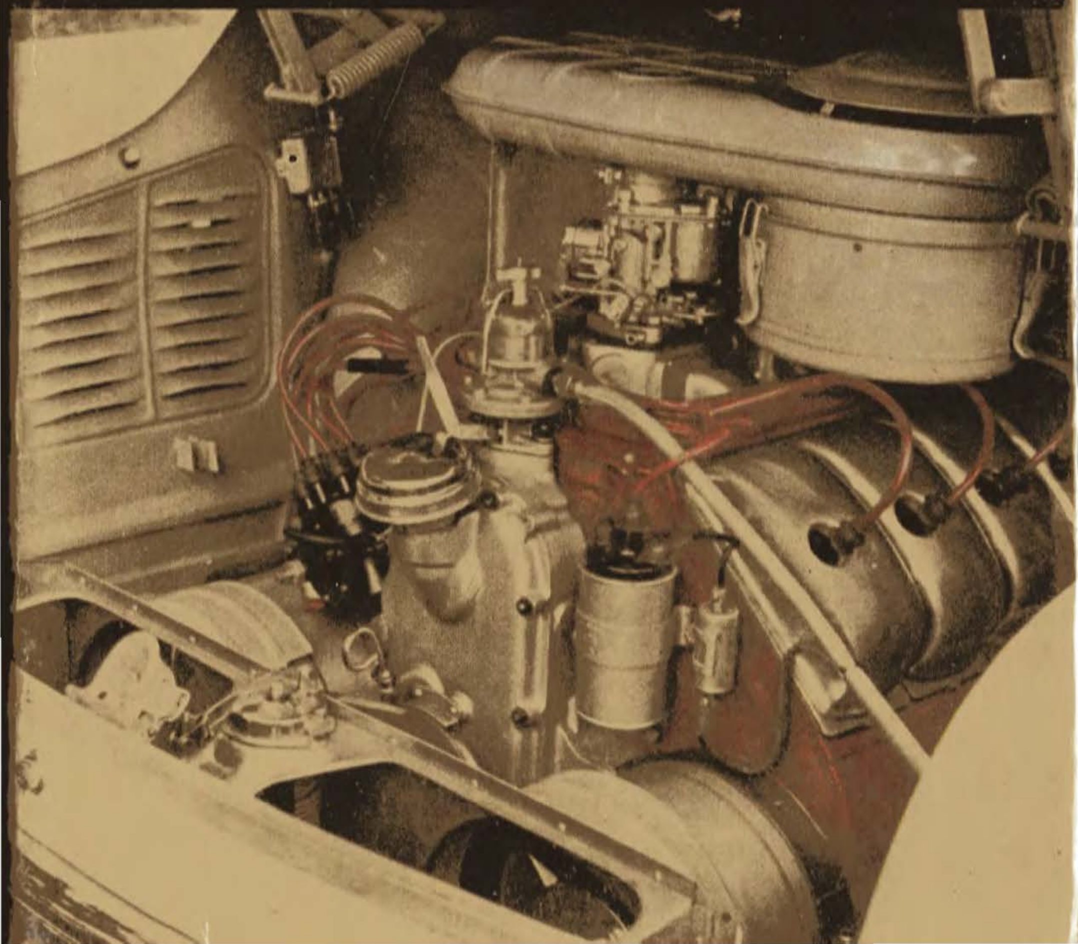
JULIUS MACKERLE, M. E.

AIR-COOLED MOTOR ENGINES



AIR-COOLED MOTOR ENGINES

By Julius Mackerle, M. E.

Direct cooling by air is one of the most effective means for reducing engine weight besides securing many other advantages. Its use in certain vastly successful small cars is prompting a lot of new study of its possibilities. Yet until now a comprehensive reference book has been lacking and the facts have been scattered in technical journals in many countries.

The first part of Mr. Mackerle's timely work deals with fundamentals of heat transfer, and heat removal from finned cylinders in particular. Special attention is paid to cooling control and utilization of waste heat. The second part of the book goes into design details of individual parts of the engine, choice of parameters, and the general conception of the engine design. All conclusions are based upon the long and varied practical experience of the author, and represent a valuable guide for the solution of problems encountered by designers.

The first, theoretical, and the second, practical, parts are complemented by an exhaustive survey of air-cooled engines produced by leading manufacturers of all countries, and the copious illustrations and diagrams are an outstanding feature.

While this will be a precious aid for those actually working on engines of this type, it contains a vast amount of detail valuable to all designers and draughtsmen in the motor car, motor cycle and aero industries. As the only book of its kind on a topic of immense technical importance it will earn its shelf-space in any engineering library.

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THE AUTHOR

Julius Mackerle, born in 1909 in Moravia, Czechoslovakia, took his degree in mechanical engineering at the Technical University in Brno and followed a career as a designer of internal combustion engines successively with Škoda Works in Pilsen and Prague, Dr. Porsche in Stuttgart, and Tatra at Kopřivnice.

In 1948 he was nominated the head of the design department of the Tatra Works, a pioneer factory of air-cooled engines, who produced their first passenger car so equipped in 1923. His projects included both petrol and oil engines as well as passenger cars and trucks.

After ten years of fruitful activities as chief designer of Tatra engines and cars he has taken over the engine department at the Research Institute for Motor Vehicles in Prague. Here he devotes his lifetime's experience to the research problems and development of new engine types and prime-mover units, and other design problems, in close co-operation with the Czechoslovak motor car manufacturers.

Mr. Mackerle has published numerous articles in European technical journals, and his book on Valve Gears for Internal Combustion Engines reached three editions. **Air-cooled Motor Engines** has appeared in two Czech editions, and has been translated into English, Russian and Polish.

AIR-COOLED MOTOR ENGINES

JULIUS MACKERLE M. E.

Demy 8vo, 476 pages, 415 illustrations,
inc. 1 folding plate and a nomogram

Authorized English Edition by K. Čáslavský
English Translation by K. ČÁSLAVSKÝ and H. ŠNÁBL
Edited by E. H. MACLACHLAN, *Transport Journal*

SNTL–Publishers of Technical Literature, Prague
and Cleaver–Hume Press Ltd, London

Printed in Czechoslovakia for
Cleaver-Hume Press Ltd, 31 Wright's Lane London W 8

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PREFACE

Direct cooling by air is one of the most effective means for reducing engine-weight in automobiles and commercial vehicles besides securing many other advantages. Its use in certain widely successful European small cars is prompting much new study of its possibilities, notably in Britain and America. But although much work has been done on air-cooled aircraft-engines a comprehensive reference book on these engines for cars has hitherto been lacking and the facts have been scattered in technical journals in many countries. It was to meet this need that the Author prepared the present book.

The first part deals with the fundamentals of heat transfer from hot gases through cylinder walls, and its removal from finned cylinders to the atmosphere. Special attention is paid to cooling control and utilization of waste heat. The second goes into design details of individual parts of the engine (especially where they differ from the corresponding features with liquid-cooling); choice of parameters, and the general concept of the engine design. All conclusions are based upon long and varied practical experience, and will it is hoped prove of value in the solution of problems encountered by designers. Many of the specific details may well interest a wide circle of draughtsmen in the motor car, motor cycle and aero industries.

The first, theoretical, and the second, practical, parts are complemented by an extensive survey of air-cooled engines produced by leading manufacturers of all countries and illustrations and diagrams are very freely used to elucidate the discussion. The tables, formulae and nomogram should make the volume additionally useful for reference.

In most instances British equivalents of the original metric units have been added, but it is felt that for certain detailed dimensions relating to engine specifications the British engineer is sufficiently familiar with metric units to leave these unconverted. A table of metric conversion factors is included.

J. Mackerle
E. H. Maclachlan

1 mm	0.039 in
1 cm	0.39 in
1 m	3.28 ft or
	1.094 yd
1 km	0.621 miles
1 cm^2	0.155 in^2
1 m^2	10.763 ft^2
1 cm^3	0.061 in^3
1 m^3	35.314 ft^3
1 litre	0.219 gal (Imperial) or
	0.264 gal (U.S.)
1 kg	2.204 1b
1 kg/cm^2	14.223 lb/in^2
1 kgm	7.233 ft-1b
1 kcal	3.968 B.t.u.
1 kg/m	0.671 lb/ft
1 litre/100 km	0.354 Imp. gal/100 miles
1 km/litre	2.87 miles/Imp.gal
1 metric horse power	0.987 b.h.p.

FIRST PART

THEORY

CHAPTER I

GENERAL PRINCIPLES

1. Advantages of air-cooled engines

It is correct say that all vehicle engines are air-cooled. Even in water-cooled engines heat is transmitted first from cylinder to water and afterwards, in the radiator, from water to air. This method of cooling by water is easy to accomplish, because the heat taken off the hot cylinder walls by water can be distributed without difficulty upon the large cooling surface of the radiator, and so the easy transmission of heat to air is made possible. Most vehicle engines are cooled by water; however, the great advantages of cooling engines by air are gradually being recognized.

Reciprocating engines used in aircraft are almost entirely air-cooled. Aircraft engines cooled by air are manufactured to-day in sizes ranging from 50 to 3500 h. p. and they have superseded water-cooled engines. The principal advantages of air-cooled aircraft engines are low weight, and greater reliability in operation. Nearly all transport air-liners are equipped with air-cooled engines. Their minimum fuel consumption at economical speeds amounts to about 0·39 to 0·48 lb/b.h.p.-h (180 to 220 g/b.h.p.–h.).

Modern motor cycles also are designed almost exclusively with air-cooled engines; with them, as with aircraft, sufficient cooling air is provided by the flow due to the motion of the vehicle, and no special cooling fans are required.* Air-cooled motor cycle engines have high specific performances, and they are equal to the best water-cooled engines. New designs of air-cooled vehicle engines are notable for their easy maintenance, reliability and economical operation.

In the internal combustion engine only one third of the heat energy contained in the fuel is converted into work, one third is lost in the exhaust gases, and one third is carried off by cooling. This distribution of available energy is approximate only, the exact relation depends on specific factors such as engine design, type of fuel, cooling system, etc. Heat carried off by cooling must be considered as a definite loss, because apart from the fact that no useful work can be obtained from it, part of the engine performance is frequently used for its removal. Therefore, every endeavour must be made to keep this loss of heat energy at a minimum.

* Recently special cooling fans have been mounted in some aircraft engines (see Chapter VI.).

Cooling of the engine is necessary for the following reasons. The maximum temperature of the cylinder wall is determined by lubricating conditions. After a certain temperature has been reached lubricating conditions begin to deteriorate rapidly, and increased wear, or even seizure of pistons and piston rings, are likely to occur. The maximum permissible temperature depends upon the quality of the lubricating oils; it ranges from 160 °C to 200 °C.

Strength of conventional materials limits the temperature of cylinder heads to approximately 220 °C — 260 °C. Cylinder heads of air-cooled engines are usually produced of light alloys whose strength decreases rapidly at temperatures above 200 °C. Additionally, if the temperature of the cylinder head is excessive the inducted air becomes too hot and volumetric efficiency decreases.

If, for instance, the external cylinder wall temperature in an air-cooled engine is 180 °C and the temperature of cooling air 20 °C, the temperature difference is 180 — 20 = 160 °C. In a water-cooled engine the cylinder is surrounded by 80 °C hot water, and the temperature difference amounts to 100—80 = 20 °C only. As the overall coefficient of heat flow from the cylinder wall into water is favourable, the actual heat transfer is not affected adversely by the decreased temperature difference. Heat transfer from radiator to air is less favourable; here the same conditions prevail as in the case of cylinder fins. However, the temperature difference is only 80 °C — 20 °C = 60 °C. The reduced temperature difference must be compensated by an enlarged cooling surface. Figure 1. gives a diagrammatic illustration of temperature differences in air- and water-cooled engines.

With water-cooled engines the heat transferred from the engine can be distributed over the sufficiently large surface of the radiator, hence the flow of air caused by the movement of the vehicle is adequate for cooling purposes, and it is often possible to dispense with a fan.

When the vehicle is climbing on a steep hill the road speed is reduced, and consequently the quantity of air passing through the radiator is not sufficient to convey the required amount of heat. The quantity of heat carried off by air is less than that conveyed from the engine to the radiator and the difference between both quantities is consumed for heating the cooling water. The temperature difference between radiator and air is thus increased, and accordingly the quantity of heat transmitted to the air is increased also. If at this stage no balance between the heat conveyed to and that taken off the radiator is attained, the temperature of the cooling water rises to boiling point.

To prevent coolant boiling it is necessary to use fans in water-cooled systems, so as to ensure a sufficient velocity of air flowing through the radiator at low speeds during uphill travel.

In modern cars with streamlined bodies the front of the car is lowered and consequently the radiator face reduced. In order to maintain an equal area of the cooling-surface, the depth of the radiator must be increased which means an increased resistance. Direct air flow is not sufficient and the increased

resistance must be overcome by the fan which thus requires a higher power input.

Water cooling can be improved by operating the radiator at overpressure. The boiling point of water is increased and so is the temperature difference between radiator and air. However, due to the higher pressure in the radiator, sealing of joints is more difficult.

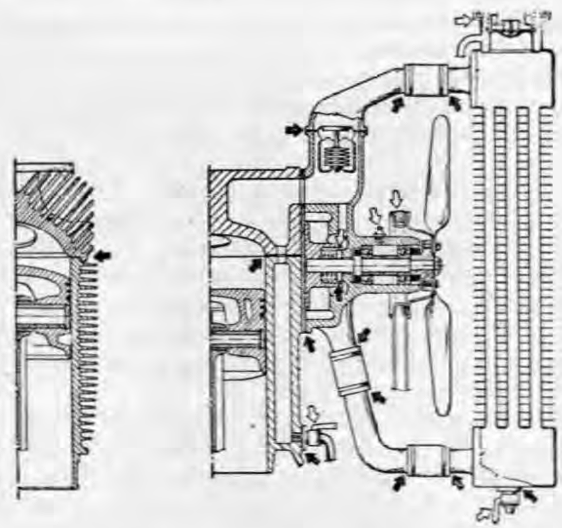


FIG. 1. In air-cooled engines heat is transferred directly to air. Temperature difference is considerable (160 °C approximately). In water-cooled engines heat is transferred first to the water and then from the radiator to air. The temperature gradient is smaller (radiator to air 60 °C). White arrows indicate points requiring maintenance, black arrows possible points of leakage.

The radiator in a water-cooled system, being made of non-ferrous metals, is an expensive component part of the vehicle. It is easily damaged, and, therefore, protected against shocks by flexible mounting. Water cooling requires a water pump whose stuffing boxes are usually not tight enough to prevent leaking, and need attendance. The rubber tubing connecting engine and radiator must permit positional variations and thus presents another source of insufficient tightness. Temperature control requires a thermostat, usually with a by-pass, i.e. with additional tubing and thus a potential source of failures. A fan is needed for air movement. Pressure cooling requires, in addition to the above equipment, a safety valve (in the radiator cap). Deposits of scale in the radiator have an adverse effect on heat transfer and their removal from the radiator is practically impossible; various cleansing agents denude minute fissures and cause leakages. According to statistical data 20% of all engine failures are caused by faults of the water-cooling system.

A direct air-cooling system is free from this type of deficiency. The main advantages of an air-cooled engine are the following.

No water is required and this means the elimination of 20% of engine failures caused by leakages of radiators, packings, tubing, etc., by faults of thermostats, safety valves, etc. Most important of all, serious breakdowns caused by frozen water cannot occur. Antifreeze cooling mixtures do not prevent all difficulties of this type. Apart from the high price of anti-freeze mixtures the inconvenience of their use must also be considered. Many of these mixtures attack rubber tubing and cause deposits in the radiator. A soiled radiator must be replaced by a new one, while soiled fins of an air-cooled engine can be easily cleaned.

By the elimination of water the operation of the vehicle is facilitated, especially in winter time; there is no emptying and filling up of the cooling system. Running a water-cooled vehicle in winter is difficult for short distance journeys. The water is not warmed up properly and cools off rapidly. In this connection it has to be borne in mind that the cylinder wear during a single starting of a cold engine is the same as during a 31 mile (50 km) journey. The absence of water is especially welcome in tropical countries and in places with water shortage.

Greater temperature difference between cooling air and cylinder has a particularly beneficial effect on cooling in hot weather. Consider a typical instance: in a water-cooled engine water temperature in the radiator is 80 °C, in an air-cooled engine the cylinder temperature is 180 °C. By an increase of the temperature of the surrounding air from 0 °C to 40 °C, the temperature difference in the water-cooled engine is reduced to half of the original value, whereas the drop in the air-cooled engine amounts to only 22%. An air-cooled engine therefore operates satisfactorily under tropical conditions, where water-cooled engines are liable to overheating.

Due to the high average cylinder temperature in the air-cooled engine thermal losses are small; the quantity of heat carried off by the cooling system is less than in the case of water-cooled engines and consequently the thermal efficiency of the air-cooled engine is higher. This explains also the smaller specific fuel consumption. A minimum specific fuel consumption of 0·33 lb/b.h.p.-h (150 g/b.h.p.-h) has been reached with air-cooled Tatra compression-ignition engines.

Less cooling air. Owing to the greater temperature difference one unit of cooling air carries off more heat, and, therefore, less cooling air is required for direct air-cooling of the cylinders than in a water-cooling system. A properly designed air-cooled engine requires only one half of the quantity of cooling air necessary for a water-cooled engine of similar performance. Air channels and ducts have smaller dimensions which is of great advantage particularly in the case of armoured vehicles.

