpose. Figure 13–12 shows a deflector device to be interposed between the exhaust duct and manifold, that is supposed to offer little resistance to the outward flow of the exhaust gas. The return flow, however, caused by the rebound of the gas column from the end of the pipe, encounters a gaseous *plug* formed by the whirling gases in the toroidal cavity 9, located as close to the cylinder as possible.

Somewhat similar devices are shown in Fig. 13–13 and 13–14.

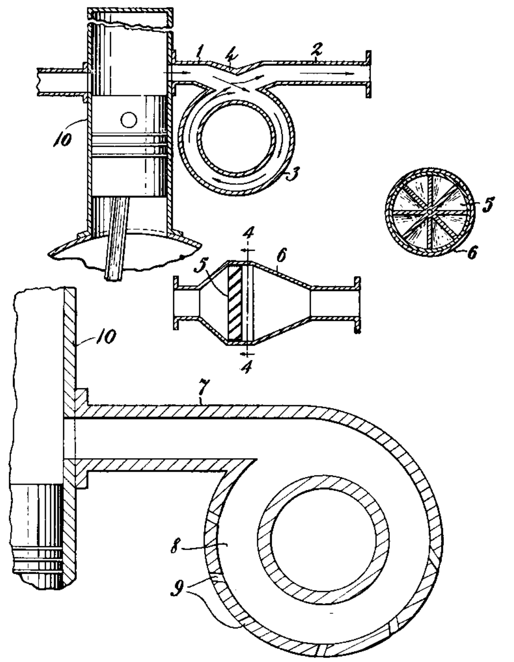

Fig. 13-13. U. S. Patent No. 2,167,303, M. Kadenacy. Means to minimize the rebound of the exhaust gas column by prolonging its outward motion and delaying its reversal and/or reducing its intensity.

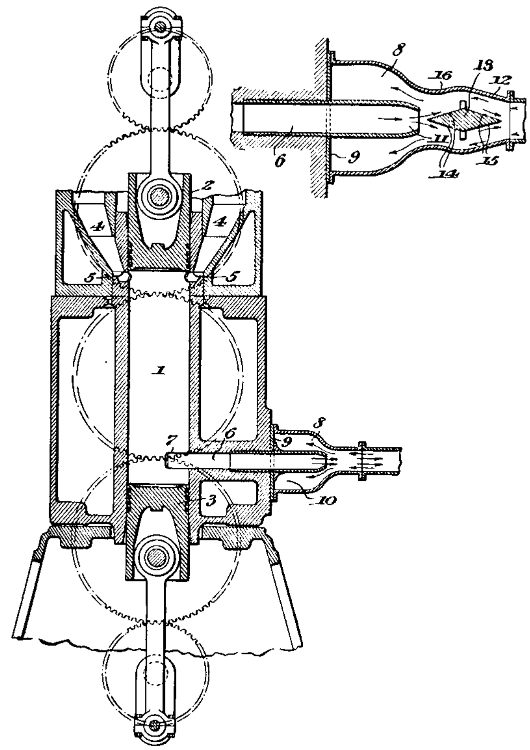

Fig. 13-14. U. S. Patent No. 2,168,528, M. Kadenacy. Means to prevent a too early return of the exhaust gas column that would reach the cylinder when the charging process is still in progress.

Reports of the successes of the above-described devices are not sufficiently conclusive to have them recorded.

CHAPTER 14

A COMPLETE PORTING CALCULATION

14.1 This chapter is devoted to the complete porting calculation of a simple port-scavenged engine. The only difference between these calculations and those presented in the preceding chapters is that here they are dovetailed into each other and integrated into a whole.

The scavenging system of an engine begins with the air intake bell or silencer and terminates at the exhaust stack. It is a chain in which all components, such as silencer, blower, pipes, manifolds, intake and exhaust ports and ducts, and exhaust muffler interact with each other. Strictly speaking the dimensioning of one component should influence the dimensioning of all. The difficulty of the analysis prevents, however, simultaneous consideration of the complete system. Nevertheless the interaction of such closely related components as cylinder and blower cannot be ignored. A numerical example serves to describe the recommended procedure in detail. The example is a simplified version of an actual porting calculation worked through by the author.

Cylinder bore 20.5 in.
Stroke 27.5 in.
Length of connecting-rod 62 in.
Engine speed 240 rpm
Blower: Roots two-lobe, 10 in. by 34 in.
One for each two cylinders

The above data are considered as fixed and the engine is to be a simple heavy-duty, port-controlled, loop-scavenged engine with a fair output and good fuel economy.

PRELIMINARY CALCULATION

14.2 As a first step a preliminary calculation is made. In view of the fact that a large engine is being dealt with and aiming at low fuel consumption the delivery ratio is set at 1.3 (see section 7.3). Next an estimate is made of the width of the ports.

As the use of water-cooled port bridges is contemplated that would take approximately 30 per cent of the periphery, 70 per cent of the periphery is left for ports. In using Curtis-type loop scavenging, a suitable division between inlet and exhaust port width is 4:3, which allows 40 per cent of the periphery for the inlet ports and 30 per cent for the exhaust ports.

14.2a Inlet Ports.

The approximate scavenge pressure is selected on the basis of Fig. 7–1. With

$$\frac{Dn}{\beta} = \frac{20.5 \times 240}{0.4} = 12400$$

the tentatively selected scavenge pressure is

$$p_{sc} = 2.75 \text{ psig.}$$

Estimating the blower discharge temperature at 125 F, with a pressure ratio of

$$\frac{14.5 + 2.75}{14.5} = 1.197,$$

from Fig. 7–2

$$\frac{w_i}{S} = 555 \text{ ft per sec.}$$

In view of the intention to use cast ports with rounded edges, from Table 7–I the scavenge factor is selected as $S = 0.4$, which makes

$$w_i = 222 \text{ ft per sec.}$$

The mean inlet velocity being known, the required inlet time-area is calculated by equation (7–1)

$$A_{im}\alpha_i = \frac{L}{2w_i} V_{disp}n$$

$$= \frac{1.3}{2 \times 222} \frac{20.5^2\pi}{4} 27.5 \times 240$$

$$= 6480 \text{ sq in.-deg.}$$

The total width of the rectangular inlet ports is

$$b_i = 0.4 \times 20.5 \times \pi = 25.7 \text{ in.}$$

and the required amount of per cent-degrees from equation (7–4) is

$$F_i = \frac{6480}{25.7 \times 27.5} = 9.18 \text{ per cent deg.}$$

The corresponding port height is read from Fig. 7–5 with 4.5-to-1 connecting rod-crank ratio as

$$h_i = 14 \text{ per cent stroke} = 3.85 \text{ in.}$$

and an inlet duration of

$$\alpha_i = 98 \text{ deg.}$$

14.3 Exhaust Ports.

The necessary minimum blowdown time-area is calculated from equation (8–4)

$$A_m\alpha = 0.00023\, V_{disp}n = 0.00023 \times 9077 \times 240 = 500 \text{ sq in.-deg.}$$

The desirable blowdown time-area from equation (8–2) is

$$A_m\alpha = \frac{0.98}{\sqrt{RT}} V_e ny$$

where

$$R = 53 \times 12 = 636$$
$$T = 1710R$$
$$V_e = 9077\left(1 + \frac{1}{16 - 1}\right) = 9170$$

Expansion end pressure from Fig. 8–2 with an indicated mep of 80 psi

$$p_e = 49 \text{ psi}$$

and

$$\frac{p_e}{p_i} = \frac{49}{14.5 + 2.75} = 2.84$$
$$y = 0.42$$
$$A_m\alpha = \frac{0.98}{1045} 9170 \times 240 \times 0.42 = 876 \text{ sq in.-deg.}$$

A tentative compromise on a blowdown time-area is

$$A_m\alpha = 600 \text{ sq in.-deg.}$$

The total width of the exhaust ports is

$$b_e = 0.3 \times 20.5 \times \pi = 19.33 \text{ in.}$$

Therefore, 600 sq in.-deg corresponds to

$$\frac{600}{19.33 \times 27.5} = 1.127 \text{ per cent deg,}$$

which gives from Fig. 9–1 an exhaust lead of

$$\alpha = 18.5 \text{ deg.}$$

This makes the exhaust port open at

$$180 - \frac{98}{2} - 18.5 = 112.5 \text{ deg after top center}$$

and the height of the exhaust port upper edge (from Fig. 7–5) 26.35 per cent stroke which corresponds to

$$0.2635 \times 27.5 = 7.25 \text{ in.}$$

or 3.4 inches above the upper edge of the inlet ports.

With these data the preliminary port time-area diagram looks like Fig. 14–1.

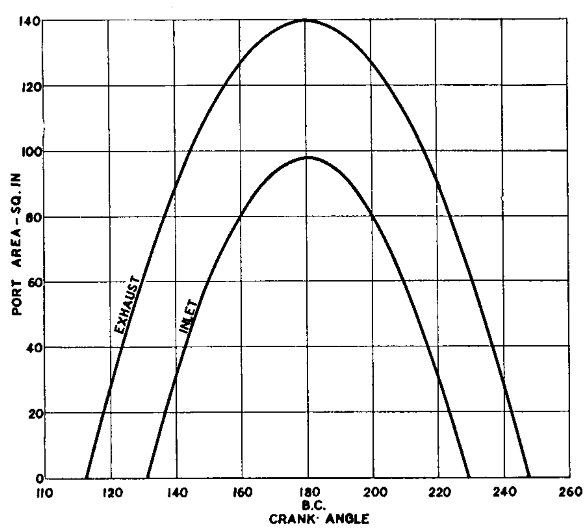

Fig. 14-1. Preliminary Port Area Diagram.

14.4 Blower.

The next step is to select the blower speed or blower-crankshaft gear ratio.

The blower must deliver approximately

$$\frac{V_{de}n}{1728} = \frac{1.3 \times 9077 \times 244}{1728} = 1665 \text{ cfm}$$

of free air against 2.75 psig back pressure.

Testing the blower at 1100 rpm blower speed for delivery and horsepower input resulted in the curves shown in Fig. 14–2. Since neither the volumetric efficiency nor the adiabatic efficiency changes appreciably with a small change in blower speed, both the air delivery and the horsepower input can be considered proportional to the blower speed within a limited range.

Against a discharge pressure of 2.75 psig the curves give an air delivery of

$$\text{Air delivery} = 1690 \text{ cfm}$$

and a horsepower input of

$$\text{Input} = 29.25 \text{ hp.}$$

In order to get 1665 cfm of air the blower speed must be

$$n_b = 1100 \frac{1665}{1690} = 1083 \text{ rpm,}$$

which calls for a gear ratio of

$$\frac{1083}{244} = 4.45 : 1$$

from the crankshaft to the blower.

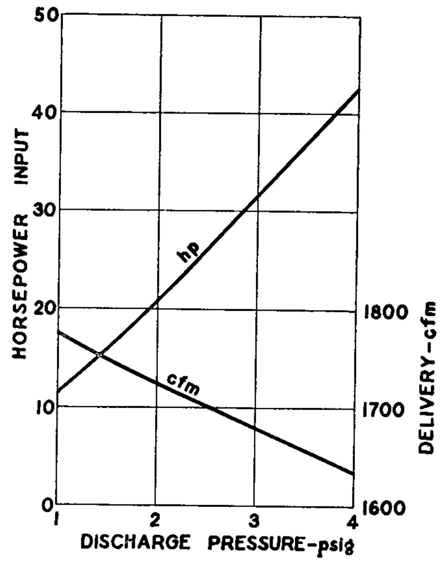

Fig. 14-2. Roots Blower Performance at 1100 Rpm.

The horsepower input is

$$\frac{1083}{1100} 29.25 = 28.8 \text{ hp,}$$

which corresponds to a blower mep

$$p_b = \frac{396000 \times 28.8}{9077 \times 244} = 5.15 \text{ psi}$$

which is an acceptable value.

14.5 Air-Box.

Continuing the preliminary calculations, the necessary receiver or air-box volume is determined. The engine is to be built with four or more cylinders, therefore, the air-box volume which is suitable for the four cylinder engine is suitable for all.

In order to keep pressure fluctuations within 10 per cent the receiver volume required is equal to that shown in Table 11–III which gives for four cylinders and 100 degree inlet duration, 0.6 × displacement volume, or

$$0.6 \times 4 \times 9077 = 21800 \text{ cu in.} = 12.6 \text{ cu ft.}$$

Theoretically that volume would suffice for four cylinders. This is, however, based on the assumption that any cylinder can draw on the entire total volume. Actually a cylinder can draw on only the volume which is adjacent to its own inlet ports because the rest of the receiver volume is too far distant to be of any help. Therefore, the air-box volume for each cylinder must not be less than 12.6 cubic feet.

14.6 Layout.

With these data a tentative layout of the cylinder block and liner is drawn up (see Fig. 14–3).

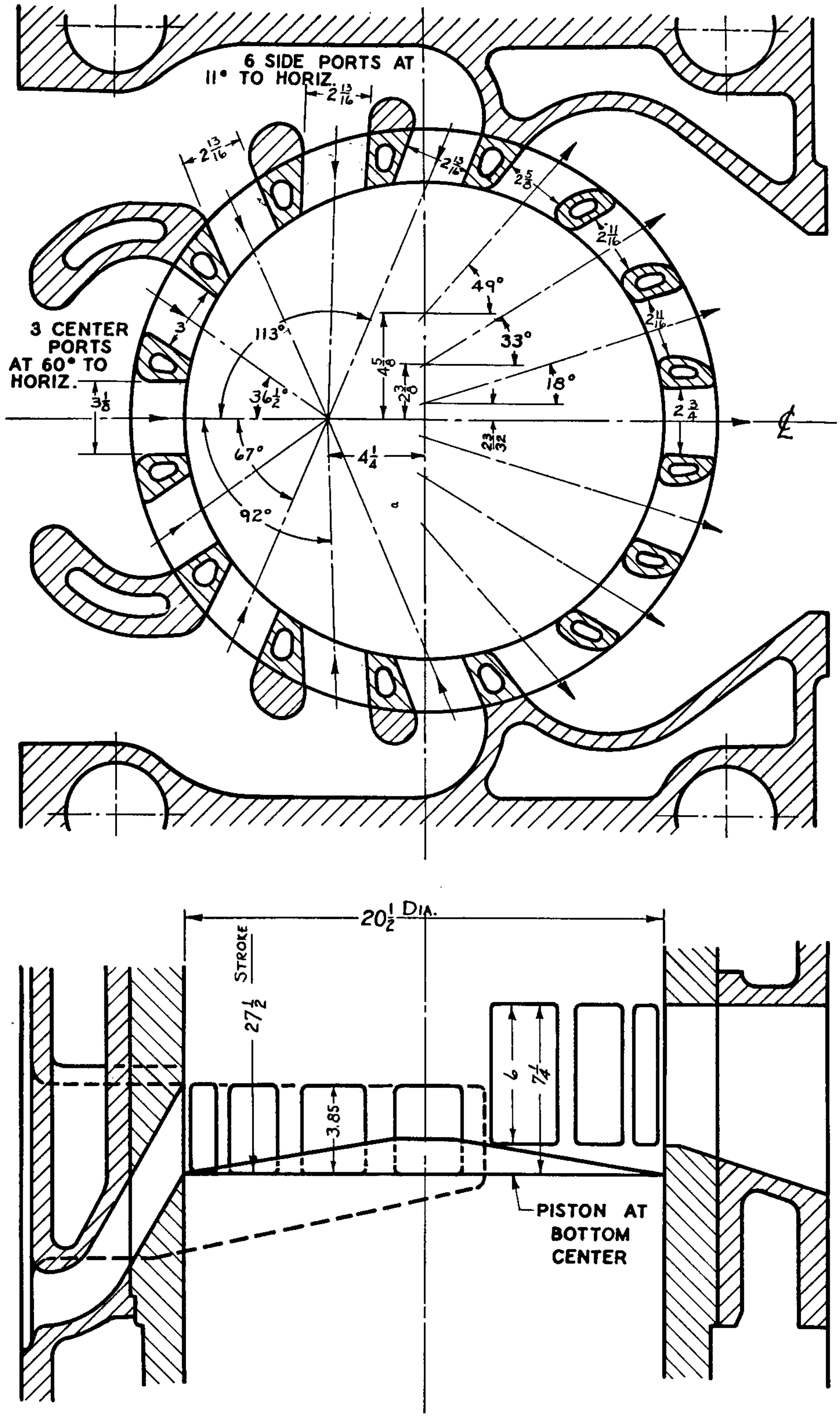

Fig. 14-3. Tentative Layout of Cylinder Block and Liner.

Sixteen ports are laid out, 9 inlet ports and 7 exhaust ports. In the horizontal cross section all ports appear to be directed to a point in the centerline $4\frac{1}{4}$ inches from the center. This is good practice in using Curtis-type scavenging. The width of the exhaust ports square to the flow are $2\frac{5}{8}$, $2\frac{11}{16}$, $2\frac{11}{16}$, $2\frac{3}{4}$, $2\frac{11}{16}$, $2\frac{11}{16}$, and $2\frac{5}{8}$ inches, which add up to $19\frac{3}{4}$ inches, which is slightly more than the assumed 19.33-inch value. With 6 inches port height the total exhaust port area is 116 square inches.

Of the 9 inlet ports the 6 side ports are made horizontal, while the 3 center ports are at 60 degrees to the horizontal. The peripheral width of all ports occupies an arc of 20 degrees, which still permits the use of nonpinned piston rings if the upper and lower edges are slightly relieved or if Gothic ports are used (see 6.7).

The square-to-the-flow widths of the inlet ports are $2\frac{13}{16}$, $2\frac{13}{16}$, $2\frac{13}{16}$, 3, $3\frac{1}{8}$, 3, $2\frac{13}{16}$, $2\frac{13}{16}$, $2\frac{13}{16}$ inches, which total $25\frac{7}{8}$ inches, and exceeds the assumed width of 25.7 inches.

For the height of the inlet ports the preliminary calculation gave 3.85 inches. The bottom edges of the inlet ports are made to coincide with the top edge of the piston at bottom center position. The top edges of the exhaust ports are 7.25 inches above the piston edge and 3.4 inches above the top edges of the inlet ports. The bottom edges of the exhaust ports may be set $1\frac{1}{4}$ inches higher than the bottom edges of the inlet ports, as that gives plenty of exhaust-port area.

COORDINATING CALCULATION

14.7 In the preceding calculation the inlet ports were determined independently of the blower and of the exhaust ports. The air flow through the inlet ports, however, is not independent of the blower and of the exhaust ports. Therefore the correct calculation must consider all of these together. Strictly speaking, the inlet and exhaust piping should also be considered but they are of such size that the pressure drops in them are negligible; therefore they are ignored.

The important part of the calculation is the determination of the optimum height of the inlet ports. It is realized that in making the ports shorter, the scavenge pressure increases, the air delivery decreases, the effective stroke increases, and the blower horsepower input also increases. By making the ports longer the opposite happens. The porting is to be designed for maximum net horsepower output.

It is also realized that as soon as a given blower has been decided upon, the port heights and the scavenge pressure no longer can be selected independently, because shorter ports automatically increase the scavenge pressure and vice versa.

The procedure to be followed envisages a number of possible inlet port heights, as 3.7, 3.85, 4, 4.15, 4.30, 4.45 and 4.60 inches. The amount of air delivered by the blower is always equal to the amount of air going through the cylinder. At the equilibrium scavenge pressure the blower delivery exactly equals the cylinder intake. For each port height that scavenge pressure must be found. Then for each port height the engine gross output, the blower input, and the net output of the engine must be found. The one which gives maximum net output is the optimum port height.

The calculation is carried out in several stages. The first stage concerns the blower.

14.8 Blower Calculation.

It is known that the correct scavenge pressure is close to 2.75 psig but may differ from it by a fraction of a pound. In Table 14–I are tabulated the blower performance characteristics for 1.75, 2.25, 2.75, 3.25, 3.75 psig discharge pressures.

Table 14-I. Blower Calculation.

10-inch by 34-inch Roots blower, 1083 rpm. One blower for two cylinders. The figures refer to one-half blower.

1. Discharge pressure (psig)	1.75	2.25	2.75	3.25	3.75
2. Blower delivery (cfm)	1710	1687	1665	1645	1616
3. Delivery ratio L	1.335	1.317	1.3	1.285	1.27
4. Blower input (hp)	18.70	23.62	28.8	34.2	41.85
5. Blower mep (psi)	3.39	4.26	5.22	6.23	7.62
6. Adiabatic horsepower	12.5	15.52	18.47	21.0	23.58
7. Adiabatic efficiency	0.67	0.66	0.64	0.61	0.57
8. Temperature rise, adiabatic (F)	15	20	25	30	35
9. Temperature rise, friction (F)	9	12.5	16	21	30
10. Total temperature rise (F)	24	32.5	41	51	65

Line 2 is read from Fig. 14–2 and corrected by a factor of 1083/1100 = 0.986.
Line 3 is line 2 divided by

$$\frac{V_{disp} \times \text{rpm}}{1728} = \frac{9077 \times 240}{1728} = 1261.$$

Line 4 is read from Fig. 14–2 and corrected by a factor of 0.986.

Line 5 is (line 4) $\times \dfrac{396000}{9077 \times 240} = 0.1818 \times$ hp.

Line 6 is read from Fig. 11–1 and multiplied by cfm, line 2.
Line 7 is obtained by dividing line 6 by line 4.
Line 8 is read from Fig. 11–21.
Line 9 is read from Fig. 11–21.
Line 10 is total of line 8 and line 9.

14.9 Inlet Velocities.

From Table 14–I the inlet velocities can be plotted. According to Fig. 7–2 they are determined by the scavenge pressure, scavenge air temperature, and scavenge factor. Estimating the last as 0.4, Table 14–II can be set up.

Table 14-II. Inlet Velocities.

11. Discharge pressure (psig)	1.75	2.25	2.75	3.25	3.75
12. Blower intake temperature (F)	100	100	100	100	100
13. Total temperature rise (F)	24	32.5	41	51	65
14. Blower discharge temperature (F)	124	132.5	141	151	165
15. Pressure ratio	1.11	1.156	1.19	1.223	1.26
16. $\frac{W_i}{S}$ (ft per sec)	411	484	532	573	618
17. Inlet velocity (ft per sec)	164.5	194	213	229	247.2

Line 11 is identical with line 1 of Table 14–I.

Line 12 represents the estimated temperature of the blower intake air.

Line 13 is identical with line 10, Table 14–I.

Line 14 is the total of line 12 and line 13.

Line 15 is obtained by (14.5 + line 11)/14.5.

Line 16 is read from Fig. 7–1 for values listed in lines 14 and 15.

Line 17 is obtained by multiplying line 16 by ($S = 0.4$).

In the next table the mean inlet velocities resulting from various inlet port heights are calculated.

Table 14-III. Inlet Ports.

18. Inlet port height (in.)		3.7	3.85	4.0	4.15	4.30	4.45	4.6
19. Total inlet port width					25.875 in.			
20. Max. port area (sq in.)		95.4	99.3	103.2	107.0	111.0	114.8	118.7
21. Rel. port height (% stroke)		13.45	14.00	14.55	15.09	15.62	16.18	16.72
22. Inlet duration (deg)		96	98	100	102	104	$105\frac{1}{2}$	107
23. F_i (% deg)		8.6	9.1	9.7	10.25	10.75	11.3	11.8
24. Inlet time area (sq in.-deg)		6120	6480	6900	7290	7650	8050	8410
25. Mean inlet velocity, ft per sec								
$P_{sc} = 1.75$	$L = 1.335$	237.5	224.5	211.0	199.7	190.0	180.7	173.2
2.25	1.317	234.0	221.5	208	197	187.5	178.0	171
2.75	1.3	231.2	218.6	205.2	194.5	185.2	176	168.6
3.25	1.285	228.4	216.0	203	192.2	183.0	173.7	166.7
3.75	1.27	226.0	213.8	201.5	190	180.8	171.9	164.7

In line 18, seven different inlet port heights are listed.

Line 19 gives the combined width of the inlet ports, measured square to the flow from Fig. 14–3.

Line 20 is the product of lines 18 and 19.

Line 21 is obtained as

$$100\frac{h_i}{s} = 100\frac{\text{Line 18}}{27.5}$$

Line 22 is read from Fig. 7–5 with line 21.

Line 23 is also read from Fig. 7–5.

Line 24 is obtained by equation (7–3).

$A_{im}\alpha_i = 25.875 \times 27.5 \times F_i$

Line 25 is obtained from equation (7–1)

$$w_i = \frac{L}{2A_{im}\alpha_i} V_{disp}n = \frac{L \times 9077 \times 240}{2A_{im}\alpha_i}$$
$$= 1{,}090{,}000\frac{L}{A_{im}\alpha_i}$$

In this equation $A_{im}\alpha_i$ is taken from line 24 and L from line 3 of Table 14–I.

The next step is the construction of the graph (Fig. 14–4). Against the scavenge pressures are plotted the inlet velocities as read from line 17 of Table 14–II. These are the air velocities produced by the blower. On this are superimposed the velocities read from line 25 of Table 14–III. These are the velocities produced by the inlet ports of various heights. The intersection of these lines give the resulting scavenge pressures and inlet port velocities for port heights 3.7 inches, 3.85 inches, etc.

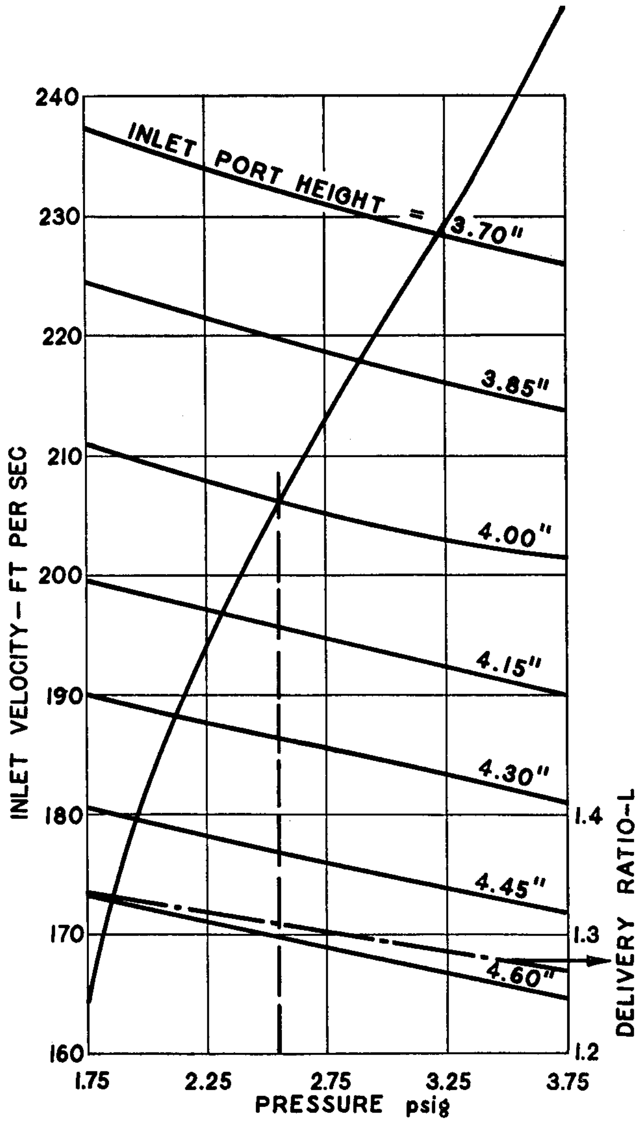

Fig. 14-4. Inlet Velocities *vs.* Scavenge Pressure.

The next step is calculation of the expected power output.

For this purpose Table 14–IV lists the expected *gross mep* based on the effective compression stroke, the *blower mep* from the experimental data represented by Fig. 14–2 and the *net bmep* dy subtracting the latter from the former.

Table 14-IV. Output Calculation.

26. Inlet port height (in.)	3.7	3.85	4.0	4.15	4.30	4.45	4.60
27. Scavenge pressure (psig)	3.23	2.88	2.56	2.31	2.13	1.97	1.85
28. Blower input (hp)	34.0	30.3	26.8	24.25	22.40	20.9	19.7
29. Blower mep (psi)	6.18	5.51	4.87	4.41	4.07	3.80	3.58
30. Effective stroke (per cent)	86.5	86	85.4	84.9	84.4	83.8	83.8
31. Gross mep (psi)	75.5	75	74.5	74	73.5	73	72.6
32. Net bmep (psi)	69.32	69.49	69.63	69.59	69.43	69.20	69.02
33. Net power output (hp per cyl)	381.0	381.9	383.5	383.2	381.6	380.5	380.0

In Table 14–IV, line 26 lists the various inlet port heights.

Line 27 is read from Fig. 14–4.

Line 28 is read from Fig. 14–2 and corrected by a factor

$$\frac{1083}{1100} = 0.986$$

corresponding to the actual blower speed.

Line 29 is obtained from line 28 by the equation

$$\text{hp} = \frac{\text{mep } V_{disp} n}{12 \times 33000} = \text{mep}\,\frac{9077 \times 240}{396{,}000} = 5.50 \text{ mep}$$

Line 30 is 100 less relative port height, and the latter is listed in Table 14–III, line 21.

In line 31 the gross mep is set equal to 75 psi for 3.85 inches port height. This value is obtained from Fig. 5–9 for an air delivery of 130 per cent. The *gross mep* for the other port heights are set proportional to the effective stroke.

Line 32 is the difference between line 31 and line 29.

Line 33 is obtained by the equation

$$\text{hp} = \text{bmep}\,\frac{9077 \times 240}{396{,}000} = 5.50 \text{ bmep}$$

Table 14–IV shows that 4 inches inlet port height gives maximum power, and the corresponding scavenge pressure is 2.56 psig. With a relative inlet port height of

$$\frac{4}{27.5} = 14.55 \text{ per cent}$$

from Fig. 7–5 the inlet port opens 130 degrees after top center and closes 230 degrees after top center.

14.10 Exhaust Ports.

With a total width of 19.75 inches for the exhaust ports, 600 square inch-degrees blowdown time-area corresponds to

$$\frac{600}{19.75 \times 27.5} = 1.107 \text{ per cent deg,}$$

which gives (Fig. 8–1) 18 degrees exhaust lead. The exhaust ports open at 112 degrees and close at 248 degrees after top center.

If the exhaust ports are made full length, they are (from Fig. 7–5) 26.4 per cent of stroke or

$$h_e = 0.264 \times 27.5 = 7.24 \text{ in.}$$

The air flow through the cylinder is reduced by only a small amount if the exhaust ports are made somewhat shorter on the bottom. The blowdown period is unaffected, and if the inlet port area is smaller than the exhaust port area the size of the exhaust port area has little effect on the flow. Shorter exhaust ports have distinct advantages inasmuch as the cooling of the cylinder is less disturbed by the exhaust ports, and the coring of the cooling passages in the bridges is simpler.

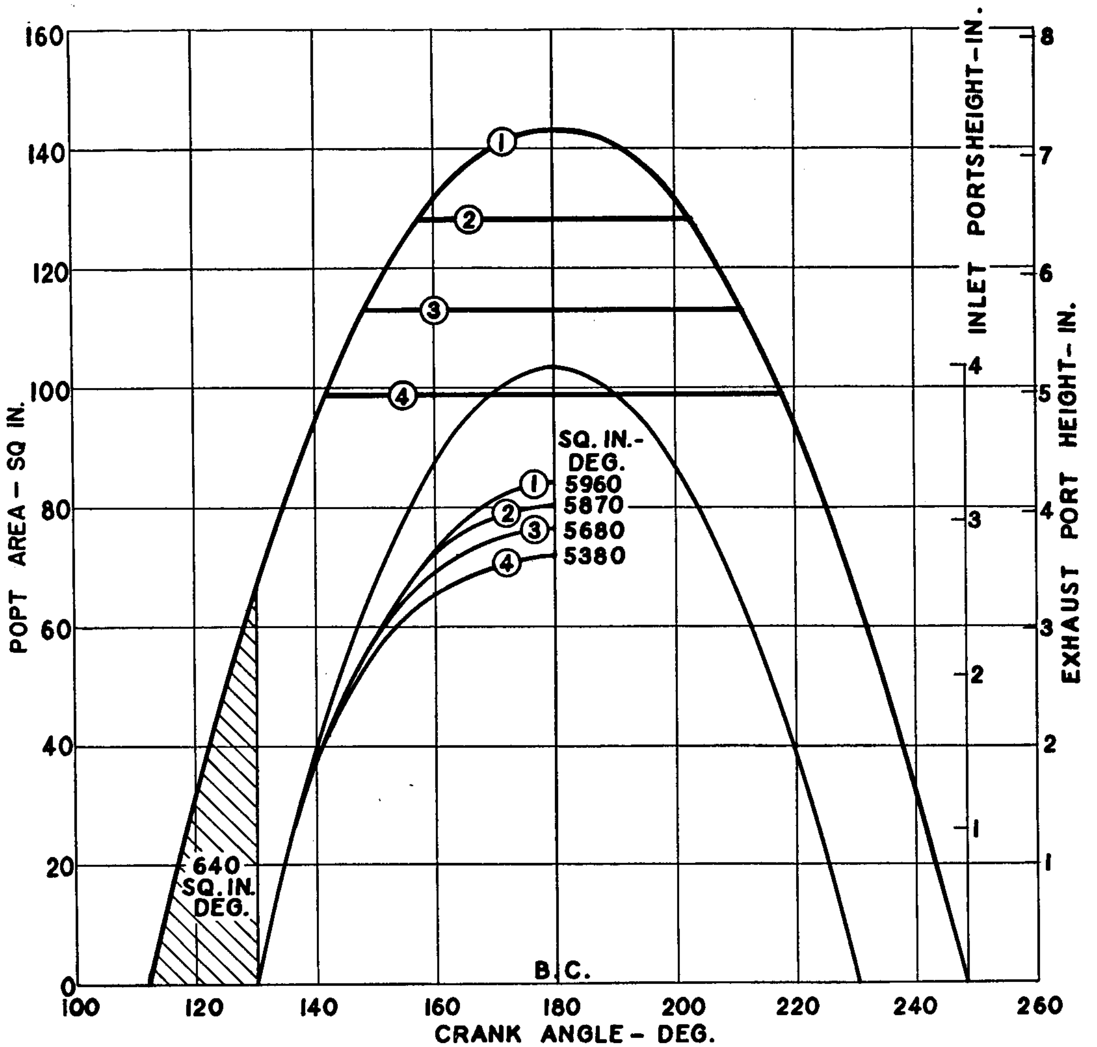

Fig. 14-5. Reduced Port Area Diagrams with Shortened Exhaust Ports. Combined width of inlet ports 20⅞ in. Combined width of exhaust ports 19¾ in.

In Fig. 14–5 the calculated optimum porting is presented. Three horizontal lines are shown on the exhaust line representing $6\frac{1}{2}$-, $5\frac{3}{4}$- and 5-inch long exhaust ports in addition to the $7\frac{1}{4}$-inch long port corresponding to full length exhaust ports. For each of the four exhaust port lengths the *reduced* time area is calculated using Hold's method as described in section 9.9. The corresponding reduced time-area diagram are shown on the figure and marked one, two, three and four. The square inch-degree area of each, as obtained by planimetration, is also marked on the figure.

It is safe to assume a scavenge factor of $S = 0.5$ for loop scavenging with cast ports if the factor relates to the *reduced* time-area diagram. The required time-area diagram is

$$(A\alpha)_r = \frac{L}{2w_i} V_{disp} n$$

$$= \frac{1.308}{2 \times 0.5 \times 520} 9077 \times 240 = 5475 \text{ sq in.-deg.}$$

The figures required to evaluate this equation were obtained in the following way.

From Fig. 14–4 the optimum inlet port length, which is 4 inches, gives a scavenge pressure of 2.55 psig. In the same figure the delivery ratio was plotted against scavenge pressure as read off Table 14–I, lines 1 and 3. The L line gives 1.308 for 2.55 psig.

From Fig. 7–2 for a pressure ratio of $(14.5 + 2.55)/14.5 = 1.176$ and an air temperature of 125 F

$$\frac{w_i}{S} = 520$$

and for S, 0.5 has been selected.

Accordingly 5475 square inch-degrees reduced time-area is adequate. Diagram three corresponding to an exhaust port length of $5\frac{3}{4}$ inches has an area of 5700 square inch-degrees. Consequently $5\frac{3}{4}$ inches exhaust port length is adopted.

From planimetration the blowdown time-area is 640 square inch-degrees which is more than the compromise figure of 600 square inch-degrees, and therefore acceptable.

The final porting, therefore, is as follows:

Exhaust port	opens	112 deg after top center
	closes	248 deg after top center
	duration	136 deg
Inlet port	opens	130 deg after top center
	closes	230 deg after top center
	duration	100 deg
Exhaust lead		18 deg
Exhaust port	length	$5\frac{3}{4}$ in.
	bottom edge	$1\frac{1}{2}$ in. above piston top edge at bottom center
Inlet port	length	4 in.
		bottom edge at piston top edge at bottom center.

APPENDIX TO CHAPTER 14

The following tables on displacement factors have been reprinted with the permission of Mr. Levi B. Smith from his pamphlet *Displacement, Velocity, and Acceleration Factors for Reciprocating Motion*, 1940.

To illustrate the use of the table in porting calculation, the following example is worked out in detail.

A $20\frac{1}{2}$ by $27\frac{1}{2}$-inch bore and stroke engine has a connecting rod length of 62 inches. Its inlet ports open at 130 degrees after top center. What is the exact height of the inlet ports?

The first step is the determination of $\frac{L}{R}$, which equals

$$\frac{L}{R} = \frac{62}{13.75} = 4.51.$$

The height of the inlet ports is the $s(1\text{-}D.F.)$ where s is the total stroke, 27.5 inches, and $D.F.$ is the displacement factor.

To find the displacement factor enter the table at the crank angle group containing the column headed by 130°. Proceed down this column until the factors corresponding 4.40 and 4.60 are found to be 0.8550 and 0.8535. By linear interpolation find the desired displacement factor for 4.51 to be 0.8541, which gives for inlet port height 27.5 (1–0.8541) = 4.01 inches.

Table 14-V. Displacement Factors for Reciprocating Motion.

Linear displacement = (Stroke) (Factor). Volumetric displacement = (Swept volume) (Factor). The decimal point is located immediately to the left of the first digit of each factor.

L/R	2°	178°
2.00	0005	9998
3.00	0004	9998
4.00	0004	9998
6.00	0004	9997
INF.	0003	9997
	4°	176°
2.00	0018	9994
3.00	0016	9992
4.00	0015	9991
6.00	0014	9990
INF.	0012	9988
	6°	174°
2.00	0041	9986
3.00	0036	9982
4.00	0034	9979
6.00	0032	9977
INF.	0027	9973
	8°	172°
2.00	0073	9976
3.00	0065	9967
4.00	0061	9963
5.00	0058	9961
6.00	0057	9959
INF.	0049	9951
	10°	170°
2.00	0114	9962
2.50	0106	9954
3.00	0101	9949
4.00	0095	9943
5.00	0091	9939
6.00	0089	9937
INF.	0076	9924
	12°	168°
2.00	0163	9945
2.50	0153	9934
3.00	0145	9927
4.00	0136	9918
5.00	0131	9912
6.00	0127	9909
INF.	0109	9891

L/R	14°	166°
2.00	0222	9925
2.40	0210	9913
2.80	0201	9904
3.40	0192	9895
4.00	0185	9888
5.00	0178	9881
6.00	0173	9876
INF.	0149	9851
	16°	164°
2.00	0289	9902
2.40	0273	9886
2.80	0262	9874
3.20	0253	9866
3.60	0247	9859
4.40	0237	9850
5.20	0230	9843
6.00	0225	9838
INF.	0194	9806
	18°	162°
2.00	0365	9875
2.20	0354	9864
2.40	0345	9855
2.80	0330	9841
3.20	0319	9830
3.60	0311	9822
4.00	0304	9815
4.40	0299	9810
5.20	0291	9801
6.00	0285	9795
INF.	0245	9755
	20°	160°
2.00	0449	9846
2.20	0435	9832
2.40	0424	9821
2.60	0415	9811
2.80	0406	9803
3.20	0393	9790
3.60	0383	9780
4.00	0375	9772
4.40	0368	9765
5.20	0358	9755
6.00	0350	9747
INF.	0302	9698

L/R	22°	158°
2.00	0541	9813
2.20	0525	9797
2.40	0511	9783
2.60	0500	9772
2.80	0490	9762
3.00	0481	9753
3.20	0474	9746
3.60	0462	9734
4.00	0452	9724
4.40	0444	9716
4.80	0437	9709
5.20	0432	9703
5.60	0427	9699
6.00	0423	9694
INF.	0364	9636
	24°	156°
2.00	0641	9777
2.20	0622	9757
2.40	0606	9741
2.60	0592	9728
2.80	0581	9716
3.00	0571	9706
3.20	0562	9698
3.40	0554	9690
3.60	0548	9683
4.00	0536	9671
4.40	0526	9662
4.80	0519	9654
5.20	0512	9647
5.60	0506	9642
6.00	0501	9637
INF.	0432	9568
	26°	154°
2.00	0749	9737
2.20	0727	9715
2.40	0708	9696
2.60	0692	9680
2.80	0679	9667
3.00	0667	9655
3.20	0657	9645
3.40	0648	9636
3.60	0640	9628
4.00	0626	9614

L/R	26°	154°
4.40	0615	9603
4.80	0606	9594
5.20	0599	9587
5.60	0592	9580
6.00	0586	9574
INF.	0506	9494
	28°	152°
2.00	0865	9694
2.10	0851	9680
2.20	0839	9668
2.40	0817	9647
2.60	0799	9628
2.80	0783	9613
3.00	0770	9600
3.20	0758	9588
3.40	0748	9578
3.60	0739	9568
4.00	0723	9553
4.40	0711	9540
4.80	0700	9530
5.20	0691	9521
5.60	0684	9513
6.00	0677	9507
INF.	0585	9415
	30°	150°
2.00	0987	9648
2.10	0972	9632
2.20	0958	9618
2.30	0945	9605
2.40	0933	9593
2.60	0913	9573
2.80	0895	9555
3.00	0880	9540
3.20	0866	9527
3.40	0855	9515
3.60	0844	9505
4.00	0827	9487
4.40	0812	9473
4.80	0800	9461
5.20	0790	9451
5.60	0782	9442
6.00	0774	9434
INF.	0670	9330

Table 14-V. Displacement Factors for Reciprocating Motion (*Continued*).

Linear displacement = (Stroke) (Factor). Volumetric displacement = (Swept volume) (Factor). The decimal point is located immediately to the left of the first digit of each factor.

L/R	35°	145°
2.00	1324	9516
2.10	1303	9495
2.20	1285	9476
2.30	1268	9459
2.40	1252	9443
2.50	1238	9429
2.60	1225	9416
2.80	1201	9393
3.00	1181	9372
3.20	1163	9355
3.40	1148	9339
3.60	1134	9326
3.80	1122	9313
4.00	1111	9302
4.40	1092	9283
4.80	1076	9268
5.20	1063	9254
5.60	1052	9243
6.00	1042	9233
INF.	0904	9096

L/R	40°	140°
2.00	1700	9361
2.10	1671	9334
2.20	1650	9310
2.30	1628	9288
2.40	1608	9269
2.50	1590	9250
2.60	1573	9234
2.70	1558	9218
2.80	1544	9204
3.00	1518	9179
3.20	1496	9156
3.40	1476	9137
3.60	1459	9119
3.80	1444	9104
4.00	1430	9090
4.20	1417	9078
4.40	1406	9066
4.80	1386	9046
5.20	1369	9030
5.60	1355	9015
6.00	1342	9003
INF.	1170	8830

L/R	45°	135°
2.00	2110	9181
2.10	2078	9149
2.20	2048	9119
2.30	2021	9092
2.40	1997	9068
2.50	1975	9046
2.60	1954	9026
2.70	1936	9007
2.80	1918	8989
2.90	1902	8973
3.00	1887	8958
3.20	1860	8931
3.40	1836	8907
3.60	1815	8886
3.80	1796	8867
4.00	1739	8851
4.20	1764	8835
4.40	1750	8821
4.80	1726	8797
5.20	1706	8777
5.60	1689	8760
6.00	1674	8745
INF.	1464	8536

L/R	50°	130°
2.00	2549	8977
2.10	2510	8937
2.20	2474	8902
2.30	2443	8871
2.40	2414	8842
2.50	2387	8815
2.60	2363	8791
2.70	2341	8769
2.80	2320	8748
2.90	2301	8729
3.00	2283	8711
3.20	2251	8679
3.40	2223	8651
3.60	2198	8626
3.80	2176	8604
4.00	2156	8584
4.20	2138	8566
4.40	2122	8550
4.60	2107	8535
4.80	2094	8522
5.20	2070	8498
5.60	2049	8477
6.00	2032	8459
INF.	1786	8214

L/R	55°	125°
2.00	3009	8745
2.05	2986	8722
2.10	2964	8700
2.15	2943	8679
2.20	2923	8659
2.30	2886	8622
2.40	2853	8588
2.50	2822	8558
2.60	2794	8530
2.70	2768	8504
2.80	2745	8480
2.90	2723	8458
3.00	2702	8438
3.10	2683	8419
3.20	2665	8401
3.40	2633	8369
3.60	2604	8340
3.80	2579	8315
4.00	2556	8292
4.20	2535	8271
4.40	2517	8252
4.60	2500	8236
4.80	2484	8220
5.00	2470	8206
5.20	2457	8193
5.60	2433	8169
6.00	2413	8149
INF.	2132	7868

L/R	60°	120°
2.00	3486	8486
2.05	3460	8460
2.10	3434	8434
2.15	3411	8411
2.20	3388	8388
2.25	3367	8367
2.30	3346	8346
2.40	3308	8308
2.50	3274	8274
2.60	3242	8242
2.70	3213	8213
2.80	3186	8186
2.90	3162	8162
3.00	3139	8139
3.10	3117	8117
3.20	3097	8097
3.30	3078	8078

L/R	60°	120°
3.40	3061	8061
3.60	3029	8029
3.80	3000	8000
4.00	2974	7974
4.20	2951	7951
4.40	2930	7930
4.60	2911	7911
4.80	2894	7894
5.00	2878	7878
5.20	2863	7863
5.60	2837	7837
6.00	2814	7814
INF.	2500	7500

L/R	65°	115°
2.00	3973	8199
2.05	3943	8169
2.10	3915	8141
2.15	3889	8115
2.20	3864	8090
2.25	3840	8066
2.30	3817	8044
2.40	3775	8002
2.50	3737	7963
2.60	3702	7928
2.70	3670	7896
2.80	3641	7867
2.90	3613	7839
3.00	3588	7814
3.10	3564	7790
3.20	3542	7768
3.30	3521	7748
3.40	3502	7728
3.50	3484	7710
3.60	3467	7693
3.80	3435	7661
4.00	3407	7633
4.20	3382	7608
4.40	3359	7585
4.60	3338	7564
4.80	3319	7545
5.00	3301	7527
5.20	3285	7511
5.40	3270	7496
5.60	3256	7482
6.00	3231	7457
INF.	2887	7113

Table 14-V. Displacement Factors for Reciprocating Motion (*Continued*).