Reduced weight. The manufacture of an air-cooled cylinder requires less

material than that of a cylinder of a water-cooled engine, besides the elimination of radiator, tubing and coolant. These savings mean a reduction of weight and smaller overall dimensions.

Reduced wear of cylinders. The cylinder of an air-cooled engine warms up more rapidly, because it is surrounded by a smaller quantity of material. This has a favourable influence upon the wear of cylinders. Maximum wear occurs

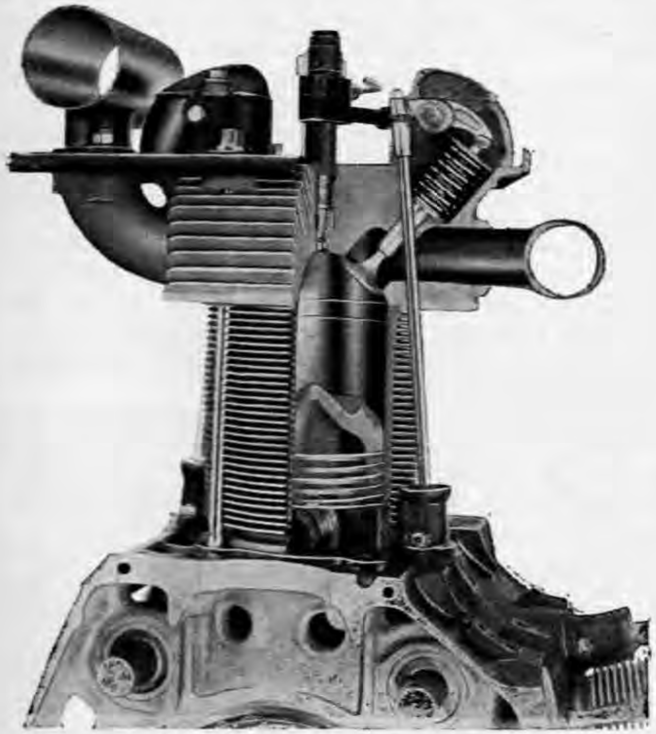


FIG. 2. Single-cylinder element of air-cooled Tatra engines.

immediately after starting, when cylinder walls are cold and inadequate lubrication takes place. Combustion products condensed on the cold cylinder walls attack the bearing surface and cause corrosion particularly in the upper third of the cylinder. This critical period is considerably shortened in an air-cooled engine; an important fact especially for vehicles operated on short runs in towns. Remarkable reduction of cylinder wear has been achieved on engines using producer gas fuel.

Unit cylinder construction. The air-cooled engine design requires a unit cylinder construction. The unit cylinder elements can be advantageously combined into engine units with various numbers of cylinders. Most of the component parts subject to wear are identical for all engines, a circumstance which permits a reduced stock of spare parts. Only crankshafts and camshafts are different. Crankcases, induction and exhaust piping, etc. are not subject to wear and do not appear often as stored spare parts. A unit cylinder construction is shown in Fig. 2.

If one unit cylinder element is to be used for both, in-line and V – types of engines, the cylinder spacing must be chosen so as to allow the proper dimensioning of crankshaft bearings in V engines, and giving no excessive cylinder distances for in-line engines. The use of roller bearings as main crankshaft bearings fulfils this condition.

A further advantage of the unit cylinder built-up construction is the possibility of rapid and less expensive repairs. A damaged cylinder can be singly dismantled and replaced. Similar damage in a water-cooled engine with no separately cast cylinders would call for replacement of the whole engine block or crankcase. An instance of an air-cooled railway engine is known, where a cylinder with piston was replaced at the terminal station in time to start the return run according to time-table; no such quick repair would be possible in the case of a water-cooled engine. An air-cooled engine can be removed from a vehicle without the difficulty of disconnecting water tubing or dismantling the radiator.

Saving of non-ferrous metals. Elimination of the radiator means considerable saving of non-ferrous metals; neither are non-ferrous metals required for gaskets and packings between cylinder heads and engine blocks (in air-cooled engines no gaskets are required for aluminium cylinder heads).

Sustained performance. Air-cooled engines do not give such a falling of in performance during operation as water-cooled engines: performance losses are caused by the build-up of carbon on the cylinder walls surrounding the combustion chamber which means a reduction in volumetric efficiency. The higher mean (not maximum) temperature of the cylinder of an air-cooled engine prevents excessive carbon deposits and thus the original performance of the engine is sustained for a long time. In water-cooled engines, particularly with side-valves, performance losses can amount to 14%. In these engines cooling conditions gradually deteriorate too, because scale is deposited from the cooling water on the walls between cylinder head, cylinder and water jacket.

Though the above advantages of air-cooled engines are significant, it is also necessary to consider their shortcomings. The absence of a water jacket surrounding the cylinders causes increased noise and special attention must be paid to pistons and their clearance in the cylinders. With the varying cylinder temperature valve clearances vary considerably with increase in noise level unless special precautions are taken. Noise inside the vehicle can be reduced by a suitable location of the engine (in the rear) and by proper acoustic insulation.

The use of oil-coolers instead of radiators is sometimes brought forward as

an argument against air-cooled engines. This reproach is not fully justified, because with an increasing specific power output (b.h.p. per unit of displacement) cooling of oil becomes very important. All contemporary water-cooled engines have water jackets along the whole length of the cylinder; formerly this type of design had been repudiated in order to leave the lower part of the cylinder without cooling and so prevent fuel condensation on the cylinder walls and the subsequent thinning of lubricating oil in the crankcase. However, high oil temperatures have prompted designers to employ the overall water jacket type "oil cooler" in water-cooled engines. High efficiency engines require special oil coolers.

A properly designed air-cooled engine operates satisfactorily with a comparatively small oil cooler or even without it. Air-cooled compression-ignition Tatra engines operate smoothly without oil coolers; lower oil temperatures are achieved in these engines by employing roller bearings for the crankshaft.

From the military point of view the reduced vulnerability of the air-cooled engine is of the utmost importance. A bullet hole in the radiator or water tubing of a water-cooled engine means a breakdown due to loss of cooling water. The loss of several fins of the cylinder head or cylinder of an air-cooled engine has practically no ill effects and the engine can continue operation.

An additional advantage of the air-cooled engine is its ease of installation. No difficulties with cooling are encountered with air-cooled engines located in the rear of the vehicle; less cooling air is required than in a water-cooled engine and the larger air pressure head is not influenced excessively by the dynamic pressure of the air in the inlet openings of the vehicle body.

The air-cooled two-cylinder engine with horizontally opposed cylinders (flat twins) is very simple and well balanced. The water-cooled version of the same engine shows a marked disadvantage caused by the complicated water installation. The air-cooled engine is not sensitive to position in the vehicle, for there is no danger of vapour locks forming and subsequent interruption of water circulation.

Air-cooled engines have proved their value also as boat engines, particularly for river craft, where contaminated water causes difficulties in a water cooling system and heat exchangers mean higher cost and increased weight.

2. A survey of the development of air-cooled engines

Air-cooled engines are as old as internal combustion engines. Direct air-cooling was used in the first types of engines. Later on, air cooling was retained for motor-cycle engines and water cooling was generally adopted for car engines. In the first stages of the development of multi-cylinder engines water cooling had been preferred by designers, because water proved to be the proper medium to transfer heat easily from inaccessible parts round the exhaust valve and to prevent local overheating. Designers could, therefore, concentrate their efforts on solution of other problems, e.g. carburation, ignition, lubrication, etc.

Despite such trends air-cooled engines persisted in some automobiles. Franklin, Tatra, Krupp, Phänomen and others are well known firms manufacturing air-cooled vehicle engines for many years. Air-cooled aircraft engines are particularly popular.

Remarkable progress in the development of air-cooled engines was made during the last world war. From the military viewpoint they possess many advantages such as easy servicing, reduced vulnerability, and low weight. Highly appreciated was the built-up (unit cylinder) construction which was responsible for the drastic reduction in the number of spare parts required for large fleets of varying vehicles. At the beginning of the second world war there were several successful vehicles with air-cooled engines in the standard equipment of the German Army, e.g. VW, Phänomen, Krupp, etc. Such vehicles have produced remarkable results in service and all factories concerned were ordered to develop an air-cooled tank engine having a performance of about 750 b.h.p.

The Simmering and MAN engines were among the best developed ones. Both had 16 cylinders. The Simmering engine had an X-type arrangement of cylinders as described on page 450. The cylinders of the MAN engine had a horizontal H arrangement with a calculated total performance of up to 900 b.h.p. Technical data of the engines are given in Table 1.

Table 1.

Technical Data of Air-Cooled Tank Engines

Manufacturer	MAN	Simmering	Continental
Bore, mm	130	135	146.1
Stroke, mm	165	160	146.1
Cylinder, number, arrangement	16 H	16 X	12 V
Displacement, cm^3	35,100	36,800	29,400
Performance, b.h.p.	690	620	750
Speed, rev/min	2,000	2,000	2,400
Weight, kg	1,545	2,250	1,700
Unit performance, b.h.p./l	19.6	16.9	25.5
Mean eff. pressure, kg/cm^2	8.8	7.6	9.55
Mean piston speed, m/s	11	10.7	11.7
Unit weight, kg/b.h.p.	2.24	3.63	2.27

Some vehicle engines of the compression-ignition type were produced during the same period. The considerable experience of the Tatra company which came under German rule after Munich, was directed towards the development of a compression-ignition engine with a cylinder bore of 110 mm and

direct fuel injection. Deutz engines were of the pre-combustion chamber type. Air-cooled engines were also made by Mercedes-Benz, Opel and several other manufacturers.

At that time fighting in North Africa and in Arctic regions increased the usefulness of air-cooled engines.

The advantages of air-cooled engines were recognized by the U.S. Army. Owing to a shortage of powerful vehicle engines, air-cooled Guiberson and Continental radial aero engines were fitted into tanks. The compression-ignition Guiberson engine which had not proved a success in aircraft, gave satisfactory service as a tank engine.

The experience gained from vehicles equipped with air-cooled engines in war service has led to an intensive development in the post-war period. The Deutz factory continued to develop independently the air-cooled chamber engine with 110/140 mm cylinders, i.e. of the same dimensions as the cylinders of their water-cooled engines. Under these conditions it was easy to compare the advantages and short-comings of both types of cooling. The result was significant: air-cooled engines were produced in larger number and production of water-cooled engines ceased almost completely. Air-cooled Deutz engines were followed by other makes, e.g. MWM, Stihl, Porsche, Güldner, in Germany; Petter, Armstrong-Siddeley, SLM, Warchalovski, Continental, Caterpillar, and many more in other countries.

Close attention is also paid to air-cooled engines in the USSR. Many designs were subjected to severe tests under the extreme climatic conditions prevailing on the vast territory of the USSR with results confirming all known advantages of these engines. Today several types of air-cooled petrol and oil engines of original design are being developed and full production is to be expected in the near future.

In the USA a rapid development of air-cooled engines for the U.S.Army was started in the post-war years. Several series, ranging in performance from 0·5 to 1400 b.h.p., were developed. (See description on page 411.) Reduction of the weight of the driving unit and the elimination of cooling difficulties made possible the development of new and more efficient vehicle types. The benefits of standardization is clearly illustrated by the following instance: the smallest series of 6 air-cooled engines containing 800 component parts covers the same range of performance as a series of 78 water-cooled engines containing 23 000 component parts used in World War II. Great efforts are being made towards the application of air-cooled engines for tractors and similar vehicles.

Air-cooled engines have been gradually introduced into passenger cars. New types have appeared following the successful application in the Tatra, VW, and Phänomen cars. Among them were the Citroën 2 CV, Panhard-Dyna, and Porsche. Recently Fiat have brought forward their 500 c.c. which has been followed by its Austrian version of Steyer. To this category belong the BMW Isetta, NSU Prinz, Goliath, Goggomobil and many others. The possibility of using an air-cooled engine in a large car has been demonstrated by the Tatra T 603 which is equipped with an air-cooled eight-cylinder rear engine developing 100 b.h.p. (Fig. 3.)

However, the introduction of air-cooled engines was not easy. One of the first pioneers was Dr. Eng. H. Ledwinka, former general manager and chief designer of the Tatra automobile works. After he had presented, in 1923, his original design for a popular car with an air-cooled twin-cylinder engine

FIG. 3. Tatra 603 car with air-cooled rear engine.

and swinging half-axles, he met a general resentment backed by all important competitors. The hard fight for the first air-cooled car engine was finally decided by the good specifications of the new car which was proved in competitions and races, and was fully accepted by the public. Many of the original Tatras are still running after a total mileage of well over 620,000 miles (1,000,000 km). The first popular car was followed by larger cars, among which the Tatra 87, the pioneer of streamlined bodies, with an air-cooled eight-cylinder 3-litre rear engine, won the greatest fame. The foresight of Dr. Ledwinka determined the line of development of the Tatra works, and the soundness of his ideas has since been proved.

In Germany a similar trend was followed by Dr. Eng. F. Porsche, who built a monument for himself by the design of the renowned Volkswagen. The VW holds the record of the largest production figure among European cars. The air-cooled four-cylinder rear engine contributed largely to the success of this car. Dr. Porsche also called to the public attention other applications of air-cooled engines. He built a series of air-cooled engines for Allgaier tractors and other purposes. The high efficiency and reliability of air-cooled engines has been proved to the world by the remarkable success of the sporting cars designed by Porsche.

Having dealt so far with the general history of the air-cooled engine, let us consider a case of actual development which clearly illustrates the development

from a comparatively simple idea of the past to a complex and highly perfected product of to-day. The author's own experience in designing air-cooled Tatra engines for many years enables him to present a detailed account of development.

The first stage came in 1923. It consisted of a small car with a central supporting tunnel, swinging half-axles and air-cooled two-cylinder engine. The

FIG. 4. The air-cooled flat twin engine of the Tatra 12 car was the pioneer of air-cooled motor engines in Czechoslovakia.

quality of the design of this popular car is proved by numerous vehicles of the original type still running after more than 35 years.

Reliability of the first Tatra was largely due to the air-cooled engine forming a part of the frame as shown in Fig. 4. The flat twin had an 82 mm bore, 100 mm stroke, and 1·1-litre displacement, giving a performance of 12 b.h.p. In spite of the solid construction the weight of the vehicle did not exceed 1499 lb (680 kg). The cast iron head and cylinder formed one unit, fitted with fins slanted in the direction of the air stream supplied by a fan. Parallel overhead valves were used.

In 1925 this car won the irksome "Targa Florio" race winning the 1st and 2nd prize in the 1100 c.c. category. In the same year she won the great

Russian Reliability Rally on the 5300 km track Leningrad – Moscow – Tiflis – Moscow, large sections of which consisted of field tracks.

After this experience the Tatra works decided to build air-cooled engines exclusively. They built a flat four-cylinder engine, T 30 with a displacement of 1·68 litres (Fig. 5) and the 1·91 litre type T 52. One of their most popular cars was the 1·16 litre 20 b.h.p. T 57 with an air-cooled four-cylinder engine. The engine had two cylinder blocks. (Fig. 6.)

FIG. 5. Air-cooled four-cylinder engine of the Tatra 30 type.

Increased engine performance was the main task in the design of an air-cooled engine intended for a fast passenger car. In 1934 the T 77 appeared as the first version of the fast passenger car with independent suspensions, stream-lined body and an air-cooled eight-cylinder rear engine. The T 77 reached speeds up to 89·9 m.p.h. (145 km/h) with an engine output of only 60 b.h.p. The eight-cylinder V type engine was of 2·97 litres. In the next version of this engine, type T 77a, the cylinder bore was enlarged to 80 mm, and so displacement increased to 3·4 litres and performance to 70 b.h.p. Experience gained from this engine has led to the design of a passenger car known as the type T 87 with an improved version of the original engine giving a performance of 75 b.h.p.

The T 87 represented a further step in development of the air-cooled Tatra engines. Cooling was improved by the application of an aluminium cylinder head with adequately designed fins. The cylinder head was removable, and was of semi-spherical type enabling the use of large valves which guaranteed a high volumetric efficiency. The cooling surface of the cast iron cylinder was enlarged by closely spaced fins. The electron metal crankcase was deeply extended underneath the crankshaft axis, and served as a rigid basis of the whole engine. Valves were operated by a camshaft located in the cylinder head. This arrangement prevented valve seat clearance from increasing with temperature and enabled a large valve incline. Cooling air was supplied by two radial fans, one for each line of cylinders.

FIG. 6. Undercarriage of the Tatra 57 car with an air-cooled four-cylinder engine.

The modern design was responsible for boosting output to 75 b.h.p., i.e. to 25·3 b.h.p./litre as compared with 18 b.h.p./litre obtained from the original design. The T 87 engine was constructed as a built-up type, i.e. a unit cylinder construction. The four-cylinder flat T 97 type engine was derived from the T 87. It had a unit cylinder construction and was mounted in the rear. Mounting of engines in the rear was made practicable by the small consumption of cooling air. An original feature has been the overhanging engine fixed to the flywheel flange, an arrangement used later on in the mounting of large oil engines (Fig. 8.).

In the following years the Tatra works used their experience with petrol engines to develop a series of air-cooled compression-ignition engines having a bore of 110 mm and a stroke of 130 mm. The best known engine of this type is a twelve-cylinder developing 200 b.h.p. mounted in the Tatra 111 truck. (Fig. 9.).

From the designer's point of view the cylinder head of the T 111 is very

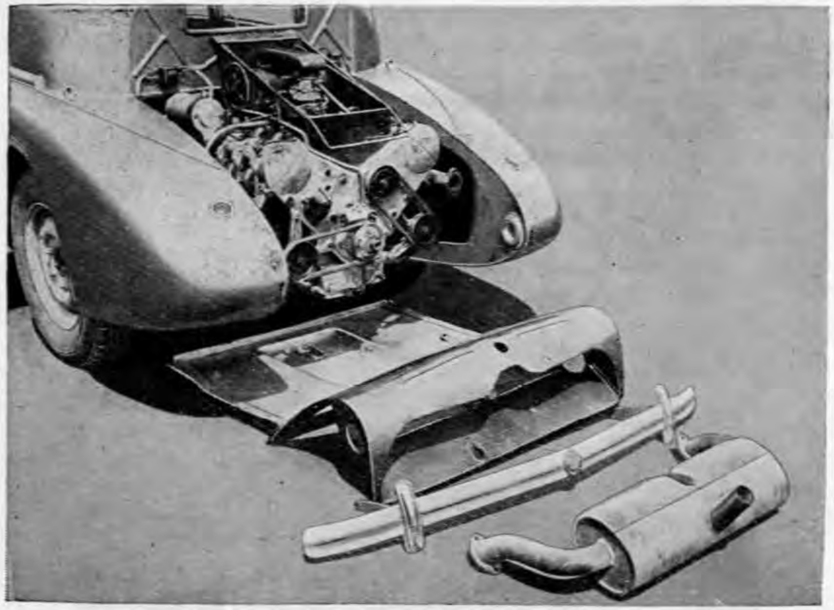


FIG. 7. The T 87 rear engine is easily accessible.

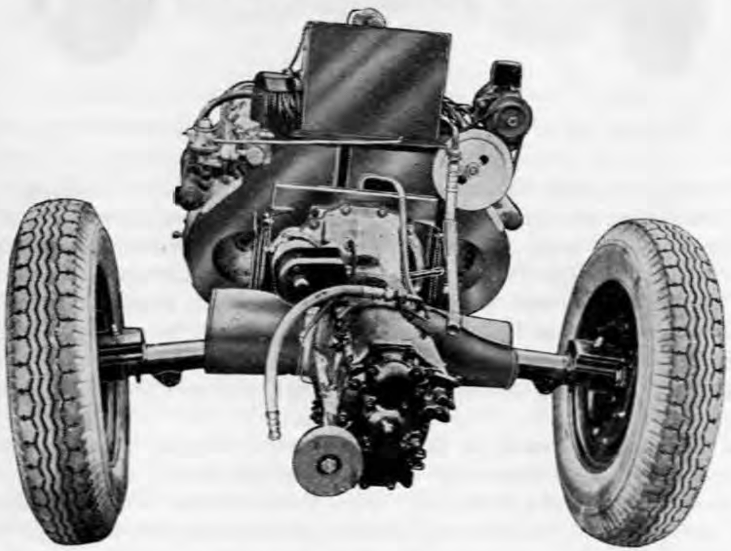


FIG. 8. Driving unit of the T 87 car. Overhung engine on the flywheel flange.

FIG. 9. The ten-ton Tatra 111 is fitted with a twelve-cylinder air-cooled engine.

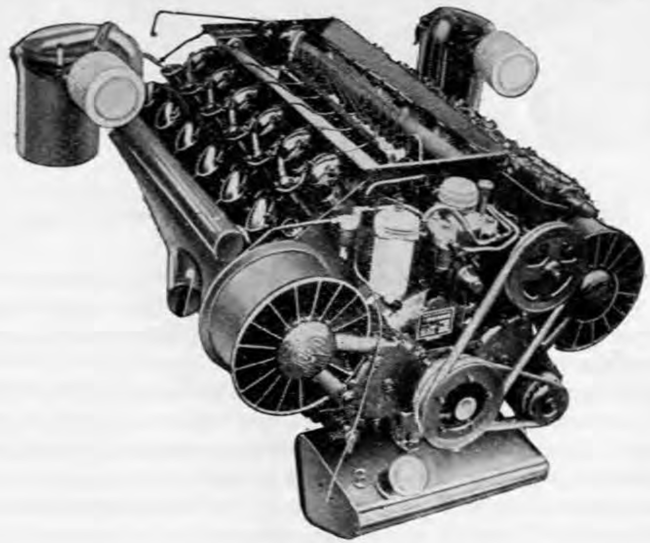


FIG. 10. The twelve-cylinder T 111 engine.

similar to that of the T 87. It has a semi-spherical combustion chamber and the same type of finning. The injection nozzle seated in the cylinder axis did not permit, for reasons of accessibility, operation of valves by a camshaft located

:r the cylinder head. Therefore, the unusual arrangement of valve operation by two camshafts, mounted in the crankcase, had to be used. Thus, the V type engine contains three camshafts. (Fig. 10.)

The T 111 engine has a cast iron crankcase and a split crankshaft with roller bearings. Giving a performance of 220 b.h.p. the engine weighs 2,173 lb (986 kg) only, i.e. a weight per b.h.p. value of 9 lb/b.h.p. (4.47 kg/b.h.p.).

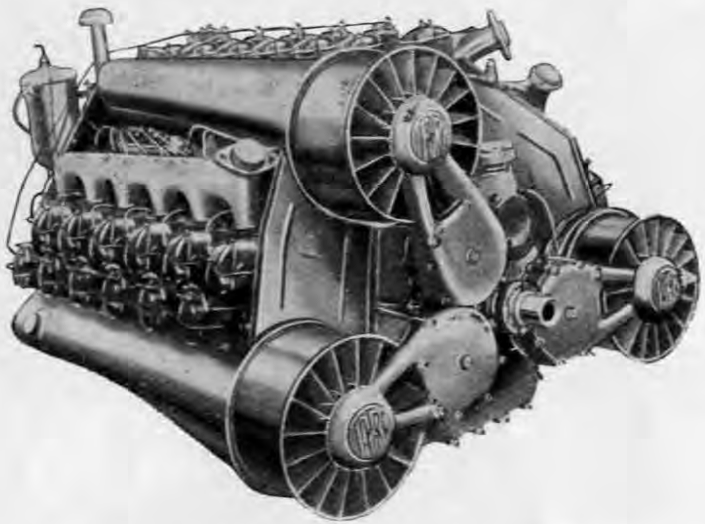


FIG. 11. Air-cooled 18-cylinder W engine of a built-up design.

Compared with water-cooled engines this is a very favourable ratio even without considering the weight of radiator and cooling water. Further advantages of the T 111 were small requirements of non-ferrous metals, and easy starting at low temperature.

The unit cylinder design of this built-up type was used for the construction of four- and six-cylinder in-line engines and eight- and twelve-cylinder V engines. An eighteen-cylinder engine, with three lines of cylinders arranged in W formation, was also constructed. The prototype of this engine assembled and tested experimentally in 1953 developed 330 b.h.p. at 200 rev/min. (Fig. 11.) This series of 110 mm bore was complemented by a five-cylinder in-line engine in 1945.

After 1945 it was decided to enlarge the bore to 120 mm and thus reduce the number of cylinders and improve conditions of combustion. The semi-spherical combustion chamber which proved its value in petrol engines, was not as satisfactory for diesel engines. In order to remove combustion roughness and fumes, the cylinder head had to be redesigned. A toroidal piston face

was used as a combustion chamber and air turbulence in the cylinder brought under control. Before applying the new design in the 120 mm bore series, tests were carried out in the 110 mm bore cylinder.

The good results of the re-designed cylinder head justified its use in the 110 mm bore series, manufacture of which continued due to the necessity to produce spare parts for existing vehicles in the post war years. The develop-

FIG. 12. Development of piston combustion chambers in the Tatra oil engine of 110 mm bore.

ment of combustion chambers with semi-spherical piston faces for Tatra diesel engines is shown in Fig. 12. The re-designed head of the 110 mm bore cylinder produced a smooth combustion, eliminated fumes and reduced specific fuel consumption. Bench test measurements on an eight-cylinder V-type engine indicated a specific fuel consumption of less than 0·33 lb/b.h.p.–h. (150 g/b. h.p.–h).

Following the successful application of air-cooled Tatra engines in passenger cars and trucks, it was decided to use them in the M 131.1 type railroad cars of the Czechoslovak railways. (Fig. 13.) This first example of using air-cooled engines for railway traction was the natural result of the extensive development work carried out by the Tatra works. For the extremely heavy conditions of railway operation, the output of the twelve-cylinder engine was reduced to 160 b.h.p. at 1600 rev/min. and an oil-cooler was fitted.

The air-cooled engine proved its value in the railway service. Breakdowns caused by water freezing were eliminated, traction cars could be laid off at stations without heated sheds, and repairs presented no difficulties. After the

FIG. 13. M 131.1 type Tatra railroad car driven by an air-cooled engine.

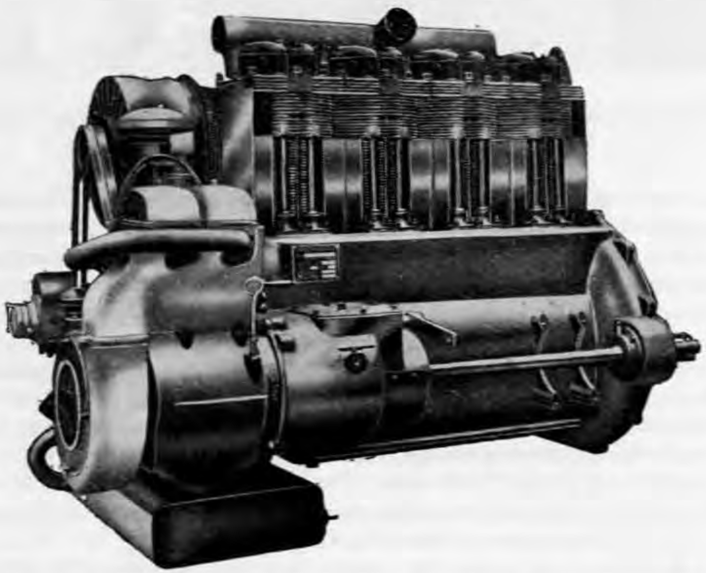


FIG. 14. Four-cylinder T 904 air-cooled engine with two-cylinder petrol engine for starting used in ČKD tractors.

introduction of the redesigned cylinder head, fuel consumption was reduced by 15%. With proper maintenance this railway car with 48 seats consumed 27 litres of oil per 100 km.

The construction of compression-ignition engines of the 120 mm bore series commenced in 1945. A flat six-cylinder intended for a seven-ton truck was the first unit produced. This had a bore 120 mm, stroke 150 mm, output

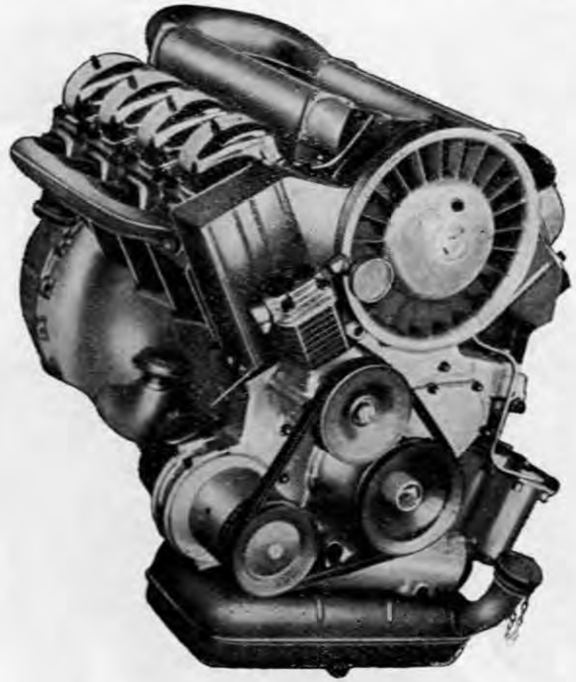


FIG. 15. Eight-cylinder Tatra 928 engine with 120 mm bore and 130 mm stroke.

130 b.h.p. at 2000 rev/min. The four-cylinder in-line type of this series used for the ČKD caterpillar tractor is of interest. The diesel engine is started by means of a two-cylinder petrol engine mounted on its side. No storage battery is needed and the tractor can be readily started even after a standstill of several months. (Fig. 14.)

The new series of the Tatra air-cooled engines (120 mm bore, 130 mm stroke) was exhibited for the first time in 1956. Improvement of specific values and technical simplification were fully incorporated in the development of the new series of compression-ignition engines. Due to consistent standardization the new engines were composed of similar cylinder groups, common crankshaft components, unified fans, oil pumps, etc.

Simplification of design is further illustrated by all in-line and V engines having one type of camshaft and fan. Reduction in number of cylinders due to the enlarged bore, from 110 to 120 mm, made possible a considerable weight reduction. The new eight-cylinder T 928, for instance, developing 180 b.h.p. weighs 1,311 lb (595 kg) as compared with the 2,160 lb (970 kg) weight of the twelve cylinder T 111 having the same output of 180 b.h.p. The

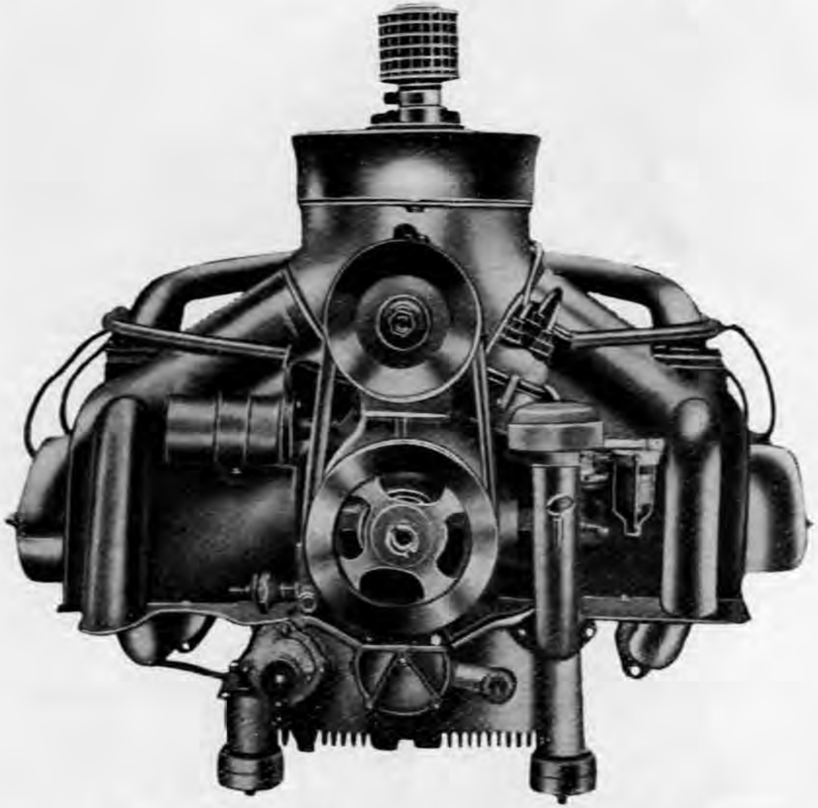


FIG. 16. Engine of the Tatraplan car with fan on vertical shaft.

weight per b.h.p. value, calculated for 1800 rev/min, was reduced from 11·5 lb/b.h.p. (5·4 kg/b.h.p.) of the Tatra 111 to merely 6·9 lb/p.h.b. (3·3 kg/b.h.p.) of the T 928.

Great care was given to the control of cooling. The fan drive includes a fluid coupling which is not filled with oil until the engine warms up, when a thermostat, placed in the stream of hot air beyond the cylinders, opens

the oil inlet to the coupling. In the cold season fuel consumption is reduced due to less power being required for driving the fan.

Compression-ignition engines are produced with the following numbers of cylinders: 1, 2, 4, 6, 8, and 12. An eight-cylinder engine is shown in Fig. 15. The 8V and 12V types are being developed as supercharged engines which means a further improvement of specific values. Further improvement is

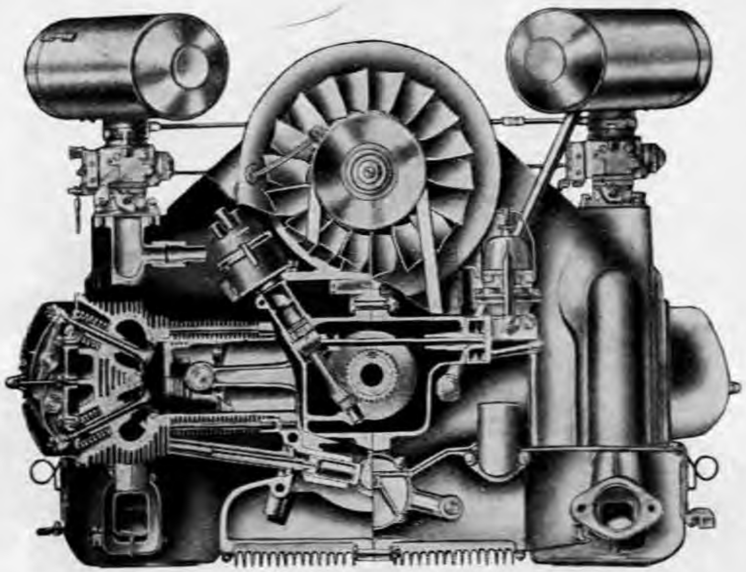


FIG. 17. Cross section of the T 600 engine with fan on horizontal shaft and two carburettors.

possible by developing types with higher speeds. A description of these engines can be found on page 461.