Linear displacement = (Stroke) (Factor). Volumetric displacement = (Swept volume) (Factor). The decimal point is located immediately to the left of the first digit of each factor.

L/R	70°	110°
2.00	4462	7883
2.05	4430	7850
2.10	4400	7820
2.15	4371	7791
2.20	4344	7764
2.25	4318	7738
2.30	4294	7714
2.40	4248	7668
2.50	4207	7627
2.60	4169	7589
2.70	4134	7554
2.80	4102	7522
2.90	4072	7492
3.00	4045	7465
3.10	4019	7439
3.20	3995	7416
3.30	3973	7393
3.40	3952	7372
3.50	3932	7353
3.60	3914	7334
3.80	3880	7300
4.00	3850	7270
4.20	3822	7242
4.40	3797	7218
4.60	3775	7195
4.80	3754	7174
5.00	3735	7156
5.20	3718	7138
5.40	3702	7122
5.60	3687	7107
5.80	3673	7093
6.00	3660	7080
INF.	3290	6710
	75°	105°
2.00	4949	7538
2.05	4915	7503
2.10	4883	7471
2.15	4852	7440
2.20	4823	7411
2.25	4795	7384
2.30	4769	7357
2.35	4744	7333
2.40	4721	7309
2.50	4677	7265
2.60	4636	7225
2.70	4599	7188

L/R	75°	105°
2.80	4565	7154
2.90	4534	7122
3.00	4505	7093
3.10	4478	7066
3.20	4452	7040
3.30	4429	7017
3.40	4406	6995
3.50	4386	6974
3.60	4366	6954
3.70	4347	6936
3.80	4330	6918
4.00	4298	6886
4.20	4269	6857
4.40	4243	6831
4.60	4219	6807
4.80	4197	6785
5.00	4177	6765
5.20	4158	6747
5.40	4141	6730
5.60	4126	6714
5.80	4111	6699
6.00	4097	6685
INF.	3706	6294
	80°	100°
2.00	5428	7165
2.04	5399	7136
2.08	5371	7108
2.12	5345	7081
2.16	5320	7056
2.20	5295	7032
2.25	5267	7003
2.30	5239	6976
2.35	5213	6950
2.40	5189	6925
2.50	5142	6879
2.60	5100	6837
2.70	5062	6798
2.80	5026	6763
2.90	4993	6730
3.00	4963	6699
3.10	4935	6671
3.20	4908	6645
3.30	4884	6620
3.40	4860	6597
3.50	4839	6575
3.60	4818	6555
3.70	4799	6536
3 80	4781	6517

L/R	80°	100°
4.00	4747	6484
4.20	4717	6454
4.40	4690	6426
4.60	4665	6402
4.80	4642	6379
5.00	4621	6358
5.20	4602	6339
5.40	4585	6321
5.60	4568	6305
5.80	4553	6289
6.00	4539	6275
INF.	4132	5868
	85°	95°
2.00	5893	6765
2.04	5863	6735
2.08	5835	6706
2.12	5807	6679
2.16	5781	6653
2.20	5757	6628
2.25	5727	6599
2.30	5699	6570
2.35	5672	6544
2.40	5647	6518
2.45	5623	6494
2.50	5599	6471
2.60	5556	6428
2.70	5517	6388
2.80	5480	6352
2.90	5447	6318
3.00	5415	6287
3.10	5386	6258
3.20	5359	6231
3.30	5334	6206
3.40	5310	6182
3.50	5288	6160
3.60	5267	6139
3.70	5247	6119
3.80	5229	6100
4.00	5194	6066
4.20	5163	6035
4.40	5136	6007
4.60	5110	5982
4.80	5087	5958
5.00	5065	5937

L/R	85°	95°
5.20	5046	5917
5.40	5028	5899
5.60	5011	5882
5.80	4995	5867
6.00	4981	5852
INF.	4564	5436
	90°	
2.00	6340	
2.04	6310	
2.08	6281	
2.12	6253	
2.16	6227	
2.20	6202	
2.25	6172	
2.30	6144	
2.35	6117	
2.40	6091	
2.45	6067	
2.50	6044	
2.55	6021	
2.60	6000	
2.70	5960	
2.80	5923	
2.90	5889	
3.00	5858	
3.10	5829	
3.20	5801	
3.30	5776	
3.40	5752	
3.50	5729	
3.60	5708	
3.70	5688	
3.80	5670	
4.00	5635	
4.20	5604	
4.40	5576	
4.60	5550	
4.80	5527	
5.00	5505	
5.20	5485	
5.40	5467	
5.60	5450	
5.80	5434	
6.00	5420	
INF.	5000	

CHAPTER 15

HIGH SUPERCHARGING OF TWO-STROKE CYCLE ENGINES

15.1 The natural trend to increased engine output by increased supercharging met its first limitation in the case of two-stroke cycle engines at a charging pressure of 4 to 7 psig where the power absorbed by the scavenging blower began to exceed the power increase of the engine. It was shown (Chapter 11) that the power absorbed by the blower increases about proportionately to the scavenge pressure, but the power output increases more slowly than does the scavenge pressure. In consequence even engines designed for maximum output, as example 2 of section 7.12, allow only a rather moderate scavenge pressure. When scavenge pressures of 4 to 7 psig (depending on the engine and on the blower) are reached, further increase in scavenge pressure no longer increases the net power output. If the engine is designed for best fuel economy the optimum scavenge pressure turns out to be still lower.

One reason for the poor economy with higher scavenge pressures is the relatively low efficiency of the scavenging blowers. Roots blowers that are most frequently used have adiabatic efficiencies of approximately 60 per cent, and the efficiencies of other current types are not much better.

A contributing factor is that the two-stroke cycle engine uses considerably more air than is involved in combustion, that extra air merely passing through the cylinder to the exhausts. If the charging pressure is raised, the pressure of this extra air must also be raised. This extra or short-circuited air is not to be confused with the 50 to 80 per cent *excess* air required for clear combustion (more at part load). Therefore, the actual amount of air handled by the engine is double or more of what is really burned. The extra amount of air increases the load on the blower so much that peak net power is reached at a comparatively low scavenge pressure.

15.2 Sulzer Method.

The work spent on compressing the air can be materially reduced by an arrangement introduced by Sulzer where low-pressure air is used for scavenging and high-pressure air for supercharging. Such an arrangement is shown in Fig. 15–1. The low-pressure air is admitted through the lower ports which are equipped with automatic nonreturn valves. The cylinder has been scavenged and charged with air up to scavenging pressure by the time the exhaust ports close. Then a mechanically actuated valve opens into the upper ports and compressed air flows into the cylinder until the ports close. This valve remains closed the rest of the time after admitting only the quantity of additional air required for high supercharging. In one 1000-hp Sulzer engine the scavenging air pressure was 3.5 psig, the high supercharge air pressure was 9.7 psig, and the engine developed 105 psi bmep with 140 per cent delivery ratio at 121.5 revolutions per minute.

Scavenging with low-pressure air has an additional advantage. High pressure difference between scavenge and cylinder pressure is likely to result in turbulence, and interferes with the orderly scavenging of residual gases from the cylinder space. This is particularly noticeable with cross- and loop-scavenged engines. When the scavenge pressure, even momentarily, exceeds the cylinder pressure by over 3 or 4 psig, turbulence and separation of the stream from the guiding walls is likely to set in (see sections 5.2 to 5.4), and the efficiency of the scavenging is impaired.

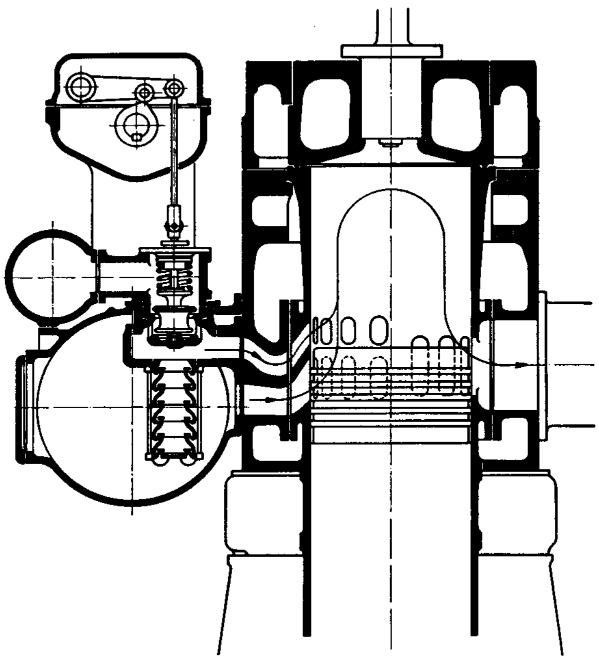

Fig. 15-1. Sulzer Method of Supercharging with Separate High-Pressure Air. (Courtesy Sulzer.)

The most fundamental reason why high supercharging is impractical in its conventional form lies in the energy loss suffered during the exhaust blowdown. While it takes considerable energy to raise the pressure of the air to charging pressure, when upon opening the exhaust, the cylinder pressure is brought into communication with the atmosphere, the energy contained in the high-temperature, high-pressure gas goes largely to waste. With an increase of charging pressure, the exhaust release pressure is raised in about the same proportion and the energy wasted in the exhaust blowdown is increased correspondingly.

EXHAUST TURBINE

15.3 In order to make high supercharging economical, some utilization of the exhaust gas is indispensable. Limited use of the exhaust energy is being made by tuning the exhaust pipe and by the Kadenacy effect (Chapter 13). Compounding of the engine by allowing the exhaust gas to expand in a second low-pressure cylinder has not been a success because of the large dimensions required to handle the exhaust gas. The exhaust-gas turbine was found to be the only practical means of recovering considerable energy from the exhaust of an engine. Therefore, high supercharging of a diesel engine ordinarily involves the use of a gas turbine.

Exhaust-gas turbines are used in conjunction with two-stroke cycle engines in three ways: (1) turbine-driven blowers, (2) geared turbines, and (3) power gas generators.

15.4 Turbine-Driven Blower.

In this arrangement the useful power is derived from the engine, and the exhaust turbine merely drives the supercharging blower. The turbine has no mechanical connection with the engine but only with the blower. The power required by the blower and the power developed by the turbine must balance each other. This corresponds to the well known Buchi supercharging of four-stroke cycle engines with one significant difference. At starting and light load the turbine output is very small because the temperature of the exhaust gas is low. This is of no consequence for the four-stroke cycle engine because, being self-breathing, the engine can operate without any air delivery by the blower. Supercharging is not needed except at high load when turbine power is available. The two-stroke cycle engine, however, always depends on air furnished by a blower and in the absence of hot exhaust gases to drive the turbine the machine **would not even start.** Therefore, if a free floating exhaust turbine is used to supercharge a two-stroke cycle engine, it must be supplemented by another blower which is **mechanically** connected to the engine. A preferred arrangement is use of a turboblower as the first stage and a reciprocating or Roots blower as the second stage. The engine starts on the air furnished by the mechanically driven positive-displacement blower, and as the load increases that blower is precharged by the turboblower. At high load all of the work may shift to the turboblower and the reciprocating or Roots blower may merely "float" on the line, consuming very little power. Figure 15–2 shows such an arrangement.

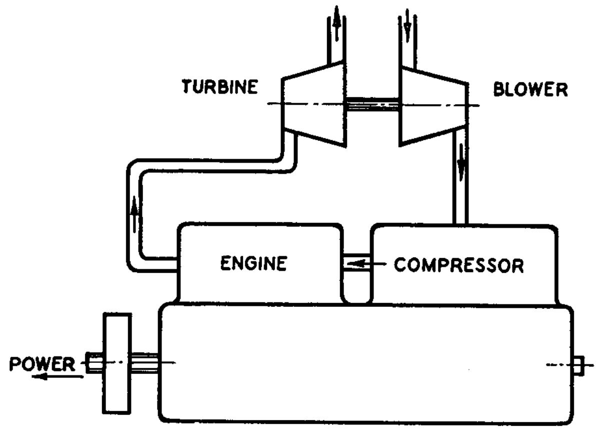

Fig. 15-2. Two-Stage Supercharging. Turbine-driven blower constitutes the first stage and engine-driven compressor the second stage of air compressor.

15.5 Geared Turboblower.

The two-staged compression may be dispensed with if the blower is geared to the engine. With a multistage blower of the centrifugal or preferably the axial flow type, a scavenge pressure of several atmospheres can be produced. By placing a turbine on the exhaust line an appropriate exhaust back pressure is created. The turbine utilizes the energy contained in the exhaust gas as well as the energy contained in the high-pressure short-circuited air. Such an arrangement is shown in Fig. 15–3.

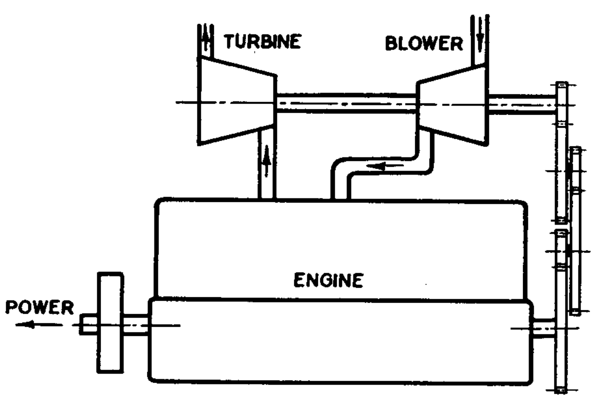

Fig. 15-3. Geared Turboblower.

At starting and low load the blower drives the turbine. The blower, of course, obtains its drive from the engine. As the load increases the temperature of the exhaust gas increases, the turbine power increases and the turbine takes over. At high load the turbine not only is able to drive the blower without mechanical aid from the engine, but may even have a surplus power which it would deliver through the blower gearing to the engine shaft and so increase the useful power of the engine.

The indicator diagram shown in Fig. 15–4 is somewhat idealized but it shows the thermodynamics of this arrangement. The engine has a lower compression ratio, approximately 10:1, and a correspondingly larger clearance volume. The blower compresses the charging air to 2 atmospheres (14.7 psig) and furnishes a quantity of free air which is equal to approximately 2.6 times engine displacement volume. That allows adequate amount of air for short-circuiting and still leaves 200 per cent air for charging the diesel cylinder. The work absorbed by the blower is shown by the area 1–2–3–4.

The exhaust gas drives the turbine which delivers work corresponding to area 5–6–3–7 which may be less than or may exceed the area 1–2–3–4. In the latter case the turbine delivers the difference to the engine shaft, adding to the useful power of the engine. The useful power is the total of the engine power and turbine power less the power absorbed by the blower. The principal source of power

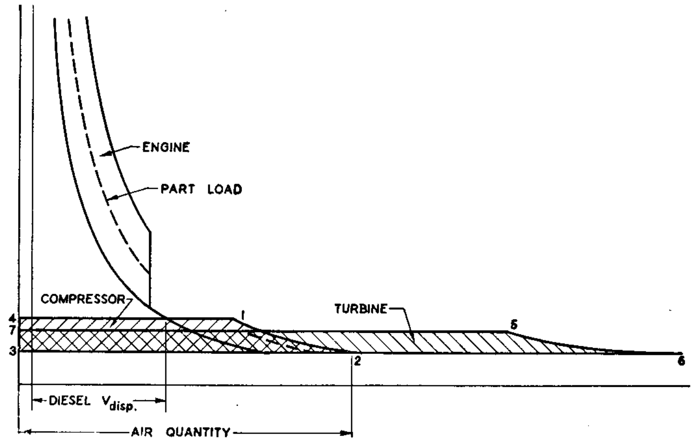

Fig. 15-4. Ideal Indicator Diagram of a Two-Stroke Cycle Diesel Engine Turbocharged to 2 atmospheres (14.7 psig). Upper part of diagram not shown.

may be the engine, as would be the case with moderate supercharging, or the turbine if the air is supercharged to several atmospheres. The supercharging may even be increased to such magnitude that the net engine power (engine output less compressor input) becomes negative and the turbine must assist the engine to drive the compressor.

When a geared turbine is used to drive the blower that furnishes supercharging air of moderate pressure, the problem of starting and light-load running does not arise. For starting all that is necessary is to crank the engine. The blower then furnishes air whether the exhaust turbine is active or not. Likewise at light-load running, the turbine may contribute nothing to the power, it may even constitute a drag, but the blower functions nevertheless, being gear-driven from the engine shaft.

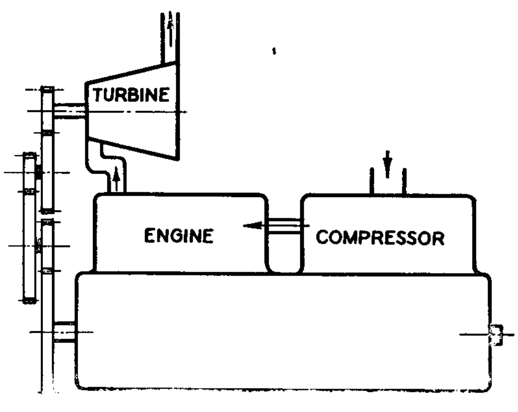

Fig. 15-5. Directly Coupled Reciprocating Compressor and Geared Exhaust Turbine.

15.6 Engine-Coupled Reciprocating Compressor.

In an alternative, but in principle identical, arrangement a reciprocating compressor is coupled directly to the engine crankshaft while the turbine is geared to it (see Fig. 15–5). Thermodynamically this arrangement is identical with the one previously described and is represented by the same indicator diagram (Fig. 15–4). While the power absorbed by the compressor (blower) is essentially independent of the engine load, the power output of both the engine and the turbine is reduced at part load. In any case, the turbine output supplements the engine output and improves the fuel economy.

An engine of this type has been built by Sulzer Brothers in Switzerland [Sulzer, 1941]. The engine had four horizontally arranged cylinders, each fitted with two opposed pistons. The scavenging and charging air was supplied by reciprocating compressors coupled to the crankshaft. The single stage exhaust turbine was mounted overhung to the end of the exhaust manifold and gear-connected to the crankshaft.

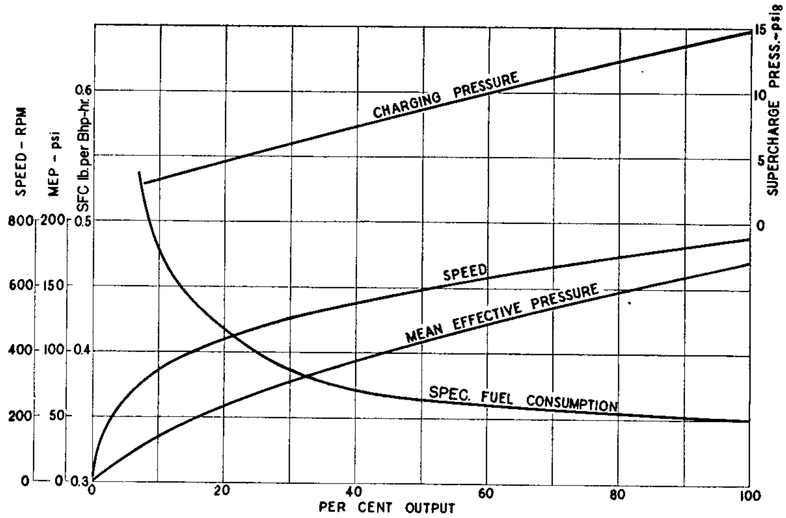

Fig. 15-6. Performance of Sulzer Supercharged Opposed-Piston Engine. Plotted against output when working according to propeller load.

The engine had 190-millimeter (7.45-inch) bore, 2 × 300-millimeter (11.8-inch) stroke, and 750 rpm, which corresponds to 1475 feet per minute piston speed. Under one-hour rating the engine delivered 1370 bhp, which corresponds to 170 psi mean effective pressure, including of course the turbine and the compressor. The fuel consumption was 0.35 pound per bhp-hr, which is an excellent performance. Figure 15-6 reproduced from the Sulzer report shows the principal performance figures of this engine.

The Sulzer engine was supercharged up to 2 atmospheres and gave encouraging results both as to power output and fuel consumption. The supercharge may, however, be raised much higher. Figure 15-7 shows the calculated performance of the engine with geared turboblower for various degrees of supercharge. The computations have been made under favorable but realistic assumptions. One hundred per cent excess air was assumed to insure clear combustion. For each supercharge ratio an engine compression ratio was chosen which insures starting from cold.

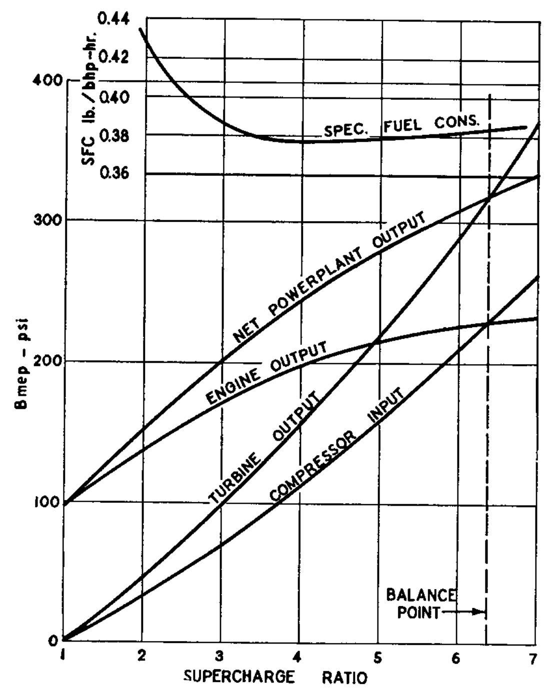

Fig. 15-7. Calculated Performance of Compound Power Plant with Geared Exhaust Turbine.

It is evident that at low supercharge ratios the turbine output is small compared to the engine output, but while the increase of the engine output grows less and less, the turbine output increases in an accelerated manner at increased supercharge ratios. The power plant output, consisting of engine output plus turbine output less compressor input, increases in less than straight-line relation with the supercharge ratio but nevertheless reaches 312 psi bmep when the inlet air is supercharged to 6 atmospheres.

The fuel consumption is around 0.38 pound per hbp-hr at supercharge ratios of 3 to 7 but much higher at supercharge ratios less than 2. This explains why it is uneconomical to turbocharge two-stroke cycle engines to less than 2 atmospheres. The gain in power is then relatively small but the increase in fuel consumption is appreciable.

15.7 Power Gas Process.

As the supercharge increases so does the power absorbed by the supercharging compressor or blower, in fact somewhat faster. At a certain supercharge ratio a point is reached where the compressor input equals the engine output. For the conditions represented in Fig. 15–7 that point is reached at 6.35 supercharge ratio. With more excess air than 100 per cent that balance is reached at a lower supercharge ratio, with less excess air (injecting more fuel) at a higher supercharge ratio.

When the power delivered by the diesel engine becomes equal to the power absorbed by the supercharging compressor, the effective output of the power plant corresponds to the output of the exhaust-gas turbine. The turbine may, therefore, be disconnected from the engine without affecting the energy balance. Thus the turbine becomes the sole source of useful power. The unit composed of diesel engine and compressor serves exclusively to supply exhaust or power gas to the turbine. This is called the *power gas* process. The air produced by the compressor is used only to supercharge the engine up to 5–8 atmospheres (60–120 psig). The engine produces hot exhaust gas at a slightly lower pressure, which gas in the turbine produces the power.

The engine is nothing more than a hot-gas generator furnishing gas to the turbine. It differs from the stationary combustion chamber of the conventional gas turbine in two respects: (1) With equal air-fuel ratio it furnishes colder gas because the gas cools while it expands inside of the diesel cylinder. (2) The power derived from the expansion of the gas in the diesel cylinder furnishes all of the power needed by the compressor.

Figure 15–8 shows a typical hot-gas generator-turbine power plant, where the compressor is direct connected to the diesel engine and the turbine is floating freely on the line. Figure 15–9 shows

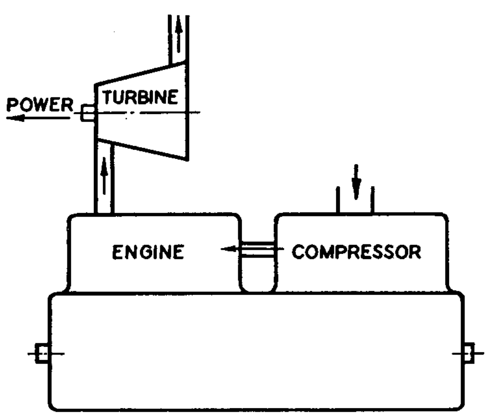

Fig. 15-8. Hot-Gas Generator-Turbine Power Plant. Engine and compressor direct-connected, turbine free on exhaust line.

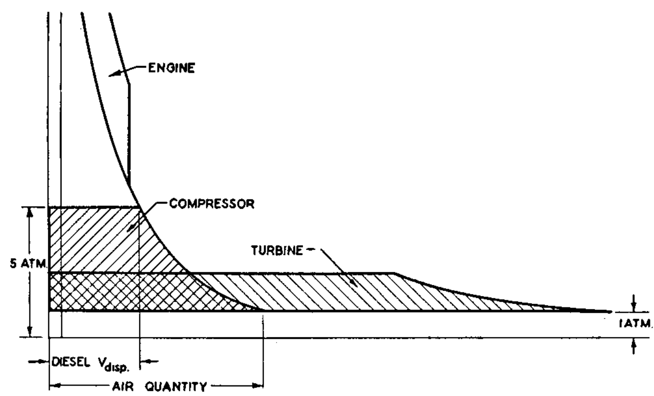

Fig. 15-9. Ideal Indicator Diagram of Power-Gas Process. Engine charged to 5 atm. Compressor work area and engine work area theoretically equal. Upper part of diagram not shown.

an idealized diagram of the power gas process. It represents a *balanced* cycle inasmuch as the compressor work area and the engine work area are theoretically equal. Actually the indicated engine horsepower must be greater to provide for the mechanical losses in the engine and in the compressor.

Figure 15–7 represents at the *balance point* the computed performance of a hot-gas generator power plant. Since the calculation was made under conservative assumptions, actual performance may exceed and has exceeded the values shown. It is evident from the curve sheet that in balanced cycle the power plant output is approximately 40 per cent greater than the output of the supercharged engine alone and corresponds to 320 psi engine bmep. The engine output which equals the compressor input corresponds to 230 psi engine bmep. The fuel consumption was calculated as 0.328 pound per bhp-hr.

By a partial utilization of the *kinetic* energy of the engine exhaust, the performance might be considerably improved (12.27).

15.8 Free-Piston Machine.

A popular form of hot-gas generator is the free-piston engine compressor. [Pescara, 1937, 1939], where the diesel pistons and compressor pistons are reciprocating in one piece. The machine has no crankshaft. Figure 15–10 shows such a power plant, consisting of an opposed-piston engine

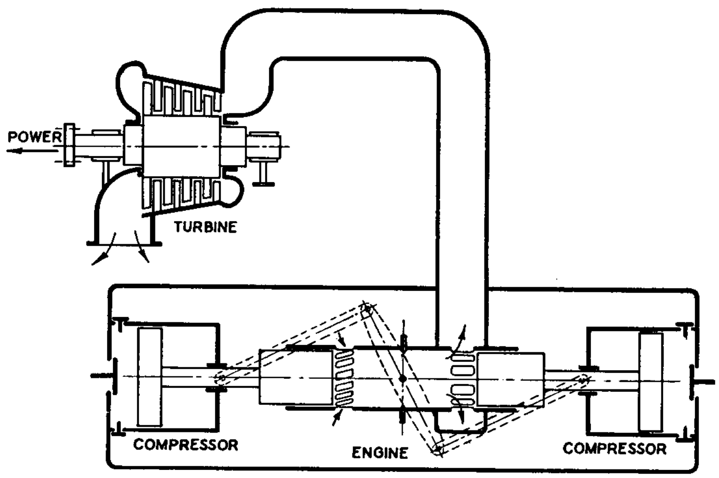

Fig. 15-10. Free-Piston Hot-Gas Generator Turbine Power Plant.

with compressor pistons directly attached to the diesel pistons. The compressor pistons furnish the high-pressure (5 to 8 atmospheres) air to the diesel cylinder, which transforms that into power gas to drive the turbine. A synchronizing linkage keeps the pistons in phase with each other.

15.9 Semifree-Piston Machine.

Many alternative arrangements are being developed. A *semifree-piston hot-gas generator* proposed by the author is shown in Fig. 15–11.

The power gas process with a balanced cycle is free from the gearing required in the coupled turbine arrangement. Because an equality must always exist between engine output and compressor input it lacks the power flexibility of the geared arrangement. It possesses, on the other hand, a geometric flexibility in the placement of the turbine. In many cases distinct advantage is derived from the possibility of placing the turbine away from the engine. For instance, the power gas generated by several engines may be conducted to a single gas turbine for ship propulsion and other applications requiring high power.

15.10 Superatmospheric Engine.

As soon as a turbine is attached to the exhaust line of a two-stroke cycle engine, the exhaust pressure ceases to be substantially atmospheric. It is this very pressure difference between the engine exhaust and the turbine exhaust that the turbine utilizes in converting heat energy into power. The

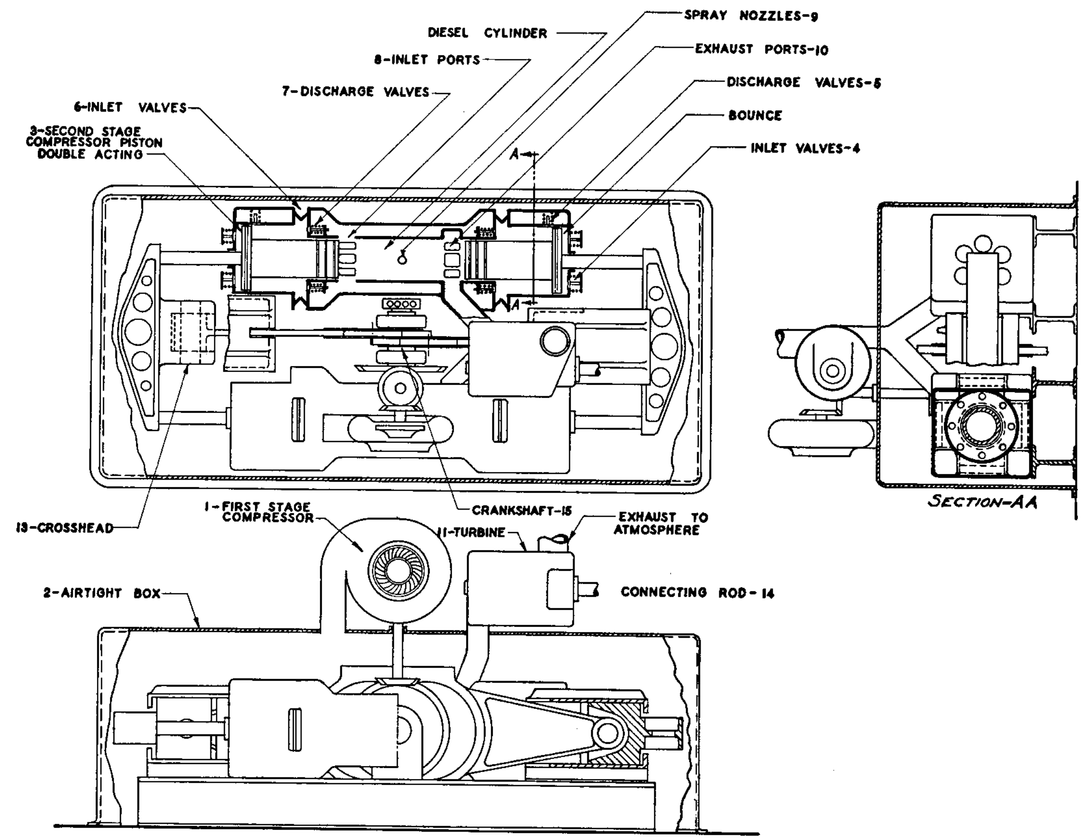

Fig. 15-11. Semifree-Piston Hot-Gas Generator. Two cylinders in tandem with crosshead and synchronizing crankshaft in between.

term *superatmospheric engine* is used to designate an engine the exhaust back pressure of which is considerably above atmospheric. The term *supercharged engine* was applied in a general manner to engines the *inlet* pressures of which are substantially above atmospheric. Since in a two-stroke cycle engine a positive pressure difference must exist between inlet and exhaust, it is obvious that every superatmospheric two-stroke cycle engine must be supercharged but not vice versa.

15.11 Porting of Superatmospheric Engines.

In the analysis of charging (appendix of Chapter 7) it was assumed that the cylinder pressure during the charging period is equal to the exhaust pressure which is substantially atmospheric. In a superatmospheric engine, equation (7–12) for the rate of airflow is replaced by

$$G = SA_{im}\frac{1}{v_i}\left(\frac{p_e}{p_i}\right)^{\frac{1}{k}}(RT_i)^{\frac{1}{2}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left[1-\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}} \qquad (15\text{–}1)$$

For the sake of consistency in terminology, mean inlet velocity is still defined as

$$w_i = \frac{Gv_0}{A_{im}} \tag{15–2}$$

although in case of a superatmospheric engine w_i has little relation to the actual air velocity through the inlet ports, which would be more truly represented by Gv_i/A_{im}. By preserving the original definition of the mean inlet velocity, in case of high charging air density, the actual gas velocity is smaller than the mean inlet velocity in the ratio of the respective air densities.

From (15–1) and (15–2)

$$w_i = S\frac{v_0}{v_i}\left(\frac{p_e}{p_i}\right)^{\frac{1}{k}}(RT_i)^{\frac{1}{2}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left[1-\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}. \tag{15–3}$$

Since $v_i = RT_i/p_i$

$$w_i = S\frac{v_0p_i}{(RT_i)^{\frac{1}{2}}}\left(2g\frac{k}{k-1}\right)^{\frac{1}{2}}\left(\frac{p_e}{p_i}\right)^{\frac{1}{k}}\left[1-\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{k}}\right]^{\frac{1}{2}}. \tag{15–4}$$

By multiplying and dividing the right side of the equation by $\frac{p_i}{p_e}$, this can be brought to the form of

$$w_i = S\frac{v_0p_e}{\sqrt{RT_i}}\sqrt{2g\frac{k}{k-1}\left(\frac{p_i}{p_e}\right)^{\frac{k-1}{k}}}\sqrt{1-\left(\frac{p_e}{p_i}\right)^{\frac{k-1}{k}}} \tag{15–5}$$

which is identical with equation (7–13) except that p_e stands in place of p_0. The significance of this is that the *mean inlet velocity* of the superatmospheric engine is determined by the pressure ratio in the same manner as that of the atmospheric engine (see Fig. 7–2) but the result is multiplied by p_e/p_0 (exhaust pressure divided by atmospheric pressure).

The required inlet time-area is still calculated by equation (7–1)

$$A_{im}\alpha_i = \frac{L}{2w_i}V_{disp}n$$

but the selection of L demands special consideration.

The delivery ratio L was defined as

$$L = \frac{V_{del}}{V_{disp}} = \frac{\text{Volume of free air delivered}}{\text{Displacement volume}}$$

and in case of an atmospheric engine usually a good filling was obtained if L was between 1.3 and 1.5 or usually about 1.4 with separately scavenged engines. In a superatmospheric engine, where the inlet pressure may be equal to several atmospheres the term *scavenge ratio* is used; it is defined as

$$\text{Scavenge Ratio} = \frac{\text{Weight of air delivered}}{\text{Weight of air of } V_{disp} \text{ at inlet density}} \tag{15–6}$$

or

$$R_s = \frac{W_{del}}{V_{disp}\,\rho_i} = L\frac{v_i}{v_0} = L\frac{p_0T_i}{p_iT_0} \tag{15–7}$$

In case of an atmospheric engine ρ_i is approximately equal ρ_0 and the scavenge ratio is equal to the delivery ratio. In a superatmospheric engine, in order to have adequate scavenge and good filling of the cylinder, the *scavenge ratio* must be about 1.3. That makes the delivery ratio

$$L = 1.3\frac{p_i}{p_0}\frac{T_0}{T_i}. \tag{15–8}$$

15.12 Example.

Calculate the inlet porting for an opposed-piston hot-gas generator of 8-inch bore, 10- + 10-inch stroke, and 720 rpm, supercharged to 6 atmospheres (88.2 psia) against an exhaust back pressure of 80.3 psia. The inlet air temperature is to be

$$T_i = 520 \times 6^{\frac{1.35-1}{1.35}} = 830\text{R}$$

According to equation (15–5) the inlet velocity is

$$w_i = S \times 144 \frac{13.28 \times 80.3}{\sqrt{53.3 \times 830}} \sqrt{2 \times 32.2 \frac{1.4}{0.4}\left(\frac{88.2}{80.3}\right)^{\frac{0.4}{1.4}}} \sqrt{1 - \left(\frac{80.3}{88.2}\right)^{\frac{0.4}{1.4}}}$$

$$= S \times 144 \frac{1065}{212} \times 15 \times 1.026 \times 0.164.$$

Assuming a scavenge factor of 0.65,

$$w_i = 0.65 \times 144 \times 12.72 = 1190 \text{ ft per sec.}$$

Assuming a scavenge ratio of 1.3, the delivery ratio is equation (15–7)

$$L = 1.3 \times 6 \times \frac{520}{830} = 4.875$$

The required inlet time-area is, from equation (7–1),

$$A_{im}\alpha_i = \frac{4.875}{2 \times 1190} 1006 \times 720 = 1480 \text{ sq in.-deg,}$$

which is almost the same as the 1390 square inch-degrees inlet time-area used for the atmospheric engine of similar dimensions (Fig. 9–9).

CHAPTER 16

EXPERIMENTAL INVESTIGATION OF SCAVENGING

16.1 In designing a two-stroke cycle engine certain formulas are used at the start. The formulas recommended in this book are simple and contain only a limited number of variables. For instance, in calculating the inlet ports no attention is paid to the exhaust ports and in designing the exhaust ports the exhaust pipe is disregarded.

The fact is that these items have an effect on each other. Each component of the engine which has a part in the gas flow has interaction with every other such component, beginning with the blower inlet and ending with the exhaust stack. The infinite variety of engine operating conditions also influences the porting. To consider all factors in calculations is impossible; to consider many more factors than have been considered already would make the formulas awkwardly complicated and still far from complete.

The answer to the problem is a continuance of development by experimentation after an engine has been built. The formulas generally refer to a "good" engine, meaning that the factors not figuring in the formulas are assumed to have about optimum values. With the wrong length of exhaust pipe the best port dimensions may differ considerably from those calculated by the formulas. However, it is easier to change the exhaust piping than to make the engine fit the wrong exhaust pipe. It is not feasible to supply a set of formulas by which a perfect engine can be designed automatically, but formulas plus experimentation and common sense should give the result.

16.2 Experimental Development.

After having been finished in the shop, the new engine is put on the test stand. It is assumed that it performs halfway satisfactorily, and naturally improvement is desired. The problem is, how to accomplish it.

If the engine fails to deliver the expected power or if its fuel consumption is high, improper porting is a possible reason but not the only one. The reason may be improper combustion or excessive mechanical losses. Those factors are not of concern here, but the experimenter should be fairly certain that the air feed is deficient before he undertakes changes on the scavenging system.

The most pertinent question that an experimental investigation of scavenging is to answer is whether or not the charge of air retained in the cylinder is adequate. A direct answer to this question would be obtained by measuring the amount of air retained in the cylinder at every cycle. No direct method is known that can be applied to an actual engine. What is required is, in effect, to stop the engine suddenly during the compression stroke and determine the air content in the cylinder.

An alternative method which has been used in development work consists of measuring the air consumption and the trapping efficiency.

In a normally aspirated four-stroke cycle engine the air consumption alone gives the desired information. In the absence of appreciable valve overlap, all air delivered to the cylinder per cycle is retained by the cylinder and partakes in the combustion. Therefore, by measuring the air consumption the adequacy or inadequacy of the air charge can be determined.

In a two-stroke cycle engine some of the air delivered escapes from the cylinder before the exhaust ports (or valves) are closed. Since the amount so short-circuited is uncertain, the air consumption may be ample and the engine still starve for lack of air. Therefore in two-stroke cycle engines the air consumption test must be supplemented by some other test that reveals what portion of the supplied air is retained in the cylinder. Naturally the same information is obtained by determining what portion of the supplied air short-circuits to the exhaust, since the difference between the delivered and short-circuited air is the trapped air. The same is true for those supercharged four-stroke cycle engines which operate with a considerable valve overlap that allows a portion of the inlet air to escape before the exhaust valves close.

The amount of retained air can therefore be determined by measuring the air consumption per cycle, which in the terminology used here is equal to V_{del}, and the trapping efficiency η_{tr} which determines what portion of the delivered air is trapped in the cylinder. Then

$$V_{ret} = \eta_{tr} V_{del}$$

or

$$V_{ret} = \eta_{tr} L V_{disp}.$$

That is the air which sustains combustion.

MEASURING AIR CONSUMPTION

16.3 Air consumption is ordinarily measured in terms of cubic feet of free air per minute. V_{del} is expressed in cubic inches of NTP air and is obtained by dividing the NTP air consumption (cubic feet per minute) by the rpm and multiplying by 1728.

The air delivery may be measured on the running engine or on the motored engine; the two figures may differ by a slight amount. The motoring values are frequently a few per cent higher but if the exhaust pipe is well tuned it may be the reverse.

16.4 Pulsations.

Air is usually measured before entering the engine or the blower. Displacement meters like gas meters or rate meters like orifice meters may be used. The former are fairly insensitive to pressure fluctuations but seldom available with a capacity large enough to measure the air consumption of good-sized engines. It is customary instead to use flow nozzles or thin plate orifices for measuring the air delivery, in which case care must be taken to eliminate or minimize pressure fluctuations. Pulsations are especially noticeable with reciprocating compressors or crankcase-scavenged engines. To damp out pressure fluctuations a receiver of sufficient volume must be used between the flowmeter and the engine (or compressor). A rubber membrane cover on the receiver tank will also help to damp out pulsations. Throttling orifices are

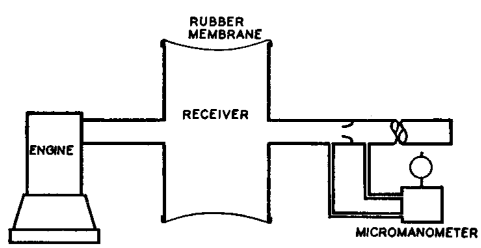

Fig. 16-1. Arrangement for Eliminating Pulsations in Air Flow.

similarly effective but their use is less recommended because it reduces the air delivery. Figure 16-1 shows a recommended arrangement.

For measuring the air flow, a standard nozzle or a thin plate orifice may be used and the flow determined from the pressure drop across the orifice with an appropriate formula found in a handbook [ASME Fluid Meters, 1937]. Figures 16-2 and 16-3 show the installation of orifice and nozzle in the

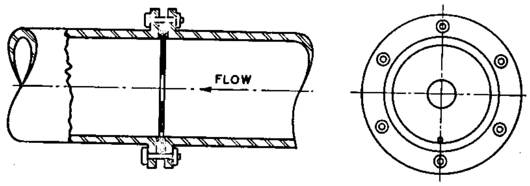

Fig. 16-2. Installation of Orifice Plate.

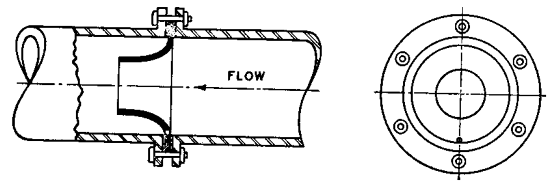

Fig. 16-3. Installation of Flow Nozzle.

pipe. If the orifice (or nozzle) diameter is not greater than one-half of the pipe diameter a straightening length of not more than 6 diameters is required between the orifice and the engine (or receiver) and a couple of diameters on the upstream side of the orifice [ASME Fluid Meters, 1933].

16.5 Micromanometer.

The selected orifice must be of sufficient size to avoid any appreciable throttling of the air flow. The pressure drop across the orifice is therefore small. Inclined water manometers or micromanometers are used to measure the pressure difference. A little known simple micromanometer with which pressures of a few thousandths of an inch water can accurately be measured is shown in Fig. 16-4 [Klinge et al., 1944]. Two of these instruments are needed, one in the upstream side and one in

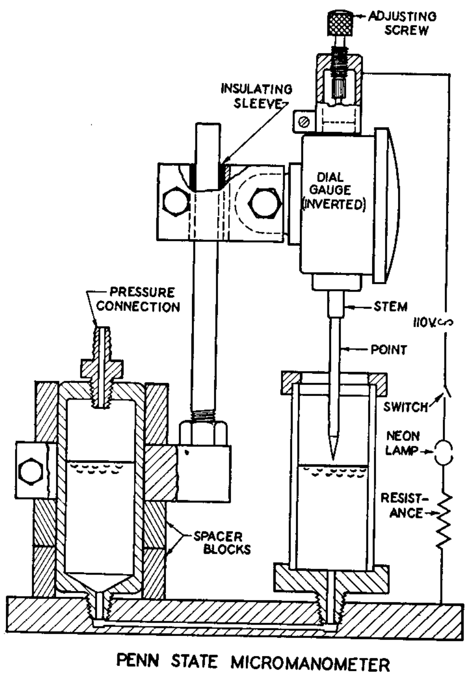

Fig. 16-4. Penn State Micromanometer.

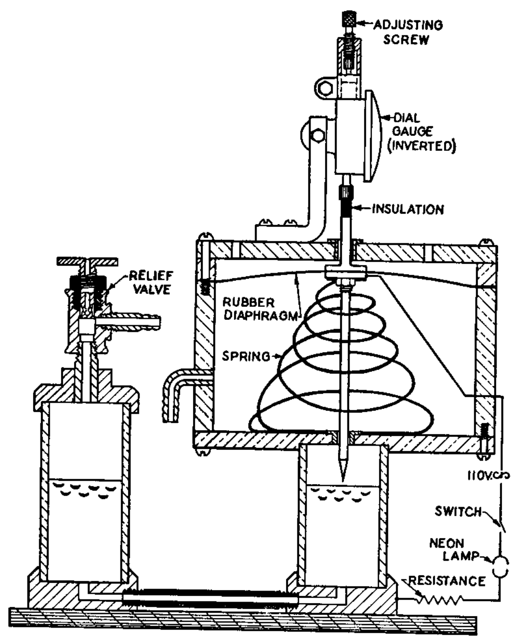

Fig. 16-5. Penn State Differential Micromanometer.

the downstream side of the orifice. Using the differential micromanometer shown on Fig. 16-5, one instrument and one reading are sufficient.