For the post-war Tatraplan passenger car a flat four-cylinder, with 1.95 litre displacement developing 52 b.h.p. at 4000 rev/min, has been designed. This petrol engine has aluminium cylinder heads and semi-spherical combustion chambers. Valves are not inclined as much as in the T 87, and they are operated by aluminium rods from the camshaft which is placed in the aluminium crankcase. This short-stroke, bore 85 – stroke 86 mm, engine develops 26 b.h.p. per litre which is the same specific output as that of the T 87. Originally the engine had an axial fan with a vertical shaft driven by a bevel gear (Fig. 16.). On the later design a horizontal fan was mounted directly to the dynamo shaft and driven by a V belt. The long manifold piping of the original design was eliminated by using two carburators and by this arrangement performance was increased to 55 b.h.p. (Fig. 17.).

Despite the results obtained, the fight between followers and opponents of air-cooled engines continued in the post-war years. One of the principal opposing arguments was the firm assertion that a specific output of 26 b.h.p. per litre must be regarded as a maximum and consequently air-cooled en-

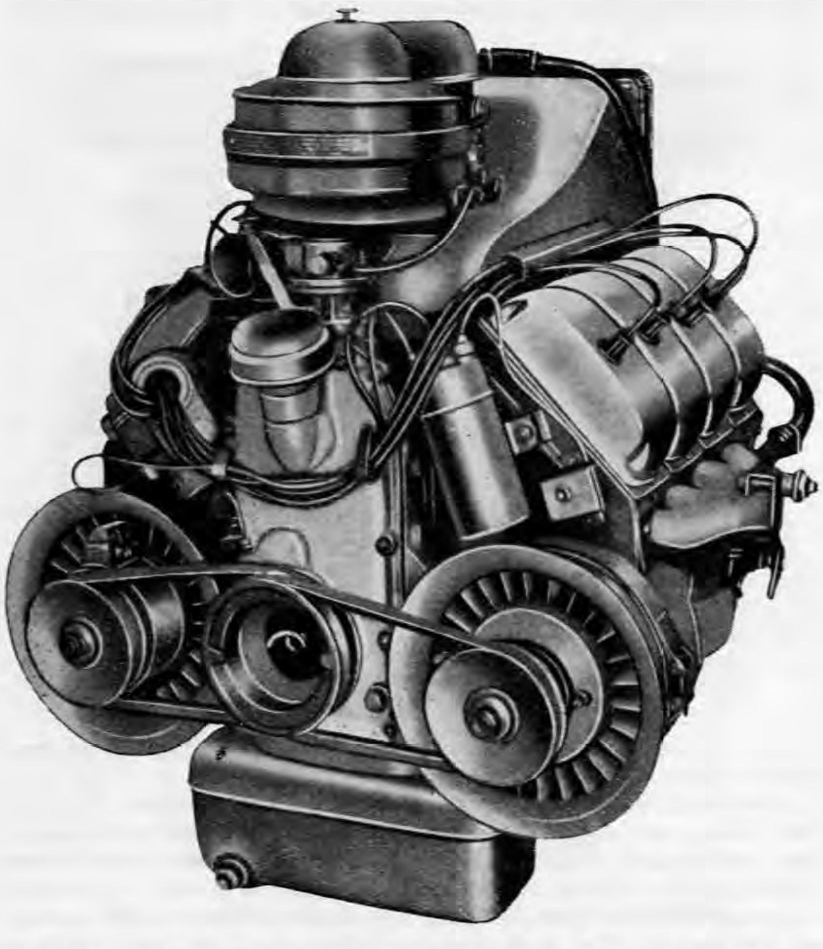


FIG. 18. The eight-cylinder T 603 petrol engine is small, light and efficient. It is fitted in Tatra racing cars.

gines are bound to be inferior to modern water-cooled engines. Further it has been pointed out that in countries with a highly developed automobile industry air-cooled engines are used only on a small scale, particularly in pas-

FIG. 19. Tatra racing car with air-cooled rear engine.

senger cars. The Tatra works decided, therefore, to design a new petrol engine to prove the advantages of an air-cooled engine.

The new Tatra T 603 engine was designed as a special rear engine for passenger cars. Requirements of optimum accessibility, low weight, and of the use of one carburettor with short induction piping necessitated the selection of an eight-cylinder V type engine. Originally cylinder dimensions were 72×72 mm, but the bore was increased to 75 mm at an early stage of development. The combustion chamber was again of a semi-spherical shape. The camshaft at the top of the crankcase enabled the use of a large angle between the valves to give high volumetric efficiency even at high engine speeds. The engine weighs only 330 lb (150 kg) and develops 85 b.h.p. at 4500 rev/min. (Fig. 18.).

The main advantages of an air-cooled engine become apparent by comparing this engine with the T 87. At an equal bore the T 603 has a stroke of only 72 mm and all advantages of a short-stroke engine are provided. Heat losses are smaller and so are the dimensions and the weight of the engine. The total displacement of the engine is 2·545 litres which means a specific output of 33·5 b.h.p./litre, i.e. a value attained by the best types of water-cooled engines. Weight per b.h.p. amounts to 3·2 lb/b.h.p. (1·76 kg/b.h.p.) which is less than corresponding figures of the best mass produced water-cooled engines (if weight of radiator and cooling water are included).

Experience gained from racing car engines is very valuable to designers of air-cooled engines. From the early stages of development up to present times it has been recognised that more deficiencies can be discovered in a single race than in thousands of miles travelled under normal conditions. Dynamic loads of the valve gear and crank mechanism increase with speed and by increasing rev/min from 4500 to 8000, stresses will increase 3·15 times. An engine having proved its value in a race can be safely assumed to be reliable for running under normal conditions. As a matter of principle the Tatra works used for racing purposes engines selected from normal production series with slight modifications.

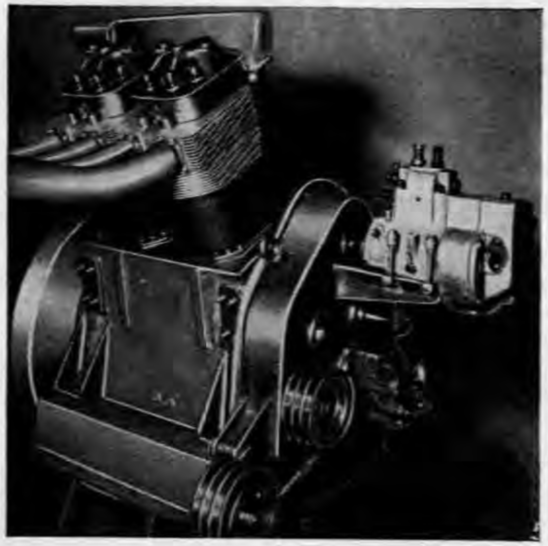


FIG. 20. Experimental twin cylinder Škoda unit, bore 140 mm. Fan and cylinder head cowlings removed.

The racing type of the T 603 engine differs from the standard engine by having two carburettors, special camshaft and stronger valve springs. Compression ratio is increased from 6 : 1 to 11 : 1 and with special fuel the engine develops 200 b.h.p. at 7500 rev/min. With increasing rev/min this performance will increase even further. (Fig. 19.).

The extensive development work carried out by the Tatra works has been followed by other Czechoslovak factories.

The Škoda works, among others, developed a twelve-cylinder compression-ignition engine having a performance of 450 b.h.p. at 2000 rev/min, bore 140 and stroke 175 mm. An experimental twin cylinder unit of this engine gave very good test results; with an output of 75 b.h.p. at 2000 rev/min specific fuel consumption amounted to about 0.35 lb/b.h.p.–h (160 g/b.h.p.–h.). The prototype cyinder head was fully machined from an aluminium block and, in a hot state, screwed on to the steel cylinder which had turned cooling fins. Two inlet and two exhaust valves were used in each cylinder head, and, between them, in the cylinder axis was the seven-hole injection nozzle for direct fuel injection. A centrifugal cooling fan was mounted sideways (Fig. 20.). After 1945 further development work on this type ceased. The Škoda works also developed several air-cooled flat six-cylinder petrol engines.

The ČKD works successfully used air-cooled four cylinder compression-ignition Tatra T 114 engines for their caterpillar tractors. After this the Tatra

T 904 engine (bore 120 mm), with a small two-cylinder starting petrol engine, was used in a larger caterpillar tractor of the same company. In view of the excellent performance of air-cooled Tatra engines in tractors, the ČKD works embarked on the development of an air-cooled compression-ignition engine of their own. Following the first tests with a 145 mm bore ex-

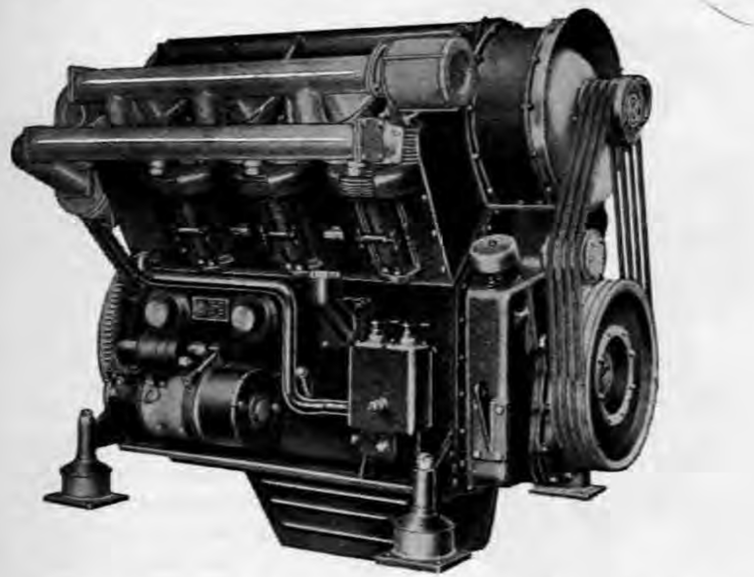


FIG. 21. Air-cooled six-cylinder ČKD V engine, bore 145 mm, for heavy duty service.

perimental single-cylinder unit, tests were carried out with variously shaped combustion chambers and four- and two-valve cylinder heads. The result was a two-valve aluminium cylinder head fixed by a flange joint to the steel cylinder having turned fins. The first engine of this type was a six-cylinder with two banks of cylinders arranged in a V inclined at 60°. (Fig. 21). The engine, having a performance of 160 b.h.p., is being developed further. The ČKD production program includes a twelve-cylinder engine.

Simultaneously developed motor-cycle engines gained a reputation based on quality and high performances. Four- and two-stroke JAWA engines lead the field with the Strakonice ČZ engines as second.

Among air-cooled aircraft engines developed in Czechoslovakia it is worthwhile to mention the remarkable design of the two-stroke compression-ignition ZOD type produced by the Zbrojovka in Brno (Fig. 22). Former manufacturers of air-cooled aircraft engines were Walter, Avia, ČKD, etc.

Today, the tendency to use air-cooled engines for stationary installations can be observed all over the world. Modest requirements of maintenance and insensitiveness towards climatic conditions are mainly responsible for this trend. Similar properties are required for tractor and popular car engines. The rapid development is bound to bring about a further refinement of air-cooled engines and their subsequent application to more types of vehicles.

FIG. 22. Air-cooled two-stroke aircraft engine of the ZOD type.

3. Comparison of air- and water-cooled engines

For comparing two engine types, suitable criteria must be established. The principal requirement in movement of persons and goods is economical operation. This is influenced not only by the specific fuel consumption, but also by other specific values, particularly specific weight and performance per unit of required space. Let us compare the individual characteristics of vehicle engines.

1. *Fuel consumption* is of utmost importance, because it has a direct bearing upon transport costs. Specific fuel consumption serves as an important comparative value and also expresses the overall efficiency of an engine. Type of fuel and its price have to be considered, too. At equal specific fuel consumption a diesel-engine is more economical than a petrol engine.

Fuel consumption for passenger cars and for trucks is even more important. This consumption is determined not by the engine only, but by the overall properties of the vehicle as well. A well designed vehicle may compensate for the shortcomings caused by a less economical engine.

2. *Weight of engine* is another important property. Vehicles are built to transport persons and goods; therefore, the weight of the vehicle must be kept at a minimum. It has to be borne in mind that the engine weight determines, to some extent, the weight of other parts of the vehicle. A low-speed engine with a large torque requires heavier transmission and driving gear. Increased dead weight means higher fuel consumption and greater wear of tyres. Engine weight can be reduced in cars and buses by using light metals for component parts; this, however, is less feasible for truck engines. Specific weight is a further comparative value. In water-cooled engines the weight of radiator and cooling water should be added to the weight of the engine unit.

3. *Engine dimensions* ought to be as small as possible. A small engine can be easily accommodated in a passenger car, and its weight can be distributed between front and rear axles without difficulty. The overall length of buses is

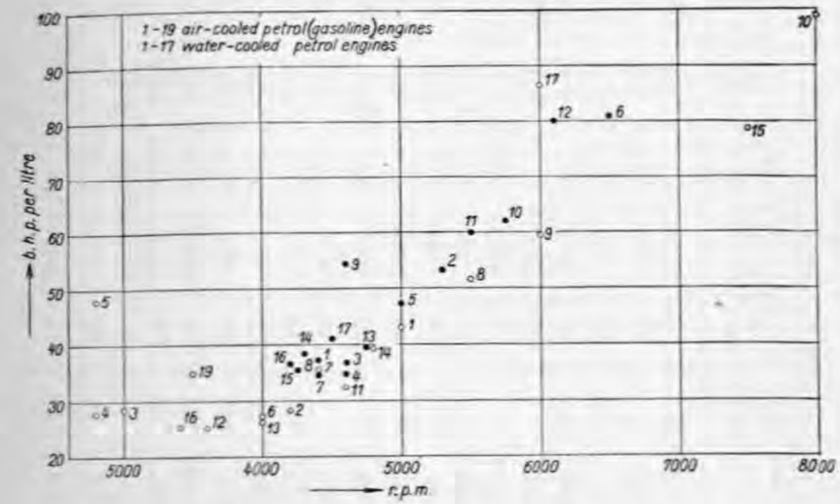


FIG. 23. Unit performance of petrol engines.
o — air-cooled engines, • — water-cooled engines, *5* — supercharged engines.

limited and the largest possible space must be made available for passengers. A small engine, located beside the driver's seat or beneath the floor, may increase the capacity of the vehicle by several seats. The same considerations apply to the utilization of the length of trucks for loadings area.

4. *Maintenance* is an important factor in overall transport costs. In addition to direct maintenance cost, the value of idle standstill time has to be calculated. Often the duration of standstill depends on the availability of spare parts. Sometimes one single spare part missing may cause the vehicle to be out of operation for a long time.

The above factors must be considered whenever engines are being compared. Performance per unit displacement is less important for comparison unless it is considered instead of another specific value, or in cases where displacement is used as a basis for taxation or determining racing categories. An increased specific performance means an improvement of other specific values. Performance per unit displacement depends largely on the rev/min value.

Engine speed can be advantageously increased up to a corresponding mean

Table 2.

Technical Data of Air-Cooled Petrol Engines

	Manufacturer	Bore mm	Stroke mm	Cyl.	Sw. vol. cm^3	Weight kg	Perf. bhp	Speed r.p.m.	Unit perform bhp/l	p_e kg/cm^2	c_m m/s	Unit weight kg/bhp	Comp. ratio
1	BMW 700	78	73	2F	696		30	5000	43	7.8	12.3		7.5
2	Citroên	66	62	2F	425		12	4200	28.2	6.05	8.7		6.25
3	Continental	117.5	101.6	8F	8808	408	250	3000	28.4	8.5	10.1	1.64	
4	Continental	146.1	146.1	12V	29300	1080	810	2800	27.7	8.9	13.6	1.34	
5	Continental	146.1	146.1	12V	29300		1400	2800	47.9	15.4	13.6		
6	Fiat 500	66	70	2L	479		13	4000	27.2	6.1	9.35		6.5
7	Chevrolet	85,7	66	6F	2286		81	4400	35.4	7.25	9.7		8.0
8	NSU Prinz	75	66	2F	583		30	5500	51.5	8.5	12.1		6.8
9	Panhard Dyna	85	75	2F	841		50	6000	59.5	8.9	15.–		
10	Porsche RSK	85	66	4F	1498		148	8000	99	11.1	17.6		9.8
11	Steyer Puch	70	64	2F	493		16	4600	32.5	6.35	9.8		
12	Tatra 87	75	84	8V	2968	194	75	3600	25.2	6.3	10.–	2.59	5.6
13	Tatra 600	85	86	4F	1950	150	52	4000	26.5	6.0	11.5	2.88	6.0
14	Tatra 603	75	72	8V	2545	160	100	4800	39.3	7.4	11.5	1.6	6.2
15	Tatra 603 Sport	75	72	8V	2545	160	200	7500	78.6	9.4	18.0	0.8	12.–
16	Volkswagen	77	64	4F	1192	78	30	3400	25.2	6.67	7.24	2.6	6.6
17	JAP	84	99	2V	1096	50	95	6000	86.5	13.0	19.7	0.6	14.–
18	Willys	69.85	57.15	4F	877	62	25		28.5			2.48	6.5
19	American Motors	82.5	82.5	4V	1770	90	62	3500	35	5.1	9.65	1.45	7.5

Table 2. cont.