If the air delivery is adequate with a reasonable scavenge pressure, the inlet ports are large enough. If the air delivery is too small, the inlet ports may be too small. However, too-small exhaust ports or a high exhaust counterpressure due to improper ducts or pipes may also be responsible for the insufficient air charge. Weak-spring indicator diagrams of the cylinder and exhaust duct reveal the existence of these conditions. The easiest remedy for inadequate delivery in separately scavenged engines is to increase the scavenge pressure by increasing the air supply. This is generally possible on the experimental engine but may not be economical for the commercial engine. In any case determination of the air delivery with various scavenging pressures at maximum speed gives information helpful in engine development.

While the engine receives a sufficient amount of air, individual cylinders may still be starved because of inadequate intake ducts. This also can be checked with a water manometer or micromanometer. The pressure drops from the scavenge air receiver to the intake ports of the individual cylinders should be substantially the same.

It was pointed out that in a two-stroke cycle engine the air delivery is not the sole deciding factor in judging the air charge because an unknown portion of it is short-circuited. But if the air consumption test is supplemented with a test that furnishes the latter information, the question on the adequacy of the air charge can be answered.

THE TRACER GAS METHOD

16.6 By far the most significant quantity in the scavenging and charging process is the relative amount of air trapped in the cylinder, because it is a measure of the success of retaining the supplied air with a minimum of waste.

The amount of trapped air is equal to the difference between air delivered and air short-circuited, and the trapping efficiency, therefore, is

$$\eta_{tr} = \frac{V_{del} - V_{short}}{V_{del}} = 1 - \frac{V_{short}}{V_{del}}.$$

The amount of short-circuited air can be determined in a relatively simple manner by the tracer gas method.

If a small amount of a suitable tracer gas is continuously fed into and mixed with the inlet air of a running engine, the gas divides in the same proportion as the air delivered. Part of it is short-circuited and part of it is retained in the cylinder. The trapped portion of the tracer gas is to burn in the cylinder during combustion. The short-circuited portion is to appear unchanged in the exhaust gases. By measuring the amount of the tracer gas in the exhaust, the trapping efficiency can be determined.

Through comprehensive tests conducted under the sponsorship of the National Advisory Committee for Aeronautics at The Pennsylvania State College [Schweitzer and De Luca, 1942] monomethylamine was found to be a suitable tracer gas because: (1) it is safe to handle; (2) its quantitative determination is easy and convenient; (3) it burns almost completely (approximately 97 per cent) when exposed to the combustion in the cylinder; (4) the short-circuited portion remains intact in the exhaust.

16.7 Methylamine.

It is easier to measure the concentration of the tracer gas in the diluent than its absolute quantity. The recommended method consists of introducing $\frac{1}{2}$ to 1 per cent monomethylamine gas into the intake pipe of the engine operating at three-quarter to full load and to withdraw, under a slight vacuum, samples of the inlet air and exhaust gas through a burette containing, say, 25 cubic centimeters of standardized sulfuric acid solution. A sulfuric acid solution retains the monomethylamine. The known amount of standardized sulfuric acid solution can neutralize a corresponding amount of monomethylamine. After passing through the burette the now amine-free gas continues on to a gas meter which measures the amount passed through the sulfuric acid. A few drops of mixed indicator solution are added to the sulfuric acid solution. The moment the acid solution becomes neutralized by the monomethylamine, the color changes from purple to green and at this point the gas meter reading is recorded. The quantity of sulfuric acid solution used being known, the percentage of ammonia in the gas mixture can be calculated.

16.8 Test Setup.

Figure 16–6 shows the setup for the test. The monomethylamine, which is commercially available, is fed from the steel flask through a flow control valve into the engine intake. If the air

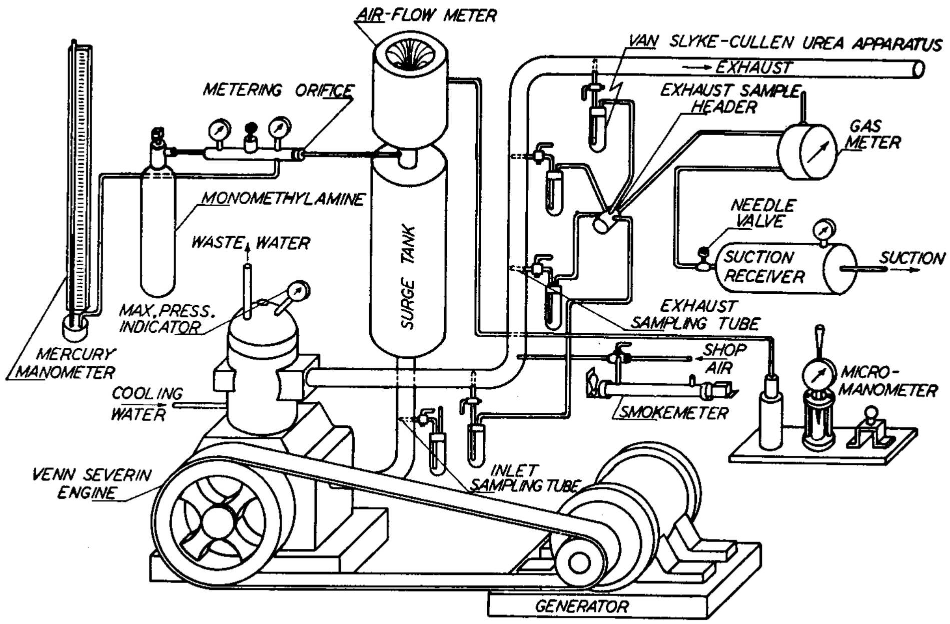

Fig. 16-6. Setup for Testing Engine Scavenge with the Tracer Gas Method. (Penn State Diesel Laboratory.)

delivery is to be measured at the same time with an orifice-type flowmeter, a surge tank of sufficient volume must be interposed between the engine and the flowmeter. The inlet sample is withdrawn from the intake pipe between the surge tank and the engine through a Van Slyke-Cullen urea burette. In order to obtain a representative exhaust sample, gas is bled from several points in the exhaust pipe simultaneously into burettes and from there through a collector header to a single gas meter, which measures the total volume of the exhaust gas samples. As the color of the sulfuric acid solution in each exhaust-sampling test tube changes, the sample flow is stopped. When the flow through the last exhaust sample tube has been stopped, the volume of the combined samples that passed through the meter is recorded. The relative amount of short-circuited air in the engine can then be calculated by the formula.

$$\alpha = \frac{V_{inl} \approx 25 \text{ cc } 0.1 \text{ } NH_2SO_4}{V_{exh} \approx 25 \text{ cc } 0.1 \text{ } NH_2SO_4}$$

where the numerator is the number of cubic feet of inlet sample that neutralizes 25 cubic centimeters of approximately 0.1 normal acid solution and the denominator is the number of cubic feet of exhaust sample that neutralizes the same amount of the same acid solutions.

The trapping efficiency then is

$$\eta_{tr} = 1 - \alpha$$

which may, however, be in error by as much as 4 per cent. Applying a correction discussed in the above cited reference,

$$\eta_{tr} = \frac{1 - \alpha}{0.962}$$

which is likely to be correct within 1 per cent.

Complete details of the testing procedure are described in the above cited reference.

By this method the trapping efficiency can be measured on the actual running engine. The amount of air retained in the cylinder is equal

$$V_{ret} = \eta_{tr} V_{del}$$

and the delivered air can be determined during the same run by an orifice type flowmeter as indicated in Fig. 16–1.

16.9 Test Results.

A $10\frac{1}{2}$ by 12-inch single-cylinder two-stroke cycle, crankcase-scavenged Venn-Severin engine was tested by this method, by taking one inlet and four exhaust samples in each test.

Two tests were made with each air delivery, and the air delivery was varied by placing a restricting orifice in the inlet-air duct next to the engine. The Van Slyke-Cullen urea apparatus was used for absorbing the monomethylamine in the inlet and exhaust samples. Figure 16–7 shows the arrangement used to withdraw an inlet sample through the Van Slyke-Cullen apparatus.

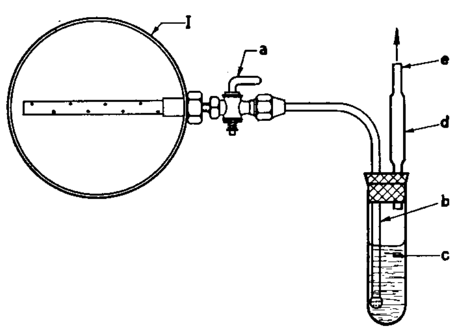

Fig. 16-7. Arrangement Used to Withdraw a Sample Through the Van Slyke-Cullen Urea Apparatus. I — Inlet air duct. *a.* Sampling tube stopcock. *b.* Aeration tube. *c.* Test tube containing sulfuric acid solution. *d.* Trap to separate any entrained liquid. *e.* Outlet tube to gas flowmeter. (Penn State Diesel Laboratory.)

With no throttling of the intake air, the trapping efficiency was 84 per cent; with a 3-inch-diameter orifice used in the inlet duct, it was 82 per cent and, with a 2-inch-diameter orifice, 86.8 per cent according to the tests.

As a further step, the delivery ratios were determined. This procedure involved measurement of air deliveries. A flowmeter nozzle of 3.145-inch diameter was inserted in the intake side of a surge tank. The pressure drop across the nozzle was measured by a micromanometer.

The measured air delivery was 125 cubic feet per minute at 381 rpm or 0.328 cubic foot per cycle. This value gave a delivery ratio of 0.544 with no throttling.

The low delivery ratio obtained in these tests is not typical of the engine but was the result of an inadequate surge-tank capacity and a short 2-inch-diameter tube inserted between the tanks to minimize the pressure fluctuations. This small tube restricted the air delivery to below normal. Subsequently, the air-delivery tests were repeated with a larger surge tank (about 27 times displacement volume), free of restrictions and having two 20-inch-diameter rubber membranes in addition, to minimize pressure fluctuations. The air delivery thus measured corresponds to a delivery ratio of 0.72. If the trapping efficiency were the same (0.84), this setup would give an air charge of 60 per cent of the displacement volume.

If the amount of air retained in the cylinder is more or less than the displacement volume, the bmep is also more or less by about the same percentage, compared to the bmep of a four-stroke cycle engine using the same air-fuel ratio and having the same thermal efficiency. Ways to improve trapping efficiency of the engine have been discussed in Chapters 5 to 8.

SAMPLING TESTS

16.10 Before the tracer gas method was developed, the sampling valve method was used almost exclusively for determining the quality of scavenging. By a synchronized valve that opens during every cycle for a short period between port closure and injection, gas samples are drawn from the cylinder and analyzed for oxygen and/or carbon dioxide content.

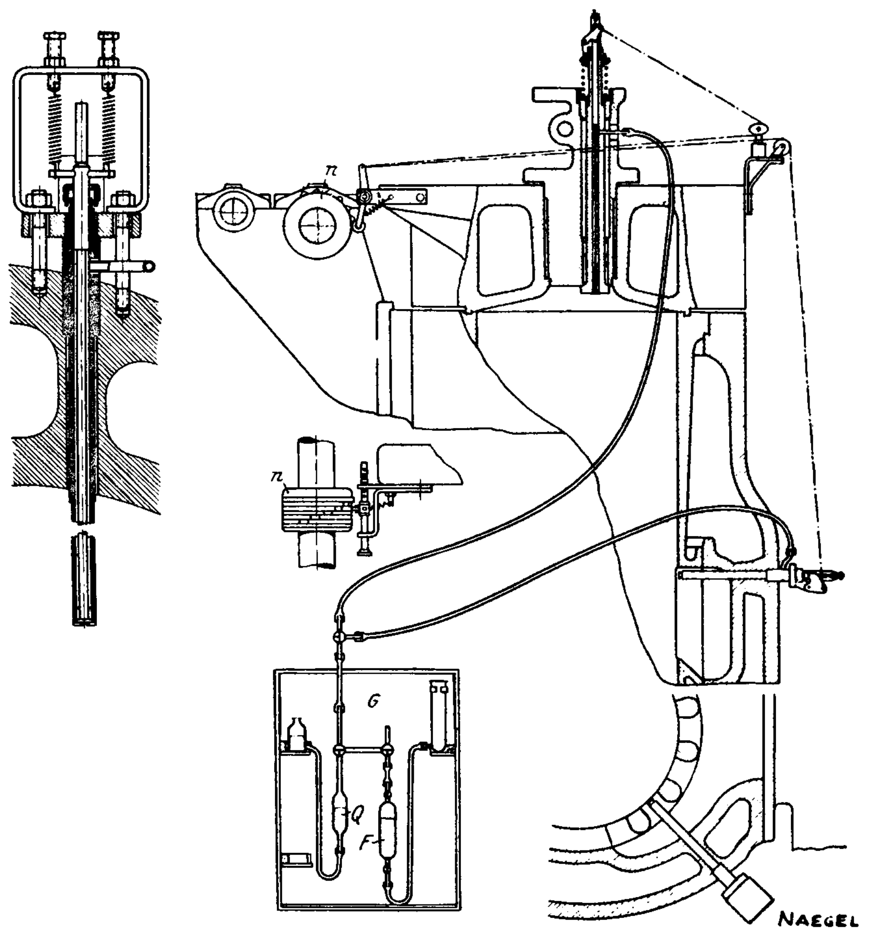

Fig. 16-8. Test Arrangement with Details for Taking Gas Samples from a Sulzer Engine. (Naegel, A., "Dieselmaschine der Gegenwart" *VDI Zeitschr.*, **67**, 733, 1923.)

16.11 Sulzer Tests.

An arrangement for taking gas samples is shown in Fig. 16–8 and some of the test results obtained in Fig. 16–9. The test refers to a 1600-hp Sulzer two-stroke cycle engine reported by Naegel

[Naegel, 1923] and shows both the oxygen and carbon dioxide content not only at the completion of the scavenging but during the entire process. The purity of the charge η_p is obtained by dividing the final oxygen concentration percentage by 21.

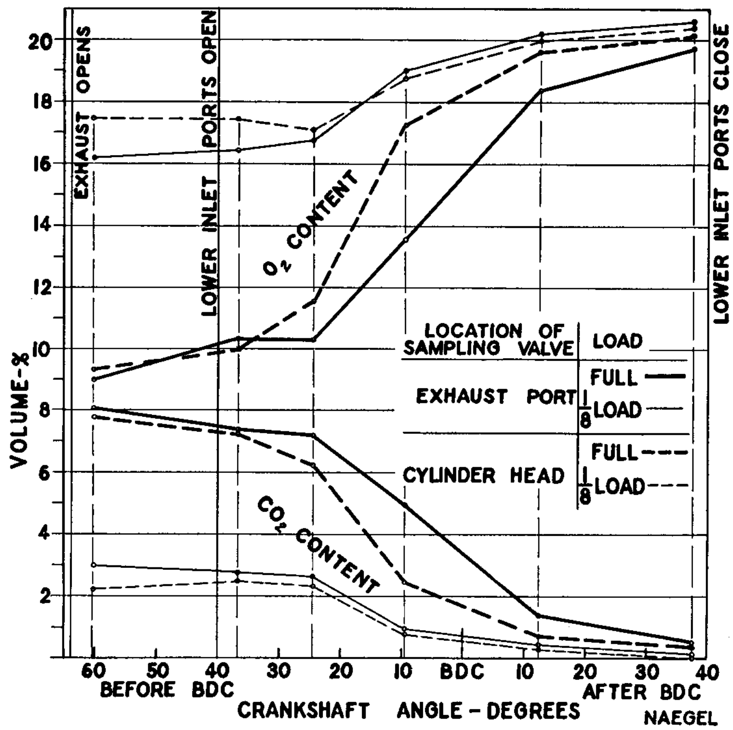

Fig. 16-9. Sample Gas Analysis from a 1600 Hp Sulzer Engine. (Naegel, A., "Dieselmaschine der Gegenwart," *VDI Zeitschr.*, **67**, 733, 1923.)

If it is assumed that combustion is virtually complete, which is a safe assumption for a diesel engine that is not overloaded, the carbon dioxide content may also be used for the determination of the purity of the charge by making use of the Ostwald combustion diagram.

Calculated in either way, the purity of the cylinder charge represented by Fig. 16–9 seems to have varied between 94 and 98 per cent. It is evident that the purity varied not only with the load, which is plausible, but also with the location of the sampling valve, which casts suspicion on the validity of the tests.

16.12 Scavenging Efficiency.

In using the carbon dioxide content in scavenging study, it is more often the practice to make two readings, one sample being taken from the combustion products just before the scavenging air flow begins, and the other during the compression stroke, after scavenging has been completed. Then

$$1 - \frac{\text{Per cent } CO_2 \text{ in trapped charge}}{\text{Per cent } CO_2 \text{ in combustion gas}}$$

gives the scavenging efficiency η_{sc} as defined in Chapter 4. This can be seen by the following reasoning.

Since no carbon dioxide is added during the scavenging period, whatever carbon dioxide was contained in the residual gas is diluted by the scavenging air in the proportion of

$$\frac{\text{Per cent } CO_2 \text{ in } V_{ch}}{\text{Per cent } CO_2 \text{ in } V_{res}} = \frac{V_{res}}{V_{ch}}.$$

On the other hand from equation (4–6)

$$\eta_{sc} = \frac{V_{ret}}{V_{ch}} = \frac{V_{ch} - V_{res}}{V_{ch}} = 1 - \frac{V_{res}}{V_{ch}}.$$

Comparing this with that above

$$\eta_{sc} = 1 - \frac{\text{Per cent } CO_2 \text{ in } V_{ch}}{\text{Per cent } CO_2 \text{ in } V_{res}}.$$

However, the concentration of carbon dioxide in the residual gas is the same as its concentration in the combustion gas which is determined by sampling the cylinder in or next to an exhaust port just **before** inlet opens. Therefore, the ratio of the two carbon dioxide readings deducted from 1 gives the scavenging efficiency.

Scavenging efficiency tests by this method were conducted by Lindner [Lindner, 1933] who measured a scavenging efficiency of approximately 90 per cent for cross scavenging of the type shown on Fig. 6–3 with a delivery ratio of 1.15 and 450 rpm.

16.13 Motoring Tests.

Scavenging efficiency was determined by motoring tests by DeJuhasz [DeJuhasz, 1935] by artificially introducing carbon dioxide before scavenging and withdrawing samples before and after scavenging as in the firing engine. Again the uneven distribution of carbon dioxide after the scavenging precluded accurate results. The composition of the combustion gas is fairly homogeneous near the bottom end of the piston stroke, but **it is next to impossible to obtain a sample representing the average composition of the charge after scavenging, no matter where the sampling valve is mounted on the running engine.** The difficulty is not eliminated by using ammonia in place of carbon dioxide in motoring tests as was done by Lutz [Lutz and Noeggerath, 1939].

16.14 Method of List.

Appreciating the difficulty of obtaining representative gas samples from a running engine, List [List and Niedermayer, 1938] developed a method which was equivalent to stopping the engine in the middle of the compression stroke and taking the sample after the gas became uniform in composition by diffusion. He realized this effect by suddenly detaching the piston head from the rest of the piston connected to the crank mechanism. During regular running, the piston head was held on the piston body only by the gas pressure. By a sudden opening of a large valve in the cylinder head, the gas pressure acting on the piston was removed. Therefore the piston head stopped at top center. The cylinder content was then transferred through the valve into a vessel where it could be leisurely examined quantitatively, and yielded dependably accurate results. The method of List can be used only on a specially built test engine that has little similarity to any production engine, and the results are in consequence of a rather general nature.

A more practical method of estimating the scavenging performance of an actual engine is from the ratio of its power output when it is run normally as a two-stroke cycle engine, and its power output when it is run as a four-stroke cycle engine by injecting at every other stroke only, but leaving other operating conditions unchanged. It may be assumed that if combustion is followed by a scavenging cycle with no combustion, the subsequent charge has 100 per cent purity. On this basis, from equation (4–5)

$$\frac{\text{bmep (2)}}{\text{bmep (4)}} = \eta_p$$

provided f, λ, and C_{rel} can be considered unchanged.

It may be necessary to limit the injections to every sixth or every eighth stroke to assure a charge at 100 per cent purity.

MODEL TESTS

16.15 Model tests on scavenging have been used extensively to study flow of air in the cylinder during the critical scavenging period. The most popular model tests use two- or three-dimensional

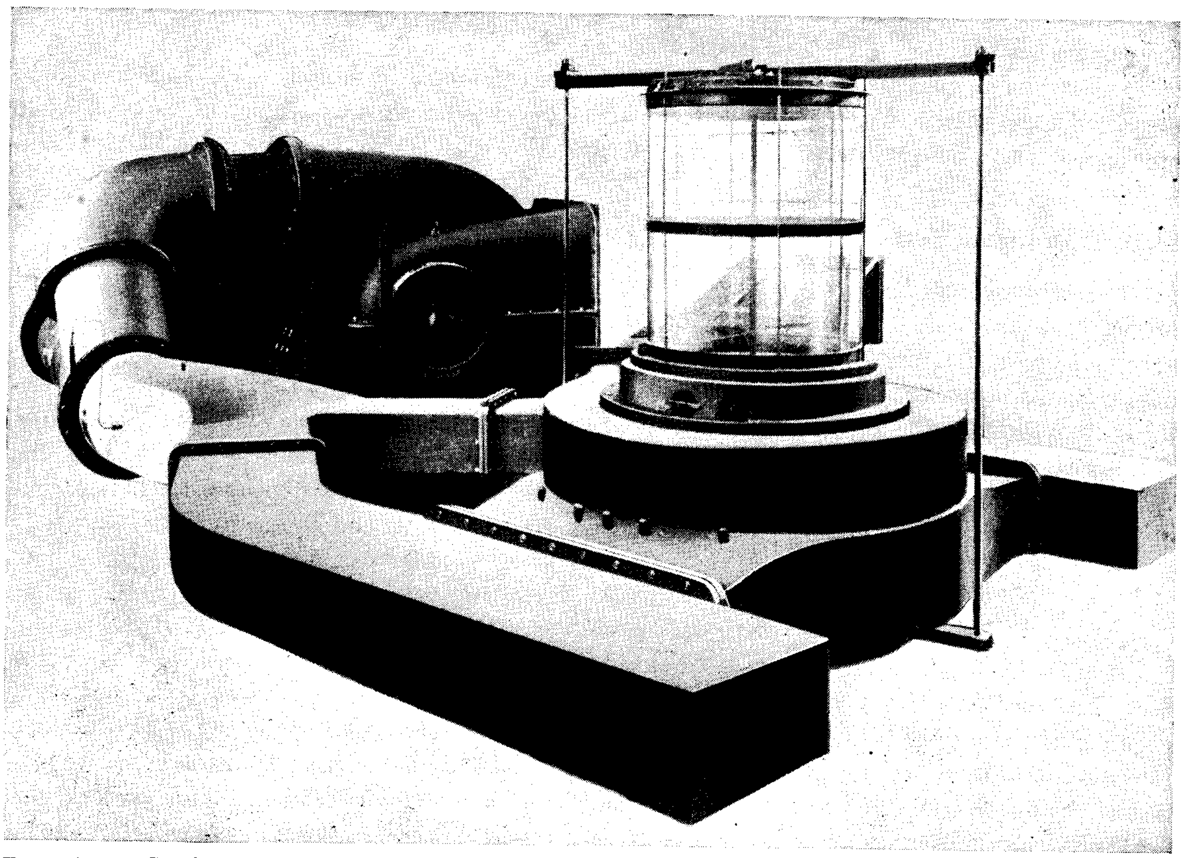

Fig. 16-10. Sass's Outfit for Testing the Scavenging in a Transparent Cylinder. (F. Sass, *Kompressorlose Dieselmaschinen.* Copyright, 1929, by Julius Springer in Berlin.)

transparent models of the cylinder and determine the nature and direction of the air flow through the cylinder by smoke or light flags of thread located at a number of points in the cylinder. The outfit used by Sass [Sass, 1929] is shown in Fig. 16–10, and Fig. 16–11 shows a test in progress.

More recently Rogowski, Bouchard and Taylor [Rogowski et al., 1940] have used this method in a modified form on a small sized engine. Magnesium powder and ammonia fog made by mixing silicon tetrachloride ($SiCl_4$) with ammonia [Lutz and Noeggerath, 1939] have also been used in transparent models for indicating the air flow.

The last step in this direction consists of making the scavenging visible by flakes of tissue paper stamped out from a ribbon and fed into the inlet stream at a certain point of the inlet ports [Riedel, 1942]. In combination with stroboscopic illumination the method can be applied to a model engine with a running piston and transparent cylinders. By changing the phase of the light flashes relative to the piston motion, the spatial development of the scavenging air stream can be determined. The time effect can be observed by phasing the feed of the flakes into the air stream relative to the crank rotation.

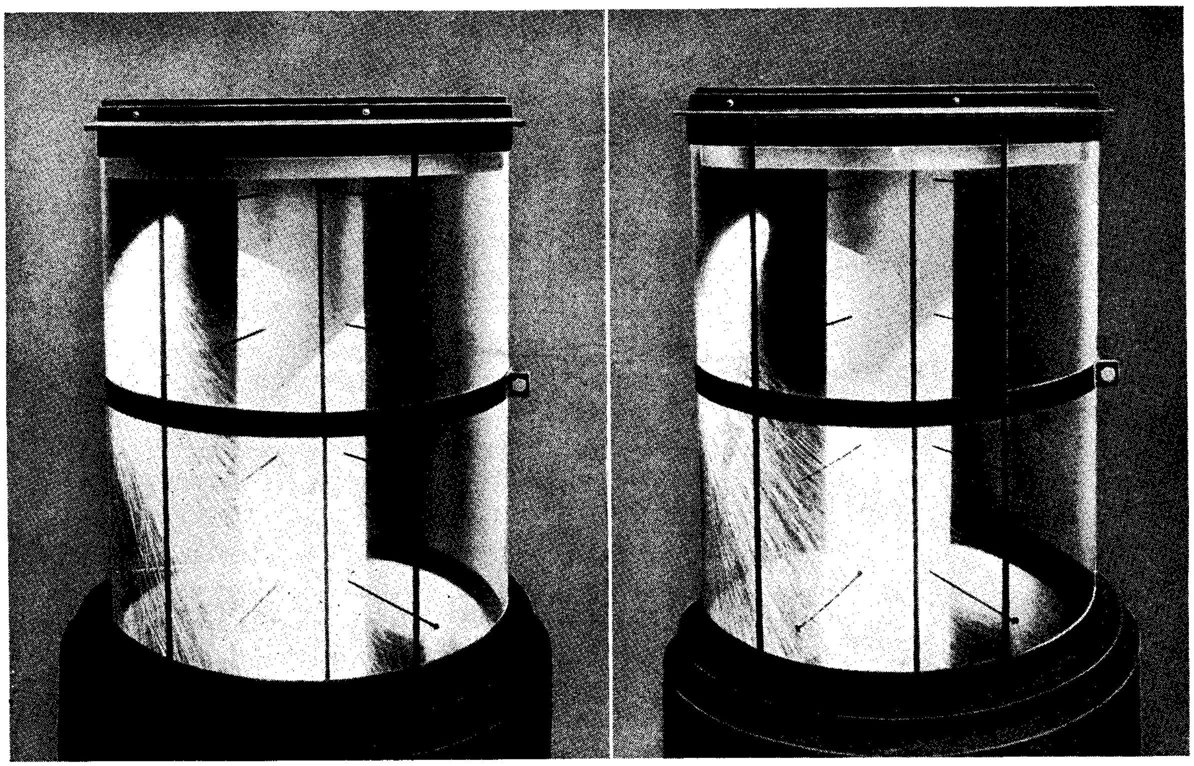

Fig. 16-11. Scavenging Test in a Full Scale Glass Cylinder. (F. Sass, *Kompressorlose Dieselmaschinen.* Copyright, 1929, by Julius Springer in Berlin.)

Pitot tubes have also been used for studying the air flow in the cylinder, although the method is rather laborious.

Model tests of the type described are intended principally for qualitative study and are unsuitable for determining the scavenging efficiency or cylinder charge purity. Curtis' model tests, however, were free from this objection and are therefore described in somewhat more detail.

16.16 Curtis' Tests of Scavenging Efficiency.

With an ingenious apparatus, Curtis [Curtis, 1933] tested the scavenging efficiency of various porting arrangements on model cylinders. He used full scale wooden models complete with pistons. The piston did not move during the test but was either at bottom center or at another selected position, partly covering the ports. Curtis filled the space with atmospheric air contaminated with a small percentage of carbon dioxide. Then he simulated the scavenging process by suddenly admitting a known amount of fresh air which purged the cylinder of part of the contaminated air through the

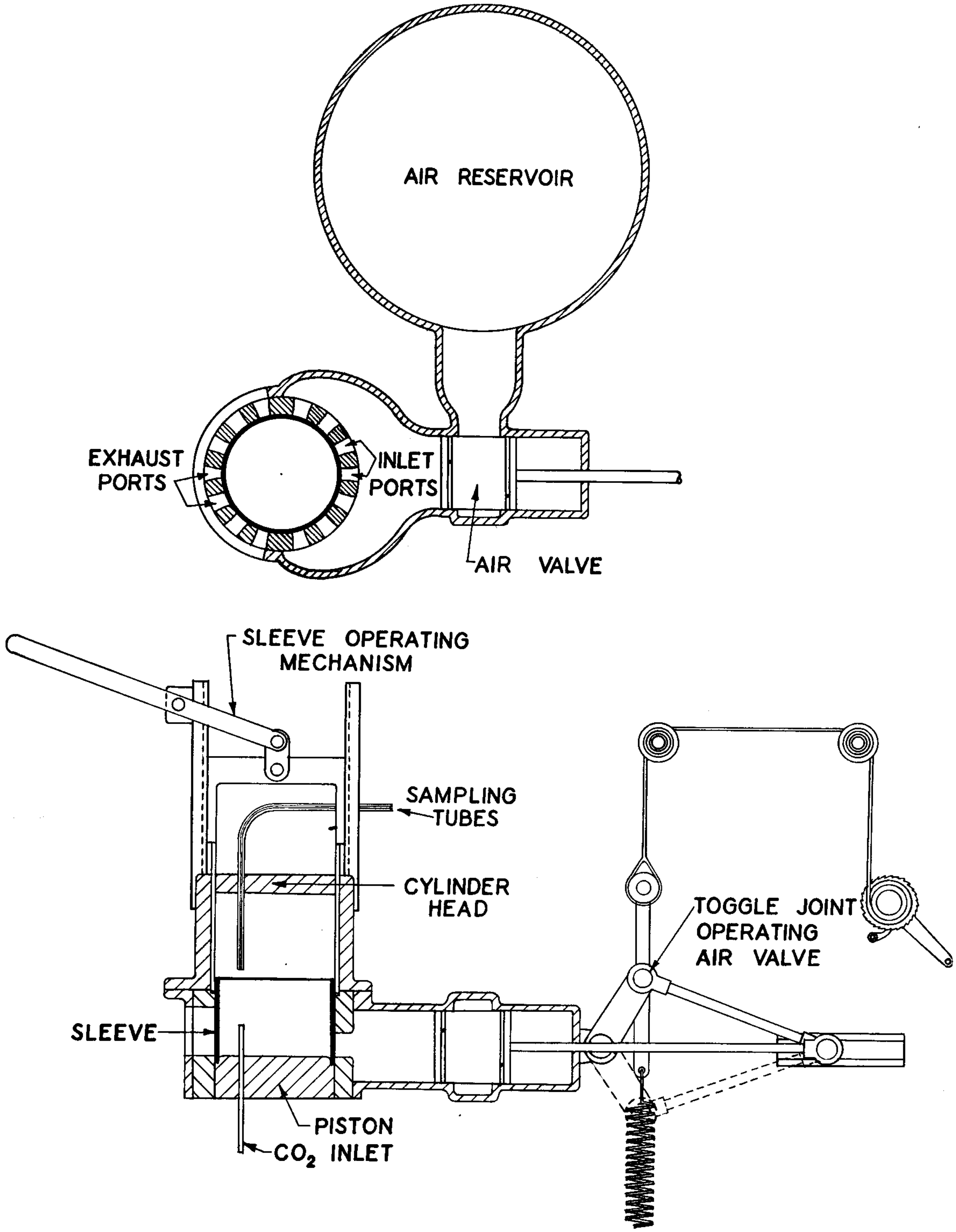

Fig. 16-12. Curtis' Testing Apparatus for Determining the Efficiency of Scavenging.

exhaust ports. By comparing the carbon dioxide concentration before and after the scavenging he determined the scavenging efficiency with various amounts of air supplied and also in various piston positions.

Figure 16–12 shows the Curtis apparatus consisting of an air reservoir of considerable capacity to feed scavenging air to the model cylinder through a quick-acting piston valve. A small pipe was provided for introducing carbon dioxide from an ordinary carbonic gas flask. The carbon dioxide was introduced shortly before, and was well distributed by the time the scavenging was set in motion. To determine the initial carbon dioxide concentration a sample was then drawn through a sampling tube into the burette of a carbon dioxide analyzer.

The scavenging process was set in motion by tripping the spring-loaded toggle joint of the air valve which allowed a certain amount of air of known pressure to enter through the inlet ports and out through the exhaust ports. A second sample was then drawn from the cylinder through a second sampling tube leading into another burette, determining the carbon dioxide concentration after scavenging.

In order to prevent an outflow of carbon dioxide through the exhaust ports or a backflow through the inlet ports into the scavenging belt, a thin metal sleeve was provided with mechanism for raising and lowering it. This sleeve was lowered before introducing the carbon dioxide into the cylinder but was raised immediately before the scavenging, so as not to interfere with the scavenging air flow. The quantity of scavenging air was determined from pressure readings in the air reservoir before and after scavenging, by means of a water manometer. This gave the air delivery or number of cubic feet of free air used after allowing for the temperature change.

Curtis' test is free from most objections to the gas sampling method. The introduction of carbon dioxide takes place comparatively slowly into stationary air and uniform distribution is thus assured when the first sample is taken. Sufficient time is allowed after the scavenging is completed before the second sample is taken. The samples are taken from the middle of the cylinder instead of from the vicinity of the cylinder wall. These circumstances assure representative samples.

It is worth noting that with the equation

$$\eta_{sc} = 1 - \frac{\text{Per cent } CO_2 \text{ after scavenge}}{\text{Per cent } CO_2 \text{ before scavenge}}$$

the Curtis test gives exactly what has been defined as scavenging efficiency (3.3).

16.17 Test Results.

Figures 16–13, 16–14, and 16–15 show Curtis' results with cross, Curtis-loop, and uniflow scavenging.

The results obtained with cross scavenging (Fig. 16–13) differ little from those that represent perfect mixing (Fig. 3–3).

On the type of loop scavenging developed by himself, Curtis' model tests gave results shown in Fig. 16–14. The upper curve was obtained with no exhaust hood, the lower with exhaust hood added but with no header.

Results of model tests with uniflow scavenging are shown in Fig. 16–15. The upper curve refers to four exhaust valve arrangements with no exhaust ducts or elbows added, but providing a free flow for the air after leaving the valve passages. The lower curve was obtained with an exhaust hood of rather large area.

The results obtained were not only reproducible but agreed well with the engine tests of List

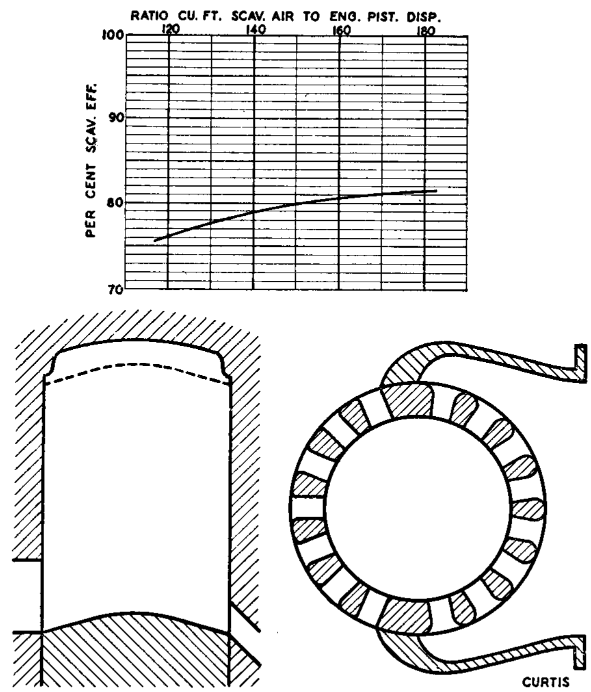

Fig. 16-13. Curtis' Model Test on Cross Scavenging.

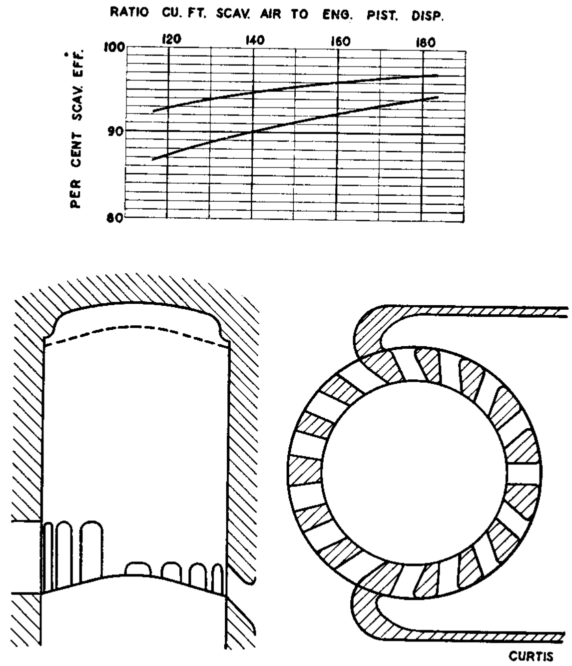

Fig. 16-14. Results of Model Tests on Curtis-Type Loop Scavenging. Lower curve with exhaust hood, upper curve without hood.

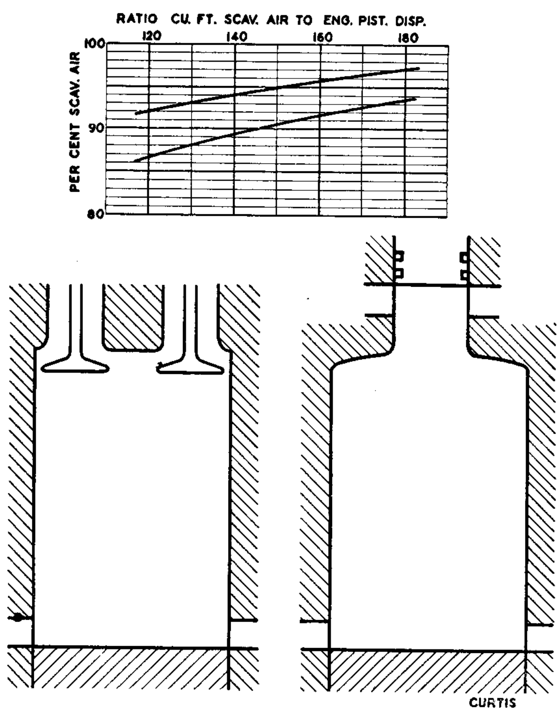

Fig. 16-15. Results with Uniflow Design. Upper curve, no exhaust elbows or hood; lower curve, exhaust hood.

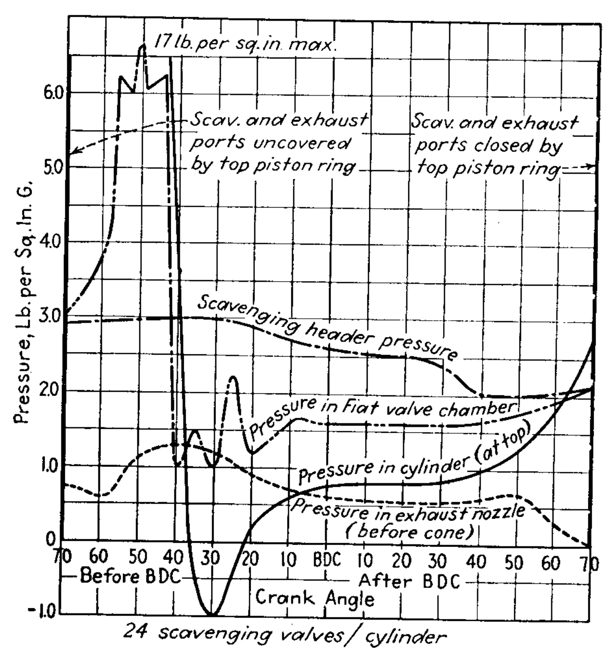

Fig. 16-16. Weak-Spring Diagram of an Engine with Sufficient Exhaust Lead. (By permission of Nordberg Mfg. Co.)

(see Chapters 7, 8, 9) and also with subsequent performance test on engines built according to Curtis designs.

CHECKING THE EXHAUST LEAD

16.18 The exhaust lead which gives best scavenging is that which permits a blowdown to the scavenge pressure before the inlet port opens. If the exhaust lead is considerably less, a back blow of exhaust gases takes place into the inlet port. If the exhaust lead is too great, it is at the expense of the effective stroke or of the inlet period or both.

16.19 Weak-Spring Diagram.

The best way to check the exhaust lead is with a weak-spring indicator. For slow-speed engines, mechanical indicators may be used, but for high-speed engines, electrical or optical indicators are recommended. Balanced pressure-type indicators such as the Bureau of Standards, Farnborough, or M.I.T., connected with mercury manometers are suitable for recording the charging process.

It is advantageous to take time-pressure cards from both the cylinder and the inlet manifold.

Figure 16–16 shows a record from an engine with ample exhaust lead. The exhaust lead of the engine from which the record of Fig. 16–17 was taken is too short.

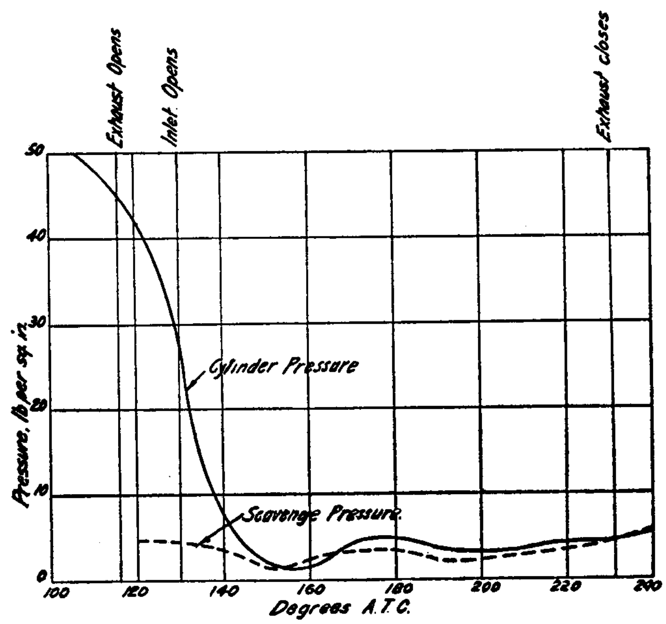

Fig. 16-17. Weak-Spring Diagram of an Engine with Insufficient Exhaust Lead.

16.20 Back Blow.

If the exhaust lead is insufficient, exhaust gas is blown into the inlet ports for a short period. This can be observed with a transparent window in the intake duct. Lubricating oil deposits in the inlet ports are also evidence of insufficient exhaust lead.

If the exhaust lead is found inadequate, again the easiest remedy is to boost the scavenge pressure. This, however, is not recommended if the scavenge pressure is already fairly high. Then it may be impractical to boost the scavenge pressure for two reasons: (1) A high scavenge pressure may spoil the orderly flow of scavenge air in the cylinder and set up eddies which reduce the scavenging efficiency; this is especially likely in nonuniflow engines. (2) The production of higher scavenge pressure absorbs proportionately more power and the net power output is correspondingly reduced.

If it is impractical for any reason to boost the scavenge pressure at the point of inlet opening, the exhaust lead must be increased. The exhaust lead may be increased either by opening the exhaust earlier or by opening the inlet later or by doing both. If the designer is willing to shorten the effective stroke, the simplest way of increasing the exhaust lead is to file the upper edges of the exhaust ports.

CHECKING THE CLOSURE OF THE INLET PORTS

16.21 This problem is without significance in case of symmetrically scavenged engines, where the exhaust remains open longer than the inlet. In supercharged engines a later closure gives a longer supercharge, but a longer supercharge does not represent a gain if more is lost in cylinder pressure by the delay in beginning of the compression than is gained by the pressure charge. The conditions of balance are not easy to calculate, but this can be determined experimentally if the closure of the inlet port can be varied. The timing that gives the highest compression pressure for a given compression ratio is the best.

16.22 Late Inlet Closure.

Late inlet-port closure can sometimes be recognized by a considerable reverse flow in the inlet ports before the end of the supercharging period. If the weak-spring card shows cylinder pressures higher than scavenge pressure while the inlet port is still open, it indicates air flow out of the cylinder. Obviously the pressure of the cylinder charge rises after exhaust closure for two reasons: first, because air is introduced through the inlet ports; second, because the air in the cylinder is compressed by the piston. If the pressure reached in the cylinder exceeds the scavenge pressure, the flow in the inlet ports reverses and the cylinder loses air during the remainder of the supercharge period. There is no object in keeping the inlet port open beyond the point where the cylinder pressure balances the scavenge pressure, because the cylinder charge would be reduced rather than increased thereby, a result certainly contrary to the objective of supercharging. Therefore, if superimposed pressure records show an excess of cylinder pressure over the inlet-manifold pressure during a substantial part of the supercharging period, it may be inferred that the supercharging period is too long and the inlet closure relatively too late.

It would be incorrect to conclude from the above that no reverse flow in the inlet ports can be tolerated or that inlet closure must always be sufficiently advanced to prevent a pressure excess to develop in the cylinder if maximum cylinder charge is sought. While it is true that sudden closure of the inlet port at the reversal point would benefit the charge, sudden closure is not possible. If closing of the inlet port is started earlier, the inlet time-area prior to closure is reduced accordingly, which adversely affects the pressure build-up in the cylinder during the early part of the supercharging period. Only experiments can definitely determine the most favorable closing point for the inlet port.

It is safe to state, however, that if the cylinder pressure considerably exceeds the inlet-manifold pressure during a substantial part of the supercharging period, the inlet port closure is too late. The easiest remedy is to increase the scavenge pressure. If such increase is impracticable or uneconomical, a change in porting is required.