1	Austin 55	73	88.9	4L	1489		56	4400	37.6	7.7	13		8.3
2	Alfa Romeo 2000	84.5	88	4L	1975		105	5300	53.2	9.—	15.5		8.25
3	Čajka	92	92	8V	4890		180	4600	36.8	7.2	12.6		
4	Fiat 600	60	56	4L	633	71.4	22	4600	34.8	6.8	8.6	3.24	7.5
5	Fiat 1800	72	73.5	6L	1795		85	5000	47.3	8.5	12.3		8.8
6	Ferrari 410	88	68	12V	4962		400	6500	80.7	11.2	14.7		9.0
7	Ford Zephyr	82.5	79.5	6L	2552	196	90	4400	35.3	7.2	11.7	2.18	7.8
8	Hillman Minx	76.2	76.2	4L	1390		52	4400	37.4	7.65	11.16		8.0
9	Chrysler LC3H	101.6	99	8V	6423		350	4600	54.5	10.6	15.2		10.0
10	Jaguar	83	106	6L	3442		213	5750	62	9.7	20.3		8.0
11	Mercedes Benz 300	85	88	6L	2996	256	180	5500	60	9.8	16.1	1.42	8.55
12	Mercedes Benz 300 SL	85	88	6L	2996		240	6100	80	11.8	17.9		9.5
13	Morris 1000	62.9	76	4L	948		37.5	4750	39.6	7.5	12.0		8.3
14	Opel Kapitän	85	76.5	6L	2605		100	4300	38.4	8.03	10.95		7.8
15	Renault Dauphine	58	80	4L	845		30	4250	35.5	7.52	11.3		7.25
16	Škoda 440	68	75	4L	1089	103	40	4200	36.75	7.88	10.5	2.57	7.0
17	Triumph Herald	63	76	4L	948		39	4500	41.1	8.25	11.4		8.0

Cylinder arrangement: V, L in-line, F flat.

piston velocity of 29 to 36 ft/s (9 to 11 m/s). Mean piston velocities in engines of passenger cars reach values up to 50 ft/s (15 m/s) and even more. In such cases, however, the risk of a considerable reduction of the mechanical efficiency of the engine occurs. Low mean piston velocities have an adverse effect upon the lb/b.h.p. (kg/b.h.p.) weight of the engine. In some cases, however, optimum conditions can be attained at smaller mean piston velocities; especially in robustly built engines where the accelerating forces of the heavy pistons and connecting rods cause a reduction of the mechanical efficiency even at low speeds. Specific fuel consumption increases with reduced mechanical efficiency, and, therefore, speeds of compression-ignition engines are kept at 1800 rev/min or less. The economic advantage of higher mechanical efficiency is partially counterbalanced by a higher specific weight.

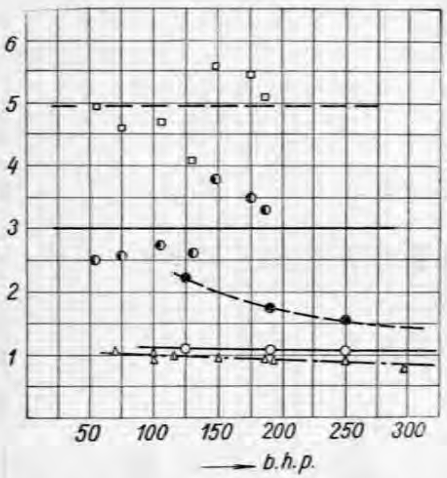


FIG. 24. Comparison of weights in kg/b.h.p.
□ — weight of water-cooled engines with accessories for lorries and buses, ◐ — basic weight of engines for lorries and buses, ● — weight of air-cooled vehicle engines with accessories, ○ — basic weight of air-cooled vehicle engines, △ — basic weight of air-cooled aircraft engines.

Petrol engines. When comparing air-cooled and water-cooled engines it must be remembered that the weight of water-cooled engines is given in tables without the weight of radiators and cooling water. Technical data of the best known vehicle engines are given in Table 2. Specific performances of engines listed in Table 2 are shown in the diagram Fig. 23 where the full lines indicate maximum engine performances attainable without the use of superchargers. [58]. Mean effective pressures are also shown in the diagram.

The benefit of smaller weight is even more marked in the case of large size air-cooled petrol engines. The Continental Company converted many of their aircraft engines into large size petrol engines suitable for buses and trucks. Weights of these engines approach very closely those of aircraft engines. Fig. 24 shows weights of these engines, net and including accessories, compared with weights of water-cooled engines. It is clearly illustrated that air-cooled engines weigh approximately only 1/3 of the weight of water-cooled engines.

Compression-ignition engines. Similar considerations apply to this type of engine. Technical data of proved compression-ignition engines are given in Table 3 and Fig. 25. Table 4 contains data of two-stroke and supercharged engines including dimensions and enables, therefore, a more complete compar-

Table 3.

Technical Data of Air-Cooled Oil Engines

	Manufacturer	Bore mm	Stroke mm	Cylind	Sw. vol. cm³	Perf. bhp	Speed r.p.m.	Unit perform bhp/l	p_e kg/cm²	c_m m/s	Weight kg	Unit weight kg/bhp
1	Continental	146.1	146.1	12V	29400	750	2400	25.5	9.55	11.7	1700	2.27
2	ČKD	145	180	6V	17800	160	1600	9.0	5.05	9.6	1300	8.1
3	Deutz F 4L 514	110	140	4L	5300	90	2300	17.0	6.68	11.8	476	5.38
4	Deutz F 6L 514	110	140	6L	8000	130	2250	16.25	6.40	10.58	625	4.80
5	Deutz F 8L 614	110	140	8V	10640	175	2250	16.50	6.68	10.58	850	4.85
6	Deutz F 6L 612	90	120	6L	4580	84	2800	18.3	5.9	11.2	360	4.3
7	Guiberson	130	150	9R	16730	250	2200	15.0	6.15	10.25	290	1.15
8	MAN	130	165	16H	35100	690	2000	19.6	8.8	11.0	1545	2.24
9	MWM	105	120	8V	8312	140	2500	16.9	6.1	10.0		
10	Paxman	152.4	167.6	12V	3650	446	2100	12.8	5.5	11.7		
11	Petters	76.2	76.2	4L	1390	20	3000	14.4	4.32	7.6	228	11.35
12	Petters	80	110	2L	1106	9	1500	8.1	4.9	5.5	255	28.0
13	Robur GD4	90	125	4L	3180	52	2600	16.3	5.6	10.8	355	6.8
14	Robur LD8	100	125	8V	7850	150	2600	19.1	6.6	10.8	550	3.7
15	SLM	110	140	12F	16080	220	2000	13.70	6.16	9.3		
16	SLM	110	140	8F	10600	150	2000	14.0	5.75	10.3		
17	Simmering	135	160	16X	36800	620	2000	16.9	7.6	10.7	2250	3.63
18	Tatra 111	110	130	12V	14820	200	2000	13.50	6.05	8.7	986	4.93
19	Tatra 928	120	130	8V	11752	180	2000	15.3	6.9	8.7	595	3.3
20	Tatra 928K	120	130	8V	11752	220	2000	18.7	8.4	8.7	630	2.85

ison. The most important values are performance per unit of required space in b.h.p./ft^3 (b.h.p./m^3) and specific weight in lb/b.h.p. (kg/b.h.p.) Other values, with the exception of specific fuel consumption and life, are less important from the consumer's point of view.

Air-cooled compression-ignition engines, compared with water-cooled, are conspicuous once again by low specific weight and high performance per unit

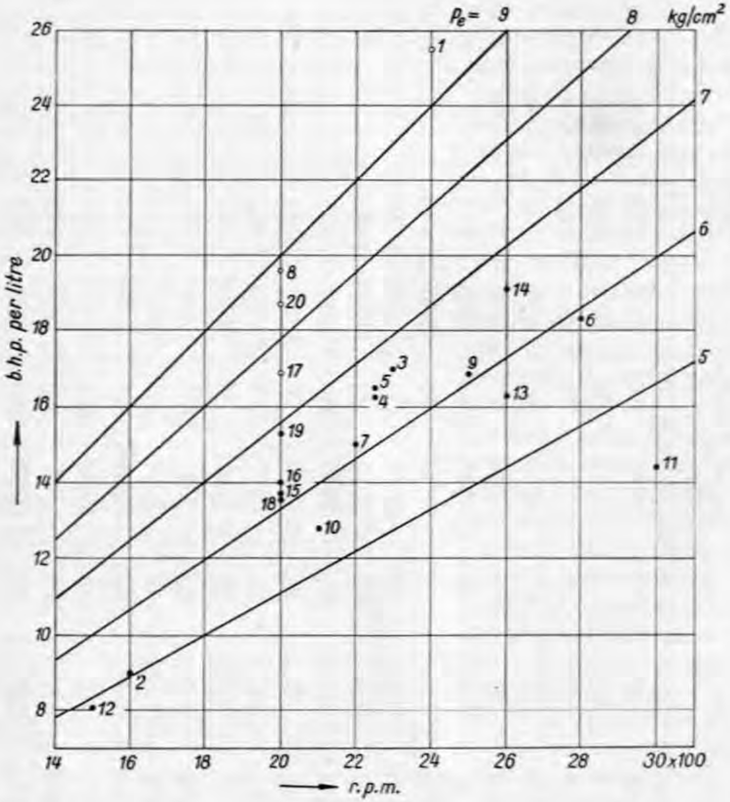


FIG. 25. Unit performance and mean effective pressure of air-cooled compression-ignition engines.

of required space. The benefit of high rev/min is demonstrated in the case of the Hercules engine having a performance of 6·1 b.h.p. per ft^3 (215 b.h.p. per 1 m^3) of space and a specific weight of 7·5 lb/b.h.p. (3·85 kg per b.h.p.) The corresponding values would be still better in an air-cooled engine having the same speed.

Good specific values can be attained with supercharged engines. The same possibility exists with air-cooled engines. It is apparent from the table of technical data that air-cooled engines without superchargers are closely approaching values of supercharged water-cooled engines. A remarkable

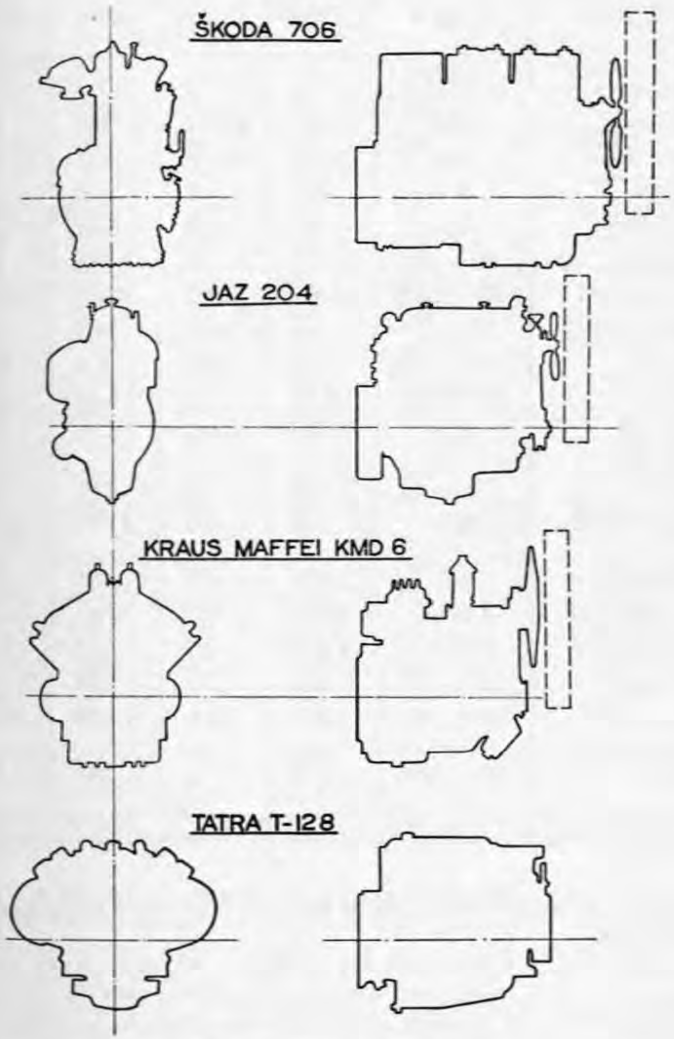


FIG. 26. Comparison of engine size.
Škoda 706 — water-cooled four-stroke, JAZ 204 — water-cooled two-stroke, Krauss-Maffei KMD 6 — water-cooled two-stroke, Tatra T-128 — air-cooled four-stroke

Table 4a

Comparative Table of Oil Engines for Motor-Cars

Manufacturer	Krauss-Maffei	Gräf & Stift	G. M.	JAZ 206	JAZ 206	FODEN	Tatra	Škoda	Praga	Hercules	Saurer	Cummins	Soviet Diesel
Type	KMD 6	6 ZT 180	4—51	series	max.	FD-6	111	706	RND	DIX 60	CVDL	NHS 600	W2
2- or 4-stroke	2	2	2	2	2	2	4	4	4	4	4	4	4
Bore, mm	120	120	104.14	108	108	85	110	125	105	92.08	110	130.2	150
Stroke, mm	130	140	104.14	127	127	120	130	160	130	101.6	140	152.4	180
Cylinders	4 V	6 V	4	6	6	6	12	6	4	6	12 V	6	12 V
Displ., litres	5.88	9.49	3.68	6.98	6.98	4.09	14.82	11.78	4.5	4.11	15.96	12.12	38.86
Performance, bhp	145 160	180	90	165	200	126	220	145	60	92	300	275	500
Speed, rev/min	2,200 2,500	2,000	2,800	2,000	2,000	2,000	2,250	1,800	2,000	3,000	2,200	2,100	1,800
Unit perf. bhp/l	25 27.6	19	24.7	24.0	28.7	30.8	14.9	12.3	13.3	22.3	18.8	22.7	12.8
p_e, kg/cm²	5.1 4.96	4.3	3.98	5.4	6.45	6.9	5.96	6.15	6.0	6.7	7.7	9.75	6.45
c_m, m/s	9.5 10.8	9.35	9.7	8.45	8.45	8.0	9.75	9.6	8.7	10.1	10.3	10.65	11.2
Weight, kg	735	695	442	750	750	499	986	930	415	354	1,150	1,590	750
Unit weight, kg/bhp	5.2 4.7	3.86	4.9	4.55	3.75	3.95	4.47	6.4	6.9	3.85	3.83	5.8	1.5
Max. length, mm	887	865	990	1,383	1,383	1,280	1,332	1,387	990	970	2,082 (1,778)	1,526	1,558
Max. width, mm	910	920	650	738	738	740	1,150	662	575	535	914	823	1,116
Max. height, mm	1,110	1,195	780	1,041	1,041	990	840	1,167	1,010	825	914	1,229	1,072
Total space, m³	0.895	0.955	0.5	1.06	1.06	0.94	1.25	1.07	0.575	0.427	1.740	1.545	1.87
Perf. bhp/m³	162 179	189	180	156	188	134	160	135	104	215	172	178	267
Supercharger	MC MC	MR	MR	MR	MR	MR	—	—	—	—	Bü	MR	—
Spec. fuel cons. g/bhp h	177 190	180	.	230	230	184	180	190	212	200	165	.	180

MR = Mechanically Driven Roots Blower, MC = Mechanically Driven Centrifugal Compressor, Bü=Büchi Turbo-Compressor, ( )=Dimension without Fan or Büchi.

improvement of specific values can be expected from air-cooled engines with superchargers.

Two-stroke engines are prominent by high performances per unit displacement and small weight, and compete successfully with four-stroke air-cooled engines. It must be borne in mind that two-stroke engines have double the number of power strokes and are partially supercharged. The charging pressure is not high, but it is always higher than back pressure of the exhaust gases.

As far as specific weight and required engine space are concerned standard air-cooled four-stroke engines exhibit satisfactory values. It should be remembered that dimensions and weights of water-cooled engines listed do not include radiators and weight of coolant.