16.23 Checking the Balance of Inlet and Exhaust Port Areas.

In a symmetrically scavenged engine, the inlet and exhaust ports cannot be balanced. The exhaust ports are generally larger than necessary, but their height is dictated by the necessary exhaust lead. Air at appreciably higher than atmospheric pressure cannot be trapped in the cylinder. For an exception see section 12.26.

An unsymmetrically scavenged engine is designed to trap as much air as possible. If the exhaust ports are too large, the inlet air blows through the cylinder without much resistance. However, if the exhaust opening becomes small before the scavenging period ends, pressure begins to build up earlier, and more fresh air is trapped.

It is difficult to determine experimentally the best ratio between the mean inlet- and exhaust-port areas. A simple test, however, may show up the unbalance in the exhaust-port area of a supercharged engine. If throttling the exhaust of the engine, when running at full load, increases the output, it is evidence of excessive exhaust port area. Of course, it would be a wrong remedy to operate the engine continuously with throttled exhaust. The proper remedy is to reduce the exhaust time-area by either closing the exhaust earlier or reducing the port area, or doing both. If the exhaust ports are to be reduced, it is ordinarily better to reduce their height than their width. Reducing the width or lowering the upper edge of the exhaust port may interfere with the necessary exhaust lead but raising the lower edge is not objectionable.

The rule is not reversible. If experiment fails to show higher output when the exhaust is throttled, the conclusion is not justified that the exhaust ports are properly balanced or that a reduction of the exhaust time-area would not eventually increase the output. This can be determined only by actually varying the exhaust ports and selecting the one which gives the highest compression pressure or highest power output.

16.24 Checking Exhaust Pipe Tuning.

The length of the exhaust pipe has a considerable effect on the output of a two-stroke cycle engine. An unfavorable exhaust-pipe length may reduce the output by as much as 30 per cent. It is not always possible to calculate the optimum pipe length accurately, and to get best results it is advisable to try various lengths of exhaust pipe and select the one which gives the best power output. A weak-spring indicator can be of great help in these experiments, as may be seen from Fig. 16–18 and 16–19.

Figure 16–18 shows typical diagrams of the scavenging and exhaust of a properly operating system, and Fig. 16–19 shows those of a poorly operating exhaust system. They were taken from a one-cylinder crankcase-scavenged engine of 12.6-inch bore and 15-inch stroke. The volume of the exhaust pot in both cases was 14,900 cubic inches and the inside diameter of the exhaust pipe was 5.9 inches.

Naturally, the pressure in the exhaust column oscillates in accordance with a sine wave. It is desirable to have scavenging take place when there is a "low tide" in the exhaust.

The course of the exhaust pressure is favorable in Fig. 16–18 (256-inch pipe length, 309 rpm) because the opening of the exhaust coincides with a decline in the exhaust pressure. This promotes quick decompression in the cylinder. True, the exhaust impulse makes the pressure shoot up temporarily afterwards, but the free vibration asserts itself promptly, and *low tide* (a partial vacuum) rules during the port-closing period. This is important because it permits entry of abundant scavenge air into the cylinder.

On the other hand, in the case of Fig. 16–19 (exhaust pipe 356 inches long, 310 rpm) the exhaust occurs while the pressure is rising. The *high tide* will be maintained and fortified by the exhaust impulse. The ports close while positive pressure exists in the exhaust pot, which impedes the entry of

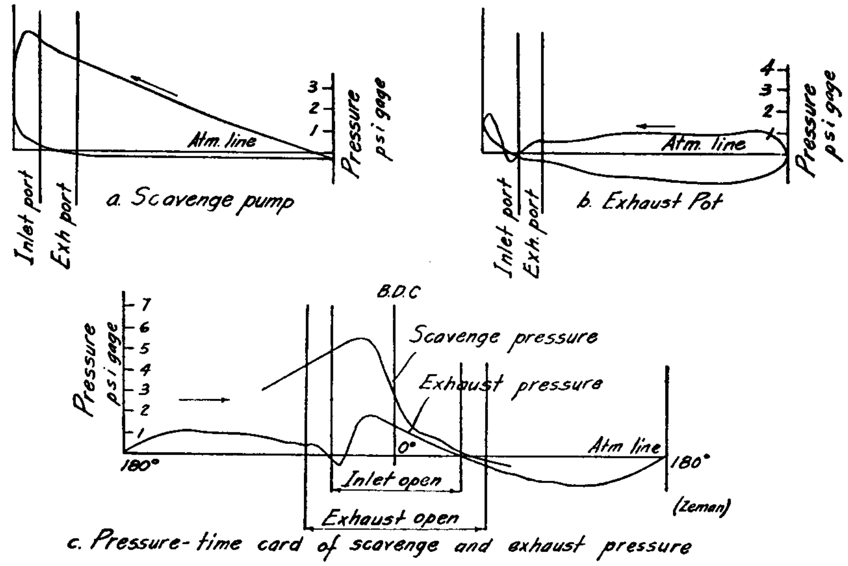

Fig. 16-18. Weak-Spring Diagrams with a Properly Functioning Exhaust System. 12.6 in. bore, 14.6 in. stroke. 309 rpm; exhaust pipe 5.9 in. dia., 256 in. long; exhaust pot 8.7 cu. ft. (J. Zeman, *Zweitakt Dieselmaschinen.* Copyright, 1935, by Julius Springer in Vienna.)

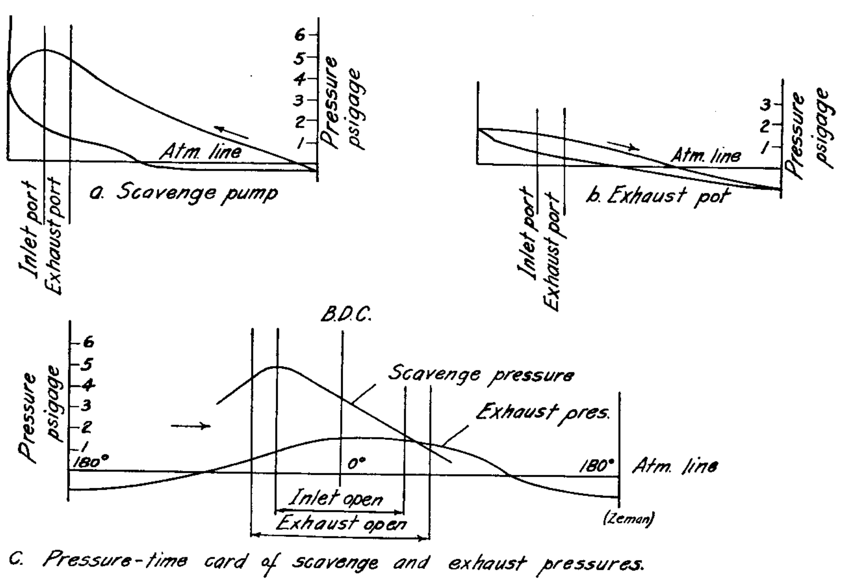

Fig. 16-19. Weak-Spring Diagrams with Improperly Functioning Exhaust System. 12.6 in. bore, 14.6 in. stoke, 310 rpm; exhaust pipe 5.9 in. dia., 356 in. long; exhaust pot 8.7 cu. ft. (J. Zeman, *Zweitakt Dieselmaschinen.* Copyright, 1935, by Julius Springer in Vienna.)

the scavenge air. The indicated volumetric efficiency of the scavenge pump, which was 80 per cent in the first case, dropped to 50 per cent. The attainable engine output was reduced in the same proportion.

16.25 Measuring the Scavenge Pressure.

An alternative, and in many instances simpler, way of checking exhaust tuning is by measuring the scavenge pressure at varying engine speed. If the air is supplied by a blower which is mechanically coupled to the engine, a plot of scavenge or air-box pressure against rpm will appear like curve *A* in Fig. 16-20, provided exhaust pipe effect is absent. This would occur with exhaust pipes completely removed from the engine.

With exhaust pipe mounted, the plot is a wavy line instead of a smooth parabola, because at resonant speeds the interaction between the exhaust and the gas column vibration results in increased resistance to the flow. At other speeds the interaction produces a reduced resistance and greater flow. With a well-tuned exhaust the graph may be a line like curve *B* which shows a relative minimum in scavenge pressure at normal speed. Good tuning manifests itself in low scavenge pressure and is associated with **high** air delivery. If the exhaust system is poorly tuned the result is a line like curve *C* which shows high scavenge pressure at the normal speed. The same exhaust pipe would be just right for an engine speed some 20 per cent lower, but for normal speed it is very poor. In order to correct the tuning, the natural frequency of the exhaust system must be reduced, which can be accomplished by making the exhaust pipe longer.

Optimum speed is the one at which a tangent to curve *B* parallel to curve *A* is lowest. In order to plot curve *A*, it would be necessary to run the engine without any exhaust pipe, which is impractical in a closed building. However, the parabola *A* is practically the same if it is obtained from a motored engine without any fuel injection. The test engine can be motored with an electric dynamometer or by other suitable means, with the exhaust pipe and ducts removed. The parabola so obtained can be used for comparison with the wavy line which is obtained with the exhaust system mounted and the engine loaded to approximately full load. The relation of the two curves discloses whether the exhaust tuning is right or off, and in the latter case whether the pipe should be lengthened or shortened.*

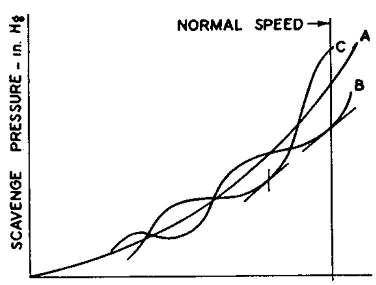

Fig. 16-20. Scavenge Pressure *vs.* Rpm Shows Exhaust Tuning or Lack of It.

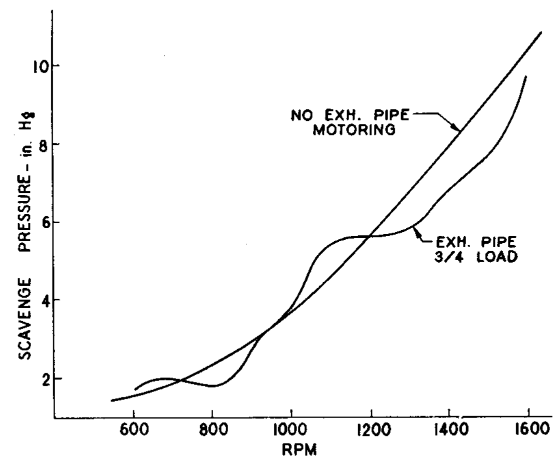

Fig. 16-21. Scavenge Pressure *vs.* Rpm of a Three-Cylinder GM 3-71 Engine. The exhaust system mounted shows favorable tuning from 1300 to 1500 rpm and unfavorable tuning around 1100 rpm. (Penn State Diesel Laboratory.)

* It should be borne in mind that maximum air delivery does not always mean maximum air charge. If the length of the exhaust pipe affects the trapping efficiency (short-circuited air), the tuning which is optimum for air delivery will not be optimum for the air charge.

16.26 Test Results.

Figure 16–21 shows the results of an actual test on a General Motors 3–71 diesel engine and it shows that with the given exhaust system the exhaust tuning was favorable between 1300 and 1500 rpm and unfavorable around 1100 rpm. If that engine with that exhaust system were to run regularly at 1100 rpm the exhaust pipe should be lengthened somewhat to obtain optimum conditions.

The advantage of good tuning usually manifests itself not only in greater power but also in reduced fuel consumption and lower exhaust temperature. In the test mentioned above there was actually no reduction in the specific fuel consumption, but normally the fuel consumption is greater at the higher speed because of the increased friction losses. The exhaust temperature on the other hand actually dropped from 750 to 720 F while speed rose from 1200 to 1400 rpm. The exhaust temperature is so sensitive to exhaust tuning that a plot of exhaust temperature against rpm could itself be used as an index of tuning.

16.27 Tuning the Intake.

A similar method can be used for tuning the intake pipe or checking its tuning. This refers, of course, to engines with reciprocating types of scavenging blowers. With rotary-type blowers no intake tuning is necessary or possible.

If *one* reciprocating blower (air pump) is used in a separately scavenged multicylinder engine as well as in crankcase-scavenged or separately scavenged single-cylinder engines, an intake pipe of the proper length can improve the air delivery substantially, while one with bad tuning can spoil it. Even if space limitations or convenience preclude the use of the optimum pipe lengths, small variations in the length may produce distinct improvements in air delivery.

A simple scheme [Leadbetter, 1944] for checking the tuning of the intake pipe consists of motoring the reciprocating blower at various speeds in the vicinity of its normal speed. The air discharge of the pump (or the engine) is led into a surge tank which has an orifice to the atmosphere of such size that at normal speed the surge tank pressure approximately equals the scavenge pressure of the engine.

Then the surge tank pressure is plotted against the pump rpm. The resulting curve is an undulated parabola, similar to those shown in Fig. 16–20. **But now the high points are the good points.** If the normal speed is higher than a *good* speed, the natural frequency of the intake system must be raised by making the intake pipe shorter. If the normal speed is lower than the next *good* speed, the intake pipe should be lengthened.

16.28 Checking the Kadenacy Effect.

It was shown in Chapter 13 that sudden evacuation of the engine cylinder produced by a well-timed rapid opening of the exhaust port (or valve) helps the charging process by reason of the vacuum created in the cylinder through the inertia of the outrushing gases. This phenomenon is frequently described as the *Kadenacy Effect.* Well-designed two-stroke cycle engines generally enjoy the benefit of the Kadenacy effect. The Kadenacy effect may enable the engine to run even without any blower; otherwise it serves in helping the blower. It was pointed out that with Kadenacy effect present the scavenge pressure actually is lower at heavy load than at light load while without the Kadenacy effect the opposite is true.

This circumstance makes it possible to test the Kadenacy effect in the engine. Again the scavenge or air-box pressures are plotted against rpm but at various load or fuel control settings. If the scavenge pressure decreases with the load, the benefit of the Kadenacy effect is obtained; if the scavenge pressure increases with the load it is not.

In Fig. 16–22 the scavenge pressures of a General Motors 3–71 diesel engine are plotted against the rpm of $\frac{1}{4}$-, $\frac{1}{2}$-, and $\frac{3}{4}$-load. Since the $\frac{3}{4}$-load curve lies generally somewhat lower than the other two, it appears that Kadenacy effect is present but that it is not pronounced.

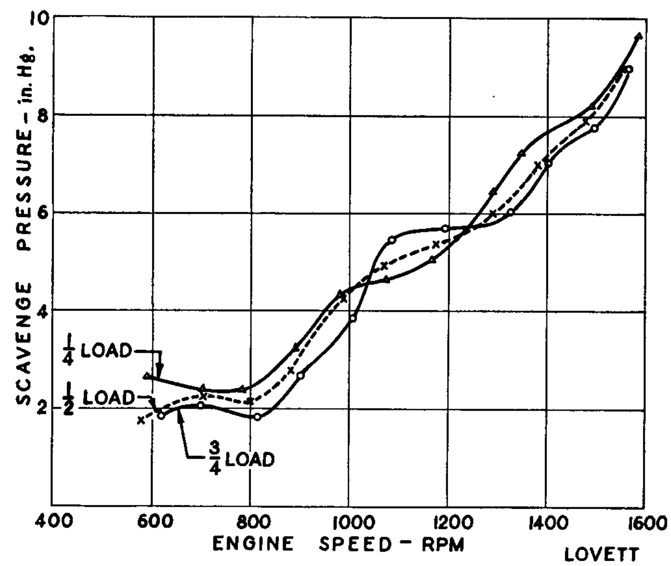

Fig. 16-22. Variation of Scavenge Pressure with Speed and Load. General Motors 3-71 Engine. (Penn State Diesel Laboratory.)

16.29 Rotary-Valve Low-Pressure Indicator.

In investigating scavenging it is frequently necessary to record the pressure fluctuations in the intake or exhaust pipes and receivers. Ordinary weak-spring indicators are suitable for this purpose if the speed is not more than a few hundred rpm. At higher speed the conventional piston indicator is too sluggish to record the pressure variations, and balanced pressure indicators or oscillographs are frequently used. The calibration of either offers certain difficulties in the low pressure range.

On the other hand, the instrument shown in Fig. 16–23 has proved itself very satisfactory for indicating exhaust pressure, scavenge pressures, and the like. It was developed at the Pennsylvania State College Diesel Laboratory [Schweitzer, 1944]. Figure 16–24 is a schematic drawing of the indicator. It consists of a rotor R, with a single distributor hole H, driven from the engine crankshaft through spline driveshaft D. The rotor rotates inside of stator S which has 36 equally spaced passages P, and at every ten degrees brings another manometer tube M into connection with the exhaust port. The connection to the exhaust (or other space the pressure of which is to be indicated) is through a tubing attached to the stationary cover plate C, above rotor R, the distributor hole of which remains in continuous connection with the connecting tubing. The connection of the indicator to the exhaust port of a 3–71 General Motors engine is shown on Fig. 16–23. To prevent soot from the exhaust from getting in the manometer tubes, a small fiberglas filter is interposed between the exhaust and the cover plate of the rotary valve, as shown on Fig. 16–23. The manometer tubes corresponding to 0, 10, 20, etc. degrees of crank rotation are identified by a rotatable band-type degree scale (Fig. 16–23) near the top of the manometer tubes, which can be adjusted to proper phase relation with the crankshaft.

The rotor is lap-fitted in the stator and cover plate, which insures a good seal and makes leakage loss negligible. Since mercury is too heavy for accurate reading of exhaust pressures, while water is too light, bromoform (specific gravity 2.87) has been used, with a drop of methyl orange added for better visibility.

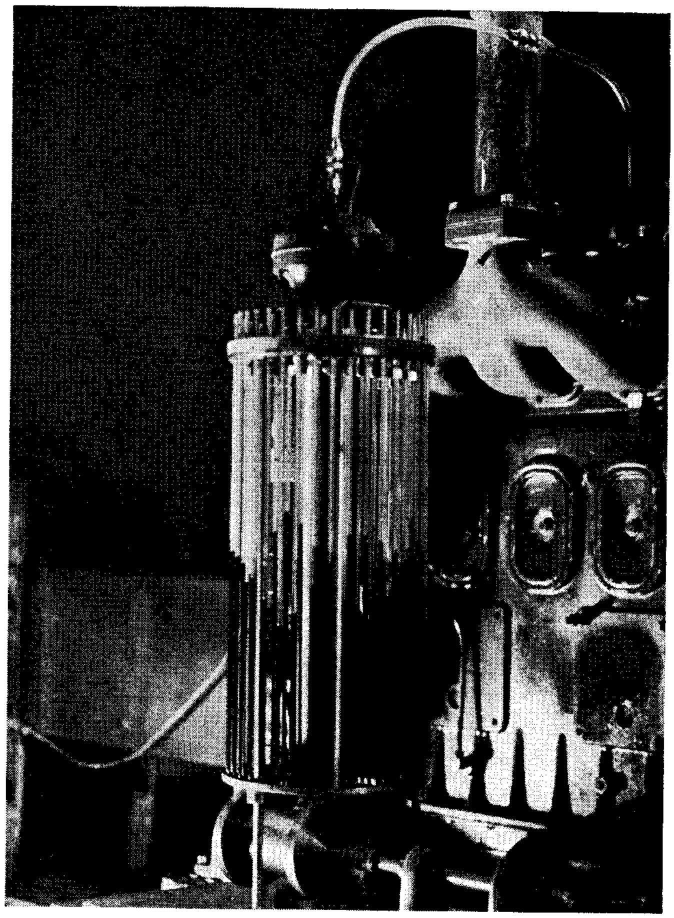

Fig. 16-23. Penn State Rotary-Valve Low-Pressure Indicator. (Penn State Diesel Laboratory.)

16.30 Pressure Record.

Figure 16–25 shows a pressure record obtained with the instrument from a General Motors 3–71 engine over 360 degrees of crank rotation. The degree scale always refers to the crank of No. 1 piston even when the pressure fluctuations in the exhaust ducts of cylinder No. 2 and 3 are recorded.

First it will be noted that the pressures in the three exhaust ducts (the ducts were tapped close to the exhaust valve) at the same instant are much alike. This is not surprising because there were only a few inches between the tapping points, and the common header was also only less than a foot from the tapping points of the ducts. The equalization of pressures was therefore almost perfect, and individual tuning of cylinders is therefore out of the question.

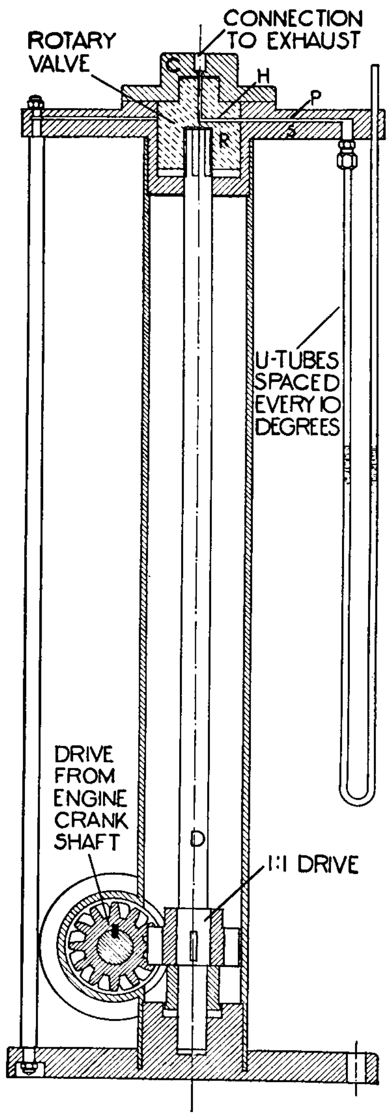

Fig. 16-24. Schematic Drawing of Penn State Rotary-Valve Low-Pressure Indicator.

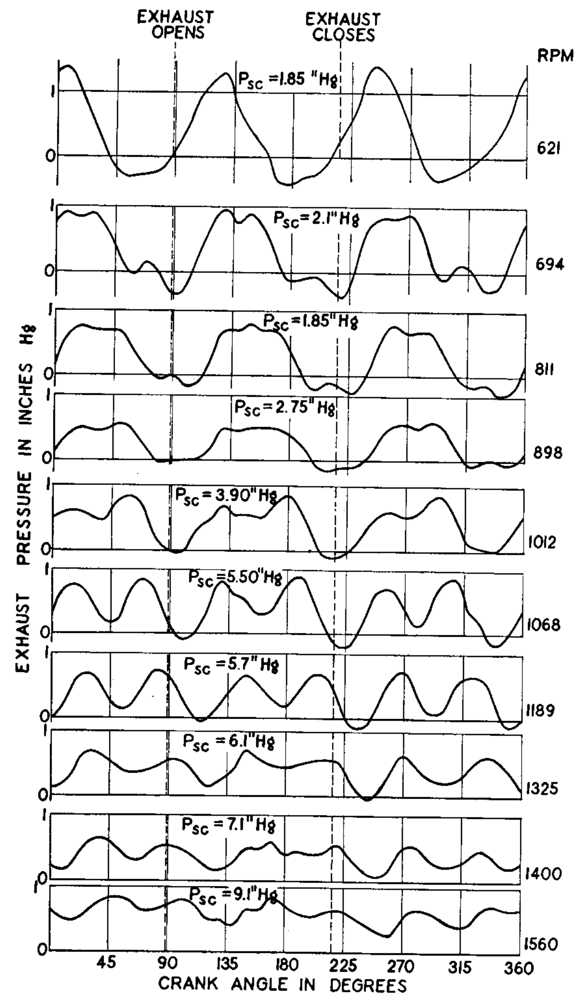

Fig. 16-25. Exhaust Pressure Fluctuations from a GM-71 Engine (Lovett), (Penn State Diesel Laboratory.)

16.31 Checking the Temperature of the Intake Air.

High air-charge temperature reduces the power output appreciably. Figure 11–21 shows that with a scavenge pressure of 6 psig, the adiabatic temperature rise amounts to 54.6 F and the friction may cause another 42 degree rise, which together would raise the temperature of the intake air by 96.6 F. This may entail a power loss of about 8 per cent.

High intake-air temperature is undesirable also because it aggravates cylinder and piston cooling and may lead to lubrication troubles or piston seizure.

The temperature of the intake air should be checked by measurement in the intake duct or the scavenging receiver. If the intake-air temperature is too high, as it is prone to be with high scavenge pressures, it may be reduced by a more efficient blower or by cooling.

Intercooling of the intake air is seldom used because of the large cooling surface required to cool the air to approximate outside temperature. However, a cooler, which would reduce the temperature of the blower air by one-third or one-half of the temperature rise, would not need to be so large, and with high scavenge pressure such an intercooler may be justified.

If an engine has ignition difficulties because its compression ratio is too low, its surface-to-volume ratio too high, or for other reasons, then, of course, the temperature rise in the blower is beneficial in promoting ignition.

PROCEDURE

16.32 Regarding scavenging, the following questions may be answered by well-planned simple experiments:

1. Is the air delivery adequate?
   See section *Measuring Air Consumption* (16.3–16.5).
2. Is the short-circuiting within acceptable limits?
   See sections *The Tracer Gas Method*, and *Sampling Tests* (16.6–16.14)
3. Is the exhaust lead sufficient?
   See section *Checking the Exhaust Lead* (16.18–16.20).
4. Is the inlet-port closure too late?
   See section *Checking the Closure of the Inlet Ports* (16.21–16.22).
5. Are the inlet and exhaust ports in balance?
   See section *Checking the Balance of Inlet- and Exhaust-Port Areas* (16.23).
6. Is the exhaust piping well-tuned?
   See section *Checking Exhaust-Pipe Tuning* (16.24–16.27).
7. *Is the temperature of the intake air not excessive?* This may be determined by thermometers in the air box or between the blower and the cylinders and by comparing the results with Fig. 11–21.
8. *Are the pressure losses in the intake system admissible?* Water or micromanometers (Fig. 16–4) furnish the answer.

The foregoing does not represent an exhaustive list of all scavenging tests, as only relatively simple tests have been described. Emphasis was placed on such tests as can be performed with instrumentation that is available in most engine laboratories. Research types of tests have either been ignored or described for reference purposes only.

Naturally not all experiments described need be made on a specific engine nor need those be made in the order listed. Selection of the tests and their order depends on the judgment of the experimenter and should suit the test equipment available. Sometimes the signs point to one item that is responsible for the unsatisfactory performance. If in concentrating on that item the situation is remedied and satisfactory performance obtained, the other tests may be omitted. Otherwise the search should continue until the source of the unsatisfactory performance is located and the desired results obtained.

16.33 Checking Port Timing.

Before making more elaborate tests, it is sensible to check the port timing to make sure that the actual port timing in each cylinder corresponds to the figures on the blueprint. Errors occur even in good shops.

For this test, as well as many others, it is necessary to have a dependable crank degree scale, preferably on the flywheel. First, mark top center position of the piston next to the flywheel. This is best done by using a dial indicator with an extension rod to reach the piston. By barring the engine over, mark a small angle on the flywheel that gives the same dial indicator readings, and top center by bisecting that angle. Then mark every degree on the flywheel from 0 to 359 in the direction corresponding to normal engine rotation. It is advisable to make the degree scale in a permanent manner by making a rigid pointer and scribing the marks on the metal face of the flywheel.

One way to check the port timing is to place an electric light inside the cylinder, remove the inlet and exhaust manifolds, and make readings on the graduated flywheel at opening and closing of exhaust and inlet ports by observing the light. Sometimes the ports are not accessible for visual observation. In that case the air-blowing method may be used. Through an opening like that of a spray nozzle, the cylinder is supplied with air of a few pounds pressure. All other openings are closed except the port to be checked. A simple venturi or a gas meter shows the escape of the air through the port (see Fig. 16–26). Naturally some little air flow takes place even with the port covered, especially after the top piston ring passes the port. The air flow increases rapidly, however, from the point where the port is uncovered by the piston edge. In this way the actual opening and closing angle of the exhaust and inlet ports can be determined.

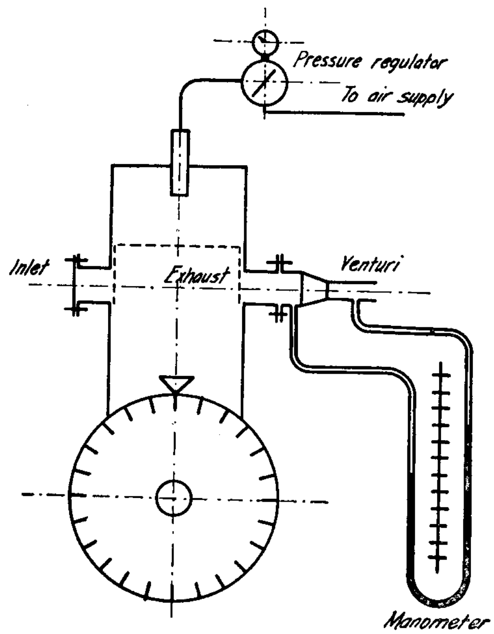

Fig. 16-26. Arrangement to Check Port Timing of an Engine.

16.34 Indirect Testing of Scavenging.

Information on scavenging is sometimes obtained by a method of elimination. If it is established that combustion is good and that the friction and pumping losses are not excessive, and the engine still fails to approximate the bmep shown on Fig. 5–8, 5–9, and 5–10, then the air charge is below par.

A good test of combustion is measuring the specific fuel consumption, preferably on the indicated horsepower basis. If the specific fuel consumption is around 0.3 pound per ihp-hr the combustion must be good. If the engine nevertheless fails to deliver the expected horsepower, deficiency in the air charge may be suspected. Direct determination of the indicated horsepower may not be convenient, in which case it may be calculated from the measured brake horsepower and the measured or calculated friction horsepower. For measuring the friction horsepower, a motoring test is ordinarily used. The actual friction horsepower may be greater than the motoring horsepower but the difference is not great. The motoring test should be run at close to operating temperature, which can be secured by having the motoring test follow a firing run at the proper load.

Determination of the horsepower absorbed by the blower alone is often useful, as together with air delivery test it reveals the efficiency of the blower. The blower horsepower is frequently obtained by subtraction after motoring the engine with and without the blower.

If motoring equipment (electric cradle dynamometer) is not available the friction horsepower may be estimated by assuming a mechanical efficiency of 70 to 85 per cent. Low-speed engines and/or those employing low scavenge pressures have high mechanical efficiencies while the low figures should apply to high-speed engines with high scavenge pressures.

The friction horsepower may also be determined from fuel consumption measurements by extrapolation. At low loads the combustion efficiency of a diesel engine is fairly constant, which makes the line of horsepower plotted against the fuel consumption (pounds per hour) virtually a straight line. By extrapolating this line to zero fuel consumption, the line intersects the ordinate axis at a certain negative horsepower which may be taken as the friction horsepower of the engine. Of course, the engine speed must be held constant during this test.

EXHAUST SMOKE

16.35 If the specific fuel consumption is excessive, it is frequently accompanied by exhaust smoke which by itself is a fairly dependable index of combustion. Discoloration of the exhaust is always caused by the presence of liquid or solid particles in the exhaust gas.

Liquid particles (fog) consist of unburned fuel or lubricating oil and occasionally of condensed water vapor. **Solid particles (soot) are generally carbon particles resulting from incomplete combustion of hydrocarbons** and occasionally powder of inorganic material such as ash.

Smoke originating from unburned liquid oil particles has a white color; that caused by droplets of lubricating oil has a bluish tinge; and solid soot particles make the smoke gray-brown varying from light gray to brown-black, depending on the amount of soot contained in the exhaust. With some reservation, it may be stated that exhaust smoke is always the product of combustion that is still in progress when the gas leaves the engine cylinder.

Exhaust smoke may be the product either of misfire or late burning. Misfire represents the case where combustion has not yet begun when the exhaust opens, while late burning combustion is still in progress when the exhaust opens. *Misfire produces cold smoke* (resulting from fuel particles ignited too late or not at all) while *late burning produces hot smoke* (resulting from sluggish combustion of the fuel particles), [Schweitzer, 1947]. These conditions frequently appear side by side because some fuel particles ignite but fail to complete combustion while others fail to ignite at all.

Late burning may be caused by either sluggish combustion or late ignition. Sluggish combustion means too low a burning rate, late ignition is the result of too late an injection or too long an ignition lag.

16.36 Sluggish Combustion.

The most important factor causing sluggish combustion is an overrich mixture. Fuel burned with a deficiency of oxygen produces soot. In a diesel engine, because of the nonuniform distribution of the fuel, it usually requires 50 to 100 per cent excess air trapped in the cylinder to obtain clear combustion.

Poor scavenging or insufficient air delivery reduces the amount of pure air trapped in the cylinder and thereby the margin of excess air, and so contributes to smoky exhaust. On the other hand, everything that improves mixing tends to reduce local overrichness and enables the engine to get along with a smaller margin of excess air with clear exhaust. Poor atomization, poor spray distribution, and insufficient turbulence cause local overrichness and exhaust smoke, particularly at heavy load, when the over-all air-fuel ratio has a smaller margin.

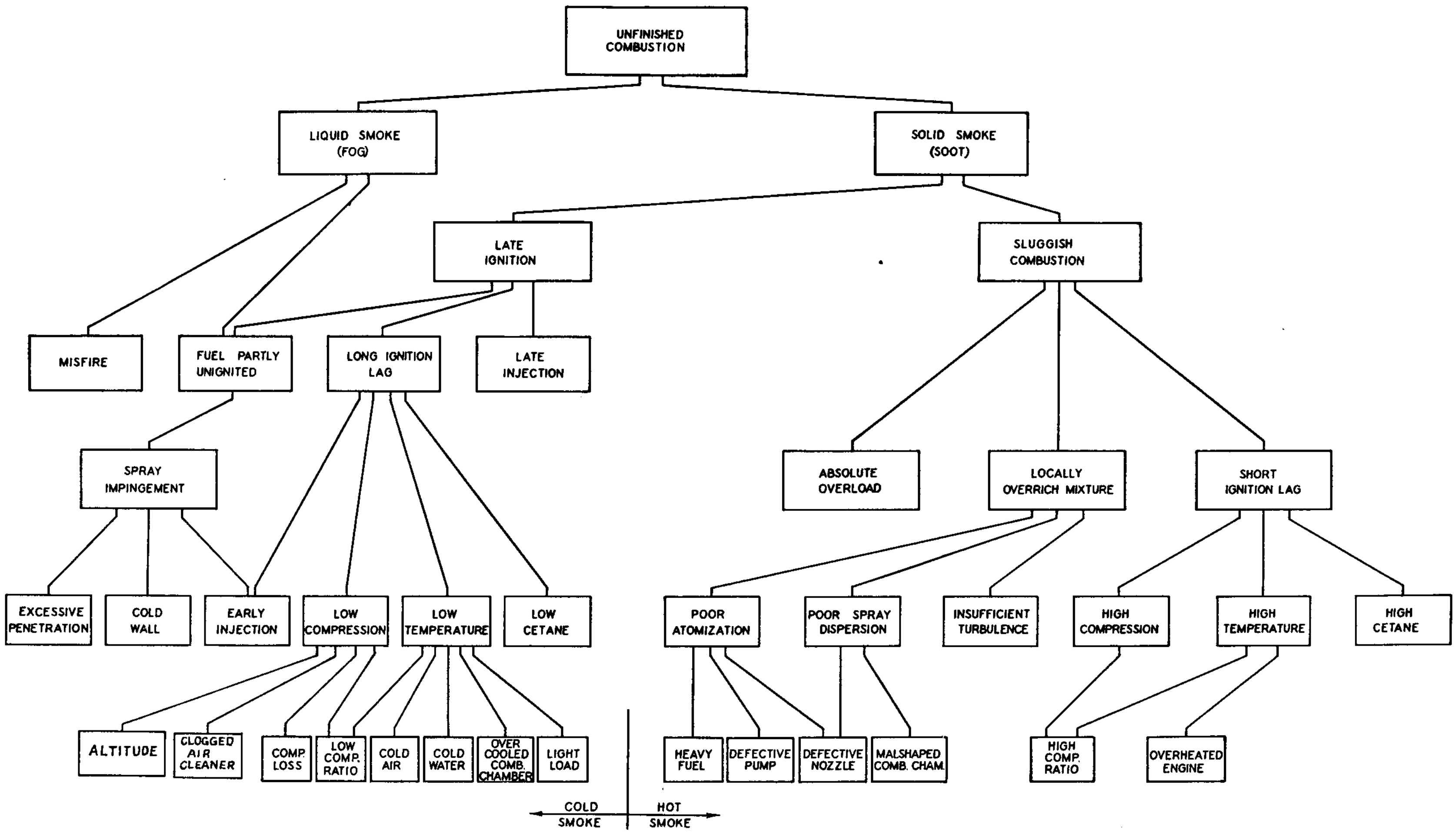

Fig. 16-27. Smoke Chart.

16.37 Late Ignition.

The other general cause of the exhaust smoke is late ignition, caused either by late injection or by long ignition lag or by the combination of the two. Late injection can be corrected by advancing the injection timing. If late ignition has been caused by too long an ignition lag, it cannot be compensated for by earlier injection. Effective remedies are increasing the compression temperature or the compression pressure, or the compression ratio, which does both. A temporary expedient is using extra-high cetane fuel, which may be the correct course in an experimental engine but is generally undesirable in a production engine.

16.38 Smoke Diagnosis.

In the course of engine development, there is ordinarily a stage when exhaust smoke looms as a major problem. The cause may be merely maladjustment such as late injection, poor spray atomization or dispersion, or a cold engine operating close to the region of misfiring. The smoke may, however, be caused by design defects that are more difficult to correct. The logical first step is a **smoke diagnosis**. The chart shown in Fig. 16–27 may be helpful in diagnosing the smoke.

If a variation of injection timing fails to produce the desired relief, the first problem is to determine which type of smoke is present, because the remedy applied must fit the case to be effective. If the difficulty is sluggish combustion resulting in hot smoke, the rational remedy is trapping more air, using a better distributed spray, or more turbulence. If the difficulty is cold smoke from late ignition, the rational remedy is higher compression ratio or a hotter combustion chamber.

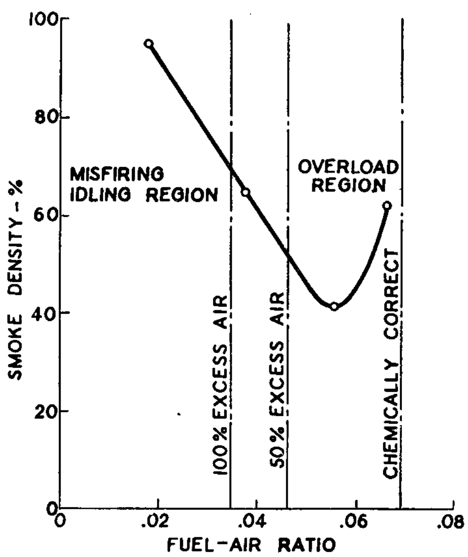

Fig. 16-28. Smoke Density Plotted against Load Indicates a "Cold" Engine. Compression ratio, 12 : 1. (Penn State Diesel Laboratory.)

16.39 Simple Tests.

One or two simple tests may prove to be effective timesavers in smoke diagnosis. In one, smoke density is plotted against load. If the smoke rise is more pronounced at the idling range, the probable trouble is cold smoke because of excessive ignition delay. Figure 16–28 shows an example of such smoke pattern. If the smoke rise is more pronounced in the heavy load range, the probable trouble is sluggish combustion. An example of this kind is shown in Fig. 16–29. Figure 16–30 shows a normal smoke pattern for comparison. Incidentally the only difference between these three smoke patterns was in the engine compression ratio. Figure 16–28 was taken with the very low compression ratio of 12:1, Fig. 16–29 with the very high compression ratio of 18:1, and Fig. 16–30 with the normal compression ratio of 16:1.

The other test is to try a more volatile fuel. Such a fuel can be concocted by mixing straight-run gasoline with diesel fuel. An experimental engine run on such a fuel ceases smoking if the smoke was caused by sluggish combustion, but smoke gets worse if it was caused by late ignition. In one case, because of poor mixture due to a defective nozzle, the smoke density was 74 per cent at full load with standard fuel. With a 50–50 blend of straight-run gasoline and 30 cetane diesel fuel the smoke was brought down to 35 per cent. With a good nozzle the standard fuel gave 35 per cent smoke. At low load the effect was if anything reversed.

Either of the two tests, or a combination of them, ought to clear up questions as to the origin of exhaust smoke and direct subsequent experimentation into the proper channel.

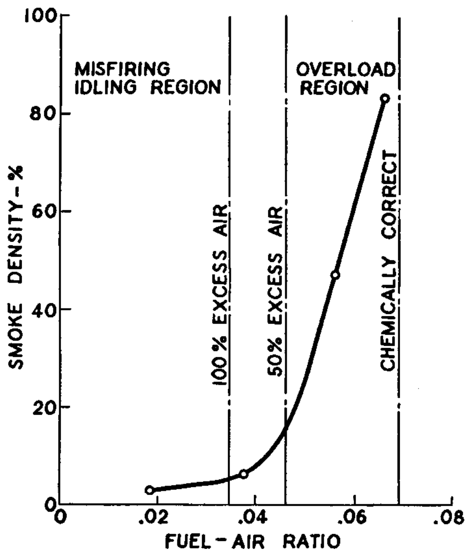

Fig. 16-29. Smoke Density Plotted against Load Indicates a "Hot" Engine. Compression ratio, 18 : 1. (Penn State Diesel Laboratory.)

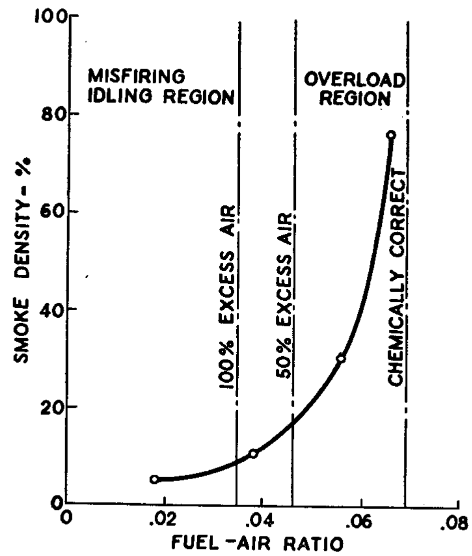

Fig. 16-30. Smoke Density against Load, "Normal" Engine. (Penn State Diesel Laboratory.)

16.40 Therapy.

To combat **delayed ignition**, the remedy must be sought outside of the scavenging. In using standard fuels, either the charge or the combustion chamber must be made hotter. Frequently a certain spot in the combustion chamber is chilled. If that happens to be the spot where ignition is to take place, the result is disappointing. A hotter piston head or a hot insert in the combustion chamber may be the answer. If the smoke is particularly bothersome at light load or idling, reduction of the air charge at light load may effect good results. This is frequently the case with small-sized, blower-scavenged engines. Decreasing the air delivery of the blower at low speed by increasing the clearance of the rotor or by-passing some of the air may produce desirable results.

The practical way to combat **sluggish combustion** is by improving the mixing. The old game of fitting sprays by cut-and-try and/or applying some form of turbulence must be played until the engine can carry its rated load with clear exhaust. Arrangements in which the fuel spray is directed **against** the air flow usually give unsatisfactory results. Only after attempts at improving the mixing have failed to produce clean combustion or there is other evidence that the air charge is inadequate, should efforts be directed at increasing the air delivery or improving the trapping efficiency of the cylinder.

16.41 Smokemeter.

Visual observation of the exhaust smoke is generally unsatisfactory and the use of a *photoelectric smokemeter* is preferable [Schweitzer, 1942]. Figure 16–31 shows a cross section of the tube unit of a smokemeter. The principle of operation is that a sample is taken of the exhaust smoke by a tube connection to the exhaust pipe and the opacity of the smoke column is measured by a photocell.

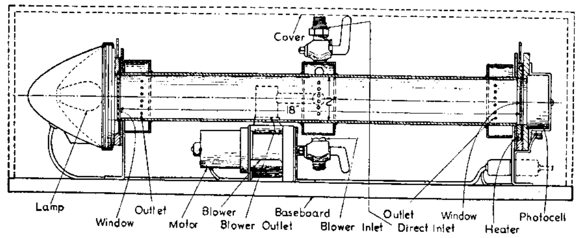

Fig. 16-31. Cross Section of CRC–Photovolt Smokemeter. (By permission of Photovolt Corp.)

The dial of the milliammeter, which indicates the current produced by the photocell, is so calibrated that per cent smoke density is read directly. With no current (light blocked completely) the dial reads 100. With clean air in the smokemeter tube the dial reads zero, and with one half of the light blocked off (by covering half of the photocell area) the smokemeter reads 50. A rheostat allows to set the no-smoke reading zero even when the windows are partially opaque because of soot deposits.

CHAPTER 17

CONSTRUCTIONAL FEATURES OF REPRESENTATIVE TWO-STROKE CYCLE DIESEL ENGINES

AUTOMOTIVE ENGINE

17.1 A popular type of automotive diesel engine is the General Motors Model 71 engine, frequently used in trucks and buses. A great number of them has also been used in small boats and landing craft. The Model 71 series of engines is generally made in three-, four-, and six-cylinder sizes, the general specifications of which are listed in Table 17–I.

Table 17-I. General Specifications of G.M. 71 Engine.

	MODEL		
	3–71	4–71	6–71
Number of cylinders	3	4	6
Bore (inches)	$4\frac{1}{4}$	$4\frac{1}{4}$	$4\frac{1}{4}$
Stroke (inches)	5	5	5
Total displacement (cu in.)	212.8	283.7	425.6
Hp — max @ 2000 rpm	82	105	160
Hp — max @ 1200 rpm	59	78	118
Hp — continuous rating @ 1200 rpm	50	67	100
Bmep — (psi @ 1000 rpm max)	93.5	94.0	94.2
Bmep — (psi continuous rating)	77.5	77.5	77.5
Max torque (ft-lb @ 1000 rpm)	264	354	532
Compression ratio, nominal	16 : 1	16 : 1	16 : 1
Piston speed (ft per min at 1000 rpm)	833	833	833
Firing order — right-hand drive	1–3–2	1–3–4–2	1–5–3–6–2–4
Firing order — left-hand drive	1–2–3	1–2–4–3	1–4–2–6–3–5
Weight of basic engine [lb (dry)]	1209	1363	1711
Lubricating oil capacity [qt (dry)]	15	17	29
Water capacity [qt (engine only)]	10	14	22

Figure 17–1 shows a six-cylinder unit and Fig. 17–2 a longitudinal cross section of it. The transverse cross section was shown in Fig. 7–10.

This series is so designed that the major parts are completely symmetrical and the engine can be assembled either hand. Thus for marine use, twin-screw propelling plants can be obtained which are true right- or left-hand units, using the same parts in the assembly, thus simplifying the spare parts problem. Similarly, other units can be made up to suit the exact requirements of the purchaser.