Dimensional sketches of engines having approximately equal performances are shown in Fig. 26: Škoda 706, JAZ 204, Krauss Maffei KMD6 and Tatra T 128. All radiators are drawn in a smaller scale than engines. It is apparent, that air-cooled engines are the shortest; their overall width is increased by the

Table 4b

Various Two- and Four-Stroke Oil Engines

Manufacturer	Cummins	Mercedes Benz	MAN	Mercedes Benz	D. B.	D. B.
Type	JSX	OM 636	—	MB 507	MB 820 Aa	MB 820 Abs
2- or 4-stroke	4	4	4	4	4	4
Application	racing	passenger car	tank	tank	railway	railway
Bore, mm	104.8	75	130	158	172	172
Stroke, mm	127.0	100	165	180	205	205
Cylinders	6	4	16H	12 V	12 V	12 V
Displ., litres	6.54	1.77	35.1	42.2	57.0	57.0
Performance, bhp	345	40	690	720	860	1,000
Speed, rev/min	4,000	3,200	2,000	2,000	1,500	1,500
Unit perf., bhp/litre	52.9	22.6	19.7	17.0	15.1	17.5
p_e, kg/cm²	11.9	6.32	8.85	7.7	9.05	10.5
c_m, m/s	17.0	10.7	11.0	12.0	10.25	10.25
Weight, kg	380	190	1,545	900	2,400	2,700
Unit w., kg/bhp	1.1	4.83	2.24	1.25	2.79	2.70
Supercharger	MR	—	—	—	MC	Bü
Cooling	water	water	air	water	water	water

Manufacturer	Fairbanks	G. M.	Eng. Electric	Maybach	D. B.	Guiberson	Junkers	ZOD
Type	38 D 8 1/8	12–278	16 SVT	MD 650	MB 502	T 1020	Jumo 205	—
2- or 4-stroke	2	2	4	4	4	4	2	2
Application	railway	railway	railway	railway	tank	tank	aircraft	aircraft
Bore, mm	206.4	222	254	185	175	130	105	120
Stroke, mm	2×254	267	305.8	200	230	140	2×160	130
Cylinders	10	12 V	16 V	12 V	16 V	9 Radial	6	9 Radial
Displ., litres	170	125	246	64.5	88.46	16.74	16.62	13.32
Performance, bhp	2,000	1,200	1,800	1,200	1,330	265	600	260
Speed, rev/min	810	750	750	1,600	1,650	2,260	2,200	1,560
Unit perf., bhp/litre	11.7	9.6	7.35	18.6	15.0	15.8	36.0	19.6
p_e, kg/cm²	6.53	5.77	8.8	10.45	8.2	6.3	7.4	5.64
c_m, m/s	6.85	6.7	7.65	10.65	12.6	10.55	11.7	6.9
Weight, kg	14,900	10,880	15,870	3,600	1,870	290	520	321
Unit w., kg/bhp	7.45	9.0	8.8	3.0	1.41	1.1	0.87	1.23
Supercharger	MR	MR	Bü	Bü	—	—	MC	MC
Cooling	water	water	water	water	water	air	water	air

MR = Mechanically Driven Roots Blower, Bü = Büchi Turbo-Compressor, MC = Mechanically Driven Centrifugal Compressor.

fans located sideways. From the assembling point of view the length is the. decisive dimension of the engine, its width is less important. As a point of interest, the air-cooled eight-cylinder Tatra V engine is shorter than the four-cylinder two-stroke Krauss – Maffei engine with radiator. The Tatra T 128 engine is also of an older design.

Interesting conclusions can be drawn from comparing the dimensions of the two-stroke GM6–71 engine with air-cooled Tatra T 111 and T 603 types and with the Rover and Boeing gas turbines. Technical data of all types concerned are given in Table 5. Gas turbines are significant for their small dimensions and weights; they have, however, a high specific fuel consumption. One-stage rotary compressors will be used as a rule in gas turbines for automotive purposes. This will mean a pressure ratio of 3·5:1 and consequently a low thermal efficiency as in all small size turbines and compressors. Specific

fuel consumption can be lowered by using an efficient heat exchanger. However, heat exchangers have large dimensions and so the advantage of small overall dimensions of the gas turbine would be lost.

The Boeing gas turbine has no heat exchanger. The Rover gas turbine uses a partial heat exchanger, and the overall dimensions of the unit are larger. Dimensions are shown in Fig. 27. By comparing the weights and specific fuel consumption given in Table 5, it can be seen that the use of gas turbines for passenger cars is not as advantageous as it seemed at first sight.

The automobile of the future must be lighter and more economical. Air-cooled engines have all properties to fulfil the requirements of future designs and therefore, it may be expected that the general application of air-cooled engines in aircraft and motor-cycles will be followed by their use in automobiles, too.

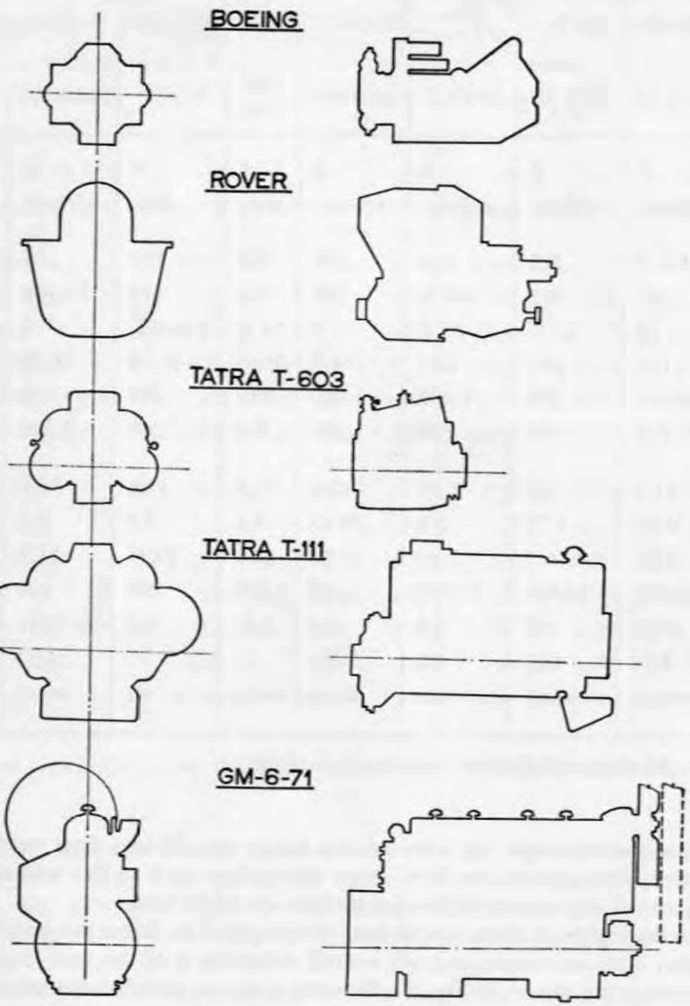


FIG. 27. Comparison of engine size with combustion turbines listed in table 5.

Table 5

Comparative Data of Engines shown in Fig. 27.

	Boeing	Rover	Tatra 603	Tatra 111	GM 6–71
Normal perf. bhp	—	150	100	195	165
at rev/min	—	—	4,800	2,000	2,000
Maximum perf., bhp	175	200	200	220	200
at rev/min	36,000	40,000	7,500	2,250	2,000
Total weight, kg	90.7	204	160	986	750
Compression ratio	2.88 : 1	3.5 : 1	12 : 1	17 : 1	16 : 1
Spec. fuel cons. g/bhp h	670	537	250	175	230
Unit weight kg/bhp	0.52	1.02	0.80	4.47	3.75

CHAPTER II

FUNDAMENTALS OF HEAT TRANSMISSION

1. Transmission of heat from a solid body to air

If an air stream flows along the surface of a solid body, individual molecules of air collide with the solid surface and thus heat is conveyed from the surface to the air stream. In order to explain this phenomenon, let us examine the character of an air stream along the solid surface. [20]

Heat transfer is a very complex process and its seemingly simple expression by the Newtonian equation does not give results corresponding to those obtained by practical measurements.

The coefficient of heat transfer from a wall surface to a gaseous medium is a function of many variables, such as the shape and dimensions of the surface, direction and velocity of gas flow, temperature, density, viscosity, specific heat and thermal conductivity of the gas.

Differential equations expressing heat transfer have been successfully solved for extremely simplified assumptions only. Empirical methods have supplied no satisfactory results either, because of the existence of many variables, and generally valid relationships have been almost impossible to establish. Today the principle of similitude is the most reliable method by which to treat and generally apply experimental data on heat transfer by convection.

In theoretical studies viscosity of the medium (air) is not considered and velocity of the air stream is assumed to be equal throughout the whole cross section of the stream. Due to the viscosity of air, however, velocity decreases in the direction towards the solid surface till it becomes zero at the interface. (Fig. 28).

Thus a thin boundary layer is formed on the solid surface in which velocity is rising from zero up to the value of v i.e. the mean velocity of the surrounding medium. Fig. 29 shows the distribution of velocities at various distances x from the solid surface. The velocity distribution curve (velocity profile) is obtained by plotting actual velocities v in the boundary layer against distance x from the wall surface.

The shear stress between two adjacent layers can be expressed by an equation based upon Newton's work on internal friction. Thus:

$$\tau = \eta \frac{dv}{dx}$$

where τ denotes the tangential force (viscous shear)
η „ „ coefficient of viscosity
$\frac{dv}{dx}$ „ „ velocity gradient

Viscosity, as a property representing the reluctance of the medium to yield to tangential forces acting between adjacent layers, has been explained theore-

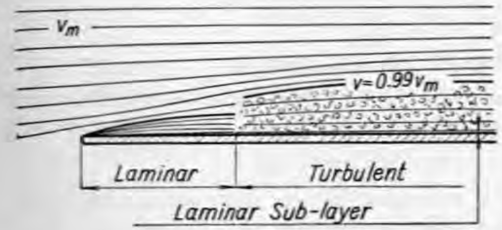


FIG. 28. Representation of the boundary layer on the surface of a solid body.

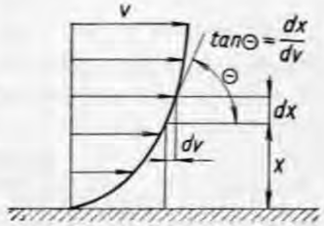


FIG. 29. Velocity profile of the boundary layer.

tically by the momentum transfer of molecules passing from one layer to another. Viscosity is, thus, physically a dissipation process of the same nature as heat convection; both are based on movements of molecules between two adjacent layers.

If the flow in the boundary layer is characterized by individual layers of the medium sliding one over the other at different velocities, in planes almost parallel to the solid surface, and with negligible transfer of molecules between the layers, we have the case of a laminar flow. In this case, heat transfer from one layer into the adjacent one, in a direction normal to the flow, is possible only by conduction.

Heat transfer from the solid surface to air [2] is in direct relation to surface friction. The correlation is expressed by the equation

$$Q = \frac{F c_p}{v_m} (t_b - t_a) \tag{1}$$

where Q denotes the heat transmitted from 1 cm^2 of the surface in one second
F „ „ surface friction (tangential force) per cm^2
c_p „ „ specific heat of air at constant pressure
v_m „ „ mean velocity of flow
t_b „ „ mean temperature of the surface
t_a „ „ mean temperature of air

It is evident from equation (1) that heat transmission, at a given velocity v_m and temperature difference $t_b - t_a$, depends on surface friction. The value of surface friction is directly related to the mean velocity of air v_m. In the majority of practical cases it is proportional to $\varrho \cdot v_m^2$,

where $\varrho = \dfrac{\gamma}{g}$ denotes the density of air (kg s²/m⁴)

γ „ „ specific gravity of air (kg/m³)

g „ „ gravitational acceleration (m/s²)

Consequently we may write $F = k \cdot \varrho \cdot v_m^2$

where k denotes a constant established by measurement.

By substituting the expression for F in equation (1), heat transmitted per unit of surface is

$$Q = k \cdot \varrho \cdot c_p \cdot v_m \cdot (t_b - t_a) \tag{2}$$

The above expression based upon surface friction represents a safe calculation of transmitted heat. Heat quantities established by practical measurements are invariably higher.

Regardless of the value of the constant k, the relations of friction, velocity, and heat transfer have been established by exact experiments. Surface friction can be expressed in relation to v_m^n, where $n = 1{\cdot}5$ for laminar flow and $n = 2$ for turbulent flow. Heat transmitted can be thus expressed in relation to v_m^{n-1}.

2. Type and thickness of the boundary layer

Several types of flow appear in the boundary layer. Laminar flow with layers sliding one over the other parallel to the solid surface exists closest to the solid body. Movement of molecules normal to the flow direction is negligible, and, therefore, heat transmission in the zone of laminar flow is unsatisfactory. As soon as an irregular transverse movement and eddy diffusion of molecules takes place in the layer, the flow becomes unsteady and is called a turbulent flow. With this type of flow an intensive exchange of molecules takes place between the zone close to the surface and the main stream. Transfer of heat from the solid surface is, therefore, considerably better.

If an air stream is directed against a very thin plate, a boundary layer is formed on the surface of the plate. The thickness and shape of the boundary layer have been studied by numerous authors. Generally the following conclusions have been reached [48]:

The thickness of the boundary layer increases gradually from the leading edge. The flow in the boundary layer is laminar. At a certain distance from the leading edge the thickness of the boundary layer increases suddenly. The abrupt increase of thickness is correlated with the transition of the laminar flow into a turbulent one. The transition occurs at the Reynolds number $\mathrm{Re} = 2$ to 3.10^5.

The Reynolds number is given by the expression $\mathrm{Re} = \dfrac{v_m \cdot l}{\nu}$

where v_m denotes the mean velocity of air (m/s)

l „ „ characteristic linear dimension, in this case the distance from the leading edge (m)

ν „ „ kinematic viscosity of air, $\nu = \dfrac{\eta}{\varrho}$ (m²/s)

η „ „ absolute viscosity of air (kg s/m²)

ϱ „ „ density of air (kg s²/m⁴)

The thickness of the boundary layer is small and depends on the velocity and kinematic viscosity of air. The actual thickness of the boundary layer is difficult to determine, because the velocity gradient decreases gradually. For further considerations in this chapter let us assume the thickness of the boundary layer as being limited by the surface and the layer where the linear velocity $v = 0{\cdot}99\, v_m$.

According to Blasius the thickness of the boundary layer with a laminar flow is given by the relation

$$x_1 = 4{\cdot}5 \left[\frac{l \cdot \nu}{v_m}\right]^{0{\cdot}5}$$

Let us divide both sides by l:

$$\frac{x_1}{l} = 4{\cdot}5 \left(\frac{\nu}{v_m \cdot l}\right)^{0{\cdot}5} = 4{\cdot}5 \left(\frac{v_m \cdot l}{\nu}\right)^{-0{\cdot}5}$$

and write the following equation:

$$\frac{x_1}{l} = 4{\cdot}5\, (\mathrm{Re}_l)^{-0{\cdot}5} \tag{3}$$

The thickness of the boundary layer with a turbulent flow is calculated by a semi-empirical formula stated by von Kármán

$$\frac{x_t}{l} = 0{\cdot}2\, (\mathrm{Re}_l)^{-0{\cdot}15} \tag{4}$$

However, a thin laminar sub-layer exists close to the solid surface in turbulent boundary layers. The thickness of the film does not depend on the distance from the leading edge, and it may be calculated from the formula

$$x_f = \frac{200\, \nu}{v_m} \tag{5}$$

In a turbulent boundary layer air velocity changes rapidly close to the solid surface, and the change becomes gradually slower towards the main air stream. The thickness of the boundary layer from the solid surface up to a layer with a velocity 0·8 V_m is thus considerably smaller as shown in Fig. 30, where the total thickness of the boundary layer $v = 0{\cdot}99\ v_m$ is depicted The values of velocities were established by actual measurements on an air foil.

Air velocities were also measured on a metal sheet 0·6 mm thick and 45 mm long. The sheet was coiled into a hollow cylinder which was placed parallel to an air stream flowing at a velocity of 39·5 ft/s (12·2 m/s).

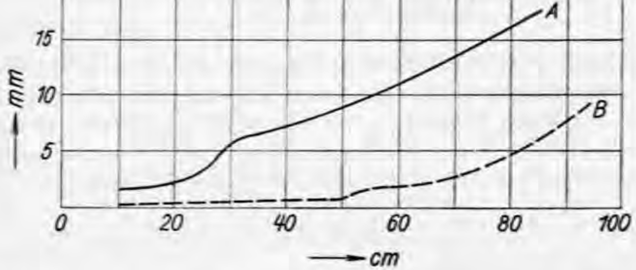


FIG. 30. Thickness of the boundary layer at a velocity of 25 m/s.
A — total thickness of the boundary layer up to $v = 0.99\ v_m$, B — distance of the layer $v = 0.8\ v_m$ from the surface.

Velocities were measured behind the sheet, not on the side, and the results are interesting in showing to what extent air velocity is influenced beyond a metal sheet having approximately the thickness of a cylinder fin of an aircraft engine. The measurements proved, that velocity was reduced in a 3·15 mm high zone. By deducting the sheet thickness of 0·6 mm, the thickness of the boundary layer was established as being 1·275 mm (measured closely behind the sheet). Halfway in the boundary layer, i.e. 0·64 mm from the solid surface, air velocity was reduced by 5% only, and at a distance of 0·3 mm from the surface it was still higher than 0·8 v_m.

The results obtained by the above measurements are almost in complete agreement with calculated values. According to equation (4) for $l = 45$ mm the thickness of the boundary layer amounts to $x_1 = 1{\cdot}05$ mm; the measured thickness $x_1 = 1{\cdot}10$ mm.

Owing to the fact that heat transfer is more intensive through a turbulent boundary layer, it is important to know at what distance from the leading edge of the fins of an air-cooled engine laminar flow changes to a turbulent one. The sooner such change occurs, the larger the fin surface area exposed to turbulent flow and the better the average heat transfer per unit of the surface area. With thicker fins and air flows oblique to the fin surface, transition from laminar to turbulent flow takes place sooner than according to calculation, and the thickness of the boundary layer at a given distance from the leading edge is larger.