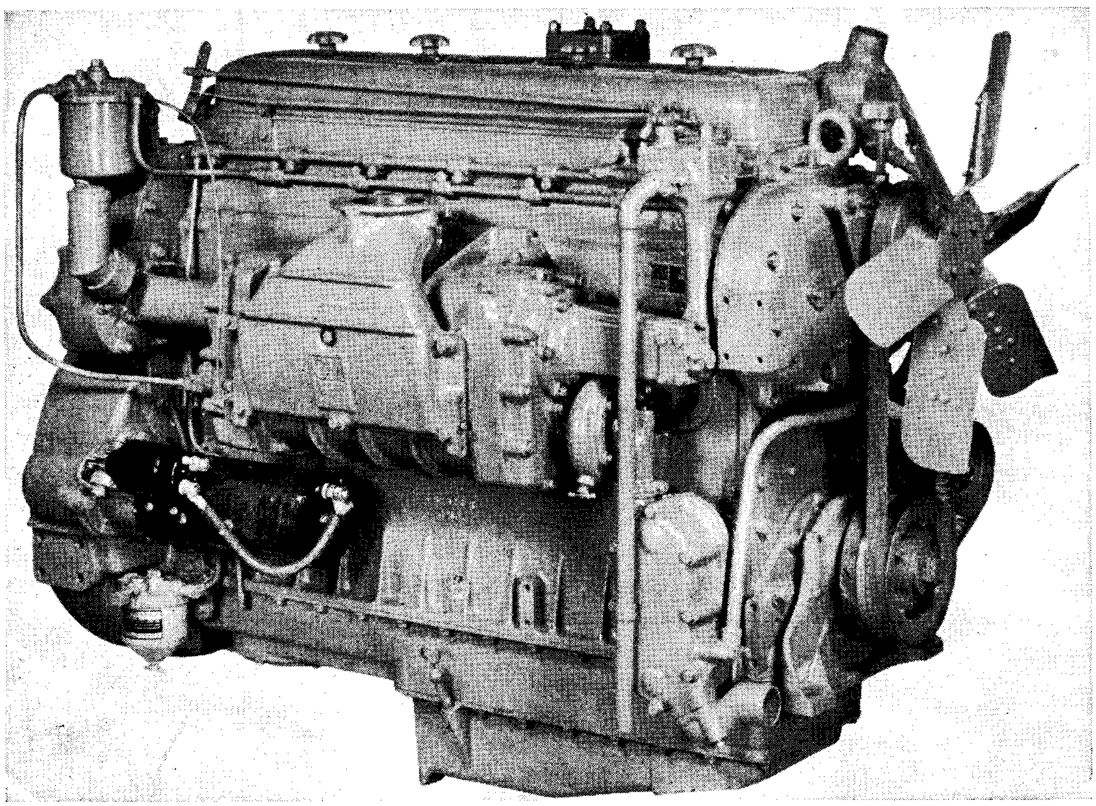

Fig. 17-1. General Motors 6–71 Engine.

17.2 Cylinder Block and Crankcase.

The cylinder block and crankcase is a one-piece casting made of cast iron (Fig. 17–3). The two ends of the block are identical so that the same flywheel housing and gear train can be put on either end of any of three models. The crankshaft is underslung in bearings at the bottom of the crankcase. The camshaft and balance shaft are located at the top of the cylinder block on either side of the cylinders, as shown in Fig. 17–4 and 17–9. The scavenging air manifold is also cast into the block (see Fig. 17–4).

17.3 Crankshaft.

The crankshaft is a drop forging with integral counterweights; all main and crankpin journal surfaces are electrically hardened by the Tocco process. Oil holes are drilled between the main and crankpin journals for lubrication. There is a main bearing between every two adjacent cylinders. Main bearing shells are interchangeable and have steel backs with antifriction metal lining.

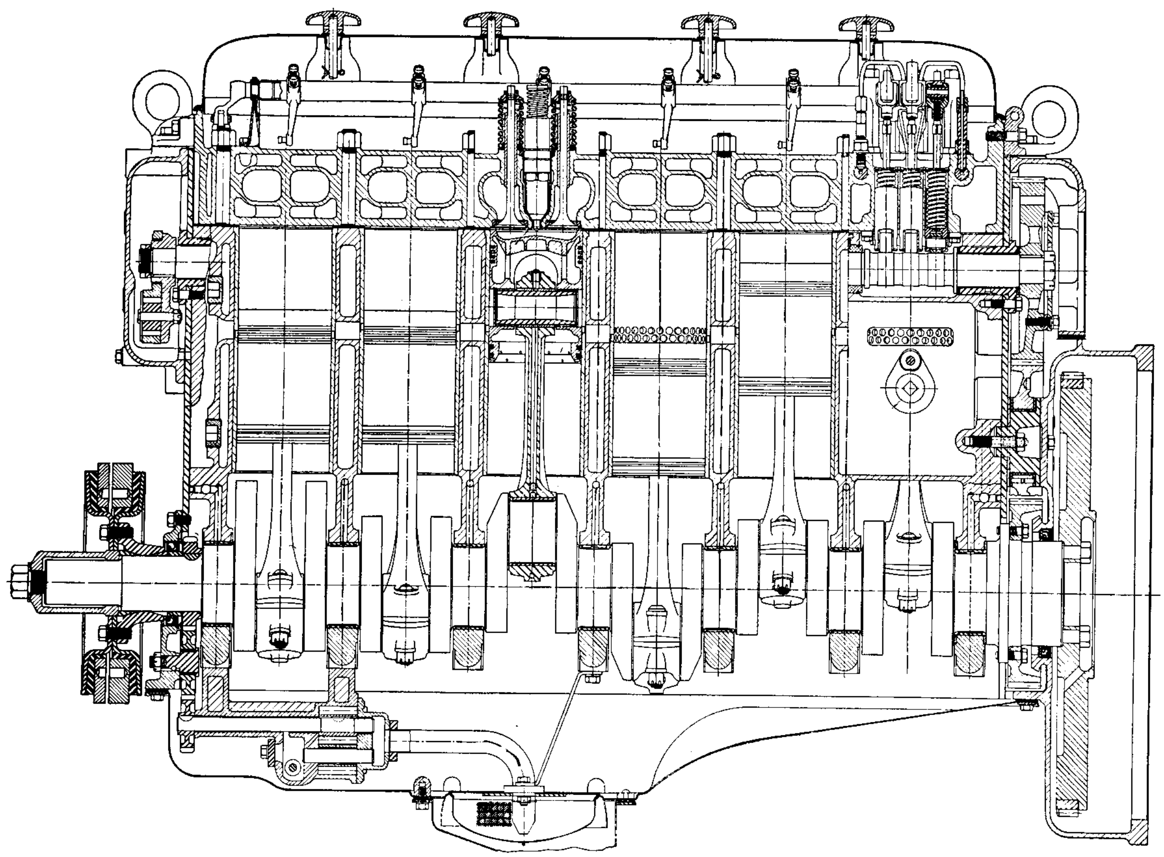

Fig. 17-2. Longitudinal Cross Section GM 6-71 Engine.

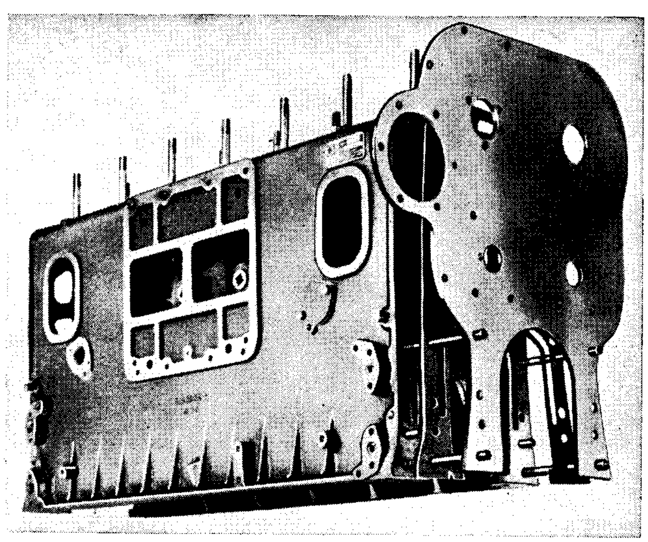

Fig. 17-3. GM 6-71 Cylinder Block. Rear end plate in front.

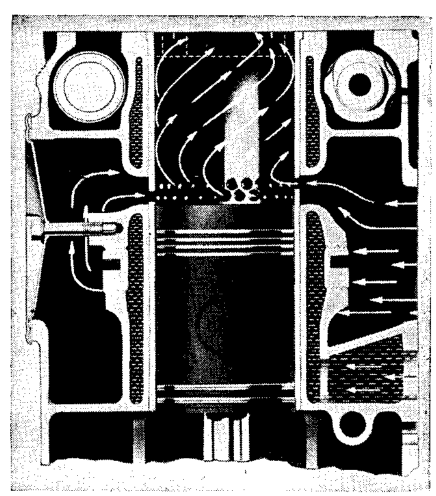

Fig. 17-4. Section of Cylinder Block Showing Air and Water Passages.

17.4 Connecting Rods and Pistons.

I-beam section, drop-forged connecting rods are used (see Fig. 17–5). They are rifle-drilled for lubrication to the wristpins and for piston cooling. Crankpin bearings are steel-backed copper-lead lined shells without shim adjustment. Pistons are of malleable iron and their crowns are shaped to the sombrero profile, as shown in Fig. 17–6.

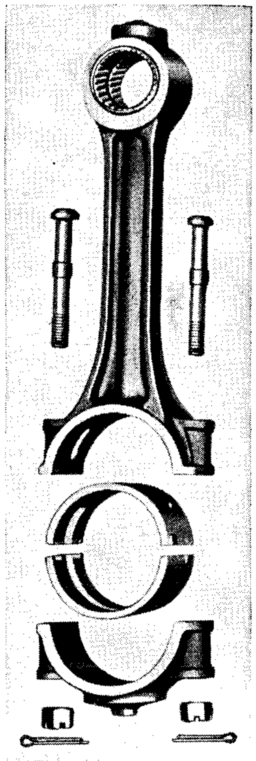

Fig. 17-5. Connecting Rod Details.

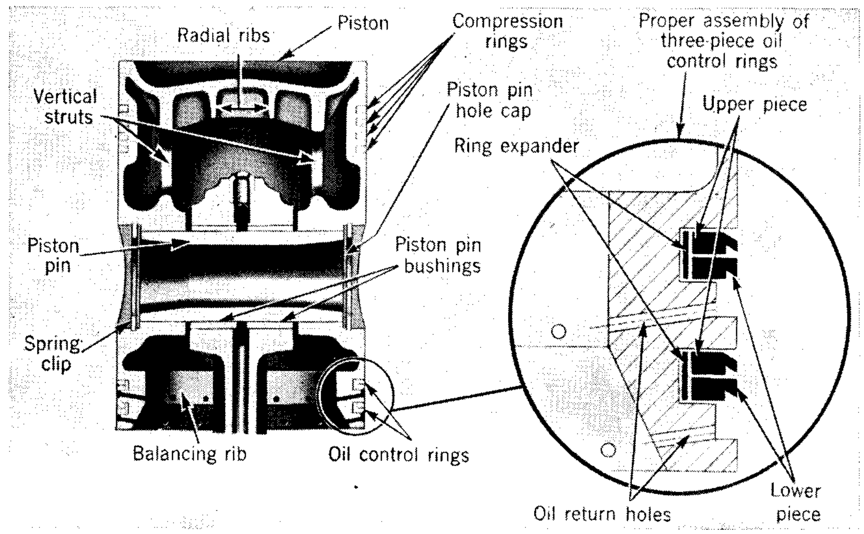

Fig. 17-6. GM-71 Piston.

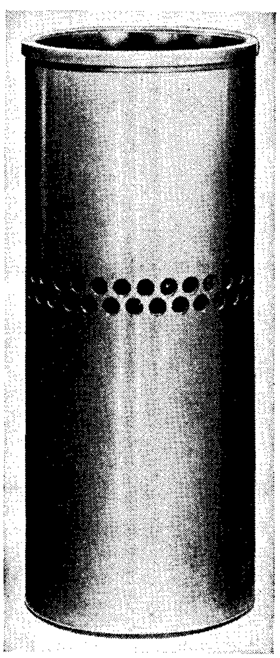

Fig. 17-7. Cylinder Liner.

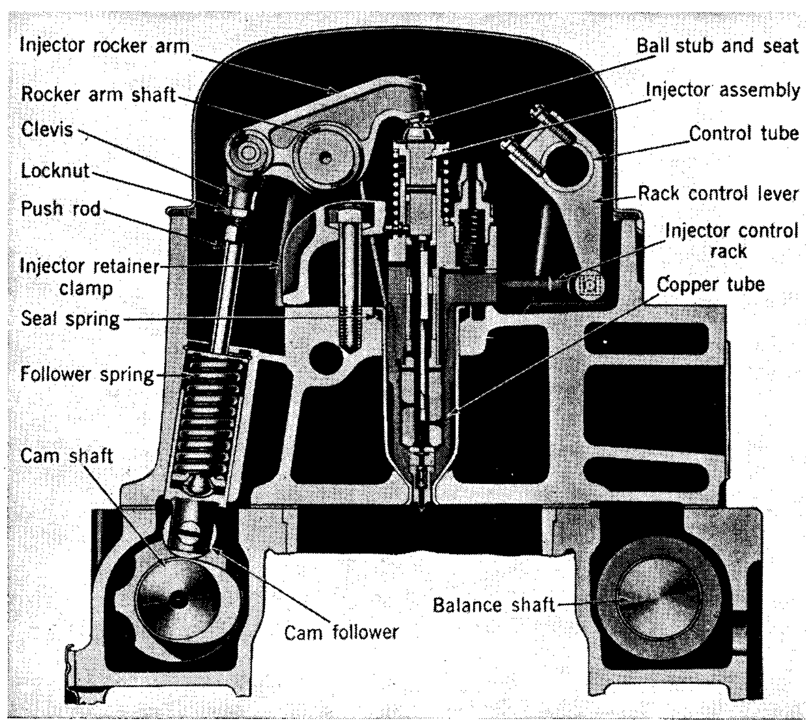

Fig. 17-8. Cylinder Head, Transverse Section.

The top of the piston forms the combustion chamber. The rim closely approaches the cylinder head leaving at top center less than $\frac{1}{16}$ inch clearance, and thus squeezes the air from the periphery into close proximity to the fuel spray. The ribbed head is cooled by lubricating oil forced from a spray jet on the top of the connecting rod. The head is so shaped that the ring-belt is protected by a layer of air from the heat of the high-temperature piston crown.

Two steel-backed bronze bushings with helical grooved oil passages are pressed into the piston to provide a bearing for the hardened, full-floating wristpin. Wristpin holes in the piston are sealed with steel caps, thus lubricating oil returning from the sprayed piston head and working through grooves in the piston bushings is prevented from reaching the cylinder walls.

17.5 Cylinder Liners.

Removable cylinder liners of the dry type are used. These are made of hardened cast iron and are accurately honed to a smooth finish. The liners fit loosely in the cylinder bore. About halfway along the length is a double row of round holes running around the liner which are the air inlet ports, Fig. 17–7.* The liners are water-cooled through their entire length except at the ports which are cooled by the scavenging air.

17.6 Cylinder Head.

All cylinder heads for one engine are cast in one piece following automotive practice. For each cylinder there are two exhaust valves operated by two rocker arms, one on each side of the unit fuel injector. The hole for the injector is above the center of the cylinder and has a copper lining pressed into it, which forms the water jacket (see Fig. 17–8).

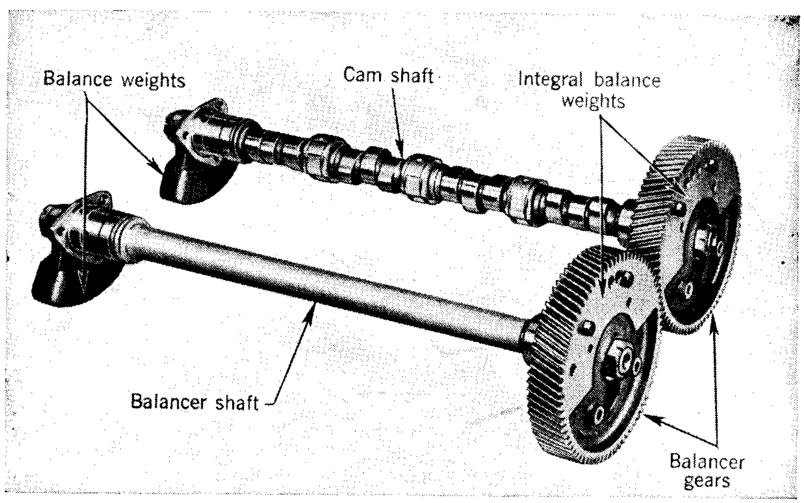

Fig. 17-9. Camshaft and Balancer Shaft.

17.7 Camshaft.

The camshaft is mounted in the top portion of the crankcase. It is made of forged steel and has integral cams (see Fig. 17–9). Roller-type cam followers are used and they transmit the cam motion by means of short push rods to the rocker arms which in turn bear on the exhaust valves and

* In the 1947 model a *single* row of inlet ports is used.

the unit injectors. The camshaft and the balancer shaft are interchangeable in the crankcase and may be assembled either left- or right-hand depending on the type of engine desired. These shafts are geared together and driven in opposite directions by a train of helical gears.

17.8 Fuel System.

There is an individual injector for each cylinder (see Fig. 17-10). The fuel injection pump and nozzle are combined in a single housing to form the unit injector. The pump and barrel are a lap fit with very close clearance to cope with the high pumping pressures characteristic of this system.

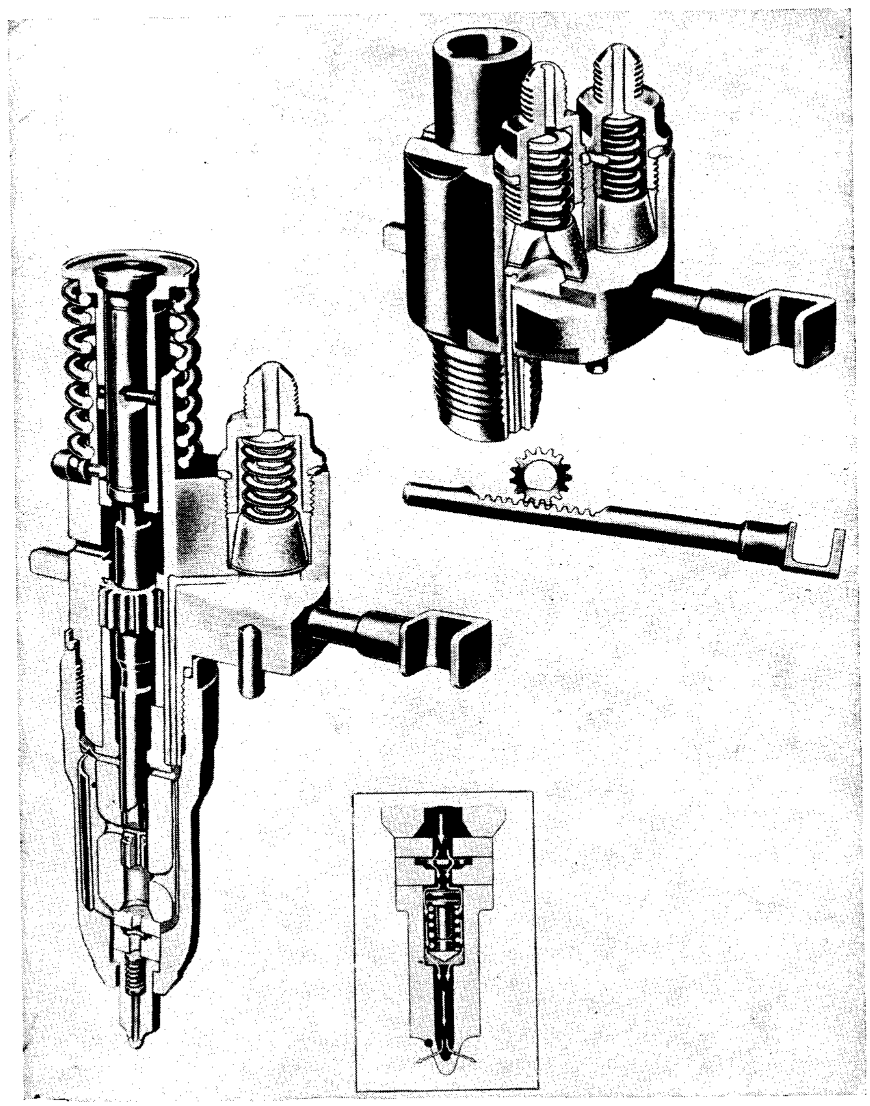

Fig. 17-10. Fuel Injector Assembly.

The amount of fuel injected is controlled by the edges of an undercut near the end of the pump plunger. The top and bottom edges of the undercut are helical shaped, and a hole is drilled radially through the undercut, with a second hole through the end of the plunger, parallel to its axis and meeting the first hole. Ports are provided in the plunger barrel, and are so arranged that the helical grooves cover them for a portion of the stroke. By turning the plunger, the ports are covered for a longer or shorter portion of each stroke and a greater or less quantity of fuel is injected. The plunger is rotated by a gear segment and rack. The racks are attached to the governor control linkage, which moves the racks to compensate for speed changes.

The injection nozzle is located in the end of the unit injector. It receives the fuel directly from the plunger, all external piping being eliminated. There are two check valves to prevent cylinder pressure from blowing back into the fuel system. The nozzle is of the outwardly opening type and the spray tip has six 0.006-inch diameter orifices which inject the fuel spray in the form of a flat cone to approximately the shape of the combustion chamber.

Fuel oil enters the injector body through a filter element and fills the annular supply chamber. This is connected to the barrel by two funnel-shaped ports. As the plunger moves downward, the fuel oil in the barrel is first displaced through the ports back into the supply chamber until the lower edge of the plunger closes the lower port. The remaining oil is then forced upward through the center passage in the plunger into the recess between the upper helix and the lower cutoff from which it can still flow back into the supply chamber, until the helix closes the upper port. The rotation of the plunger retards or advances the closing of the ports and the beginning and the ending of the injection period, at the same time increasing or decreasing the injection quantity. At increasing load the beginning of the injection slightly advances and its termination (cutoff) is retarded.

For transfer or supply pump a positive displacement vane-type pump is used which is to maintain a fuel pressure at about 20 psig in the supply line.

Model 71 engines are normally equipped with mechanical spring-and-flyball governors. Hydraulic-governor is furnished if required.

17.9 Blower.

Gear-driven, Roots-type blowers supply the scavenging air to the air box. Each Roots blower has a pair of helical, three-lobe rotors.

The upper rotor is driven at 1.95 times engine speed, and the lower rotor is driven from the upper rotor through the timing gears. A flexible coupling, formed of bundles of leaf springs, minimizes the transmission of torque fluctuations to the blower.

17.10 Lubrication System.

The lubricating system, shown in Fig. 17–11, consists of an oil pump, an oil strainer, an oil filter, and an oil cooler, with a suitable relief valve in the oil pump, and a by-pass valve between the oil pump and the oil strainer. The gear-type oil pump is driven by a roller chain from a sprocket on the crankshaft (some have gear drive) and is immersed in the crankcase oil. The gear pump delivers the hot oil through the strainers and then through the oil cooler. After leaving the cooler the stream is delivered directly to the main gallery in the cylinder block

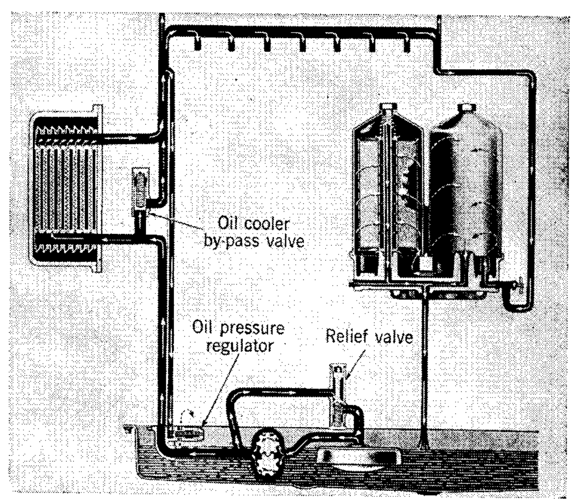

Fig. 17-11. Schematic Lubrication Diagram. Showing path of oil through system having single pressure pump, strainer, cooler, cleanable element filter, and single sump oil pan.

(see Fig. 17–12). A pipe from the main gallery continuously carries a portion of the oil to the filter for cleaning.

The gallery, located on the blower side of the crankcase, distributes the oil under about 50 psig pressure to the main bearings, and to a horizontal transverse passage at each end of the cylinder block. From these two horizontal passages, two vertical bores at each end of the cylinder block carry the oil to the end bearings of the camshaft and balance shaft, as well as to the oil passage in the camshaft, which conducts the oil to the camshaft intermediate bearings.

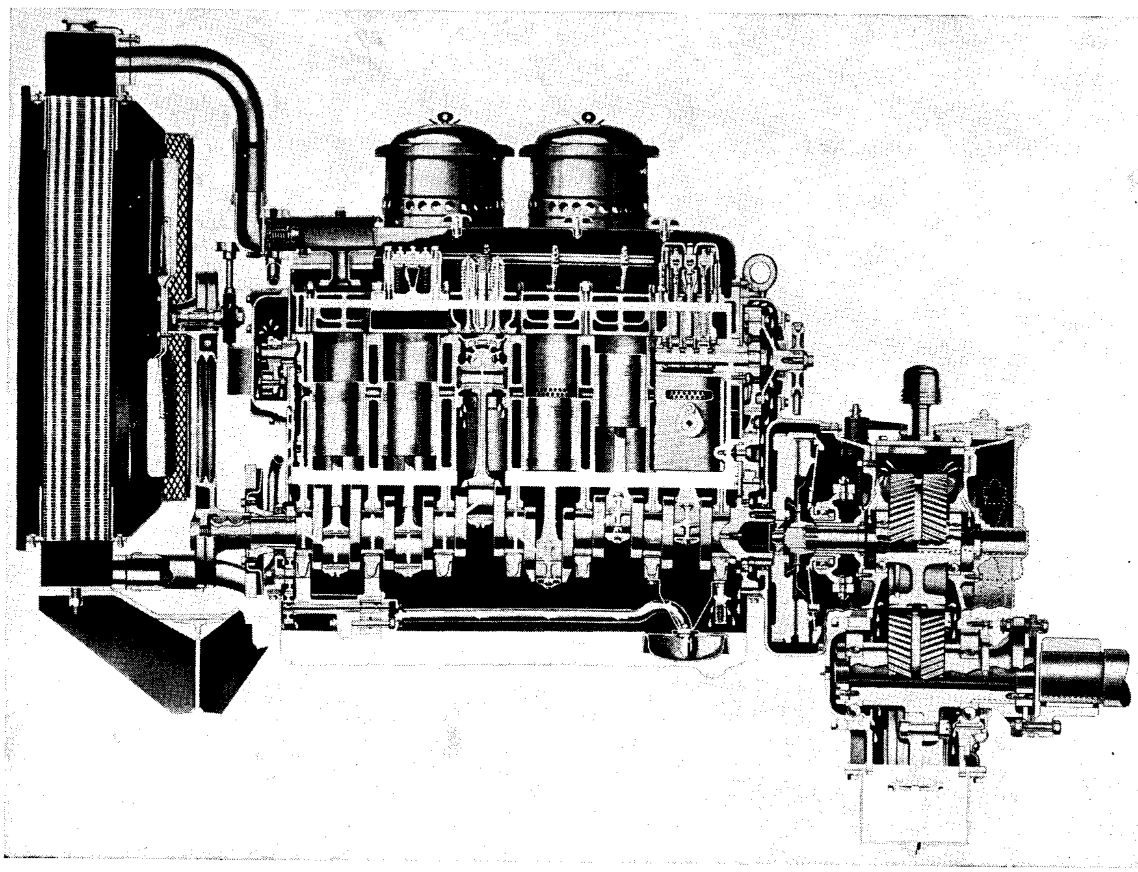

Fig. 17-12. Lubrication Passages.

Oil for lubrication of connecting rod bearings, piston pins, and cooling the piston head is provided through the drilled crankshaft from the adjacent forward main bearings. The gear train is lubricated by overflow of oil from the camshaft pocket through a communicating passage into the gear train cover. A certain amount of oil spills into the gear cover from the camshaft, balancer shaft, and idler gear bearings. The blower drive gear bearing is lubricated through an external pipe from the rear horizontal oil passage of the cylinder block.

A second longitudinal gallery is arranged on the camshaft side of the cylinder head and supplied with oil from one of the vertical bores at each end of the cylinder block. Oil from this gallery enters the hollow rocker arm shafts through the rocker shaft brackets, hollow capscrews, and lubricates the rocker arm bearings and push rod clevis bearings.

Excess oil from rocker arms lubricates the valve ends and push rods and drains to cam pockets in the cylinder head from which the cams are lubricated. After reaching a certain level, this oil over-flows through two holes at each end of the blower housing, providing lubrication for the blower drive gears at the rear end and the governor drive at the front. A dam in the blower housing cover maintains an oil level which submerges the teeth of the lower blower rotor timing gear. A slinger on the opposite end of lower rotor throws oil into the governor weight assembly. Surplus oil passes from the blower to the oil pan through drilled holes in cylinder block.

17.11 Cooling System.

Cooling of the engine is accomplished by circulation of water through the cylinder block and cylinder head by a centrifugal pump mounted on the front end of the blower and driven by the lower blower rotor shaft through a coupling.

The water pump works in conjunction with either a radiator or heat exchanger, through which the water passes in the process of cooling. In both the radiator and heat exchanger methods, the water pump draws the cooling liquid through the oil cooler and discharges it into the lower part of the cylinder block as shown in Fig. 17–13. Openings in the water jacket around the cylinder bores connect with corresponding openings in the cylinder head, where the liquid circulates around the valves and fuel injectors.

The water temperature in the engine cooling system is automatically controlled by a by-pass-type thermostat mounted in the water manifold, as shown in Fig. 17–13. The thermostat starts opening at approximately 158 F and is fully open at 185 F. After the thermostat has fully opened, water circulation takes place through the radiator or heat exchanger and engine only, no circulation taking place through the by-pass tube.

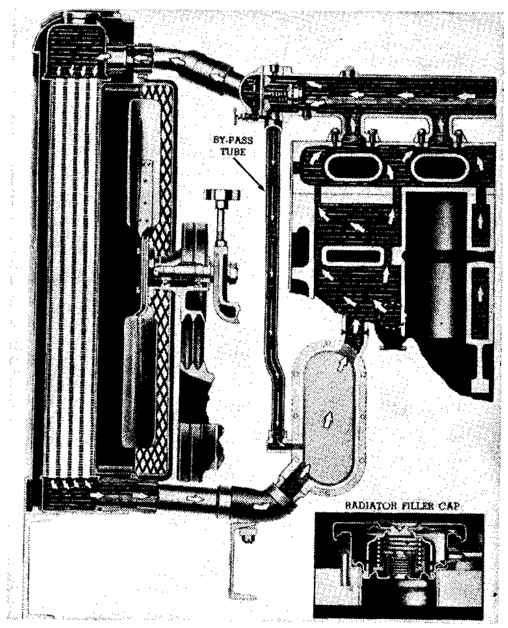

Fig. 17-13. Typical Cooling Arrangement with Radiator and Fan.

17.12 Balance.

An unusual feature of these engines is the balancing method selected for the multicylinder models. The firing order and crank sequence are so chosen that all forces and the secondary and higher unbalanced moments are inherently balanced. It was not possible, however, to select a firing order which gives complete balance for all forces and the primary and secondary couples. Therefore there remains an unbalanced primary couple. This is balanced by means of a set of four eccentric weights, one at each end of the camshaft and one at each end of the balance shaft (see Fig. 17–9). The weights at opposite ends of each shaft are set 180 degrees apart so that when the weight at one end of the engine is down, the weight at the opposite end is up. The two shafts rotate in opposite directions at crankshaft speed and their weights are so phased that both at the same end of the engine are down or up together. This causes an unbalanced couple in the vertical plane which tends to rock the engine, and opposes the primary unbalanced couple in the crankshaft. By weights of the proper size and correct phasing, all forces and couples are balanced within the engine and there is no resultant unbalance. The result is a very smooth running engine.

17.13 Starting.

A starting motor and a 12-volt battery-charging generator with voltage regulator are provided as standard equipment.

17.14 Performance.

Series 71 engines are delivered, depending on the requirements of the application, with four different sizes of injectors. Injector "S" has an approximate capacity of 50 cubic millimeters per injection, injector "A" 60, injector "M" 80, and injector "H" approximately 90 cubic millimeters per injection. The timing of the injectors and of the exhaust valves vary slightly with the different sizes of injectors. Figure 17–14 shows the timing diagram used with the "M" injectors. Figure 17–15 shows the performance characteristics of a six-cylinder engine with "M" injectors. The horsepower

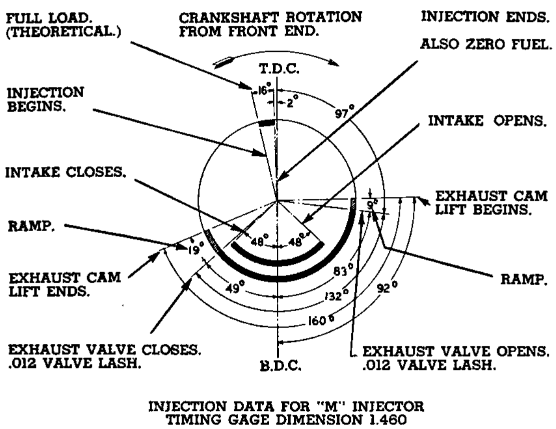

Fig. 17-14. Timing Diagram with "M" Injectors.

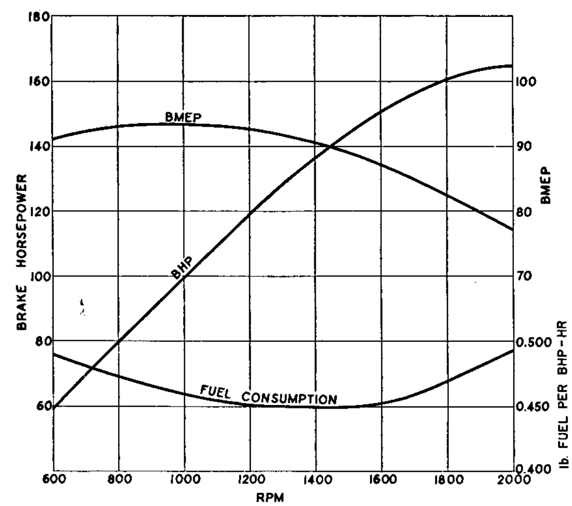

Fig. 17-15. Performance Characteristics of General Motors 6-71 Engine with "M" Injectors.

output of the engine with the smaller-sized injectors is correspondingly smaller and with the "H" injector higher. For continuous load, usually required in industrial service, an output of 100 hp is recommended at 1200 rpm, and 60 cubic millimeter injectors are used.

OPPOSED-PISTON ENGINE

17.15 Fairbanks-Morse opposed-piston engines have been used mostly in submarines and locomotives.

The earlier version of the Model 38 opposed-piston engine was made with 8-inch bore which was later increased to $8\frac{1}{8}$-inch bore when other minor changes were made. The general specifications of a ten-cylinder model of the latter 38D$8\frac{1}{8}$ series are listed in Table 17–II. For locomotive drive this engine is rated at 2000 hp at 850 rpm. Fig. 17–16 and 17–17 show the tranverse and the longitudinal cross sections of the engine, while Fig. 17–18 a view of a ten-cylinder model, to which the following description generally refers.

Table 17-II. Fairbanks-Morse 38D8$\frac{1}{8}$ 10-cylinder Opposed-Piston Engine.

Bore and stroke (in.)	$8\frac{1}{8}$ by (10 + 10)
Number of cylinders	10
Rated rpm	720
Cylinder displacement volume (cu in.)	1037
Total piston displacement at rated rpm (cu ft per min)	4371
Output at rated rpm (bhp)	1600
Bmep at rated load and rpm	84.9
Compression ratio	12.6 – 14.9
Firing order	1-8-7-3-5-9-4-2-10-6

As is common with opposed-piston engines, the combustion space is formed in the center of the cylinder between the two pistons. Two injection nozzles are located at the center of the cylinder. There are two crankshafts, one at each end of the cylinders, which are connected by a vertical shaft, with spiral bevel gears engaging the crankshafts. The exhaust ports are located near the bottoms of the cylinders, the intake ports are near the tops. The ports are controlled by the lower and upper pistons as they reach the outer positions of their travel.

The crankshafts turn in opposite directions and the exhaust crankshaft has a 12-degree angle advance over the upper shaft, permitting unsymmetrical port timing. The scavenging blower is turned by the upper shaft. For marine service the engine is built directly reversible. Naturally, port timing is not optimum during reverse operation. In the absence of cylinder heads the heat loss of the cylinder head to the jacket water is eliminated. The long cylinders with intake at one end and exhaust at the other end facilitate efficient uniflow scavenging.

It is frequently held against the double-crankshaft opposed-piston engines that the crankshafts must be removed for disassembly. By an ingenious arrangement, the lower pistons of the Fairbanks-Morse engine can be removed through the lower cylinder block cover plates, and after the lower piston has been removed the upper piston can be lowered and removed in a similar manner.

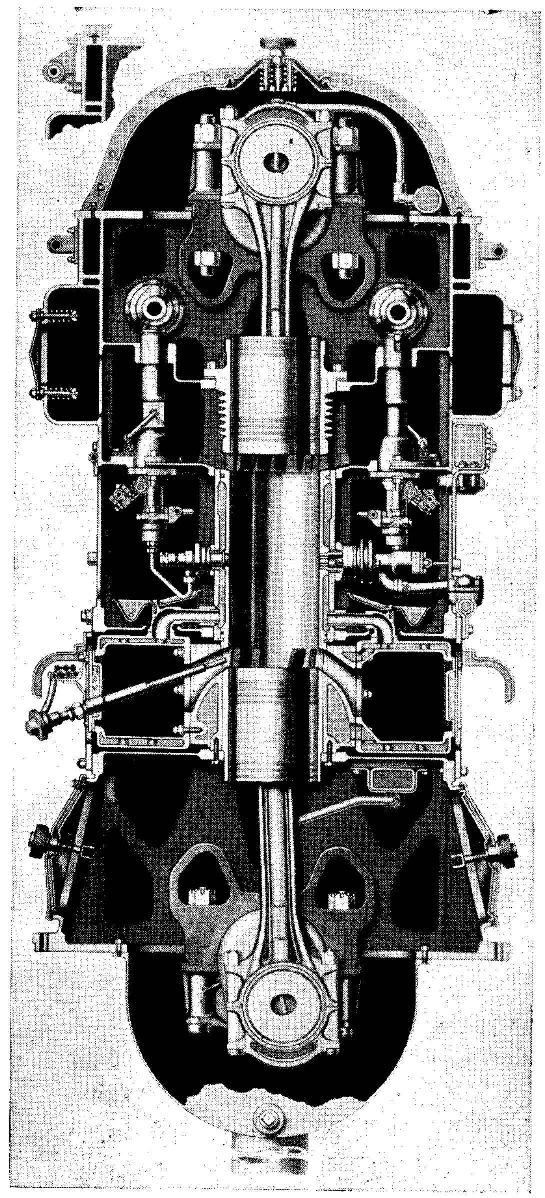

Fig. 17-16. Transverse Cross Section, Opposed-Piston Engine.

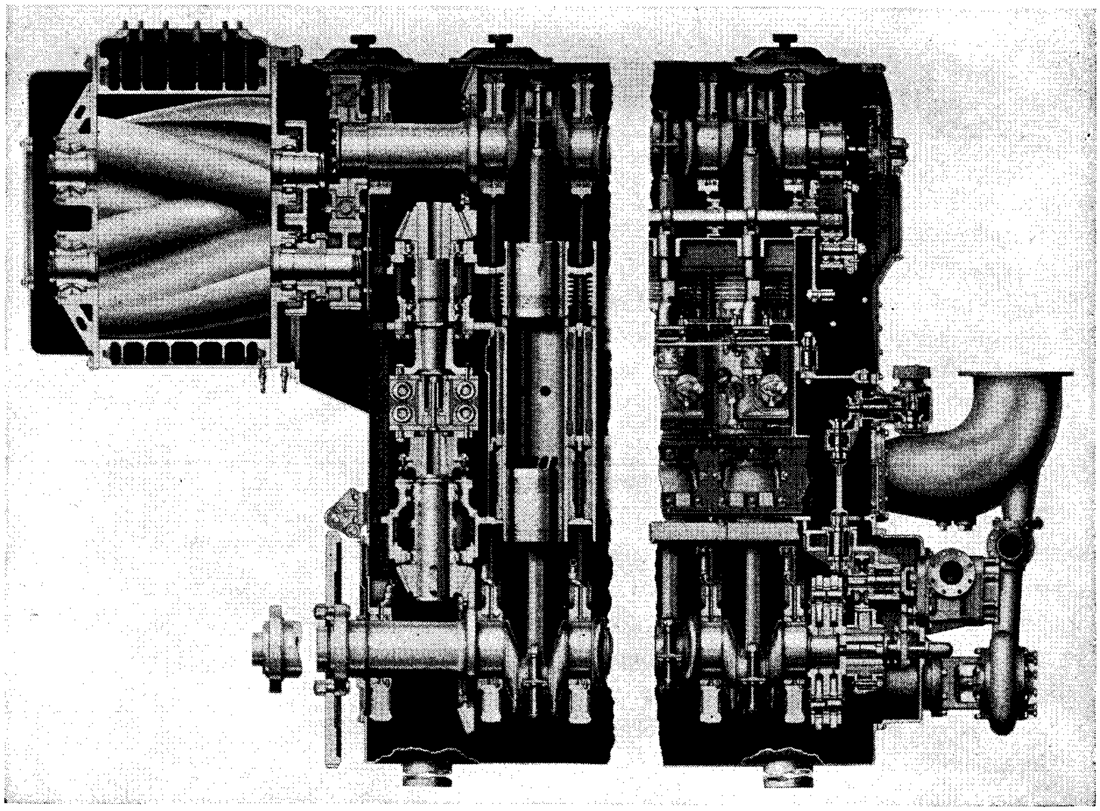

Fig. 17-17. Longitudinal Cross Section. Opposed-Piston Engine.

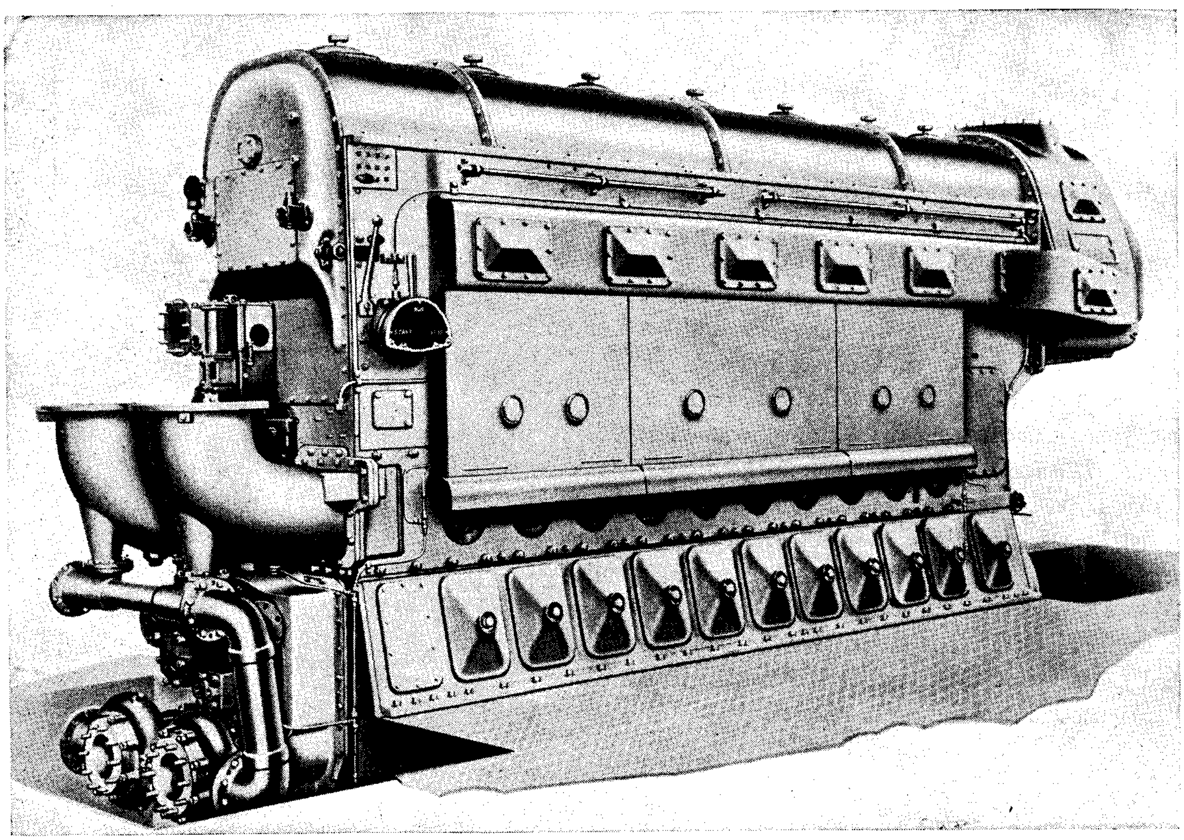

Fig. 17-18. View of a Ten-Cylinder Engine.

17.16 Cylinder Block.

The cylinder block is a welded steel structure, as shown in Fig. 17–19. It is made of steel plates, formed to shape and assembled with steel forgings in special welding fixtures. The main bearings

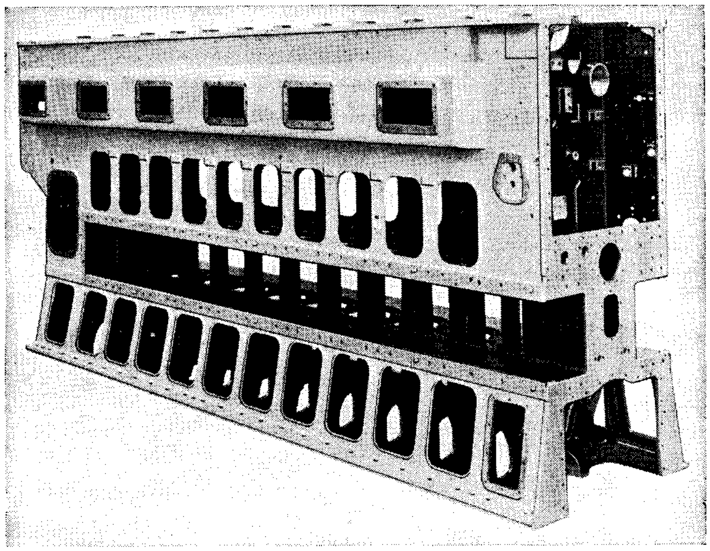

Fig. 17-19. Welded Cylinder Block of Opposed-Piston Engine.

are formed in the upper and lower portions of the cylinder block, and their caps are secured in place by bolts and nuts which fit into special recesses in the forged bearing member. Saddles carrying the bearings are welded into each end of vertical stress members and the tensile stress produced by the combustion is taken in a direct line between the bearings. The cylinder liners with integral water jackets are located through the center of the block. The two camshafts are mounted near the top of the block, one on each side of the cylinders. The scavenging air manifold is formed directly in the upper portion of the block; see Fig. 17–16. The exhaust manifold (Fig. 17–20) is separate from the block just above the lower crankshaft. The exhaust manifold is water cooled. An oil pan is located under the bottom of the block and a removable cover is located over the upper crankshaft. Spring-loaded safety covers are provided to release the pressure and thereby prevent damage in case of a crankcase explosion.

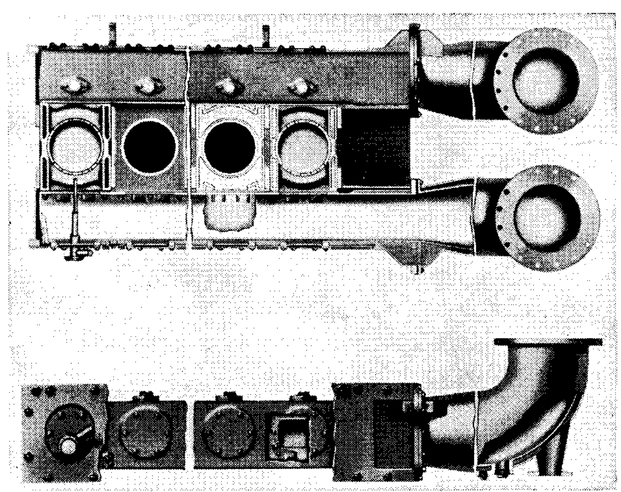

Fig. 17-20. Exhaust Manifold Arrangement.

17.17 Connecting Rods.

Alloy steel drop forged connecting rods of I-beam section are used. They are rifle-drilled to provide passage of the lubricating and piston-cooling oil from the crankpin bearings to the wristpin bearings. The caps are secured by two heat-treated alloy steel bolts. Crankpin bearings are of removable type having steel backs with white metal linings.

The designs of the upper and lower connecting rod assemblies are similar, the only difference being that the lower connecting rod is approximately four inches longer than the upper one.

17.18 Pistons.

The two pistons are practically identical. They are two-piece alloy cast iron pistons with oil-cooled heads and a cylindrical outer surface unbroken by a wristpin hole. A separate insert in which the wristpin is located is bolted to the inner side of the piston head by four studs, as is shown in Fig. 17–21. This construction permits symmetrical heat expansion and the piston skirt can remain truly

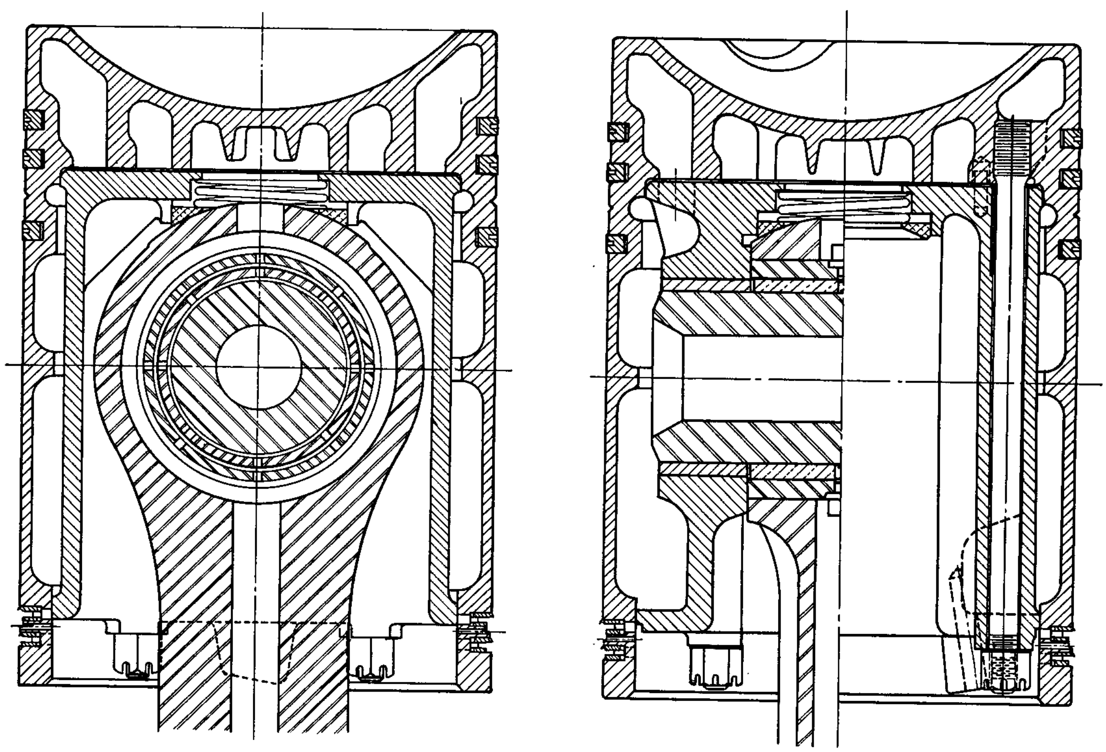

Fig. 17-21. Piston, Opposed-Piston Engine.

circular. Piston-cooling oil is supplied through a spring-loaded spherical seal at the small end of the rod. The oil fills a cavity in back of the piston crown. The hot oil is displaced by the incoming fresh oil and spills without additional piping into the crankcase.

Pistons are fitted with compression rings at the closed end and oil-control rings at the open end. Both pistons may be removed from the lower end of the cylinder through openings at the side of the crankcase.

17.19 Cylinder Liners.

Cylinder liners are made of cast iron. Ports for the exhaust and scavenging air are cast into the lower and upper ends, respectively, as shown in Fig. 17–22. Cooling fins are provided at the upper end above the scavenging air ports, which transfer the heat to the scavenging air. A steel water-jacket is pressed on to the central portion between the ports. Holes are provided at the center of the liner for the fuel-injection nozzles. A water-cooled exhaust outlet box is provided in the block into which the lower end of the liner fits. The cooling water enters the liner from the exhaust manifold which also acts as a supply header, passes upward through the liner jacket and leaves just below the scavenging air ports to enter the discharge header which is located along the side of the cylinder block. The cylinder liners are inserted into the frame from the top. The upper crankshaft must be removed to permit removal of a liner.

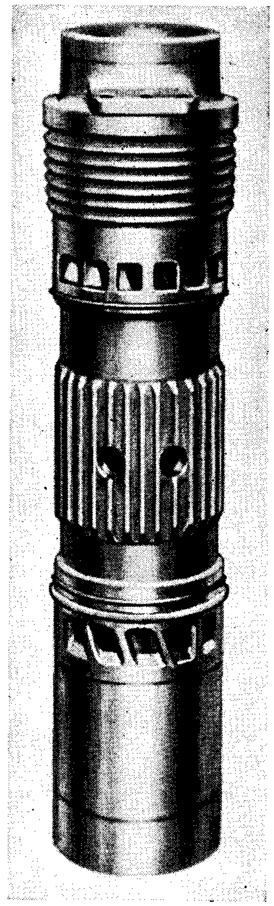

Fig. 17-22. Cylinder Liner, Opposed-Piston Engine.

17.20 Fuel System.

Two fuel injection pumps and two nozzles are provided for each cylinder. The two pumps are located on opposite sides of the engine and each is connected to an individual nozzle by high-pressure fuel injection tubing, as shown in Fig. 17–23.

The pumps are of the lapped plunger and barrel type with a helical land at the top of the plunger which passes a port in the wall of the barrel. By rotating the plunger the port is covered for a longer or shorter portion of the stroke and thus more or less fuel is injected. Rotation of the plunger is accomplished by a gear segment and rack actuated by the governor control linkage.

The fuel injection nozzles are of the closed type with a spring-loaded needle valve, as shown in Fig. 17–23. The hydraulic pressure of the injection fuel, from the fuel injection pump, opens the nozzle and the fuel is injected into the cylinder. The two nozzles are located diametrically opposite each other and the spray holes are so arranged that the spray assumes a flat pattern to fill the flat space between the cylinders at their closest point.

The fuel supply pump, gear-driven from the lower crankshaft, draws fuel from the service tank and delivers it through the strainer-filter to the header with a pressure of about 15 psig. A regulating valve built into the header allows the excess fuel to return to the tank.

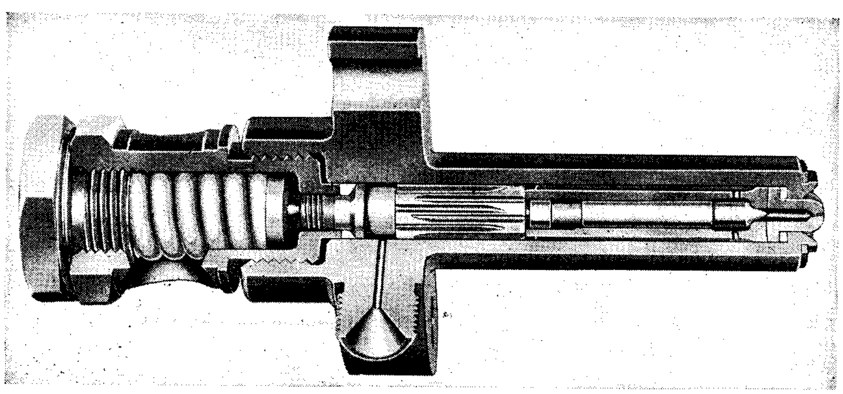

Fig. 17-23. Injection Nozzle, Opposed-Piston Engine.

17.21 Crankshafts.

Two crankshafts are used which are connected with a vertical drive. The upper and lower crankshafts are shown in Fig. 17–24 and the vertical drive with the spiral bevel gears in Fig. 17–25. A thrust bearing is provided next to the vertical drive gears and plain main bearings at each transverse member of the cylinder block.

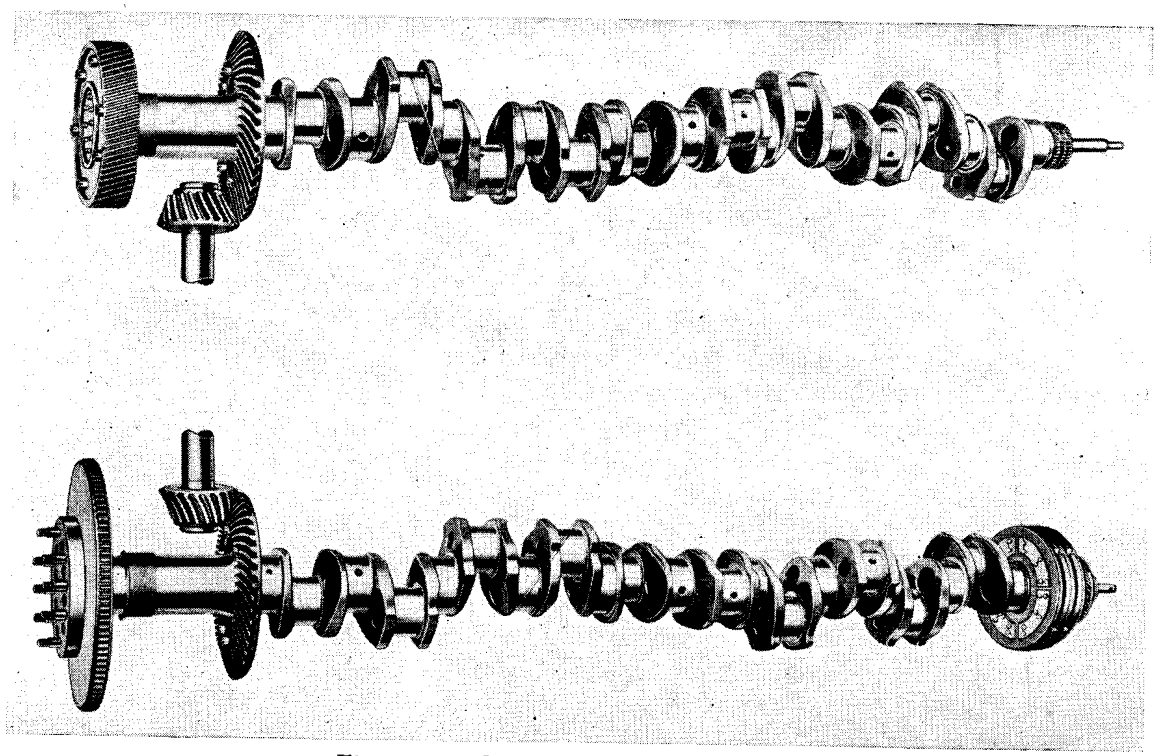

Fig. 17-24. Upper and Lower Crankshafts.

The crankshafts are cast of chrome-nickel-molybdenum alloy. They are so phased with each other that the lower (exhaust) crankshaft is 12 degrees ahead of the upper crankshaft. The blower drive is secured to the upper crankshaft.

The torsional damper is connected to the lower crankshaft at the control end. Power is taken from the engine through a flexible coupling mounted on the lower crankshaft at the blower end.

17.22 Camshafts.

Two camshafts are provided on opposite sides of the engine. These are located near the top of the cylinder block. An individual cam is provided for each fuel pump and the cams are forged integral with the camshaft and are hardened and ground. The fuel pumps are located under the camshaft and are actuated by a downward motion of the roller cam followers. The camshafts are driven by a timing chain at the front end of the engine. Means are provided to adjust slack and vary the timing of both shafts together.

For marine application the engine was made directly reversible by movement of idler sprockets in the camshaft drive which effects a change in the camshaft phase.

17.23 Scavenging System.

Scavenging air is supplied to the cylinders under a pressure of about 4 psig at full speed by a Roots-type blower with two three-lobe helical impellers. The scavenging air is conducted to the air receiver compartments extending the width of the cylinder block, and completely surrounding the cylinder liners at the intake ports.

The blower is driven from the upper crankshaft through a torsionally flexible coupling and gears which step up the speed in the ratio of 1450 :720. The displacement of the impellers is 4.8 cubic feet per revolution and the capacity of the blower is given as 6200 cubic feet per minute. This is 1.435 times cylinder displacement volume.

17.24 Lubricating System.

The lubricating oil pump is mounted on the control end of the engine and is driven by the lower crankshaft. The pump draws oil from the sump below the engine and forces it successively through the strainer and the coolers.

Entering the lower lubricating oil header from the inlet near the control end, the oil flows through the lower header toward the blower end. There a vertical pipe carries the oil to the upper header.

Through supply pipes from both lower and upper headers, oil is forced to each main bearing, then through tubes swedged into the crankshaft, to each crankpin bearing. From there oil passes to the wristpin bearings and to the piston oil cooling jackets.

The oil not used drips down into the oil pan and from the pan it drains to the sump tank.

The lubricating oil pump has 280 gallons per minute capacity at 1280 rpm pump speed. The relief valve is set at 60 psig.

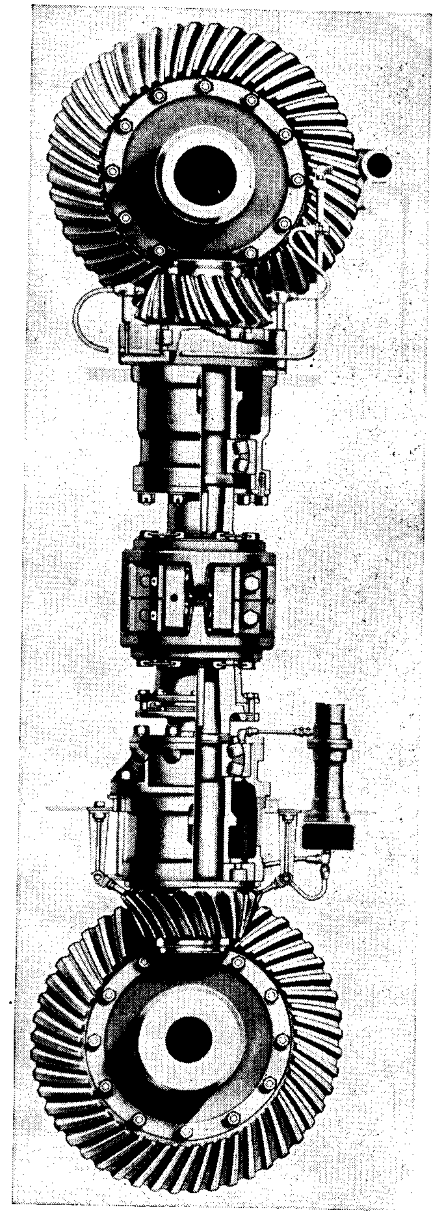

Fig. 17-25. Vertical Drive.

17.25 Cooling System.

The engine is cooled by circulating fresh water through its water passages. Figure 17–26 is a schematic diagram of the cooling system. Entering the engine through an inlet in each exhaust nozzle the water moves through passages which surround the exhaust nozzles into the exhaust manifold. The exhaust ducts and the lower part of the liner are also water-cooled.

Having performed its engine cooling functions, the water is piped to the fresh water cooler. After leaving the cooler the fresh water is used as the coolant for the lubricating oil cooler, after which it is piped back to the pump suction inlet to repeat its passage through the engine. The fresh water centrifugal pump is driven from the control end of the lower crankshaft.

A second centrifugal pump, similarly located and driven, circulates the raw water through the fresh water cooler.

The capacity of the fresh water pump is 525 gallons per minute and that of the raw water pump 600 gallons per minute both at 1745 rpm pump speed.

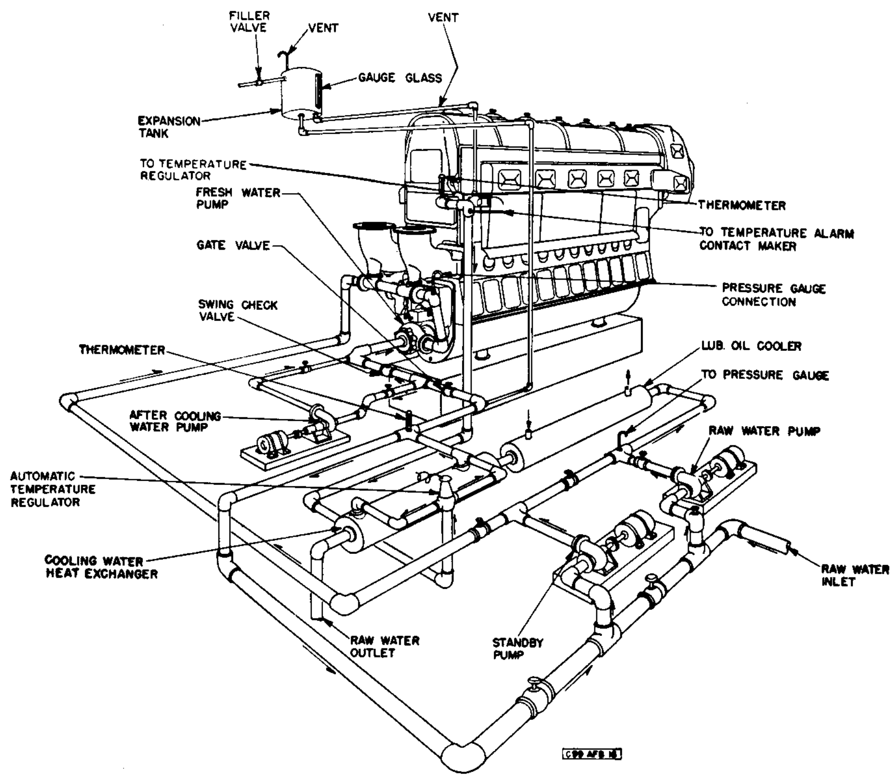

Fig. 17-26. Cooling System.

17.26 Starting System.

Compressed air of about 250 psig is used for starting.

The engine starting mechanism includes the air start control valve, air start distributor, the header, the pilot air tubing, and the air start check valves at the individual cylinders.

When the control valve is opened, starting air fills the header to each check valve. These are of the balanced pressure type pneumatically opened by air from the distributor valve. The check valves are inserted in adapters screwed into the cylinder walls. The distributor valves are in contact with the timing cam only while the main control valve is open. At other times, retriever springs hold them out of contact.

17.27 Performance.

Figure 17–27 shows the specific fuel consumption of the engine and the exhaust temperature at 720 rpm. At the rated load of 160 horsepower per cylinder the fuel consumption is 0.37 pound per bhp-hr. Since the specific fuel consumption curve is still going down at rated horsepower, the rating seems conservative and the engine could stand appreciable overload. At the rated load the bmep is 84.9 psi.

The exhaust temperatures are quite low for the brake mean effective pressure.

Figure 17–28 shows the timing diagrams of the Fairbanks-Morse opposed-piston engine. There is a lag of 12 degrees between the exhaust and inlet cranks. The exhaust lead is 16 degrees and the supercharge period 7 degrees crank angle. Fuel injection is shown to begin at 12 degrees before and end 3 degrees after top center.

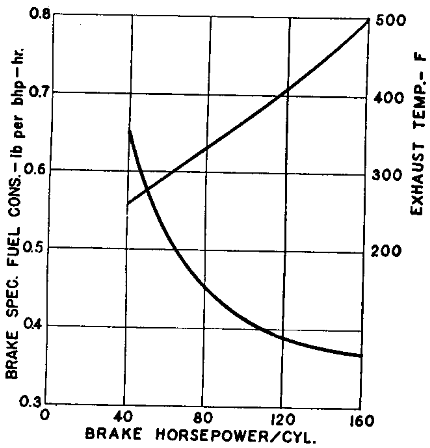

Fig. 17-27. Performance of Fairbanks-Morse Opposed-Piston Engine at 720 RPM.

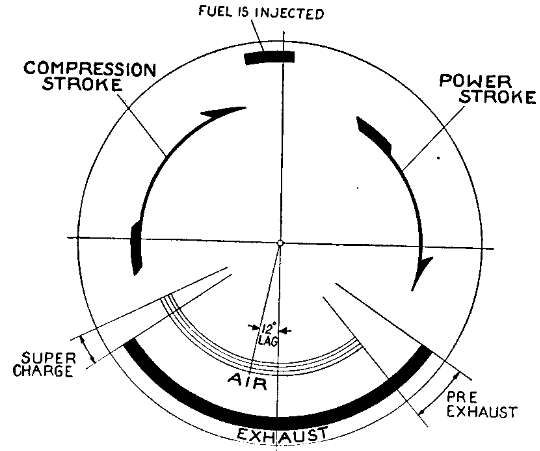

Fig. 17-28. Timing Diagram, Opposed-Piston Engine.

Figure 17–29 shows the indicator diagrams of the opposed-piston engine. Due to the 12 degrees phase advance the exhaust or lower crank diagram is considerably fatter than the upper crank diagram, and its area is more than double of the upper. From this it is sometimes inferred that the load on the inlet piston is relatively small while the exhaust piston has to stand the major portion of the burden.

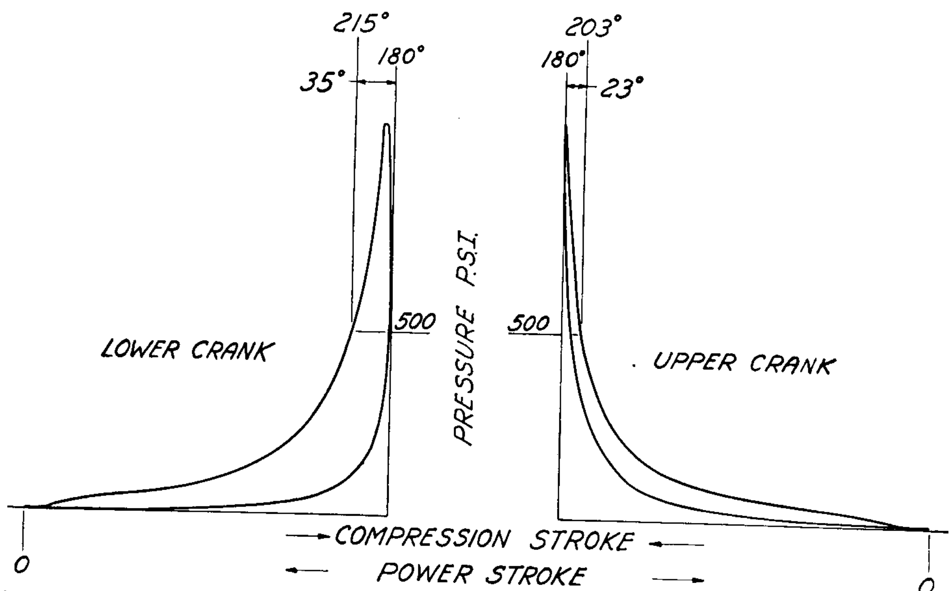

Fig. 17-29. Indicator Diagrams, Opposed-Piston Engine. Ten cylinders, 1600 hp, 720 RPM.

The truth of the matter is that the opposed-piston engine cylinder has only one combustion space and only one pressure-volume diagram instead of two. Confusion arises from the habit of failing to discriminate between **pressure-volume** and **pressure-stroke** diagrams. For a single-piston engine, this is of no consequence because the instantaneous cylinder volume is proportional to the piston travel. For the opposed-piston engine this is no longer true. The pressure-volume diagrams or the pressure-crank angle diagrams are identical for the lower and upper crank but the pressure stroke diagrams are vastly different.

Even if the power transmitted to the exhaust crank is double that transmitted to the inlet crank, it is incorrect to conclude that the exhaust piston is carrying an undue burden while the inlet piston is loafing. Actually, except for the scavenging gas flow, both pistons are exposed to exactly the same pressures and temperatures for the same lengths of time.

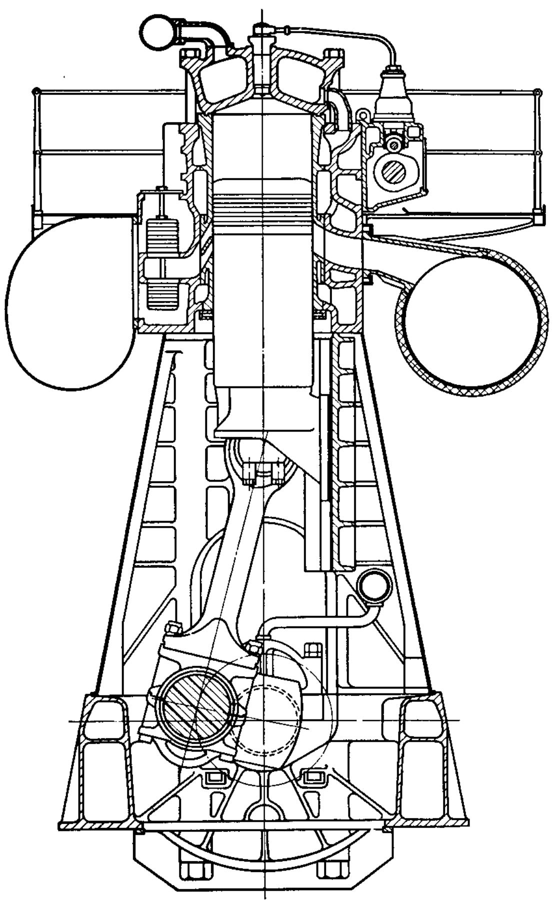

Fig. 17-30. Transverse Longitudinal Section of Nordberg Engine.

LARGE CROSSHEAD ENGINE

17.28 One of the largest engines being built in America is the Nordberg Type TS-29 engine, the general specifications of which are listed in Table 17–III.

Table 17-III. Nordberg Type TS-29 9-Cylinder Engine.

Bore and stroke (inches)	29 × 40
Cylinder number	9
Rated rpm	164
Piston speed (fpm)	1092
Cylinder displacement volume (cu in.)	26,421
Total displacement at rated rpm (cfm)	22,500
Output at rated rpm (bhp)	6400
Bmep at rated load and rpm	64

The transverse longitudinal section of the engine is shown in Fig. 17–30 and a view from the control side in Fig. 17–31.

The engine is scavenged by electric motor driven centrifugal blowers, not shown on the photograph. Curtis-type loop scavenging is used with automatic intake valves placed before the inlet ports, that effect an unsymmetrical scavenge through the delayed airflow through the high inlet ports.

The engine speed is so low that the engine is suitable for direct drive of propellers of large ships. The crosshead construction is favorable to low cylinder wear and long engine life.

17.29 Crankshaft.

The crankshaft is made in two sections, one with five cranks and the other with four cranks.

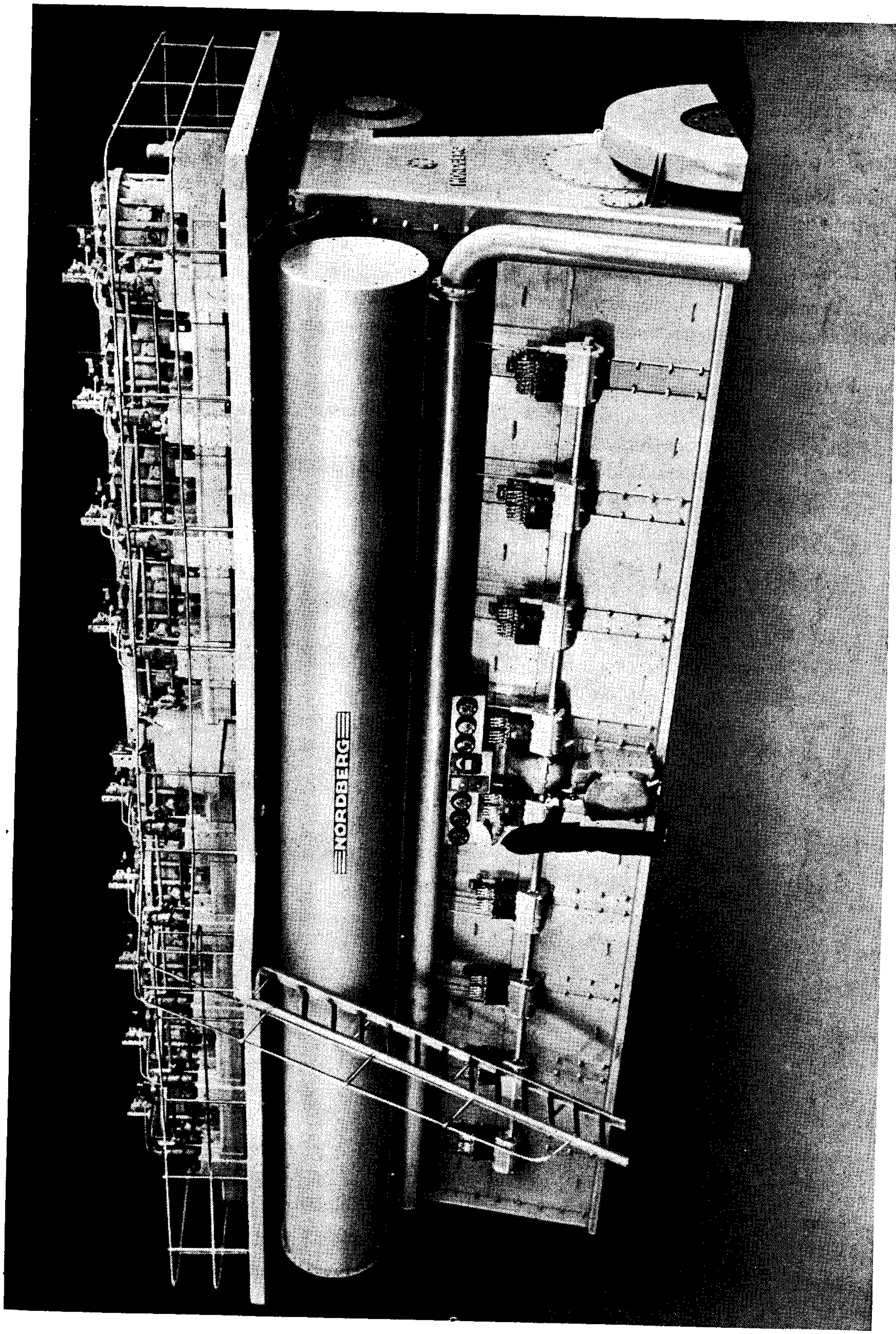

Fig. 17-31. View of 6400 Hp Nordberg Engine.

Each shaft section is a one-piece forging. The crankpins and journals are 20 inches in diameter. The complete crankshaft, finished all over, weighs approximately 80,000 pounds.

17.30 Bedplate.

The bedplate is cast in two sections which are bolted together. It is of box design and is divided into separate compartments by heavy ribbed bridges which carry the main bearings.

17.31 Bearings.

Main and crankpin bearings are removable steel-backed shells lined with centrifugally cast antifriction metal. Crosshead pin bearings are one-piece copper-base alloy bushings.

17.32 Frame.

The frame structure consists of a series of cast-iron A-frames which rest upon the bedplate. The cast-iron slides and guides for the crosshead are bolted between the columns. The A-frames are cored for tie rods which are anchored below and on each side of each main bearing housing in the bedplate and extend through the legs of the A-frames to the top of the cylinder blocks. They carry all tensile

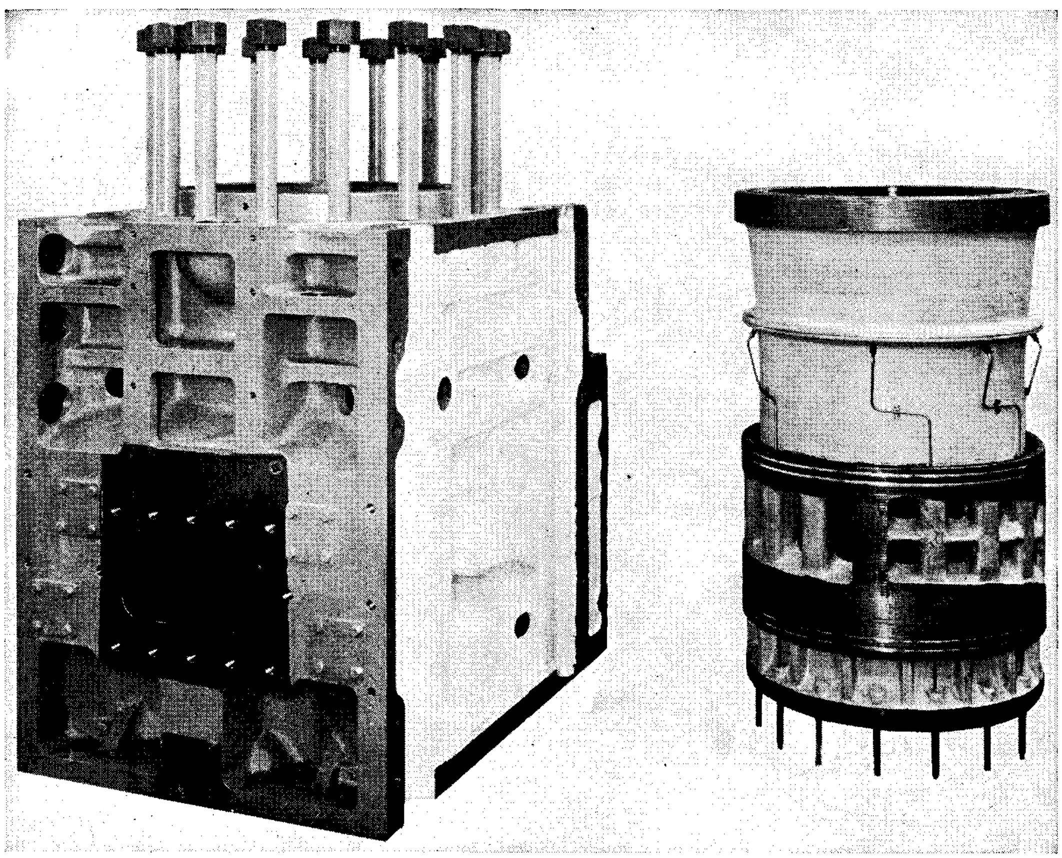

Fig. 17-32. Nordberg Cylinder and Liner.

loading resulting from gas pressure in the cylinders. To insure equal tension in the tie rods, they are elongated by a hydraulic pump which applies a predetermined force to the rods before the nuts are tightened.

The cylinder block or entablature for each cylinder is a separate casting which is fitted and bolted to the blocks adjacent to it.

17.33 Cylinder Liner.

The removable liners are of cast iron. They are held on top by a rim which fits the block and is pressed down to it by the cylinder head. The liner is completely surrounded by cooling water which passes even through the bridges between the ports. Gland rings are provided to seal the jacket water against leakage. The liner is free to expand downward.

Intake and exhaust port are cast in the liner, the general appearance of which is shown in Fig. 17–32, which also shows the connections for the lubricating oil which is introduced at seven points on the periphery near the upper end of the liner by a seven-feed lubricator.

A wiper ring assembly attached to the bottom of each cylinder consists of a casting containing two pairs of internal expanding wiper rings which fit the skirt of the piston and wipe off excess oil, returning it to the crankcase while the sludge drains into a recess in the casting, thence into a header leading to a sludge tank. The wiper rings are so arranged that the upper pair wipes the piston on its down stroke and the lower pair wipes on the up stroke.

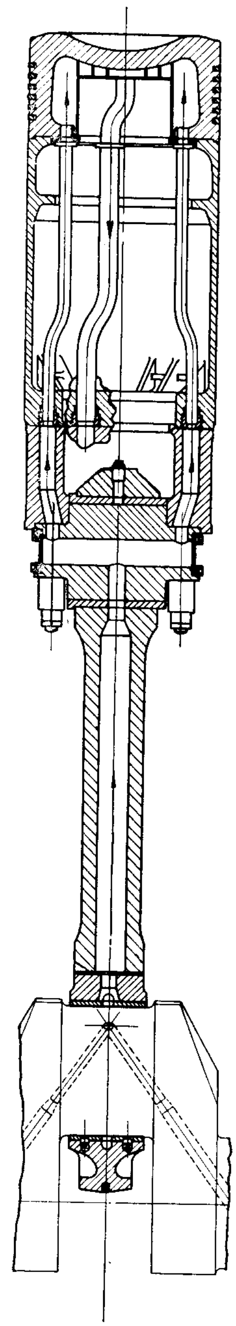

Fig. 17-33. Nordberg Piston and Connecting Rod.

17.34 Piston.

Piston is of two parts, head and skirt (see Fig. 17–33). The piston crown is slightly dished. Its underside is cooled by circulation of oil at high velocity through labyrinth passageways. The cooling oil reaches the head from the connecting rod and is discharged through a drain tube into a funnel mounted to the frame.

The piston skirt of crosshead engines rests upon a flange of the crosshead proper to which it is centered and fastened by cap bolts.

The connecting rod is a tubular steel forging finished all over, with a marine-type crankpin bearing held by four bolts. Lubricating oil is supplied to the crosshead pin through a hollow connecting rod from the crankpin bearing.

17.35 Cylinder Head.

Cast-iron cylinder heads are used. They are shown in Fig. 17–34. The head is of a simple symmetrical design, with a large continuous water space which is broken only by the starting and relief valves. The cylinder head is held by a series of studs to the block which in turn is held to the bedplate by the tie rods. The underside of the cylinder head forms a flat cone-shaped combustion space (see Fig. 17–30).

17.36 Fuel Injection.

Individual Bosch injection pumps are used and are operated from a highly placed camshaft. This permits the use of relatively short lines to the centrally located injection nozzle.

The nozzle is of the spring-loaded differential outwardly opening, multihole-type of Bosch manufacture. The spray pattern is in the shape of a flat cone which roughly approximates the shape of the combustion chamber.

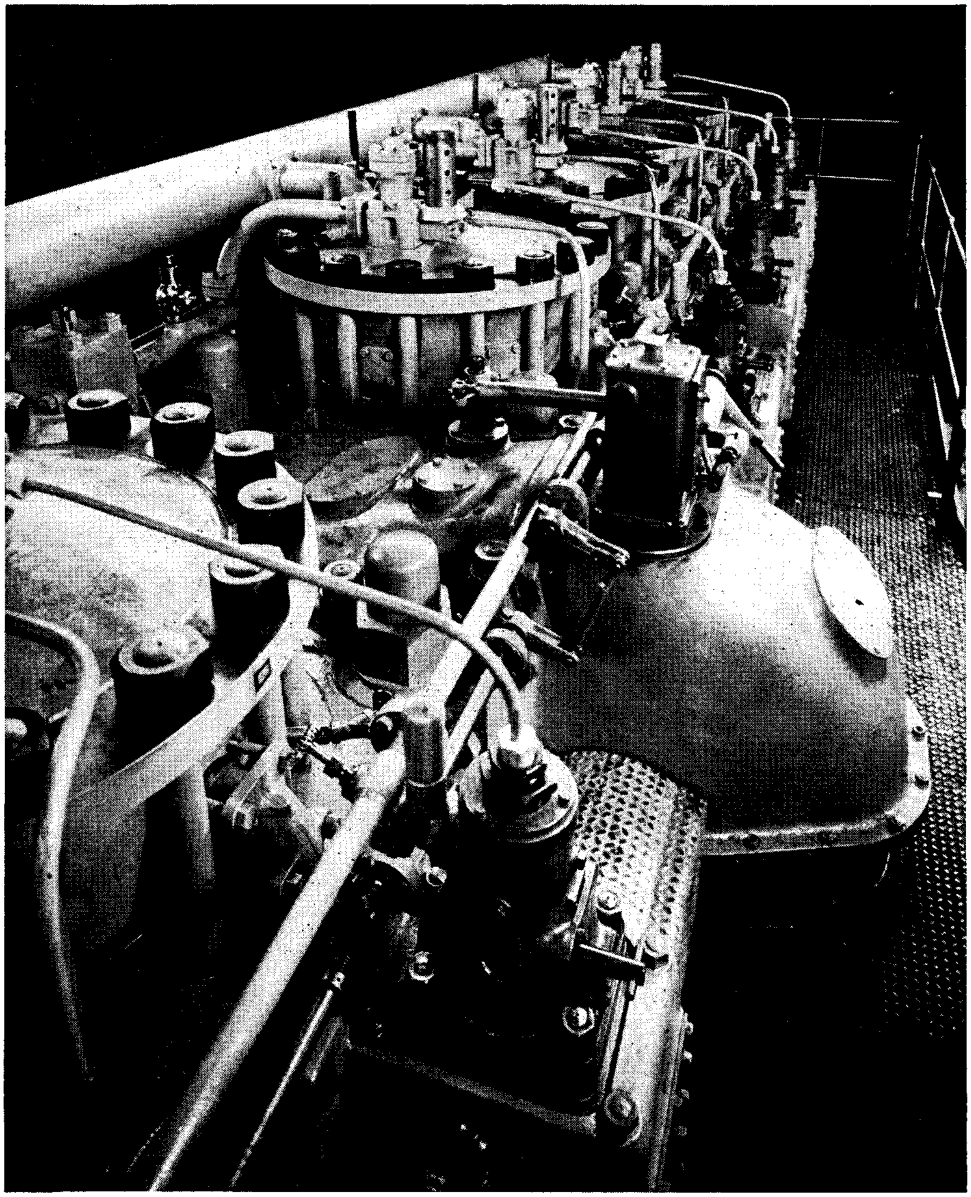

Fig. 17-34. View of Upper Gallery.

17.37 Lubrication System.

The main lubrication system is of the pressure feed recirculating type. A pressure pump delivers oil under pressure to a large header located in the crankcase with leads to all bearings, camshaft drive, and reversing gears. Both webs of each crank are drilled to conduct lubricating oil to the crankpins, crosshead pins, and to the oil-cooled piston heads through a large drilled passageway in each connecting rod. Oil from bearings, gears, and piston heads returns to the sump, from which it is delivered by a motor-driven pump through a shell-and-tube-type oil cooler and a filter to the main header in the crankcase.

There is also a force feed system for cylinder wall lubrication which consists of lubricators driven from a layshaft and having a separate pump unit for each feed. The lubricators are timed to deliver lubricating oil to each piston at a predetermined point in its stroke.

17.38 Scavenging System.

The scavenging header is located on one side of the engine, the top of which serves as a service platform. Automatic plate valves are located in the scavenging header and adjacent to the scavenging air ports of each cylinder. The automatic valve controls both the upper and lower row of ports, which are divided only for better guidance of the inlet air. The scavenging is of the Curtis loop type and is described in greater detail in 9.8.

Scavenging air is furnished by two motor-driven centrifugal blowers which discharge through separate connections to the scavenging header. At low engine speeds only one blower need be used.

17.39 Engine Control.

The engine is controlled by a single lever located on the port side of the engine. This lever controls the starting (by compressed air), fuel regulation, reversing, and stopping of the engine through a mechano-pneumatic system with interlocks which prevent damage to the engine by improper manipulation.

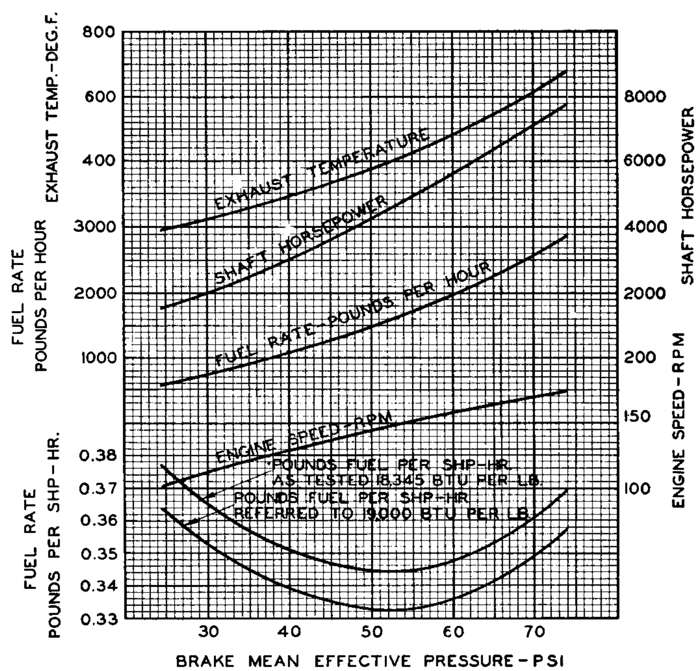

Fig. 17-35. Performance of Nordberg Engine.