3. Surface friction

Friction on the surface of a thin plate depends on the Reynolds number and also on the type of flow. For laminar flow Blasius stated that the coefficient of surface friction should be calculated from the following equations:

$$\mu_{sl} = 2{\cdot}90\ \mathrm{Re}_x^{-0{\cdot}6} \quad \text{for } 10 < \mathrm{Re}_x < 10^3$$

$$\mu_{sl} = 1{\cdot}328\ \mathrm{Re}_x^{-0{\cdot}5} \quad \text{for } 2\times10^4 < \mathrm{Re}_x < 5\times10^5 \tag{6}$$

In the case of a turbulent flow, conditions differ mainly in the following respects:

1. Surface friction with turbulent flow is greater than that with laminar flow.
2. Thickness of the boundary layer is greater than in the case of laminar flow and grows more rapidly with the distance l from the leading edge.
3. Velocity increases with the distance from the solid surface more rapidly than in a laminar flow (Fig. 31).

The coefficient of friction in turbulent flow can be calculated from the following formula (von Kármán):

$$\mu_{st} = 0{\cdot}074\ \mathrm{Re}_x^{-0{\cdot}2} - A\,.\,\mathrm{Re}_x^{-1} \tag{7}$$

where $A = 1700$ for $\mathrm{Re} = 5{\cdot}3\times10^5$

$A = 600$ for $\mathrm{Re} = 1{\cdot}9\times10^5$

$A = 300$ for $\mathrm{Re} = 10^5$

Transition from laminar to turbulent flow depends on the prevailing conditions. The place on a flat plate at which transition occurs is called the transition point, though transition takes place in a certain zone rather than in a point. The transition point determined by the Re value depends on the smoothness of the surface, the turbulence of the streaming air, the angle of the air stream, etc.; the corresponding range of Re numbers is $\mathrm{Re} = 2\times10^5$ to 2×10^6.

For a given plate the Reynolds number, $\mathrm{Re} = \dfrac{v_m\,.\,l}{\nu}$, of the transition point remains constant. Therefore, the distance l of the transition point from the leading edge decreases with increasing velocity v_m. This means also that a change of flow velocity causes an alteration of the ratio of areas exposed to laminar and turbulent

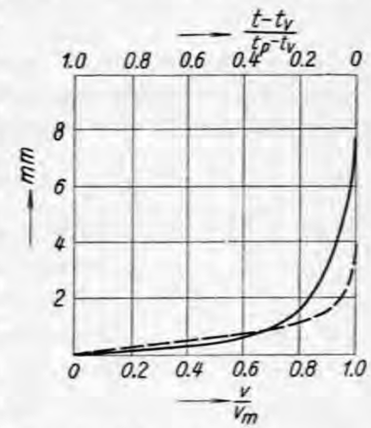


FIG. 31. Velocity and temperature gradient in the boundary layer.
Dashed curve — laminar flow, full curve — turbulent flow.

flow. The position of the transition curve depends on the degree of turbulence of the free flow. In a flow without turbulence a high value of Re is critical, and, on the contrary, in a turbulent flow a low value of Re is critical. The transition from laminar to turbulent flow occurs at the critical Re value.

Surface friction is strongly influenced by the unevenness of the surface. The correlation between surface friction and unevenness of the surface was

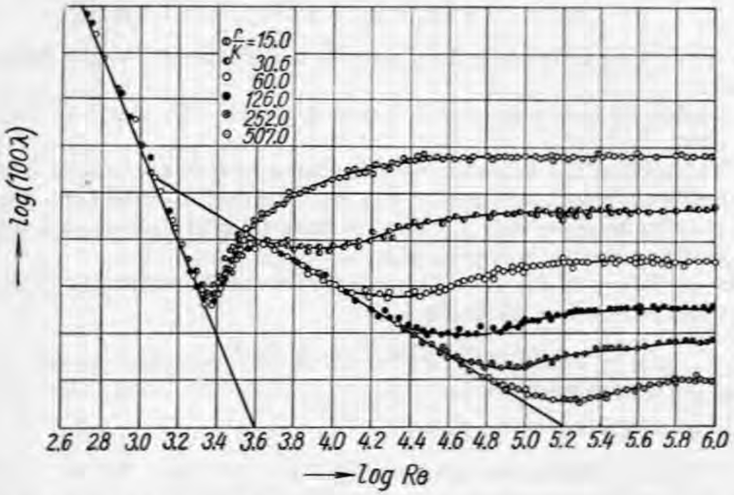


FIG. 32. Relationship between coefficient of resistance and Reynolds number for various values of relative surface roughness.

investigated by Nikuradse by examining an air stream in a tube having an internal wall of varying roughness. The loss of pressure head in the test tube can be expressed by the equation:

$$\Delta p = \frac{\lambda \cdot l}{d} \frac{\varrho}{2} v_m^2 \qquad (8)$$

where l denotes the length of the tube (m)

d „ „ internal diameter of the tube (m)

ϱ „ „ density of air (kg s²/m⁴)

v_m „ „ mean linear velocity of the air (m/s)

λ „ „ resistance coefficient

The resistance coefficient λ depends not only on the Reynolds number but also on relative roughness of the surface

$$R_r = \frac{r}{k}$$

where r denotes the radius of the tube and

K „ „ mean height of projections

This correlation is illustrated in Fig. 32 where the values of log (100 λ) are plotted against log Re for various values of relative roughness R_r.

The diagram shows that in a laminar flow up to log Re = 3·4 surface friction is not influenced by the surface roughness, because projections of the uneven wall are covered by the laminar boundary layer. If the flow becomes turbulent, the influence of roughness, with projections protruding from the laminar layer, is being felt progressively with an increasing Re. At Re = = 100,000 and larger the boundary layer is reduced to a thin film covering only projections of R_r = 507 or smaller.

As mentioned before, transmission of heat from the surface depends to a very large extent on surface friction. Theoretically, the quantity of heat Q' transferred from a plate of a width b and length l with a laminar flow on the surface, can be computed from the equation

$$\frac{Q'}{b \cdot k \cdot (t_b - t_a)} = \mu_{sl} \frac{v_m l}{\nu}$$

Measurements carried out on a very thin plate having a width b and length l with a laminar flow on the surface have shown, that the quantity of heat actually transferred is given by the relation

$$\frac{Q'}{b \cdot k \cdot (t_b - t_a)} = 1{\cdot}50 \left(\frac{v_m \cdot l}{\nu}\right)^{-0{\cdot}5} \qquad (9)$$

where k denotes the thermal conductivity of air

t_b „ „ temperature of the surface

t_a „ „ temperature of air

The tests were carried out with plates of four different lengths with l varying within the range 12·7 to 3·3 mm constant width b = 12·7 mm and mean linear velocities varying from 4·8 to 0·75 m/s.

The constant 1.50 established by measurement approximates very closely the theoretical constant in the preceding equation (see also equation 6).

The following equation is valid for turbulent flow:

$$\frac{Q'}{b \cdot c_p \cdot (t_b - t_a)} = \mu_{st} \cdot \varrho \cdot v_m \cdot l = \eta \cdot \mu_{st} \frac{v_m l}{\nu} \qquad (10)$$

where c_p denotes the specific heat of air at constant pressure

$\eta = \nu \cdot \varrho$ „ „ absolute viscosity of air

The constants μ_{sl} and μ_{st} in the respective equations depend upon the Reynolds number; for laminar flow $\mu_{sl} \propto Re^{-0{\cdot}5}$ and for turbulent flow $\mu_{st} \propto Re^{-0{\cdot}15}$. Therefore, at a given mean velocity v_m surface friction is more regular in the case of a turbulent flow, because μ_s changes with the distance l

more slowly than in the case of a laminar flow. This consideration holds for a very thin plate placed exactly parallel to the flow direction.

For cooling fins of air-cooled engines it is important to know the point at which the transition from laminar to turbulent flow occurs. Fins should be spaced so as to avoid any noticeable overlap between the boundary layers of two opposing fins.

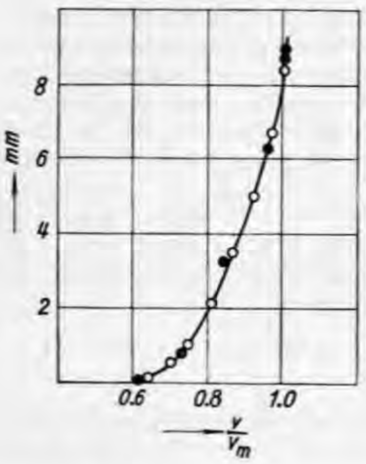


FIG. 33. Velocity and temperature gradient in the boundary layer.

As stated previously, heat transfer is correlated with the tangential force F acting on the surface (surface friction). This assumption has been confirmed by measurements of the temperature difference in the boundary layer, according to which temperature difference is in complete harmony with the velocity gradient. Results of measurements of temperature and velocity in the boundary layer are shown in Fig. 33. Full dots represent velocities, while temperatures are shown by circles. It can be seen that measurement results of both values are in complete conformity.

At the leading edge of a fin exposed to an air stream the thickness of the boundary layer is small, and the flow is laminar. Heat is transferred mainly by conduction and with increasing thickness of the boundary layer heat transmission becomes worse.

The afore described analysis of flow types leads to the following conclusion: heat transfer from a solid surface is, in spite of the greater thickness of the boundary layer, better in a turbulent flow than through the thinner boundary layer of a laminar flow.

The influence of surface roughness on heat transfer was tested on a cylinder of an air-cooled engine. The surface of the fins was roughened by punches 0.5 mm deep and 1·8 mm apart. No substantial difference in heat transfer was found as compared with a fin of smooth surface. It was also proved that, contrary to previous expectations, the rapid vibration of fins of an air-cooled engine had no influence on heat transfer.

4. The surface coefficient of heat transfer

The quantity of heat transferred from a solid surface to a surrounding gas is determined by the basic equation:

$$Q = q \,.\, \Delta t \,.\, A$$

where A denotes the area of the cooling surface (m²)
Δt „ „ temperature difference between surface and gas (°C)
q „ „ surface coefficient of heat transfer (kcal/m² h °C)

The surface coefficient of heat transfer from wall to air, or for a reversed transfer, depends upon the velocity of air, specific pressure, specific weight, specific heat and viscosity of air, the roughness of the surface and absolute temperature. Therefore, it cannot be expressed by a simple formula. Some approximate formulas are used in the following sections.

Exhaustive measurements of the surface coefficient of heat transfer were carried out on an electrically heated cylinder with copper fins having a diameter of 114·3 mm. In this case the surface coefficient of heat transfer q was found to comply with the following formula:

$$q = 1{\cdot}18\,(1 + 0{\cdot}0075\, T_m)\, v_m^{0\cdot73} \tag{11}$$

where T_m denotes the arithmetic mean of absolute temperature of the fin surface and air, i.e. the absolute mean temperature of the boundary layer (°C)
v_m „ „ mean air velocity (km/h).

Measurements were made at fin temperatures ranging from 30 °C to 155 °C and air temperatures from 15 to 18 °C. Internal diameter of fins amounted to 114 mm, outer diameter 146 mm, thickness of fins 0·55 mm and spacing 8 mm.

The correlation of the surface coefficient of heat transfer and absolute temperature was established by measurements at a constant air temperature of 15 °C and fin temperatures of 105 and 155 °C. The corresponding absolute mean temperatures are 333 and 358 °C. The increase of the temperature by 50 °C meant an increase of the temperature factor (parenthesized term in equation 11) from 3·50 to 3·68, i.e. by 5% which was exactly the relation by which the quantity of heat transferred was increased.

Fig. 34 shows values of the surface coefficient of heat transfer obtained by measurements on an even surface 152 × 152 mm having a temperature of 65 °C and exposed to an air stream of 20 °C. In this case the transferred heat was proportional to the 0·725th power of the mean air velocity v_m.

Other measurements on even fins 162 mm long gave results of coefficients being proportional to $v_m^{0\cdot747}$.

Thus the value $v_m^{0\cdot73}$ can be regarded as an entirely satisfactory

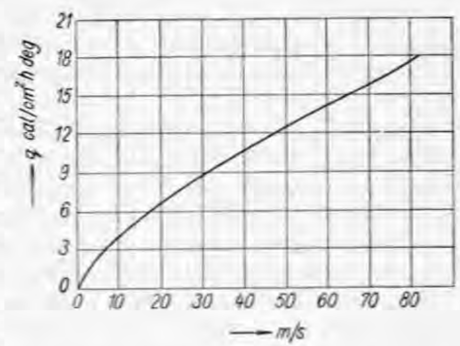


FIG. 34. Surface coefficient of heat transfer measured on an even surface in relation to air velocity.

average applicable in calculations of normal cylinders of air-cooled engines. It should be borne in mind that the heat transfer coefficient for laminar flow is proportional to $v_m^{0.5}$, for turbulent flow to v_m. In the case of fins 152 mm long, exposed to an air stream having a velocity of 65 to 213 ft/s (20—65 m/s), the surface area exposed to laminar flow varies between $^1/_4$ and $^3/_4$ of the total surface area, and therefore, the exponent 0·73 can be regarded as a satisfactory compromise.

CHAPTER III

HEAT TRANSFER FROM HOT GASES TO CYLINDER WALLS

Heat transfer from the combustion products to the cylinder walls is the basis for the calculation of engine cooling. The state of thermal balance requires that the quantity of heat removed from the cylinder walls equals the quantity conveyed to it. As stated in the preceding chapter, heat transfer from gases to the wall is determined by the equation:

$$Q = q \cdot \Delta t \cdot A \text{ (kcal)}$$

Previous considerations are confined to the influence of the absolute temperature T, which affects also air density at atmospheric pressure, and the influence of the mean air velocity v_m. It has been established that owing to the dominating influence of the above two values, heat transfer can be calculated, with sufficient accuracy, in all practical cases according to equation (11), in spite of the fact that the velocity exponent 0·73 is only a compromise and its value varies with changing conditions.

Heat transfer from hot gases to cylinder walls in an internal combustion engine proceeds under conditions far more complicated than those existing with a regular air flow along a solid wall. Basically the coefficient of heat transfer depends on the specific pressure, specific weight, and specific heat of the gas as well as on the gas velocity and roughness of the wall surface. Exact calculations should, furthermore, consider also the coefficient of heat transfer by radiation q_r.

It has been established by experiments, that, in internal combustion engines, the portion of heat transmitted by radiation amounts to about 2—6% only. Therefore, in further calculations heat transferred by radiation is not considered. The coefficient of heat transfer to the cylinder walls is difficult to measure. The influence of gas velocity, being of great importance for the calculation of the coefficient, is difficult to establish, for velocity is different at various parts of the wall and cannot be measured by conventional methods. Moreover, gas pressure in the cylinder varies within a wide range, and its effect upon the density and transfer coefficient must be taken into account. The best known equations used for calculating the coefficient of heat transfer contain relations to gas pressure p (kg/cm^2), absolute gas temperature T (°K), absolute temperature of the cylinder wall T_e (°K) and mean piston velocity

c_m (m/s). The latter is, to some extent, correlated to the velocity of gas flowing along the wall.

The quantity of heat transferred to a cylinder wall in an interval of time dt, amounts to

$$dQ = q \cdot A\,(T - T_c)\,dt$$

where, according to Nusselt,

$$q = 0{\cdot}99\,(1 + 1{\cdot}24\,c_m)\cdot\sqrt[3]{p^2 \cdot T}$$

Nusselt was aware of the limited validity of this equation, and recommended a mean piston velocity of 16 ft/s (5 m/s) as the upper limit of velocities for which the equation should be applied. The transfer coefficient is largely influenced by the turbulence of gases in the cylinder. Nusselt, Jacklisch and others have assumed that the intensity of turbulence depends on the mean piston velocity c_m. For existing automotive engines this assumption is not proved to be correct even approximately.

Turbulence in petrol engines depends largely on the width of the surface area between cylinder head and piston. In compression-ignition engines turbulence is determined by the tangential flow of air entering the cylinder during the induction stroke and further intensified by compressing the air in the recess of the piston head. In pre-combustion chamber engines intensive turbulence is caused by the combustion products entering the cylinder by the chamber port. Thus, intensity of gas turbulence in the cylinder can vary considerably regardless of mean piston velocity.

According to Briling the influence of turbulence on the transfer coefficient can be expressed by the formula

$$q = 0{\cdot}99\,(1 + d + 0{\cdot}185\,c_m)\cdot\sqrt[3]{p^2 \cdot T}$$

where $d = 0$ for petrol engines, and $d = 1{\cdot}45$ for compression-ignition engines. In this modification of the Nusselt formula the value d expresses the intensity of gas turbulence. It is evident that d is different for different types of combustion chamber, and, for a newly designed engine, it can be determined only roughly according to the theory of similitude. Bryzov performed experiments on various engines with the following results: $d = 3{\cdot}5$ for an engine of 140 mm bore at 800 rev/min, and $d = 4{\cdot}2$ for a 100 mm cylinder bore engine having a speed of 1,100 rev/min. These results of practical measurements clearly indicate the large degree of approximation which is bound to appear in calculations. Difficulties of calculations are caused also by the variability of the internal cylinder surface area exposed to hot gases; the different temperatures of combustion chamber walls, cylinder walls, piston faces and valves; the insulating effect of the oil film on the cylinder wall and the same effect of carbon deposited on combustion chamber walls and pistons.

The following method by Janevay is a good example of how to determine the heat transfer coefficient q of a petrol engine cylinder.

The variability of cylinder surface area is comparatively easy to determine from the known engine dimensions. Cylinder surface area plotted against crank position is shown in Fig. 35. The curve represents conditions of an engine of the following characteristics: bore 82·55 mm, stroke 108 mm, side valves, compression ratio 5·4 : 1 [27].