17.40 Performance.

Test results of the 6400-hp Nordberg marine engine are shown in Fig. 17-35. The engine was run for two hours at overload delivering 7300 screw horsepower at 172 rpm, 48 hours delivering 6450 horsepower at 165 rpm and 360 hours continuous endurance run developing 6000 shaft horsepower at 160 rpm. All tests were conducted with heavy fuel oil of 1076 SSU viscosity at 100 F, and 9.57 per cent Conradson carbon residue.

The fuel consumption and other figures as shown do not include the power required to drive the scavenging air blowers, which amounted to 430 bhp at normal load.

CHAPTER 18

TABULATION OF PORTING DATA

18.1 The following tabulation presents the porting data of 31 two-stroke cycle diesel engines. Most of the engines selected are of recent design and are in current production, although some experimental engines have been included in the tabulation. All of the listed engines have actually been built and with a few exceptions may be regarded as representative samples of modern porting.

Most of the information has been supplied by the manufacturers, part has been obtained from blueprints or printed drawings, and the balance from the files of the author, who helped to design several of the engines listed.

To anyone actively interested in porting design the tabulation offers a rich source of information. He will be interested in comparing engines of similar classification and finding justification for the existing differences in their porting data. The designer of a new engine may find one listed which comes close enough to his own to justify the use of the same scavenge factor (item 45), and if the listed engine is a successful one, the use of similar delivery ratio, as well as similar timing and port velocities.

Comparison of rated bmep (item 7) and calculated bmep (item 48) shows excellent agreement for the most part. Where the agreement is less good, it can probably be traced to some inaccuracy of the listed data or of the assumed specific fuel consumption, or to the fact that the respective engine is underrated or overrated. The indicated computation of item 48 permits prediction of the bmep of a designed engine on a moderately conservative basis.

EXPLANATION OF LISTED ITEMS

Bold face numerals represent the value of that item listed in Table 18–I. For example, for engine No. 1 (**9**) means 3371.

5 $V_{disp} = D^2\frac{\pi}{4}s$

6 At rated revolutions per minute

7 At rated revolutions per minute

8 Abbreviations:

Cro = cross scavenge; Loo = loop scavenge; Uni = uniflow scavenge; Cra = crankcase scavenge; Sym = symmetrical scavenge; Aui = automatic supercharge valve in inlet; Aue = automatic supercharge valve in exhaust; Mei = mechanically operated supercharge valve in inlet; Mee = mechanically operated supercharge valve in exhaust; OP = opposed piston; EPV = exhaust poppet valve; SV = sleeve valve; Rec = reciprocating blower; Roo = Roots blower; Cen = centrifugal blower.

9 If the engine is built in models with different numbers of cylinders, the data apply to the model with the number of cylinders underlined in item (3)

10 $L = \dfrac{9 \times 1728}{5 \times 4}$

11 9/6

12 Mean pressure at scavenge air receiver

13 Mean temperature at scavenge air receiver

15 Exhaust piston (sleeve, valve) ahead of inlet piston (sleeve, valve)

18 $\dfrac{17 - 16}{6 \times 4}$

20 $\dfrac{100 \times 19}{s}$

22 $\dfrac{100 \times 21}{\pi D}$

23 Refers to exhaust piston in opposed-piston engine

24 Refers to exhaust piston in opposed-piston engine

25 $\dfrac{24 - 23}{6 \times 4}$

27 $\dfrac{100 \times 26}{s}$

29 $\dfrac{100 \times 28}{\pi D}$

31 Calculated square to the flow

32 31(17 − 16)

34 Calculated square to the flow

35 34(24 − 23)

36 $\dfrac{5 \times 10}{18}$

37 $\dfrac{36}{12 \times 31}$

38 $\dfrac{5 \times 10}{25}$

39 $\dfrac{38}{12 \times 34}$

40 37/39

41 23 − 16

42 Exhaust time area up to the point of inlet port opening

43 24 − 17

44 Excess of inlet time area over exhaust time area before inlet port closure

45 $39/W_{calc}$; W_{calc} is obtained from Fig. 7-2.

46 $C_{id} = (\text{effective compression stroke}) \dfrac{14.7 + 12}{14.7} \dfrac{520}{460 + 13}$

47 From Fig. 5-8, 5-9, and 5-10

48 Calculated bmep $= 180 \dfrac{0.4}{f} \dfrac{\eta_{sc}}{1 + (\lambda - 1)\eta_{sc}} C_{id}$

where

f = specific fuel consumption
η_{sc} = 47
λ = excess air factor taken as 1.6
C_{id} = 46

SELECTION OF CYLINDER NUMBER

18.2 Table 18–I lists engines from less than $2\frac{1}{2}$ to 29 inch bore and from 30 to 30,000 cubic inch cylinder displacement. Before closing, it may be appropriate to make a few guiding remarks on the selection of the number of cylinders.

The designer knows that for a given service the engine must deliver a given horsepower. A given horsepower can be produced with a smaller number of large cylinders or with a greater number of small cylinders. He wants the best combination that will do the job.

It should be pointed out at the beginning that numerous considerations enter into the selection, such as production and service facilities, critical speeds, and many others. However, the following remarks relate to the limited problem of selecting the number and size of cylinders with regard to the effect on the scavenging and consequently the power output of the engine.

The horsepower output of a cylinder is from equation (4–1)

$$\text{Brake horsepower} = \frac{\dfrac{V_{disp}}{12} \times \text{bmep} \times n}{33{,}000}.$$

From equation (4–6)

$$\text{bmep} = 180 \frac{0.4}{f} \frac{14.5}{r_{th}} \frac{\eta_{sc}}{1 + (\lambda - 1)\eta_{sc}} C_{rel}.$$

Well-designed engines of similar scavenging arrangement have substantially identical scavenging efficiencies and their relative charge is also the same. Therefore, their bmep will be substantially the same, as shown by Fig. 5–8, 5–9, and 5–10. Consequently

$$\text{Brake horsepower} = \text{Const} \times V_{disp} \times n$$

18.3 Doubling the Cylinder Size.

It is obvious that the brake horsepower can be doubled by doubling the number of cylinders. Can it also be doubled by doubling the displacement volume (cylinder size)? From the preceding equation it is evident that this could be accomplished only if the engine speed would be kept unchanged. However, the permissible maximum engine speed is inversely proportional to the cylinder diameter; see section 10–1. By doubling the cylinder size, in maintaining the same bore-stroke ratio, the cylinder diameter must be increased by $\sqrt[3]{2} = 1.26$. Therefore the engine speed must be **reduced** to approximately 79 per cent of the original value. The result is that by doubling the cylinder size the horsepower is increased only by 59 per cent.

The nominal output of the double size engine does not increase, therefore, by 2, but only by 1.6. This holds if both engines are operated at their respective optimum speeds. On a displacement basis the small engine is some 20 per cent more powerful than the large engine.

18.4 Geometrically Similar Engines.

Similar conclusions may be drawn from considerations of mechanical similitude. Engine A is **geometrically** similar to engine B in everything including the porting, but it is larger in the linear ratio of D_a/D_b. The question is, what will be the permissible speed (from the standpoint of scavenging) of engine A and engine B.

The engines have inlet ports and exhaust ports, the lengths of which are

$$h_{ia},\ h_{ea},\ h_{ib},\ h_{eb}$$

respectively. Naturally

$$\frac{h_{ia}}{h_{ib}} = \frac{h_{ea}}{h_{eb}} = \frac{D_a}{D_b}$$

Their total widths are also

$$\frac{w_{ia}}{w_{ib}} = \frac{w_{ea}}{w_{eb}} = \frac{D_a}{D_b}.$$

The **angular** durations of the respective inlet and exhaust periods are, however, **equal**

$$\alpha_{ia} = \alpha_{ib},\ \alpha_{ea} = \alpha_{eb}$$

From equation (7–1) the permissible speed for engine A is

$$n_a = \left(\frac{A_{im}\alpha_i}{\frac{L}{2w_i} V_{disp}}\right)_a = \text{Const}\,\frac{D_a{}^2\alpha_{ia}}{D_a{}^3}$$

$$= \text{Const}\,\frac{\alpha_{ia}}{D_a}$$

Similarly for engine B

$$n_b = \text{Const}\,\frac{\alpha_{ib}}{D_b}$$

The two constants are equal, provided the two scavenge pressures are considered equal.

This results in the requirement that for various size cylinders the product nD must be constant.

What will be the power output of the engines if it is assumed that the bmep of the geometrically similar engines are equal?

$$(\text{Brake horsepower})_a = \text{Const} \times V_{disp\,a} \times n_a$$

and

$$(\text{Brake horsepower})_b = \text{Const} \times V_{disp\,b} \times n_b$$

Considering the foregoing expressions for n_a and n_b, the power ratio of the two engines is

$$\frac{(\text{Brake horsepower})_a}{(\text{Brake horsepower})_b} = \frac{\dfrac{V_{disp\,a}}{D_a}}{\dfrac{V_{disp\,b}}{D_b}} = \frac{D_a{}^2}{D_b{}^2}.$$

The power output does not vary as the displacement (D^3) but as the piston area (D^2). In order to double the displacement

$$\frac{V_{disp\,a}}{V_{disp\,b}} = \frac{D_a^3}{D_b^3} = 2$$

the bore must be increased by

$$\sqrt[3]{2} = 1.26$$

and the power output increases in consequence by

$$\frac{D_a^2}{D_b^2} = 1.26^2 = 1.59.$$

The specific outputs, bhp per cubic inch displacement, vary as

$$\frac{(\text{Bhp per cu in.})_a}{(\text{Bhp per cu in.})_b} = \frac{D_b}{D_a}$$

or the smaller the cylinder the greater the specific power output. By doubling the displacement

$$\frac{D_a^3}{D_b^3} = 2$$

$$\frac{D_a}{D_b} = \sqrt[3]{2} = 1.26$$

and the specific output goes down to

$$\frac{D_b}{D_a} = \frac{1}{1.26} = 0.79$$

which is approximately 21 per cent less than the specific power output of the small engine. This is the same result as the one obtained before.

Basing the calculation on equation (8–1) or (8–2), which express the requirements for exhaust lead, the same result will be obtained, because again the required time-area is proportional to the displacement volume and to the rotational speed.

The result obtained may be expressed in the following way. By increasing the cylinder bore 1 per cent, the engine displacement increases 3 per cent but the power output only 2 per cent. If the cylinder displacement is increased by a ratio R, the power increases by a ratio of $R^{\frac{2}{3}}$ and the specific power output **decreases** at a rate of $1/\sqrt[3]{R}$.

18.5 The Case for Small Cylinders.

Since the volume, the weight, and the cost of the engine all follow more or less the cylinder displacement, it is obvious that for a given power output a greater number of small cylinders is preferable to a smaller number of larger cylinders. This is true not only from the standpoint of porting but also from the standpoint of necessary air-box volume (see section 11.21), and even from the standpoint of the dynamic stresses.

In geometrically similar engines inertia forces increase as the fourth power of the bore and the square of the rpm; the bearing surfaces, etc. increase as the square of the bore. Therefore the stresses increase as the square of the (bore $\times$ rpm), unless the engine speed is reduced. The speed reduction must be in inverse ratio to the bore in order to keep the stresses unchanged.

All of these considerations favor small-bore multicylinder engines. They show conclusively that it is unjustifiable to expect the same specific output from a large cylinder as from a small cylinder, because the specific output decreases as the cubic root of the displacement volume.

18.6 Example.

The advantage of the small multicylinder engine can best be illustrated by an example. A loop-scavenged cylinder of 100 cubic inches displacement may be rated at 20 bhp at 1200 rpm. This corresponds to a bmep of 66 psi and a specific output of 0.2 bhp per cubic inch. By doubling the displacement volume the rotational speed must be decreased by $\sqrt[3]{2} = 1.26$ to 952 rpm. With the same bmep the power output will then be 0.159 bhp per cubic inch. By cutting the displacement in half to 50 cubic inches, the engine speed can be increased to 1512 rpm and the power output becomes 0.252 bhp per cubic inch.

The following table shows the comparison over a wide range of cylinder sizes. It is assumed that the stroke-bore ratio is 1.4:1. It has already been assumed that for this type of engine a fair bmep rating is 66 psi. The permissible speed is set, with regards to Fig. 10–1,

$$n = \frac{5400}{D}$$

where D is bore in inches and n the rpm. This speed requirement in combination with the above stroke-bore ratio is equivalent to a piston speed of 1260 foot per second for any size engine.

Table 18-II. Principal Data for Various Size Cylinders. Stroke-Bore Ratio 1.4:1, Bmep = 66 psi[1].

1. Displacement volume per cylinder (cu in.)	10	25	50	100	200	400	1000	2000	4000	10,000	20,000	40,000	100,000
2. Engine speed (rpm)	2585	1905	1512	1200	952	756	557	442	351	259	205	163	120
3. Bore (in.)	2.09	2.83	3.57	4.50	5.66	7.14	9.69	12.20	15.36	20.88	26.30	33.10	44.95
4. Stroke (in.)	2.92	3.97	5.00	6.29	7.93	10.00	13.57	17.08	21.50	29.23	36.82	46.34	62.93
5. Horsepower per cylinder	4.31	7.94	12.60	20.00	31.75	56.40	92.83	147.36	234.00	430.88	684.00	1085.76	2000
6. Horsepower per cu in.	0.431	0.318	0.252	0.200	0.159	0.126	0.093	0.074	0.059	0.043	0.034	0.027	0.020
7. Number of cylinders for a 2000 hp engine	465[2]	252[2]	159[2]	100	63	40[2]	22[2]	14[2]	9[2]	5[2]	3[2]	2[2]	1
8. Total piston displacement of 2000 hp engine (cu in.)	4650	6300	7950	10,000	12,600	16,000	22,000	28,000	36,000	50,000	60,000	80,000	100,000

[1] These data correspond to a piston speed of 1260 fpm and result in a power output of 1.26 hp per square inch piston area.
[2] Gives somewhat more than 2000 hp.

Table 18–II shows that while the cylinder displacement increases from 10 to 100,000 cubic inches, the specific power output decreases from 0.431 to 0.02 hp per cubic inch, because of the fact that the permissible engine speed decreases from 2585 to 120 rpm. The cylinder horsepower is 4.31 for the 10 cubic inch and 2000 for the 100,000-cubic-inch cylinder. The power output per piston area is, however, constant — 1.26 hp per square inch.

If a 2000 horsepower power plant is required, line 7 shows how many cylinders are necessary to supply that much power. It takes 465 10-cubic-inch cylinders or one 100,000-cubic-inch cylinder. However, the total displacement of the one large cylinder is 21.5 times greater than the 465 small cylinders as shown in line 8. Obviously the few large cylinders will weigh more and cost more than the more numerous small cylinders.

In Fig. 18–1 the data of Table 18–II are presented in graphical form with logarithmic coordinates. It is worth bearing in mind that while Fig. 18–1 is valid only for the specified stroke-bore ratio, bmep, and piston speed, it is necessary only to draw parallel lines to those shown in proper distances and the chart can be used for any engine with any combination of stroke-bore ratio, bmep and piston speed.

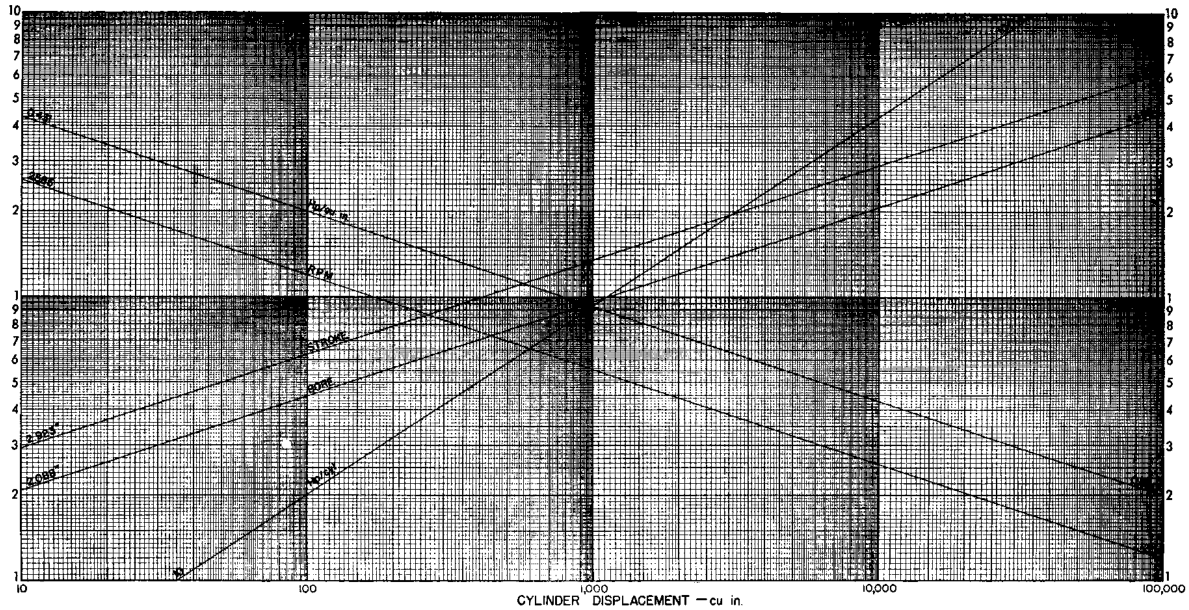

Fig. 18-1. Principal Data for Various Size Cylinders with a Stroke-Bore Ratio of 1.4 and a Bmep of 66 psi. These data correspond to a piston speed of 1260 fpm and result in a power output of 1.26 hp per sq. in. piston area.

18.7 The Case for Large Cylinders.

From the foregoing it would appear that the large cylinder has no case at all. While it is true that on the basis of specific power output or weight their case is weak, it would be wrong to conclude that large power plants should be built only of a multiplicity of small cylinders.

In the first place, diesel engines of less than about 5-inch bore have as a rule less efficient combustion because the combustion space is inadequate to accommodate the fuel spray. The result is higher fuel consumption, which is frequently accompanied by exhaust smoke. Diesel engines of less than 4-inch bore have, in addition, increasing difficulties in cold starting.

The specific fuel consumption generally improves with cylinder size, but it begins to flatten out at about 10-inch bore. The improvement in fuel consumption with very large engines is insignificant.

From these considerations it may be concluded that the most competitive diesel cylinder size is around 5-inch diameter, if fuel economy is of little consequence, and around 10-inch diameter if fuel economy is important. In practice it is found that two-stroke cycle engines are most popular in the very small,* the small, and the very large sizes.

Most diesel engines in road vehicles have cylinder bores between 4 and 6 inches and most locomotive engines have cylinder bores 8 and 12 inches. The latter have, as a rule, better fuel economy than the former.

* The very small size economy asserts itself most fully in the field of port-scavenged *spark-ignition* engines. There the starting problem is absent, and the absence of valves makes the production of small two-stroke cycle engines most economical. The widespread adoption of this type of engine for passenger automobiles is predicated upon the perfection of a method that would allow the scavenging of the cylinders without thereby wasting fuel.

Relatively few two-stroke cycle engines have been designed recently between 10- and 20-inch bore. The reason is evident from the foregoing analysis. Any possible improvement in fuel economy is insignificant with larger engines, while weight and first cost are decidedly lower with smaller size engines.

In the field of very large engines, with several thousand horsepower output, two-stroke cycle engines of 20- to 40-inch bore are frequently used. In case of ocean-going vessels the direct connection of the propeller with the engine offers an advantage which is appreciated in the marine field. The economical speed of large propellers is between 100 and 200 revolutions per minute, which fits the largest cylinder sizes developing up to 1000 brake horsepower or more per cylinder. Because of its simplicity, the valveless two-stroke cycle engine is supreme in this field. Similar engines are frequently used for stationary power plants because of their durability and because a larger number of small engines would complicate the parallel connections of electric generators.

In the range of 10- to 20-inch bore, or 1000- to 10,000-cubic-inch cylinder displacement, the economical speeds lie between 250 and 600 rpm. This engine size encounters increasing competition from the smaller bore engines.

As it was demonstrated in the foregoing analysis, small cylinder displacement engines are inherently more economical in first cost because they require less total displacement volume for a given power output. The very small cylinders at present suffer from higher fuel consumption (and starting difficulties in cold weather), but engineering progress may correct these shortcomings while the arithmetical superiority of the small cylinders will remain unchanged.

18.8 Confirmation.

Table 18–I naturally confirms the foregoing conclusions. Among the engines listed the specific output varies from 0.0218 to 0.62 hp per cubic inch of cylinder displacement, generally increasing with decreasing cylinder size. The ratio between the "best" and the "worst" is more than 28:1. The horsepower output per square inch of piston displacement varies from 0.255 to 3.55, but if the crankcase-scavenged engines are ignored, only from 0.745 to 3.55. On the piston area basis this variation is only 3.8:1, and more than half of the listed engines deliver between 1 and 2 horsepower per square inch piston area.

BIBLIOGRAPHY

ACKERET, J., "Grenzschichtabsaugung," *VDI Zeitschr.*, **70,** 1153 (1926).

"Allen Two-Stroke Blower Scavenged Diesel Engine," *Engineering,* **159,** 27–8 (January 12, 1945).

ASCHENBRENNER, J., "Einfluss der Gastraegheit auf den Liefergrad," *Forsch. Ing. Wes.*, **8,** 285–94 (1934).

—— "Nichtstationaere Gasstroemungen in Leitungen mit veraenderlichem Querschnitt," *Forsch. Ing. Wes.*, **8,** 118 (1937).

ASME Fluid Meters, Description of Meters, 1931.

—— *Fluid Meters, Their Selection and Installation*, 1933.

—— *Fluid Meters, Their Theory and Application*, 4th edition, 1937.

—— *Test Code for Displacement Compressors, Vacuum Pumps, and Blowers*, 1939.

BACHLE, C. F., "Turbine Compounding with the Piston Engine," *SAE Journal*, **53,** 345–51 (June 1945).

BELILOVE, S., "Improving Two-Stroke Cycle Engine Performance by Exhaust Pipe Tuning," *Thesis, The Pennsylvania State College*, August 1941.

—— "Improving Two-Stroke Cycle Engine Performance by Exhaust Pipe Tuning," *Diesel Power*, **21,** 608–13 (July 1943).

BIANCHI, L., "Arbeiten an Zweitaktmotoren," *Deutsche Luftfahrtforschung, Zwischenbericht 4.*

BORTH, W., "Schwingungs- und Resonanzerscheinungen in den Rohrleitungen von Kolbengeblaesen," *VDI Zeitschr.*, **60,** 565 (1919).

BUCHHOLZ, F., L. HOFFMANN, AND K. JOACHIMSOHN, "Untersuchungen ueber den volumetrischen Wirkungsgrad an einem Zweizylinder-Zweitaktmotor mit Kurbelkammer-Gemischpumpe," *A. T. Z.*, **33,** 505 (1930).

BUCHI, A. J., "Supercharging of Internal Combustion Engines with Blowers Driven by Exhaust Gas Turbines," *ASME Trans.*, **59,** 85–96 (February 1937).

—— "Turbocharging and Gas Turbines," *Journal of the American Society of Naval Engineers*, **60** (3), 261–291 (August 1948).

CALDERWOOD, J., "Some Researches on Internal Combustion Prime Movers," *North East Coast Inst. Engrs.* (1946).

CAPETTI, A., "Effect of Intake Pipe on the Volumetric Efficiency of an Internal Combustion Engine," *NACA Tech. Mem., No. 501* (February 1939).

CARTER, H. D., "The Loop Scavenged Diesel Engine," *Inst. of Mech. Engineers Proceedings*, **154,** No. 4, 386 (1946).

CAUNTER, C. F., *The Two-Cycle Engine*, London, Sir Isaac Pitman and Sons, Ltd., 1932.

CHIPLEY, A. S., *Diesel-Engine Exhaust Silencing*, Burgess Battery Co., Acoustic Division, 1941.

CLERK, DUGALD, *The Gas, Petrol, and Oil Engine*, London, Longmans, Green and Co., 1909.

Curtis, C. G., "Improvement in the Scavenging and Supercharging Two-Cycle Diesel Engines," *Diesel Power*, **11**, No. 9, 545–9 (September 1933).

—— "Two-Cycle *vs.* Supercharged Four-Cycle," *Diesel Power*, **19**, 134–6 (February 1941).

Dale, C. B. M., "Recent Development in Medium-Speed Two-Cycle Oil Engines," (British) *Motor Ship*, **25**, No. 300, 346–8 (January 1945).

Danckwortt, T. B., "Engine Data on Small Two-Cycle Diesel Engines up to 1938," *Mimeographed Communication*, 1939.

Davies, S. J., "The Characteristics of Engines of Kadenacy Design," *Engineering*, **143**, 685, 715 (1937).

—— "An Analysis of Certain Characteristics of a Kadenacy Engine," *Engineering*, **149**, 515–57, 617 (1940).

—— "Tests of Kadenacy High-Duty, Moderate Speed Experimental Unit," *Engineer*, **171**, No. 4448, 238–40 (April 11, 1941).

DeJuhasz, K. J., "Measuring the Scavenging Efficiency of Two-Stroke Diesel Engines," *Autom. Ind.*, **73**, No. 26 (December 1935).

DeJuhasz, K. J., O. F. Zahn, and P. H. Schweitzer, "On the Formation and Dispersion of Oil Sprays," *The Pennsylvania State College, Engineering Experiment Station, Bulletin No. 40* (1942).

Dennison, E. S. "Inertia Supercharging of Engine Cylinders," *Proc. 6th Oil Power Conference, The Pennsylvania State College, Technical Bull. No. 16* (1932).

"Design for Welded Exhaust Manifold Wins Lincoln Award," *Diesel Progress* (February 1948).

Drabert, F., "Untersuchung eines obengesteuerten Zweitakters mit Gleichstromspuelung," *Diss. T. H. Muenchen* (1939).

Earle, S. L., and F. J. Dutee, "Performance of a Two-Cycle Compression-Ignition Engine," *NACA Tech. Note, 610* (1937).

Ehmsen, E., "Zweitakt Krupp Kreuzkopf Dieselmotoren fuer d. Schiffsantrieb," *M. T. Z.*, **I**, 9, 56 (1939).

Eichelberg, G., "Zweitakt-Schiffs-Dieselmotor von 5500 PS_e," *VDI Zeitschr.*, **79**, 130 (1935).

Farmer, H. O., "Exhaust Systems of Two-Stroke Engines," *Inst. of Mech. Engineers Proceedings*, **143**, 396 (1938).

Fezer, E., "Der Entspannungsvorgang bei Zweitakt-Motoren," *M. T. Z.*, **3**, 399–402 (1941).

Foeppl, O., "Berechnung der Kanallaengen von Zweitaktoelmaschinen mit Schlitzsteuerungen," *VDI Zeitschr.*, **57**, 1939 (1913).

Foeppl, O., H. Strombeck, and L. Ebermann, *Schnellaufende Dieselmaschinen*, 4th edition, Berlin, Springer, 1921.

Foettinger, H., "Ueber einige Forschungsarbeiten auf dem Gebiete der Stroemungslehre," *Schiffbautech. Jahrbuch*, **39**, 240 (1938).

Foettinger, H. and R. Wille, "Beitrag zur Klaerung des Spuelvorganges der Zweitaktmaschinen," *Dtsch. Kraftfahrtforsch. Zwischenbericht 86* (1940).

Froede, W., "Zweitaktmotoren ohne Spuelgeblaese," *VDI Zeitschr.*, **82**, 119 (1938).

Gaesterstadt, J., "Development of the Junkers-Diesel Aircraft Engine," *NACA Tech. Mem. 565* (1930).

GAESTERSTADT, J., "Junkers Heavy Oil Aero Engine," *The Engine*, **163**, 167–70 (February 5, 1937).

GEYER, E. W., "Theory of Kadenacy System," *Engineering*, **151**, No. 3935, 463–4 (June 13, 1941).

GIFFEN, E., "Rapid Discharge of Gas from a Vessel into the Atmosphere," *Engineering*, **150**, 134–6, 154–5, 181–3 (1940).

GOODE, C. T., "Diesel Engine Exhaust Measurement and Analysis of Smoke and its Uses," *Automob. Eng.*, **36**, 293 (July 1945).

GUTMANN, E., "Untersuchungen zur rechnerischen Bestimmung der Lufteinlaszschlitze bei Zweitaktverbrennungskraftmaschinen," *VDI Zeitschr.*, **48**, 785 (1914).

HARRIS, L. S., "Theoretical Analysis of the Performance Characteristics of a Hypothetical Hot Gas Generator Turbine Power Plant," *Thesis, The Pennsylvania State College*, August 1946.

HARTMANN, J., "On the Kadenacy Motors," *Ingenioren*, Kopenhagen (April 22, 1944).

HEISE, M., "Entwicklung eines Gegenkolben-Zweitakters," *Motor-Kritik*, **22**, 117 (1942).

HELDT, P. M., *High Speed Diesel Engines*, 5th edition, Nyack, N. Y., P. M. Heldt, 1947.

HEROLD, R., "Automotive Diesels of the Two-Stroke Horizontal-Opposed Piston Design," *Autom. Ind.* (July 1, 1945).

HOLD, P., "Untersuchungen von Spuelsystemen an der zuendenden Zweitaktmaschine," *M. T. Z.* **2**, 347–54 (1940).

—— "Reduzierter Zeitquerschnitt und Durchflusszahl bei Spuelsystemen von Zweitaktmaschinen, *M. T. Z.* **3**, 83 (1941).

HOLM, O., "Untersuchungen ueber den Einfluss der Spuel- und Auspuffschlitzabmessungen auf den Spuelvorgang an einem 15 PS-Gluehkopf-Motor," *Diss. T. H. Danzig* (1926).

—— "Crankcase Scavenging," *NACA Tech. Mem. 446* (1927).

—— "Entwicklungsmoeglichkeiten der Kurbelkastenspuelung von Zweitaktmotoren," *A. T. Z.*, **36**, 507 (1933).

HOPKINSON, B., *Trans. N. E. Coast Inst. Engrs.*, **XXX** (1914).

HUBER, E. W., "Arbeiten an Zweitaktmotoren," *Deutsche Luftfahrforschung Forschungsarbeit, No. 1526/3* (January 1942).

JALBERT, J., "Two-Stroke Cycle Engines for Airplanes," *NACA Tech. Mem. 347* (1925).

KADENACY, M., U. S. Patent 2,102,559 (1937).

—— U. S. Patent 2,110,986 (1938).

—— U. S. Patent 2,113,480 (1938).

—— U. S. Patent 2,123,569 (1938).

—— U. S. Patent 2,130,721 (1938).

—— U. S. Patent 2,131,957 (1938).

—— U. S. Patent 2,131,959 (1938).

—— U. S. Patent 2,144,065 (1939).

—— U. S. Patent 2,147,200 (1939).

—— U. S. Patent 2,168,520 (1939).

KAISER, E. R., "Pressure Waves in the Exhaust Pipes of Two-Stroke Cycle Internal Combustion Engines," *Thesis, University of Wisconsin* (1935).

Karde, K., "Die Verbesserung des Arbeitsverfahrens der einfachen, raschlaufenden Zweitakt-Vergasermaschine durch Einfuehrung neuer Spuelverfahren," *A. T. Z.*, **38,** 421 (1935), and *VDI Zeitschr.*, **80,** 428 (1936).

Kastner, L. J., "Exhaust Pipe Pressure Waves," *Engineering*, **149,** 301 (October 18, 1940).

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

Klinge, E. R., P. H. Schweitzer, and F. C. Stewart, "Two Easily-Constructed 0.002″ Micromanometers," *Instruments*, **17,** 12 (December 1944).

Kluesener, O., "Das Arbeitsverfahren raschlaufender Zweitaktvergasermaschinen," *VDI Forsch.*, *334* (1930).

—— "Versuche ueber den Einfluss von Saug- und Auspuffrohrlaenge auf den Liefergrad," *A. T. Z.*, **35,** 299 (1932).

—— "Saugrohr und Liefergrad," *VDI Dieselmaschinen*, **V,** 107 (1932).

Kluge, H., K. v. Sanden, and W. Spannhake, "Einrichtung fuer die Untersuchung des Spuelvorganges in kleinen Zweitakt-Schnellaeuferzylinder," *Kraftfahrtechn. Forsch.*, **7,** 1–5 (1937), and *VDI Zeitschr.*, **82,** 329 (1938).

Kluge, H. and W. Spannhake, "Einrichtung fuer die Untersuchung des Spuelvorganges in kleinen Zweitakt-schnellaeufern und damit erhaltene Ergebnisse," *Dtsch. Kraftfahrtforsch., Zwischenbericht, Nr. 86* (1940).

Kluge, M., "Problem der Daempfung des Auspuffschalles der Kraftfahrzeugmotoren," *A. T. Z.*, **36,** 194, 244 (1933).

Koerner, K., *Der Bau des Dieselmotors*, 2nd edition, Berlin, J. Springer, 1927.

Kornacker, P., "Untersuchung ueber die Ausnuetzung der Auslassenergie zur Leistungsteigerung von Motor und Abgasturbine," *Jahrbuch der deutschen Luftfahrtforschung*, **11** (1940).

Kreglewski, A., "Die Spuel- und Auspuffvorgaenge bei Zweitakt-Verbrennungsmaschinen mit besonderer Beruecksichtigung der schnellaufenden Oelmotoren," *Oelmotor*, **2,** 553 (1913–4).

Kress, H., "Investigation regarding the Rate of Delivery by Overloading," *M. T. Z.*, **4,** 175 (1942).

Laudahn, W., "Die Nuernberger Groszoelmaschine," *VDI Zeitschr.*, **67,** 1093, and **68,** 1171 (1923).

—— "Die Entwicklung der Germania-Groszoelmaschine," *VDI Zeitschr.*, **68,** 1202 (1924).

Leadbetter, R. L., "Exhaust and Intake Systems for Best Performance and Quiet Operation," *Proc. Nat. Conference Oil and Gas Power Div.*, ASME, Tulsa (1944).

Ledertheil, H. *Die Zweitaktmotoren und ihr Anwendungsgebiet*, 3rd edition, Berlin, Rich. Carl Schmidt and Co., 1925.

Lee, D. W., "A Study of Air-Flow in an Engine Cylinder," *NACA Tech. Report*, *653* (1939).

Leonhardt, R., "Scavenging Air Flow in Two-Cycle Diesels," *Diesel Power*, **11,** No. 7, 442 (July 1933).

Lindner, W., "Untersuchung ueber den Spuelvorgang an Zweitaktmaschinen," *Forsch. Ing. Wes.*, *363* (1933).

List, H., "Kurbelkastenspuelung bei Zweitaktmotoren," *VDI Zeitschr.*, 73, 226 (1929), Trans. "Crankcase Scavenging of Two-Stroke Cycle Engines," *NACA Tech. Mem.*, *516* (June 1929).

—— "Der Spuel und Ladevorgang bei Zweitaktmaschinen," *VDI Zeitschr.*, **74,** 1274 (1930).

LIST, H., "Spueluntersuchungen an einem kleinen Zweitaktzylindermodell," *Mitt. tech. Inst. Woosung,* (China), **6** (1932).

—— "Die Erhoehung des Liefergrades durch Saugrohre bei Dieselmotoren," *Mitt. tech. Inst. Woosung,* (China), *Report No. 4* (April 1932), Trans. "Increasing the Volumetric Efficiency of Diesel Engines by Intake Pipes," *NACA Tech. Mem. No. 700* (March 1933).

—— "Untersuchungen am Zweitaktmotor," *Mitt. tech. Inst. Woosung,* (China), **1** (1933).

—— "Leistungsgrenzen von Zweitakt Dieselmaschinen," *Forsch. Ing. Wes.*, **8,** No. 2, 57 (1937).

—— *Die Verbrennungskraftmaschine,* Berlin, J. Springer, 1939.

—— "Die Stroemung in Saugrohren von Verbrennungskraftmaschinen," *VDI Zeitschr.*, **85,** 301–5 (1941).

—— "Ueberblick ueber Arbeiten des Institutes fuer Kolbenmaschinen," *Institut fuer Kolbenmaschinen, I. Bericht* (August 12, 1945).

LIST, H. AND E. NIEDERMAYER, "Versuche an der Zweitaktdieselmaschinen," *Forsch. Ing. Wes.*, **8,** No. 6, 265 (1935), and **82,** 504 (1938).

LOVETT, P. B., "Engine Performance with Different Exhaust Systems on a GM 3–71 Two Stroke Cycle Diesel Engine, " *Thesis, The Pennsylvania State College,* September 1946.

LUTZ, O., *Untersuchungen ueber die Spuelung von Zweitaktmotoren,* Stuttgart, K. Wittwer, 1931.

—— "Aehnichkeitsbetrachtungen bei Brennkraftmaschinen," *Ingenieur-Archiv,* **IV,** 373 (1933); Trans. "Dynamic Similitude in Internal-Combustion Engines," *NACA Tech. Mem.* 978 (1941).

—— "Grundsaetzliche Betrachtungen ueber den Spuelvorgang bei Zweitaktmaschinen," *Forsch. Ing. Wes.*, **5,** No. 6, 275 (November–December 1934).

—— "Einfluss von Schwingungen auf den Spuelvorgang," *Dtsch. Kraftfahrtforsch., Zwischenbericht, 86* (1940).

—— "Die Graphische Ermittlung von Ausstroemvorgaengen," *Luftfahrtforschung,* **17** (8), 332 (1940); Trans. "Use of Charts for Flow Discharge Calculations," *NACA Tech. Mem.* 972 (1941).

LUTZ, O. AND W. MAIER, "Untersuchungen ueber die Spuelung von Zweitaktmotoren," *Berichte aus dem Laboratorium fuer Verbrennungskraftmaschinen der T. H. Stuttgart,* **1** (1931).

LUTZ, O. AND W. NOEGGERATH, "Spuelvorgang bei Zweitaktmaschinen," *Dtsch. Kraftfahrtforsch.*, **23** (1939), and *VDI Zeitschr.*, **84,** 136 (1940).

LYSHOLM, ALF., B. SMITH AND W. A. WILSON, "The Elliott-Lysholm Supercharger," *SAE Journal* (June 1943).

MAGG, J., *Dieselmaschinen,* Berlin, J. Springer, 1928.

MAIER, W. AND O. LUTZ, "Untersuchungen ueber die Spuelung von Zweitaktmotoren," *Berichte aus dem Laboratorium fuer Verbrennungskraftmaschinen der T. H. Stuttgart, 2* (1933).

—— "Resonanzschwingungen in den Rohrleitungen von Kolbenmaschinen," *T. H. Stuttgart, 3* (1934).

MARTIN, H., "Daempfung des Auspuffschalles an Kraftfahrzeugmotoren," *Diss. T. H. Dresden* (1934), and *VDI Zeitschr.*, **78,** 1257 (1934).

MARTIN, H., U. SCHMIDT, AND W. WILLMS, "Der Entwicklungsstand der Auspuffschalldaempfer," *M. T. Z.*, **2,** 377–84 (1940), and **3,** 11–16 (1941).

MARX, G., "Integration der Ausfluszgleichung," *Angewandte Math. und Mech. Zeitschr.*, **2,** 155 (1931).

MEHLIG, H., "Ermittlung von Zeitquerschnitten fuer Zweitakt-Verbrennungskraftmaschinen," *A. T. Z.*, **36,** 610 (1933).

MERTZ, R., "Probleme des Zweitaktmotors," *A. T. Z.*, **45,** 149 (1942).

MEYER, P., "Grundlagen fuer die Untersuchung von Zweitaktmaschinen," *VDI Zeitschr.*, **21,** 1615 (1912).

MILLER, R. H., "Analysis of Intercooled Supercharger," *ASME Trans.*, **68,** 735–9, Discussion, 739–41 (October 1934).

MOHR, K., "Entwicklung der Spuelung fuer Zweitaktdieselmaschinen," *Kruppsche Monatshefte*, **12,** 230 (1931).

MUCKLOW, G. F., "Kadenacy System of Scavenging," *Engineering*, **148,** No. 3840, 187–9 (August 18, 1939), No. 3845, 330 (September 22, 1939), No. 3846 (September 29, 1939), No. 3848, 421 (October 13, 1939), and No. 3854, 589–90 (November 24, 1939).

—— "Exhaust Pipe Effects in a Single-Cylinder Four-Stroke Engine," *The Institution of Mech. Engineers, Proceedings*, **143,** 109 (1940).

—— "Communications on Exhaust Pipe Effects in a Single-Cylinder Four-Stroke Engine," *The Institution of Mech. Engineers, Proceedings* (November 1940).

—— "Exhaust Pipe Phenomena," *The Manchester Association of Engineers*, **21** (1942), Abstract in *The (British) Motorship* (January 1943).

NAEGEL, A., "Dieselmaschine der Gegenwart," *VDI Zeitschr.*, **67,** 733 (1923).

—— "Dieselmaschinen in Amerika," *VDI Dieselmaschinen*, **I** (1923).

NEUMANN, K., "Die dynamische Wirkung der Abgassaeule in den Auspuffleitungen von Kolbenmaschinen," *VDI Zeitschr.*, **63,** 89 (1919).

—— "Untersuchungen an der Dieselmaschine. Der Spuel- und Ladevorgang bei Zweitaktmaschinen," *VDI Forsch.*, **334** (1930).

—— "Die thermodynamischen Grundlagen des Spuel- und Ladevorgangs bei Zweitaktmaschinen," *Forsch. Ing. Wes.*, **2,** 273 (1931), and *VDI Zeitschr.*, **76,** 468 (1932).

—— "Junkers-Freikolben Verdichter," *VDI Zeitschr.*, **79,** 155 (1935).

NIEDERMAYER, E., "Untersuchung des Spuelvorganges an Zweitaktdieselmaschinen," *Forsch. Ing. Wes.*, **7,** 227–39 (1936).

—— "Zusammenhang zwischen Ladungswechsel und Leistung," *Dtsch. Kraftfahrtforsch., Zwischenbericht, 103* (1941).

NUSSELT, W., "Die Stroemung von Gasen durch Blenden," *Forsch. Ing. Wes.*, **3,** No. 1 (January–February 1932).

OEDERLIN, F., "Die Aufladung des Zweitakt-Dieselmotors," *Schweitzerische Bauzeitung*, **119,** No. 13, 147–52 (March 28, 1942), and 166–70 (April 4, 1942).

OPPITZ, A., "Schwingungserscheinungen in den Auspuffleitungen der Dieselmotoren," *Werft, Reederei und Hafen*, **11,** 27 (1930).

PATON, C. R. AND C. KEMPER, "Power Output and Air Requirements of a Two-Stroke Cycle Engine for Aeronautical Use," *NACA Tech. Report, 239* (1926).

PESCARA, R., "Les machines à pistons libres," *J. Soc. Ing. Automot.*, **10,** 423 (1937).

—— "La combustion dans les chambres à volume variable," *Chal. et ind.*, **20,** 143 (1939).

—— "Generateur à piston libres et turbine à gaz," *Chal. et ind.*, **20,** 211 (1939).

PETTER, B. W., "Induced Air Scavenge for Two-Stroke Engines," *Engineer*, **158,** 157 (1934).

PETTER, H., "The Escape of Exhaust Gas in Two-Stroke Engines," *Engineering* (January 4, 1918).

"Petter 'Harmonic Induction'," *Engineer*, **143,** 685, 715 (1937).

"Petter New Diesel Embodying a Uniflow Scavenging System," *Automob. Eng.*, **28,** 279–81 (August 1938).

PFLAUM, W., "Zusammenwirkung von Motor und Geblaese bei Auflademaschinen," *VDI Dieselmaschinen*, **VII** (1938).

PIGOTT, R. T. S., "Analysis of Positive Supercharger Losses," *SAE Annual Meeting Paper* (January 1947).

PINKEL, B., "NACA Study of Utilization of Exhaust Gas of Aircraft Engines," *SAE Trans.*, **54,** 714 (1946).

PISCHINGER, A., "Bewegungsvorgaenge in Gassaeulen insbes. beim Auspuff- und Spuelvorgang von Zweitaktmaschinen," *Forsch. Ing. Wes.*, **6,** 245, 273, 430 (1935).

PITCHFORD, J. H., "Aufladung kleiner Dieselschnellaeufer," *M. T. Z.*, **1,** 156 (1939).

PYE, D. R., *The Internal Combustion Engine*, Vol. II, Oxford at the Clarendon Press, 1934.

PYK, H., "Some Developments in Diesel Engine Design: Especially Referring to Scavenging Problems," *Trans. N. E. Coast Inst. Engineers and Shipbuilders*, **64,** 71 (1948).

REGENBOGEN, C., "Der Dieselmotorenbau auf der Germaniawerft," *Schiffbautechn. Jahrbuch*, **14,** 254 (1913).

REINSCH, A. AND U. SCHMIDT, "Untersuchungen an Zweitaktmotoren 1," *Kraftfahrtechn. Forsch.*, **5,** 6–17 (1936).

—— "Untersuchungen an Zweitaktmotoren 2," *Kraftfahrtechn. Forsch.*, **5,** 18–21 (1936).

RESCHEK, O., "Wirtschaftlichkeit und Leistung raschlaufender Kurbelkammer-Zweitakter," *A.T.Z.* **45,** 86 (1942).

REYL, G., "Untersuchungen an Saugrohren," *Dtsch. Kraftfahrtforsch.*, 74 (1943).

REYNOLDS, B., H. SCHECTER AND E. S. TAYLOR, "The Charging Process in a High-Speed Single-Cylinder Two-Stroke Engine," *NACA Tech. Note, 675* (1938).

RIDE, J. L., "The Positive Displacement Supercharger," *SAE Journal* (July 1942).

RIEDEL, W., "Arbeiten an Zweitaktmotoren," *Deutsche Luftfahrtforschung, Zwischenbericht 1* (1942).

—— "Arbeiten an Zweitaktmaschinen," *Zentrale fuer wiss. Berichtwesen der Luftschiffart, Forschungsbericht*, 1526/1 (1942).

RIEHM, W., "Die doppeltwirkende Zweitakt-Schlitzspuelmaschine der MAN," *Werft, Reederei und Hafen*, **12,** 272 (1924).

RINGWALD, M., "Der Auspuff- und Spuelvorgang bei Zweitaktmaschinen," *VDI Zeitschr.*, **67,** 1057, 1079 (1912).

ROGOWSKI, A. R. AND C. L. BOUCHARD, "Scavenging a Piston-Ported Two-Stroke Cylinder," *NACA Tech. Note 674* (November 1938).

ROGOWSKI, A. R., C. L. BOUCHARD, AND C. F. TAYLOR, "The Effect of Piston Head Shape and Exhaust Restrictions on the Performance of a Piston-Ported Two-Stroke Cylinder," *NACA Tech. Note 756* (March 1940).

SASS, F., "Doppeltwirkende kompressorlose Zweitaktdieselmotoren fuer Schiffsantrieb," *Schiffbautechn. Jahrbuch*, **29**, 292 (1928).

—— *Kompressorlose Dieselmaschinen*, Berlin, J. Springer, 1929.

SCHMIDT, E., "Arbeiten an Zweitaktmotoren," *Deutsche Luftfahrtforschung, Zwischenbericht 2*, (1942).

SCHMIDT, F. A. F., *Verbrennungsmotoren*, Berlin, J. Springer, 1939.

SCHMIDT, TH., "Schwingungen in Auspuffleitungen von Verbrennungsmotoren," *Forsch. Ing. Wes.*, **5**, 226 (1936).

—— "Schwingungen in Auspuffleitungen von Verbrennungsmotoren," *VDI Dieselmaschinen* **VI, 79** (1939).

SCHMIDT, U., "Untersuchungen an Zweitaktmotoren," *3. Bericht, Kraftfahrtechn. Forsch.*, **7**, 14–25, and **8** (1937).

—— "Untersuchungen an Zweitaktmotoren," *4. Bericht, Kraftfahrtechn. Forsch.*, **8**, 1–10 (1937).

—— "Leistung und Wirtschaftlichkeit des gemischgespuelten Zweitaktmotors," *A. T. Z.*, **40**, 605–13 (1937).

—— "Schwingungen auf der Ansaugseite von Zweitaktmotoren mit Kurbelkastenpumpe," *Dtsch. Kraftfahrtforsch.*, **19** (1938).

—— "Die Aenderung des Spuelerfolges bei erhoehtem Spuelmittelaufwand bei Zweitaktmotoren," *Dtsch. Kraftfahrtforsch., Zwischenbericht 86* (1940).

—— "Die aeussere Beeinflussung des Spuelverlaufes bei gemischgespuelten Zweitaktmotoren," *A. T. Z.*, **44**, 120–4 (1941).

SCHUETTE, A., "Schwingungen in Auspuff und Spuelluftleitungen von Zweitaktmotoren," *Werft, Reederei und Hafen*, **12**, 322 (1931).

—— "Untersuchung der Spuelung einer doppeltwirkenden Zweitaktmaschine." *Dieselmaschinen* **V**, 118, 141 (1932).

—— "Der Druckverlauf im Zylinder eines Zweitaktdieselmotors waehrend der Spuelperiode," *A. T. Z.* (March 25, 1935).

—— "Die Spuelung bei Auflademaschinen," *Mitt. Forsch. Anst. Gutehoffnungshuette*, **6**, 65 (1938).

SCHULTZ-GRUNOW, F., "Modellversuche ueber den instationaeren, ebenen Spuelvorgang im Zweitaktmotor," *Forsch. Ing. Wes.*, **9**, 235 (1938).

—— "Modellversuche zum Spuelvorgang in Zweitaktmotoren," *VDI Zeitschr.*, **82**, 403 (1938).

—— "Eine neue Auffassung der Expansions- und Kompressionsstroemung in Zylinder," *Dtsch. Kraftfahrtforsch.*, **86** (1940).

SCHULTZ-GRUNOW, F. AND K. WIEGHARDT, "Der Spuelvorgang auf Grund einer neuen Auffassung der Expansionsstroemung," *Dtsch. Kraftfahrtforsch.*, **6** (1941).

SCHWEITZER, P. H., "How Much Can Be Gained by Exhaust Turbines?" *Mech. Engrg.*, **47**, 860 (October 1925).

—— "Is the High Speed Two Cycle Diesel Possible?" *Power* (June 19, 1928).

—— "Factors in Diesel Spray-Nozzle Design in the Light of Recent Oil Spray Research," *ASME Trans.*, **52**, 121–9 (December 1930).

—— "Combustion Knock in Diesel Engines," *Proc. of Sixth Oil Power Conference, Tech. Bulletin 16, Eng. Exp. Station, The Pennsylvania State College*, 73–82, 98–101 (April 1933), Abstract in *Autom. Ind.*, **2**, 848–51 (June 11, 1932).

SCHWEITZER, P. H., "Penetration of Oil Sprays in Dense Air," *Autom. Ind.*, 290–300 (September 8, 1934).

—— "Diesel Fuel Pumps with Declining Delivery-Speed Characteristic Best in Vehicle Service," *Autom. Ind.*, **73**, 136–8 (August 3, 1935).

—— "Penetration of Oil Sprays," *Bulletin 46, Eng. Exp. Station, The Pennsylvania State College* (August 1937).

—— "Mechanism of Disintegration of Liquid Jets," *Journal of Applied Physics*, **8**, 513–21 (August 1937).

—— "Methods of Rating Diesel Fuels," *Chemical Reviews*, **22**, 107–42 (February 1938).

—— "Injection of Diesel Fuel Into Flame Cuts Ignition Lag Only Moderately," *Autom. Ind.*, **78**, 838–54 (June 25, 1938).

—— "Smokemeter of Simple Design Tests Diesel Combustion," *Autom. Ind.*, **79**, 238 (August 20, 1938).

—— "What Can Be Gained by Pilot Injection?" *Autom. Ind.*, **79**, 533–4 (October 9, 1938).

—— "Penetration of Oil Sprays," *Journal of Applied Physics*, **9**, 735–41 (December 1938).

—— "Cathode-Ray Engine Indicators," *Autom. Ind.*, **80**, 612, 613, 621 (May 15, 1939).

—— "The Limits of Performance of Automotive Diesel Engines," *Autom. Ind.* (December 6, 1939).

—— "Where Does Interpretation Begin? — Analysis of the Oil-Engine Power Cost Report," *ASME Trans.*, **62**, No. 2 (February 1940).

—— "Oxygen Boosting of Airplane Diesel Engines," *Mech. Eng.*, **62**, 719–22 (October 1940).

—— "Use of the Oscillograph for Testing Fuel Injection," *Autom. Ind.*, **86**, 26–30 (January 1, 1942).

—— "Knock-to-Smoke Range — A Measure of Diesel Engine Flexibility," *Autom. Ind.*, **86**, 28–9 (March 1942).

—— "The Improved Penn State Smokemeter," *Instruments*, **15**, 346–50 (September 1942).

—— "The Definition of Volumetric Efficiency," *Engineering* (February 26, 1943).

—— "Excess Air and Brake Mean Effective Pressure," *ASME Trans.*, **63**, No. 3 (April 1943).

—— "Servicing of Injection Nozzles," *Diesel Power and Diesel Transportation* **21**, 706–12 (August 1943).

—— "More HP per Lb. of Diesel," *Power*, **87**, 79–81 (September 1943), and **87**, 80–91 (October 1943).

—— "Injection Pump Test Stand," *Autom. Ind.*, **89**, 32–3 (October 15, 1943).

—— "Stroboscope Unmasks Fuel Injection," *Autom. Ind.*, **89**, 32–4 (October 15, 1943).

—— "Porting of Two-Stroke Cycle Diesel Engines," *Diesel Power and Diesel Transportation* (May, June, July, August, September, October, November, December, 1942; January, February, March, December, 1943).

—— "Improving Engine Performance by Exhaust Pipe Tuning," *Journal Am. Soc. Naval Engineers*, **56**, No. 2, 185 (May 1944).

—— "Trouble Shooting with an Indicator," *Diesel Progress*, Part I, 44 (October 1946); Part II, 51 (November 1946).

—— "Indicator Problems," *Diesel Progress*, 74 (December 1946).

—— "Horsepower from the Indicator," *Diesel Progress*, 47 (April 1947).

—— "Must Diesel Engines Smoke?" *SAE Quarterly Trans.*, **1**, No. 3 (July 1947).

SCHWEITZER, P. H. AND J. S. CHANDLER, "Rating of Diesel Fuels. A Study of their Ignition Quality in Commercial Engines," *Automob. Eng.*, **30**, 135–41 (May 1940).

SCHWEITZER, P. H. AND F. DELUCA, "The Tracer Gas Method of Determining the Charging Efficiency of Two-Stroke Cycle Diesel Engines," *NACA Tech. Note 838* (January 1942).

SCHWEITZER, P. H. AND T. B. HETZEL, "Cetane Rating of Diesel Fuels," *SAE Journal*, **38**, 206–13 (May 1936).

SCHWEITZER, P. H. AND E. R. KLINGE, "Oxygen Boosting of Diesel Engines for Take-Off," *Eng. Exp. Station Bulletin, No. 54, The Pennsylvania State College* (April 1941).

SCHWEITZER, P. H., C. W. VAN OVERBEKE, AND L. MANSON, "The Kadenacy Effect," *Engineering*, 241–4 (September 28, 1945).

—— "Taking the Mystery out of the Kadenacy System of Scavenging Diesel Engines," *ASME Trans.* (October 1946).

SEIFERT, G., "Untersuchung einer Doppelkolben-Zweitakt-Maschine mit Ueberladung," *Diss. T. H. Muenchen* (1946).

SEYERLE, W., "Die Stroemung in Kanaelen mit seitlichem Einlass," *Diss. T. H. Stuttgart* (1926).

SHEPHERD, F. H., "Correct Exhaust Piping Improves Engine Performance," *Power*, **75**, 401 (1932).

SHOEMAKER, F. G., "Automotive Two-Cycle Diesel Engines," *SAE Journal*, **43**, 485–95 (December 1938).

SMITH, P. H., "Kadenacy Tests," *Gas and Oil Power* (March 1945).

SOERENSEN, E., "Stroemungsfragen am Dieselmotor," *VDI Zeitschr.*, **84**, 877 (1944).

SPANOGLE, J. A. AND E. G. WHITNEY, "A Description and Test Results of a Spark-Ignition and Compression-Ignition Two-Stroke Cycle Engine," *NACA Tech. Report 495* (1934).

STIER, E., "Spuelung und Aufladung bei Zweitaktmotoren," *VDI Zeitschr.*, **73**, 1389 (1929).

"The Scavenging of Two-Cycle Engines," *Sulzer Tech. Review*, **4** (1933).

"The Supercharging of Two-Stroke Cycle Diesel Engines," *Sulzer Tech. Review*, Spec. No. (December 1941).

SWISSTUNOFF, J. W. AND A. A. KURITZ, "Konstruktive Entwicklung eines Zweitakt-Dieselmotors fuer Verkehrszwecke," *Dieselestrojenije*, **4** (1939), and *M. T. Z.*, **3**, 47–53 (1941).

TAENZLER, U., "Spuelgeblaese fuer Zweitaktmotoren," *VDI Zeitschr.*, **82**, 1153 (1938), and **83**, 818 (1939).

—— "Triebwerke fuer Zweitaktmotoren mit Gleichstromspuelung und ihre Eignung fuer schnellaufende Hoechstleistungsmotoren," *A. T. Z.*, **47**, 47 (1944).

TAYLOR, C. F., "Effect of Engine Exhaust Pressure on Performance of Compressor-Engine-Turbine Units," *SAE Journal*, **54** (1946).

—— "An Analysis of the Charging Process in the Two-Stroke Engine," Paper presented at *SAE Summer Meeting* at White Sulphur Springs, W. Va., June 9–14, 1940.

TAYLOR, C. F. AND E. S. TAYLOR, *The Internal Combustion Engines*, International Textbook Company, 1938.

VENEDIGER, H. J., "Steigerung der Leistung und Wirtschaftlichkeit bei Zweitakt-Vergaser-Fahrzeugmotoren," *Diss. T. H. Muenchen* (1931), and *Forsch. H. z. Autotechnik* (1931).

—— "Wertung der Spuelung von Zweitaktmotoren," *A. T. Z.*, **36**, No. 12, 301 (1933), Trans. "Evaluation of Scavenging in Two-Stroke-Cycle Engines," *NACA Tech. Mem. 732* (1933).

VENEDIGER, H. J., "Untersuchungen an schnellaufenden Auflade-Drehkolbenverdichtern," *A. T. Z.*, **36**, 579, 619 (1933), and **41**, 334 (1938).

—— "Planung und Aufbau Schnellaufender Zweitaktmotoren," *A. T. Z.*, **37**, 495 (October 10, 1934), and 529 (October 25, 1934).

—— "Grundsaetzliche Betrachtungen zum Zweitaktverfahren," *A. T. Z.*, **39**, 273 (1936).

—— "Stand und Entwicklung der Zweitaktmotoren fuer Kraftwagen," *VDI Zeitschr.*, **81**, 187–99 (1937).

—— "Zweitaktmotoren," *Automobiltechn. Handbuch*, 14th edition, 990, Berlin, M. Krayn, 1941.

—— *Zweitaktspuelung insbesondere Umkehrspuelung*, Stuttgart, Franck'sch Verlagshandlung, 1947.

VOISZEL, P., "Resonanzerscheinungen in der Saugleitung von Kompressoren und Gasmotoren," *VDI Forschungsheft*, **106** (1911).

WAGENER, A., "Ueber die Vorgaenge des Ausspuelens und Ladens bei Zweitaktgasmotoren," *Gasmotortechnik Zeitschr.* (1905).

WALDRON, C. D., "Flow Coefficients of Monosleeve Valves," *NACA Tech. Report 717* (1941).

WARE, M. AND E. E. WILSON, "The Comparative Performance of Roots Type Aircraft Engine Supercharger," *NACA Tech. Report 284* (1928).

WARMING, T., "Polar Diagram for Tuning Exhaust Pipes," *ASME Trans.*, **68**, 31–3 (January 1946).

WHITEFIELD, K. C., "Improving Diesel Engine Operation by the Selection of Proper Exhaust Pipe Length," *Thesis, University of Wisconsin* (1933); Abstract, *Power*, 77, 695 (December 1933).

WHITFIELD, J. E., "Hamilton-Whitfield Blower," *SAE Journal* (1943).

WILLE, R., "Neuere Untersuchungen ueber die Spuelstroemung bei schlitzgesteuerten Zweitaktverbrennungskraftmaschinen," *A. T. Z.*, **44** (1941).

—— "Ueber quantitative Auswertung von Zeitlupenaufnahmen des Spuelstroemungvorganges," *Dtsch. Kraftfahrtforsch., Zwischenbericht 103* (1941).

WILSON, W. A. AND J. CROCKER, "Fundamentals of the Elliott Lysholm Compressor," *ASME Paper 45-A-45*, Annual Meeting (November 1945).

WINTTERLIN, K., "Beitraege zur Erforschung der Ausspuelung des Zylinders von Zweitaktmotoren mit Schlitzspuelung," *Diss. T. H. Stuttgart* (1926).

—— "Spuelung und Leistung bei Zweitaktmotoren," *VDI Zeitschr.*, **75**, 165 (1931).

WISNER, R., "High Speed Two-Stroke Compression-Ignition Engines," *The Institution of Automob. Eng.* (June 1944).

ZEMAN, J., *Zweitakt Dieselmaschinen*, Wien, J. Springer, 1935.

—— "Baugrenzen von Zweitakt Dieselmaschinen mit Kurbelkasten-Spuelpumpe," *VDI Dieselmaschinen*, **VI**, 142 (1936).

—— "Zweitaktmaschinen mit unsymmetrischen Steuerdiagrammen," *A. T. Z.*, **41**, 420 (1938).

—— "Grundlagen fuer die Berechnung von Zweitaktmaschinen," *M. T. Z.*, **3** (1941).

—— "Durchfuehrung der Berechnung von Zweitaktmaschinen," *M. T. Z.*, **3** (1941).

—— "Die neuere Entwicklung der Zweitaktmotoren," *VDI Zeitschr.*, **87** (1943), Trans. "Recent Development of the Two-Stroke Engine," *NACA Tech. Mem. 1082* (1945).

ABBREVIATIONS

American Standard Association's abbreviations have been used. In addition, the following:

top (dead) center — T. C.
bottom (dead) center — B. C.
after top center — A. T. C.
before top center — B. T. C.
after bottom center — A. B. C.
before bottom center — B. B. C.
crank angle — c. a.
Saybold second universal — SSU

ORGANIZATIONS

American Society of Mechanical Engineers — ASME
National Advisory Committee for Aeronautics — NACA
Society of Automotive Engineers — SAE
The Institution of Mechanical Engineers — Inst. of Mech. Engineers
Maschinenfabrik Augsburg Nuerenberg — M. A. N.
Société des Ingénieurs Automotive — Soc. Ing. Aut.
Verein Deutscher Ingenieure — VDI

PERIODICALS

ASME Transactions — ASME Trans.
Automobile Engineer — Automob. Eng.
Automobiltechnische Zeitschrift — A. T. Z.
Automotive and Aviation Industries — Autom. Ind.
Automotive Industries — Autom. Ind.
Chaleur et Industrie — Chal. et Ind.
Deutsche Kraftfahrtforschung — Dtsch. Kraftfahrtforsch.
Forschungsarbeiten auf dem Gebiete des Ingenieurwesens — Forsch. Ing. Wes.
Journal Société des Ingénieurs Automotive — J. Soc. Ing. Automot.
Kraftfahrtechnische Forschungsarbeiten — Kraftfahrtechn. Forsch.
Mechanical Engineering — Mech. Engrg.
Mitteilungen aus den technischen Instituten der Staatlichen Tung Chi. — Universitaet, Woosung, China — Mitt. tech. Inst. Woosung, China.
Motortechnische Zeitschrift — M. T. Z.
NACA Technical Memorandum — NACA Tech. Mem.
NACA Technical Note — NACA Tech. Note.
NACA Technical Report — NACA Tech. Report.
Transactions North East Coast Institute of Engineers — Trans. N. E. Coast Inst. Engrs.
VDI Zeitschrift — VDI Zeitschr.

INDEX

TABLE 18-I. SC

		1	2	3	4	5	6	7	8	9	10	
1	Engine make and type	Sulzer 8SD72 Dorset	Stork HODT 7 × 72/110 D.A.	Nordberg TS & TSG-29	Nordberg TS 21-31	National-Supply	Fairbanks-Morse 32 E 14	Fairbanks-Morse 33 E 14	Atlas Polar KM	Cooper Bessemer G.M.V.	Sulzer 6TS29	Se
2	Bore and stroke (in.)	28.35 × 49.20	28.375 × 43.462	29 × 40	21 × 31	12 × 15	14 × 17	14 × 17	12.37 × 22.44	14 × 14	11.41 × 19.68	10.
3	Number of cylinders	8	7	5 to 12	5 to 9		2, 3, 4, 5, 6	5, 6, 7	4, 8, 6	4, 6, 8, 10	6	1, 2
4	Rated rpm	125	130	164	225	250	300	300	300	300	300	425
5	Displacement volume per cyl. (cu in.)	30,800	27,240 (upper) 22,400 (lower)	26,400	10,750	1,696.5	2,617	2,617	2,653	2,145	2,012	1,0
6	Rated bhp per cyl. (hp)	670	1,150	775	429	44.5	75	115	210	100	98.7	30
7	Rated bmep, (psi)	68.8	71	71	70.5	41.6	37.8	58	86.6	61.5	64.7	33.
8	Type or scavenging	CRO, AUI	LOO, CEN, SYM	LOO, AUI, CEN	LOO, (Curtis) AUI, CEN	CRA, CRO, SYM	CRA, CRO SYM	CRO, SYM, REC	LOO, CEN	CRO, REC, SYM	CRO, AUI	CR
9	Blower delivery (cfm free air per cyl.)	3371	5,985	3,400	1,850	233	363	607	812		455.5	179
10	Delivery ratio	1.51	1.6	1.356	1.325	0.95	0.80	1.335	1.48		1.3	0.7
11	Specific air consumption (cfm per hp)	5.03	5.15	4.39	4.32	5.23	4.85	5.28	3.9		4.62	5.97
12	Scavenging pressure at rated speed (psig)	2.27	1.75	2.5	2.35	...		1.3	2.1		4.15	
13	Scavenging air temp. at rated speed (F)	104	20 above amb.	100	100	...	150	100	120		135	
14	Connecting rod — crank ratio	4	4	4	3.94	5.333	5.02	5.97	4.74		4.5	4.35
15	Crank (or cam) advance for exhaust (deg)	...	...	...	...	...	...	...	...		...	
16	Exhaust opens (deg ATC)	124°	113.75° (upper) 125.5° (lower)	115.9°	116°	126°	112°	113°	116.13°	"A" bank 117° "B" bank 119°	136°	117
17	Exhaust closes (deg ATC)	236°	246.25° (upper) 234.5° (lower)	244.1°	244°	234°	248°	247°	243.87°	"A" bank245.5° ,"B" bank225.5°	224°	243
18	Exhaust duration (sec)	0.149	0.17 (upper) 0.14 (lower)	0.130	0.0948	0.0720	0.0758	0.0745	0.071		0.049	0.04
19	Exhaust port height (in.)	7.96		9.25	7.25	2.625	4.54	4.52		3.875	1.65	2.75
20	Exhaust port height (% stroke)	16.15	24.6 (upper) 25.0 (lower)	23.1	23.4	17.5	26.7	26.6		38.8	8.4	22.9
21	Exhaust port width, total (in.)	27.22		19.2	14.625	7.56	10.1	9.9	13.01	10.625	11.41	6.5
22	Exhaust port width, total (% circumference)	30.2		21.1	22.2	20.1	22.96	22.5	34.5	24	31.9	19.7
23	Inlet opens (deg ATC)	109°	127.25° (upper) 137° (lower)	144°	141°	140.4°	122°	124°	116.13°	"A" bank 134° "B" bank 137°	120°	132
24	Inlet closes (deg ATC)	251°	232.75 (upper) 223 (lower)	244.1°	244°	219.6°	237°	236°	243.87°	"A" bank229.5° "B" bank 237°	240°	228
25	Inlet duration (sec)	0.191	0.135 (upper) 0.11 (lower)	0.102	0.0763	0.0527	0.0633	0.0623	0.071		0.067	0.03
26	Inlet port height (in.)	7.12 (upper) +2.92 (lower)		8.13	7.25	1.4375	3.29	3.27		2.625	2.36 (upper) +1.38 (lower)	1.62
27	Inlet port height (% stroke)	14.5 (upper) +6 (lower)		20.3	23.4	9.6	19.4	19.2		18	12 (upper) +7 (lower)	13.5
28	Inlet port width, total (in.)	33.1 (upper) +13.88 (lower)		28.6	21.5	8.13	19.3	20.3	8.75	18.75	300 (upper) 204 (lower)	7.25
29	Inlet port width, total (% circumference)	37.2 (upper) +17.8 (lower)		31.4	32.6	21.6	43.9	47.3	23.3	42.5	33 (upper) 22.4 (lower)	21.9
30	Exhaust, maximum port (valve) area (sq in.)	170.5		153.8	98.3	18.88	26.3	44.7		41.1	18.45	17.9
31	Exhaust, mean port (valve) area (sq in.)	114		132.2	65.2	13.23	17.5	16.9			14.43	11.8
32	Exhaust, time-area (sq in.-deg)	12,780		16,900	8,350	1,430	2,380	2,265			1,270	1,48
33	Inlet, maximum port area (sq in.)	114.4 (upper) +30.22 (lower)		182	104.2	11.68	36.5	66.4		49.3	13.34 (upper) +3.25 (lower)	11.8
34	Inlet, mean port area (sq in.)	89.2 (upper) +22.17 (lower)		138.5	76.5	7.28	24.3	25.3			11.94 (upper) +2.45 (lower)	7.87
35	Inlet, time-area (sq in.-deg)	12650 (upper) +3150 (lower)		13,860	7,880	577	2,770	2,834			1445 (upper) +297 (lower)	755
36	Exhaust, mean cubic rate of flow (cu in. per sec)	314,000		275,000	150,100	22,360	27,600	46,900			53,400	14,7
37	Exhaust, mean port velocity (ft per sec)	229		173.5	192.5	141	131.2	231			308	104.
38	Inlet, mean cubic rate of flow (cu in. per sec)	245,000		352,000	186,700	30,500	33,100	56,100			39,100	19,41
39	Inlet mean port velocity (ft per sec)	202		210.5	204	350	113.5	184.5			226.5	205.
40	Mean port velocity ratio, exhaust/inlet	1.135		0.83	0.943	0.403	1.16	1.25			1.36	0.507
41	Exhaust lead (deg crankangle)	15°	13.5° (upper) 11.5° (lower)	28.1°	25°	25°	11°	11°	0°		16°	15°
42	Blowdown time-area (sq in.-deg)			1.840	800	74	39.8	38.6				61.8
43	Supercharge period (deg crankangle)	...	...	...	...	...	...	...				
44	Supercharge time-area (sq in.-deg)		...	...	180	...	...	...			...	
45	Scavenge factor	0.41		0.41	0.40			0.475			0.345	
46	Ideal relative charge (%)	87.5	76.5	72.6	72.2	82.5	73.3	73.5	71.5		91	77.1
47	Calculated scavenging efficiency (%)	70	84	78	78	53	48	66	80		63	42
48	Calculated bmep (psi)	77.6 ($f = 0.4$)	77 ($f = 0.4$)	69.5 ($f = 0.4$)	69 ($f = 0.4$)	47.6 ($f = 0.4$)	43.7 ($f = 0.45$)	55.5 ($f = 0.45$)	69.5 ($f = 0.4$)		75 ($f = 0.4$)	37.1 (

18-I. SCAVENGING DATA OF TWO-STROKE CYCLE DIESEL ENGINES

	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
r	Sulzer 6TS29	Venn-Severin M	National-Supply	Atlas Polar KE	Petter SS2-6 Series V, Mark II	Fairbanks-Morse 38D8-⅛	G.M. 278A	G.M. Model 567	Sulzer 2ZG9	Junkers Opposed Piston	Steel Pro. Eng. Co. Slant Eng.	Harnisch-feger Model 387-C	Ricardo Sleeve Valve, Exp.	Clark-Daco Horiz. Exp.	Junk Jumo
	11.41 × 19.68	10.5 × 12	6.5 × 8	7.09 × 11.81	8.5 × 13	8.125 × (10 + 10)	8.75 × 10.5	8.5 × 10	3.54 × (4.72 + 4.72)	2.36 × (3.54 + 4.72)	6 × (8 + 8)	4.5 × 5.5	5.5 × 7	4.125 × (5.25 + 5.25)	4.72 × (8.27 +
	6	1, 2, 3, 4		2–8, 6	2, 3, 4, 5, 6	6–10	6, 8, 12, 16	6, 8, 12, 16	2		9	3	1	1	6
	300	425	500	600	600	720	750	800	1,000	1,005	1,200	1,200	1,370	1,500	1,700
	2,012	1,040	265.1	465.8	735	1,037	631	567	92.7	36	452.32	87.5	167	140.6	290
	98.7	30	13	57	96	160	100	100	19.7	8	111	20	50.5	41.7	120
	64.7	33.6	33.8	81.1	85.9	84.9	83.8	87.3	84.5	87	81	75.5	87.5	78.5	96.5
),	CRO, AUI	CRA, SYM	CRA, SYM	LOO, CEN	UNI, EPV, ROO	UNI, OP, ROO	UNI, EPV, ROO	UNI, EPV, ROO	UNI, REC, HORIZ, OP	UNI, OP, REC	UNI, OP, CEN	UNI, EPV, ROO	UNI, SV	UNI, HORIZ, OP, ROO	UNI, CE
	455.5	179	72.8	244	358	600	352	420	69.9	35.5	465	88	108.5 (?)	179	370
	1.3	0.7	0.95	1.51	1.4	1.387	1.285	1.6	1.3	1.69	1.47	1.45	0.82	1.465	1.3
	4.62	5.97	5.6	4.3	3.73	3.75	3.52	4.2	3.55	4.43	4.19	4.4	2.15	4.3	3.08
	4.15			2.6	2.75	3.2	3.0	3.93	4.26	5.93	2.06	5.5	6.0	4.0	4.0
	135			150	100	150 (est.)	140 at 70 F amb.	35 above amb.	140	160	125	160	200 (est.)	136.2	150 (est
	4.5	4.35	4	4.75	4.62	5.4 (lower)	4.14	4.4	...	4, 8.5		4.32		...	3.21 exh 3.35 inl.
	...	...	...	...	...	12	16.25	12	11	15	10	17.5	35	...	9
7° 9°	136°	117°	122.4°	116.63°	106°	124°	101°	96°	124°	126°	107°	91°	105°	111°	104.5°
.5° .5°	224°	243°	237.6°	243.37°	224°	236°	226.5°	240°	226°	234°	253°	234°	235°	235°	255.25°
	0.049	0.0493	0.0382	0.035	0.0328	0.0259	0.0278	0.03	0.017	0.0179	0.0203	0.0198	0.0158	0.0138	0.0138
	1.65	2.75	1.5		0.663 valve lift (2 valves)	1.81	0.671 valve lift	0.686 valve lift	0.55	...	2.830	0.518 valve lift		0.969	1.9
	8.4	22.9	18.8			18.1	34.6 equiv.	44.2 equiv.	11.7	...	35.4	48.5 equiv.		18.4	30.5
	11.41	6.5	5	6.66	3.25 valve dia.	12.9	4 valves — 2.77 dia.	4 valves — 2.5 dia.	7.26	...	10.60	One 2.5 dia. valve		8	8.9
	31.9	19.7	24.5	30		50.6	...	...	65	...	56.3			61.6	60
4° 7°	120°	132°	137.8°	116.63°	130°	140°	134°	135°	141°	144°	125°	134°	133°	136°	122.25°
5° 7°	240°	228°	222.2°	243.37°	230°	244°	226°	225°	291°	246°	255°	228°	235°	241°	255.25°
	0.067	0.0375	0.0281	0.035	0.0278	0.0241	0.0204	0.019	0.015	0.0169	0.018	0.0131	0.0124	0.0117	0.0130
	2.36 (upper) +1.38 (lower)	1.625	0.8125		1.875	1.5	1.281	1.312	0.51	...	2.312	0.750		0.936	2.05
	12 (upper) +7 (lower)	13.5	10.2		14.42	15	12.3	13.12	10.8	...	29	13.6		17.8	24.8
	300 (upper) 204 (lower)	7.25	6.46	3.94	14.25	17.8	15	13.75	8	...	10.5	7.5		8	8.9
	33 (upper) 22.4 (lower)	21.9	31.7	17.8	53.4	69.7	54.7	51.5	72	...	55.6	53.2		61.6	60
	18.45	17.9	7.3		13.52	23.4	23.3	17.64	3.72	1.473	28.9	2.59	5.1	7.73	16.9
	14.43	11.8	4.88		7.8	15.6	8.7	8.2	2.898	1.25	18.4	1.42	3.0	5.46	13.12
	1,270	1,487	561		921.5	1,747	1,090	1,180	296.	135	2,790	251.4	390	677	1977
	13.34 (upper) +3.25 (lower)	11.8	5.25		25.45	26.7	18.87	17.9	3.10	1.893	21.6	7.07	7.3	7.48	18.25
	11.94 (upper) +2.45 (lower)	7.87	3.46		12.3	17.8	12.9	10.6	2.217	1.323	14.3	3.81	4.2	4.55	12.64
	1445 (upper) +297 (lower)	755	292		1,230	1,851	1,188	954.5	199	135	1,840	351.4	428	477.5	1680
	53,400	14,770	6,590		31,370	55,600	29,150	30,250	7,080	3,400	32,700	6,410	8,650	4,950	25,470
	308	104.2	112.5		335	297	279.0	307.5	203	226	148	376	240	247	160.5
	39,100	19,410	8,950		37,020	59,700	39,750	47,750	8,020	3,600	36,950	9,700	11,000	17,600	29,000
	226.5	205.5	215		250.5	279.5	256.5	375.0	301	226	215	212	218	323	190.5
	1.36	0.507	0.523		1.337	1.06	1.09	0.82	0.675	1	1.44	1.77	1.1	0.77	0.843
	16°	15°	15.4°	0°	24°	16°	33°	39°	17°	18°	18°	43°	28°	25°	...
		61.8	28		81	93.6	148		28	14.6	110	41	86.5	89.4	89
	...	...	...		6°	8°	0.5°	...	6°	13°	2°	6°	...	6°	...
		...	...		276	39.2			4.4	29.5		165.4	175	32.4	1.0
	0.345				0.453	0.49	0.461	0.59	0.456	0.29	0.47	0.282	0.29	0.498	0.298
	91	77.1	81.2	75.3	94	87.5	92	94	99.8	97.6	72	96	91.5	86	77.8
	63	42	54	82	92	90	90	92	90	92	91	89	78	92	89
	75 ($f = 0.4$)	37.1 ($f = 0.5$)	47.6 ($f = 0.5$)	74.5 ($f = 0.4$)	100.2 ($f = 0.4$)	92 ($f = 0.4$)	86.2 ($f = 0.45$)	88.2 ($f = 0.45$)	85.8 ($f = 0.5$)	94 ($f = 0.45$)	76.25 ($f = 0.4$)	94 ($f = 0.45$)	88 ($f = 0.4$)	81.5 ($f = 0.45$)	81.5 ($f = 0$

DIESEL ENGINES

18	19	20	21	22	23	24	25	26	27	28	29	30	31	
Sulzer 2ZG9	Junkers Opposed Piston	Steel Pro. Eng. Co. Slant Eng.	Harnisch-feger Model 387-C	Ricardo Sleeve Valve, Exp.	Clark-Daco Horiz. Exp.	Junkers Jumo IV	Atlantic Sleeve Valve, Exp.	N.A.C.A. Poppet Exh. Valve, Exp.	M.I.T. Curtis Scav, Exp.	Martin Sleeve Valve, Exp.	G.M. Model 71	Kloeckner-Humboldt-Deutz-DZ 700	Kloeckner-Humboldt-Deutz-DZ 710	1
3.54 × (4.72 + 4.72)	2.36 × (3.54 + 4.72)	6 × (8 + 8)	4.5 × 5.5	5.5 × 7	4.125 × (5.25 + 5.25)	4.72 × (8.27 + 8.27)	6 × 7	4.625 × 7	4.5 × 6	5.25 × 7	4.25 × 5	3.145 × 3.93	6.299 × 6.299	2
2		9	3	1	1	6	1	1	1	3	2, 4, 6	8	16	3
1,000	1,005	1,200	1,200	1,370	1,500	1,700	1,700	1,800	1,800	1,800	2,000	2,500	2,670	4
92.7	36	452.32	87.5	167	140.6	290	197.5	117.5	95.5	151.6	70.8	30.5	196.2	5
19.7	8	111	20	50.5	41.7	120	60	73	24	75	20	12.5	110	6
84.5	87	81	75.5	87.5	78.5	96.5	71	137	55.3	109	56	65	83.2	7
UNI, REC, HORIZ, OP	UNI, OP, REC	UNI, OP, CEN	UNI, EPV, ROO	UNI, SV	UNI, HORIZ, OP, ROO	UNI, OP, CEN	UNI, SV, CEN	UNI, EPV	LOO, SYM	UNI, SV, CEN	UNI, EPV, ROO	LOO, SYM, ROO	LOO, CEN, SYM	8
69.9	35.5	465	88	108.5 (?)	179	370	272	147	139.3	241	107	73.8	481	9
1.3	1.69	1.47	1.45	0.82	1.465	1.3	1.4	1.2	1.4	1.525	1.31	1.67	1.6	10
3.55	4.43	4.19	4.4	2.15	4.3	3.08	4.54	2.02	5.8	3.21	5.37	5.88	4.37	11
4.26	5.93	2.06	5.5	6.0	4.0	4.0	4.33	7.37	3.83	6.5	6.5	2.96	5.9	12
140	160	125	160	200 (est.)	136.2	150 (est.)	150	200 (est.)	158	125	200	125 (est.)	150 (est.)	13
...	4, 8.5		4.32		...	3.21 exh., 3.35 inl.	3.86	...	4.16	4.0	4.0	4.5 (master rod)		14
11	15	10	17.5	35	...	9	15	...	...	14	18	...	...	15
124°	126°	107°	91°	105°	111°	104.5°	104°	95°	104°	105°	94.5°	106°	92°	16
226°	234°	253°	234°	235°	235°	255.25°	226°	228°	256°	227°	229.5°	254°	268°	17
0.017	0.0179	0.0203	0.0198	0.0158	0.0138	0.0138	0.012	0.0123	0.014	0.0113	0.0112	0.0099	0.011	18
0.55	...	2.830	0.518 valve lift		0.969	1.9	0.59	...	...	0.656	0.375 valve lift	1.22	2.52	19
11.7	...	35.4	48.5 equiv.		18.4	30.5	8.44	...	...	9.4	40 equiv.	31	42	20
7.26	...	10.60	One 2.5 dia. valve		8	8.9	15	...	...	12	2 valves 1.25 dia.	1.98	3.96	21
65	...	56.3			61.6	60	79.5	...	...	73	...	20	20	22
141°	144°	125°	134°	133°	136°	122.25°	135°	130°	128°	136°	132°	124°	104°	23
291°	246°	255°	228°	235°	241°	255.25°	244.5°	230°	232°	246.5°	228°	236°	256°	24
0.015	0.0169	0.018	0.0131	0.0124	0.0117	0.0130	0.0107	0.0093	0.0096	0.0102	0.008	0.00747	0.00955	25
0.51	...	2.312	0.750		0.936	2.05	0.7	1	...	0.66	2 rows 0.3125 dia.	0.717	2.02	26
10.8	...	29	13.6		17.8	24.8	10	14.3	...	9.43		18.2	32	27
8	...	10.5	7.5		8	8.9	14	5.8	...	12		3.96	7.92	28
72	...	55.6	53.2		61.6	60	74.5	40	...	72.7		40	40	29
3.72	1.473	28.9	2.59	5.1	7.73	16.9	8.85	7.4	6.3	7.9	2.62	2.4	10	30
2.898	1.25	18.4	1.42	3.0	5.46	13.12	6.0	4.16	4.18	5.15	1.43	1.58	7.0	31
296.	135	2,790	251.4	390	677	1977	732	552	635	628	193	234	1,230	32
3.10	1.893	21.6	7.07	7.3	7.48	18.25	9.8	5.8	4.4	7.9	4.9	2.84	16	33
2.217	1.323	14.3	3.81	4.2	4.55	12.64	6.6	3.88	2.96	5.17	3.31	1.86	10.95	34
199	135	1,840	351.4	428	477.5	1680	723	388	308	571	317.4	210	1,650	35
7,080	3,400	32,700	6,410	8,650	4,950	25,470	23,100	11,450	9,550	20,900	8,220	5,150	28,550	36
203	226	148	376	240	247	160.5	321	229.4	190	334	482	272	350	37
3,020	3,600	36,950	9,700	11,000	17,600	29,000	25,700	15,200	13,900	22,600	11,620	6,820	32,900	38
301	226	215	212	218	323	190.5	324	327	392	364	291	306	250	39
.675	1	1.44	1.77	1.1	0.77	0.843	0.99	0.703	0.485	0.93	1.65	0.89	1.4	40
17°	18°	18°	43°	28°	25°	...	31°	35°	24°	31°	37.5°	18°	12°	41
28	14.6	110	41	86.5	89.4	89	118	60	41	99	26.2	9.3		42
°	13°	2°	6°	...	6°	...	8.5°	2°	...	9.5°	...	...	...	43
.4	29.5		165.4	175	32.4	1.0	222	61	...	165	158.4	...	...	44
.456	0.29	0.47	0.282	0.29	0.498	0.298	0.485	0.4	0.62	0.44	0.372	0.545	0.321	45
9.8	97.6	72	96	91.5	86	77.8	84	101.2	72	95.4	98	74	69.4	46
0	92	91	89	78	92	89	90	86	78	92	88	82	90	47
5.8 ($f = 0.5$)	94 ($f = 0.45$)	76.25 ($f = 0.4$)	94 ($f = 0.45$)	88 ($f = 0.4$)	81.5 ($f = 0.45$)	81.5 ($f = 0.4$)	79 ($f = 0.45$)	103.5 ($f = 0.4$)	55.2 ($f = 0.5$)	102 ($f = 0.4$)	81.5 ($f = 0.5$)	65 ($f = 0.45$)	73 ($f = 0.4$)	48

NOMENCLATURE

SYMBOL	CONCEPT	UNITS USUALLY USED
A	reciprocal of mechanical equivalent of heat	Btu per ft-lb
A	area in general	sq in.
$A_{em} = \frac{1}{\alpha_e}\int A_e d\alpha$	mean exhaust area	sq in.
A_h	total air consumption	lb per hr
A_i	the uncovered inlet-port area at any one instant	sq in.
$A_{im} = \frac{1}{\alpha_i}\int A_i d\alpha$	mean inlet area (normal to the direction of flow)	sq in.
$A_m = \frac{1}{\alpha}\int A d\alpha$	mean blowdown area	sq in.
a	velocity of sound in an exhaust column	in. per sec, ft per sec
B	blowdown time-area	sq in.-deg
b_e	width of (rectangular or rhomboid) exhaust ports	in.
b_i	width of (rectangular or rhomboid) inlet ports	in.
bhp	brake horse power	hp
bmep	brake mean effective pressure	psi
gross bmep	brake mean effective pressure when power to drive scavenging blower is furnished separately	psi
net bmep	brake mean effective pressure calculated by deducting from the gross bmep the blower mep	psi
blower mep	engine bmep that corresponds to hp input of blower	psi
C	discharge coefficient	dimensionless
c_c	coefficient of contraction	dimensionless
c_p	specific heat at constant pressure	Btu per lb per deg F
c_v	specific heat at constant volume	Btu per lb per deg F
c_v	coefficient of velocity	dimensionless
$C_{rel} = \frac{V_{ch}}{V_{disp}}$	relative cylinder charge	dimensionless
D	cylinder diameter (bore)	in.
F	cross sectional area of pipe	sq in.
$F_a = h_m\alpha$	blowdown area	per cent degree
F_h	total fuel consumption	lb per hr
$F_i = h_{im}\alpha_i$	inlet area	per cent degree
F_0	area of an orifice	sq in.
f	specific fuel consumption	lb per bhp-hr
G	weight of air per second passing through an opening	lb per sec
G_r	weight of air at NTP in receiver	lb
G_{del}	weight of air delivered by blower during one engine revolution	lb
g	acceleration of gravity	ft per sec per sec
h	height of ports	in.
h_e	relative height of rectangular or rhomboid exhaust ports in relation to stroke	in. per in.
h_i	relative height of rectangular or rhomboid inlet ports in relation to stroke	in. per in.
h_{im}	mean relative height of rectangular or rhomboid inlet ports	in. per in.
h_m	relative blowdown port height	in. per in.
k	ratio of specific heats c_p/c_v	dimensionless
L	length of pipe	in.
$L = \frac{V_{del}}{V_{disp}}$	delivery ratio	dimensionless
mep	mean effective pressure	psi
mip	mean indicated pressure	psi
n	number of crank revolutions per minute	min^{-1}
	number of power strokes per minute	min^{-1}
NTP	normal temperature pressure conditions (60 F, 14.7 psia)	
$P = \frac{V_{pure}}{V_{disp}}$	pure air ratio	dimensionless
p_0	ambient pressure normally used in the calculations, 14.5 psia	psia
p_1	initial absolute pressure of air	psia
p_2	final absolute pressure of air	psia
p_{av}	average pressure in receiver, gage	psig
p_b	blower mep	psi
p_e	absolute pressure in cylinder at the instant the exhaust port opens (expansion end pressure)	psia
p_i	absolute pressure in cylinder at the instant the inlet port opens (blowdown pressure)	psia

NOMENCLATURE

SYMBOL	CONCEPT	UNITS USUALLY USED
p_m	absolute pressure in the contracted section	psia
p_{max}	maximum absolute pressure in receiver	psia
p_{min}	minimum absolute pressure in receiver	psia
p_{sc}	scavenge pressure, gage	psig
R	gas constant for air	ft per deg F
R_e	gas constant for combustion gases	ft per deg F
$R_s = L\frac{\rho NTP}{\rho_i}$	scavenge ratio	dimensionless
$r = \frac{W_{pure\ act}}{W_{fuel}}$	actual air-fuel ratio	lb per lb
$r_{th} = \frac{W_{pure\ th}}{W_{fuel}}$	theoretical (chemically correct) air-fuel ratio	lb per lb
S	supercharge time-area	sq in.-deg
S	scavenge factor	dimensionless
s	length of stroke of engine	in.
T_1	absolute temperature in container 1	deg R
T_2	absolute temperature in container 2	deg R
T_e, t_e	temperature of combustion gases at instant exhaust port opens	deg R, deg F
T_m	absolute temperature in the contracted section	deg R
t	time	sec
t_{sc}	temperature of scavenge air	deg F
V_{ch}	amount of cylinder charge expressed in volume NTP	cu in.
V_{comb}	amount of combustion gas in the cylinder expressed in volume NTP	cu in.
V_{cp}	amount of residual combustion products in the cylinder expressed in volume NTP	cu in.
V_{del}	amount of air delivered to the cylinder per cycle expressed in volume NTP	cu in.
V_{disp}	displacement volume	cu in.
V_e	cylinder volume at instant exhaust port opens (expansion end volume)	cu in.
V_p	volume of the exhaust port	cu in.
V_{pure}	amount of pure air in cylinder during compression expressed in volume NTP	cu in.
V_r	volume of receiver	cu in.
V_{res}	amount of residual gas in the cylinder expressed in volume NTP	cu in.
V_{ret}	amount of fresh air retained in the cylinder expressed in volume NTP	cu in.
V_{short}	amount of air short-circuited per cycle expressed in volume NTP	cu in.
V_{th}	amount of air theoretically required per cycle for combustion expressed in volume NTP	cu in.
V_{tot}	total cylinder volume	cu in.
v_m	specific volume in the contracted cross section	cu in. per lb
w_i	mean inlet velocity	ft per sec
z_0	period of exhaust column vibration	sec
α	crank angle	deg
α	exhaust lead, crank angle	deg
α_1	instant of inlet port opening, crank angle after top center	deg
α_2	instant of inlet port closing, crank angle after top center	deg
α_e	exhaust duration, crank angle	deg
$\alpha_i = \alpha_2 - \alpha_1$	inlet duration, crank angle	deg
$\beta = \frac{b_i}{D \times \pi}$	relative width of rectangular or rhomboid inlet port	in. per in.
η_b	blower efficiency	dimensionless
$\eta_{ch} = \frac{V_{ret}}{V_{disp}}$	charging efficiency	dimensionless
$\eta_p = \frac{V_{pure}}{V_{ch}}$	purity	dimensionless
$\eta_{sc} = \frac{V_{ret}}{V_{ch}}$	scavenging efficiency	dimensionless
$\eta_{tr} = \frac{V_{ret}}{V_{del}}$	trapping efficiency	dimensionless
$\lambda = \frac{V_{pure}}{V_{th}}$	excess air factor	dimensionless
$\lambda' = \frac{V_{ret}}{V_{th}}$	alternative definition of excess air factor	dimensionless
μ	flow coefficient	dimensionless
ρ	density	lb per cu ft
τ	time	sec
ψ	Nusselt coefficient	dimensionless
ψ_e	discharge coefficient	dimensionless