Temperature of the gases and that of the cylinder wall must be established as data for the calculation of the temperature difference. For an accurate determination of the gas temperature, we must consider the different exponents n of the polytropic compression and expansion curves. Fig. 36 shows computed temperatures for different exponents n plotted against various crank positions of the same engine.

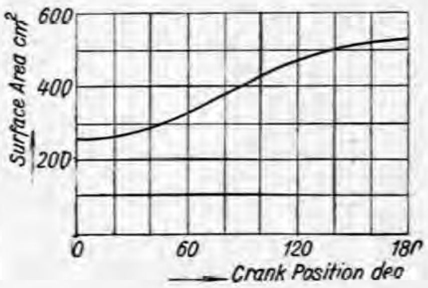


FIG. 35. Cylinder surface area exposed to hot gases plotted against crank position.

Great difficulties are encountered in determining the mean temperature of the cylinder wall. The mean wall temperature varies in the course of a single stroke as new and colder sections of the surface area are uncovered by the piston, and the wall temperature of the combustion chamber also varies within a narrow range. An accurate calculation of the mean wall temperature is thus hardly possible. Therefore the mean temperature of the cylinder wall is determined by practical estimating and measuring. If the maximum gas temperature during combustion is 1500 °C, the corresponding mean temperature of the inner cylinder wall surface is 150 °C. A fluctuation of this value within the range of, say, $\pm$ 28 °C represents an error of $\pm$ 2% in the calculated heat transfer, and this is sufficiently accurate.

Measurements at various parts of the cylinder surface indicated a mean temperature of the cylinder wall 150 °C. This temperature can be considered as being independent on the engine revolutions. At increased revolutions the quantity of heat transmitted to the walls is increased by the amount of heat produced by friction; however, due to the intensified circulation of the cooling medium heat is removed more rapidly and so the temperature of directly cooled parts remains practically constant.

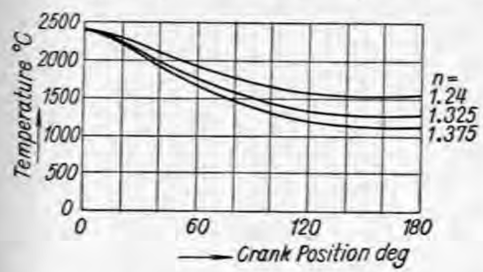


FIG. 36. Gas temperature during the expansion stroke related to crank position for various values of the exponent n.

Having determined the surface area and the temperature difference, we may enter both values into the diagram. The product of both values plotted against the crank position during the compression stroke is shown in Fig. 37. At the beginning of the compression stroke the air-fuel mixture is cooler than the walls, heat transmission is negative, i.e. heat is passed from the walls to the mixture. The overall transfer in the course of the stroke is positive and signifies the heat loss during compression.

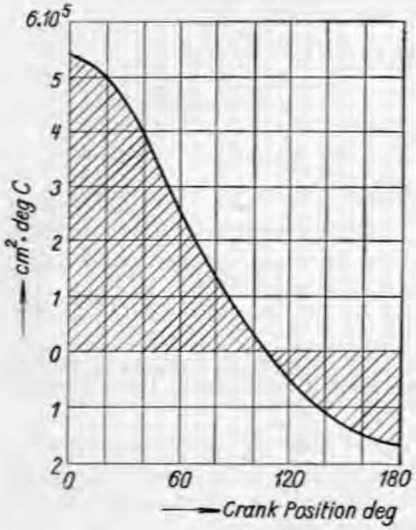


FIG. 37. Thermal losses during the compression stroke.

At a constant value of the heat transfer coefficient, the resultant positive area of the diagram represents the total heat removed.

The corresponding diagram for the expansion stroke is shown in Fig. 38. It is evident that the product of surface area and temperature difference remains practically the same during the expansion stroke, because the temperature drop in the expanding gas is compensated for by the increasing surface area of the cylinder wall. The next step is the determination of the coefficient of heat transfer from the hot gases to the cylinder wall which depends on the velocity of gas flow along the wall. The correlation between transfer coefficient and velocity originates in the diffusion of molecules in the boundary layer at the cylinder wall. Heat transfer is proportional to friction losses in the boundary layer and it is represented by the number of molecules diffused in unit time or, in other words, the product of mass velocity and density. Thus in exact calculations the heat transfer coefficient q must be considered as being dependent not only on the velocity but also on the density of gas. This fact is important for combustion conditions at dead center position, where q is consequently larger than during compression or expansion.

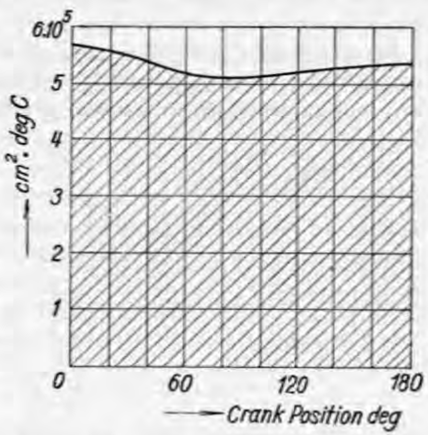


FIG. 38. Thermal losses during the expansion stroke.

The correlation between the coefficient of heat transfer q measured during expansion and speed is illustrated in Fig. 39. The curve clearly indicates how q rises with increasing speed. The exactness of the correlation is even more marked by plotting the same curve on a log-log scale. In order to include in this diagram the relation between q and gas density the diagram is drawn with the product of the speed n and the volumetric efficiency η_v entered on the horizontal axis (Fig. 40). Four computed points of the correlation lie on a straight line having a slope 0·526, and this is in accordance with the relation

$$q = \alpha\,(n\,\eta_v)^{0\cdot526} \qquad (12)$$

where n denotes the engine speed in rev/min

η_v „ „ volumetric efficiency

α „ „ constant

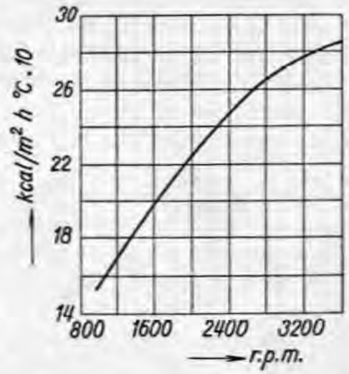


FIG. 39. Change of coefficient of heat transfer q with engine speed.

Measured values of q for compressed gases flowing through tubes under a pressure of 551·8 lb/in² (38·8 kg/cm²) are drawn in the same diagram (Fig. 40) for the sake of comparison. Results of measurements are plotted against the product of speed and density. The resultant exponent equals 0·578. In spite of the fact that the value q is in each case plotted against a different value, the correlation is evidently of the same type. The larger q value in the engine is explained by the more intensive turbulence in the engine compared with that in the tube.

In order to reduce heat lost by cooling to a minimum, the ratio of the surface area to the volume of the combustion chamber must be as small as possible. This condition is best fulfilled by a semi-spherical combustion chamber which has the additional advantages of sufficient space for large valves, good flow conditions through the valves, etc. In flat combustion chambers, particularly with side valves, heat losses by transfer to the walls are always considerable: they reduce the overall engine efficiency and increase the specific fuel consumption. Larger

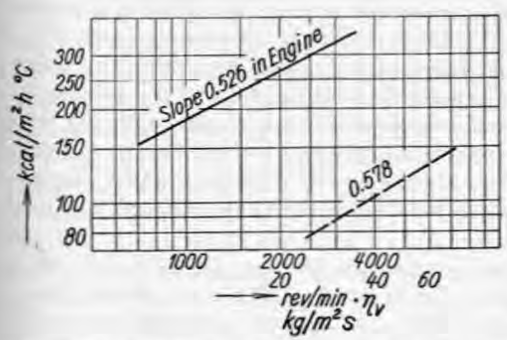


FIG. 40. Dependence of coefficient of heat transfer q on the velocity and density of gases. Full line represents tests in engine, dashed line tests with compressed air in a tube.

cooling surfaces require more power input to the cooling system. For this reason side valve designs are being abandoned in vehicle engines despite lower production cost.

Recently several F-head designs have appeared with the intake valve located in the cylinder head above the piston and the exhaust valve seated in the side wall of the cylinder in an oblong chamber. With this type of valve arrangement it is possible to design the combustion chamber for knock-free combustion even at high compression ratios. A high compression ratio ensures economical operation, particularly at partial loads, and the advantageous combustion process produces a smooth operation. However, the large surface area of the combustion chamber signifies large heat losses and a large cooling radiator.


FIG. 41. Well designed combustion chamber in piston of oil engine.

The requirement of small surface area holds for pistons and valves, too. Smallest heat losses occur with flat-head pistons. A piston with a convex face or deflector transmits more heat; its temperature is higher and the piston rings are subjected to higher stresses. This is one of the reasons for the growing unpopularity of pistons with deflectors for two-stroke engines. Convex pistons with valve recesses are equally inefficient and should be regarded as an inherent evil for engines of a high compression ratio.

Pistons of compression-ignition engines must also have compact combustion chambers with the proper surface area:volume ratio. It is desirable to remove recesses from the piston and place the valves in the cylinder head. This arrangement reduces heat losses during combustion and also during compression, giving an easy start without an auxiliary warming up of the engine. A well designed piston for a compression-ignition engine is illustrated in Fig. 41.

Valve heads, particularly those of exhaust valves, should be flat, so that the surface area exposed to hot gases is kept at the minimum. Tulip-shaped valves have a low weight, but give higher temperatures in operation and are easily distorted.

Turbulence is the second factor assisting the heat transfer from hot gases to the cylinder walls. A high turbulence of the air-fuel mixture during combustion causes a larger heat transfer to the walls of the combustion chamber. Ricardo [51] recognized this fact when introducing high turbulence into combustion chamber design. Higher heat transfer has a beneficial effect during compression, temperature is reduced towards the end of the compression stroke and the application of a higher compression ratio is made possible with

subsequent advantages. On the other hand higher heat losses occur which leads to higher specific fuel consumption and to larger dimensions of radiators.

The design and shape of the combustion chamber has an important bearing upon heat loss. In the design of combustion chambers of modern spark ignition engines increased attention is paid to the course of combustion or the variation of the flame face. In compression-ignition engines, on the other hand, turbulence is required for the proper mixing of air and fuel as well as for combustion control. Here the disadvantage of higher heat loss is partially remedied by direct fuel injection.

It is evident, that heat losses are influenced also by the temperature of the combustion gases. In high compression ratio engines and in compression-ignition engines heat losses are thus higher during combustion with the piston in dead centre position. This disadvantage is compensated for by the smaller surface area of combustion chambers operating at a higher compression ratio. Due to the increased expansion the temperature of exhaust gases is reduced and so also is the heat transfer to the cylinder walls.

Thus an increasing compression ratio means an increasing ratio of the heat transferred to the cylinder head to that transferred to the cylinder walls. However, the overall quantity of heat transferred decreases with an increasing compression ratio and the thermal efficiency of the engine is higher.

Small heat losses are particularly important for compression-ignition engines. Therefore the surface area of the combustion chamber in the cylinder head should be the smallest possible. This design leads to a larger surface area of the combustion space on the piston crown, where temperatures are consequently higher. However, this is beneficial for combustion in compression-ignition engines.

In spark ignition engines it is more desirable to have a flat-crown piston and the combustion chamber in the cylinder head only. The reverse is true for compression-ignition engines. In both cases, as stated before, the shape of the combustion chamber should give the smallest possible surface area to volume ratio.

The shape and wall surface of the exhaust duct in the cylinder head have a definite bearing on the quantity of heat removed by cooling. The exhaust gases leave the cylinder at high temperatures and a considerable quantity of heat is transferred from them to the cylinder head. Therefore, exhaust ducts in the cylinder head should be short and straight. The heat transferred from the exhaust duct wall to the cylinder head does not influence the thermal efficiency of the engine, but it does represent an extra load on the cooling system. The increased work of the cooling system means a reduction in the mechanical efficiency and higher specific fuel consumption.

The removal of this heat from air-cooled engines requires robust finning on the parts adjacent to the exhaust duct, which is a design problem often difficult to solve. In water-cooled engines the radiator area must be increased, and this means additional difficulties particularly in aircraft engines.

The heat transmitted through the walls of the exhaust duct reduces the quantity of heat removed by the exhaust gases. Removal of heat by the exhaust gases is to be preferred to removal by cooling. Part of the heat energy can be recovered from the exhaust gases, while heat carried off by cooling represents a complete loss lowering the performance of the engine.

1. The effect of wall surface quality on the quantity of heat removed by cooling

It is generally known that a polished metallic surface does not absorb heat as readily as a coarse one. Therefore, the walls of the combustion chamber ought to be as smooth as possible. Machining of combustion chamber walls is being introduced and this means lower heat loss and the additional advantage of maintaining an exact compression ratio in all cylinders of the engine. In cast combustion chambers of standard spark-ignition engines up to 500 c.c. displacement the attainable tolerance is $\pm 1{\cdot}6$ cm^3, which is a permissible tolerance for compression ratios up to 8:1.

These figures hold for cases of several combustion chambers cast integrally in one common head. In air-cooled engines each cylinder head is cast separately and so higher accuracy can be attained. Combustion chambers cast in steel moulds have very exact shapes, and high accuracy in the volume of the combustion chamber can be ensured by starting the machining of the facing between cylinder-head and cylinder from the combustion chamber wall.

Combustion chambers cast in steel moulds have, apart from smooth wall surfaces, the additional advantage of permitting thermally desirable shapes without machining problems. Conditions of heat transfer to walls are valid also for induction and exhaust manifolds, where smooth walls are preferred also for lower friction losses.

A carbon layer deposited on the walls of the combustion chamber causes a worse heat transfer with a subsequent temperature increase of the inner surface. In such cases higher octane fuels are required and volumetric efficiency may be reduced due to the higher temperature of the inducted air.

Less heat is transferred through a cast iron wall than through an aluminium wall. However, this particular advantage does not compensate for all the other disadvantages of cast iron as a material for pistons and cylinder heads.

2. The effect of engine load on heat transfer to cylinder walls

Heat transfer to cylinder walls depends also on the engine load. The maximum amount of heat is released in the cylinder of engines running at full load when the temperature of combustion products is very high. In engines operating at less than full load, the quantity of heat released and temperature of the gases are lower; consequently less heat is transferred to the walls of the combustion chambers and cylinder.

Fig. 42 illustrates the dependence of the cylinder, cylinder-head, and piston

temperature on the injected quantity of fuel and charging pressure in a compression-ignition Tatra engine having a 120 mm bore, engine speed of 1200 rev/min and the quantity of cooling air remaining constant.

The cooling fan was belt driven from a pulley mounted directly on the crankshaft. A Roots blower driven by an electric motor was used for charging.

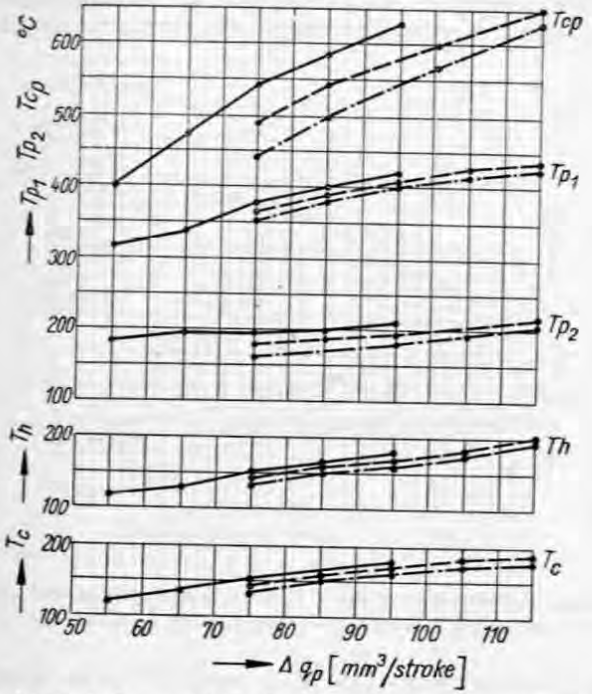


FIG. 42. Load characteristics of the T 928 engine at 1,200 rev/min.
Tcp – temperature of exhaust gases; Tp_1 – temperature of piston at point indicated in Fig. 44; Tp_2 – temperature of gases at point indicated in Fig. 44; Th – temperature of cylinder head; T_c – temperature of cylinder. Full line – without supercharging; dashed line – 140 mm w.g. supercharging; dash-and-dot line – 280 mm w.g. supercharging.

Quantity and temperature of air was measured separately for the cylinder and cylinder head. Temperatures of the cylinder head and cylinder were measured by iron-constantan thermocouples located 2 mm from the inner walls at the hottest spot of the cylinder head and cylinder. Piston temperature was measured by iron-constantan thermocouples by the compensating method in positions shown in Fig. 44. When charging at atmospheric pressure the temperature curves of the cylinder head and cylinder show that temperature, and thus the quantity of heat transferred, rises with increasing load more rapidly in the cylinder head, than in the cylinder. In this case the temperature-fuel quantity correlation represents an approximately linear function.

If supercharging is used for an increase of output at constant fuel quantity, the temperature is directly proportional to the charging pressure owing to the increased supply of air. Temperatures of the cylinder, cylinder head, piston, and exhaust gases are reduced owing to the internal cooling effect of scavenging.

At a charging pressure of 280 mm Hg (0·37 kg/cm²) fuel quantity can be increased by about 18% without the danger of increasing the temperature, and

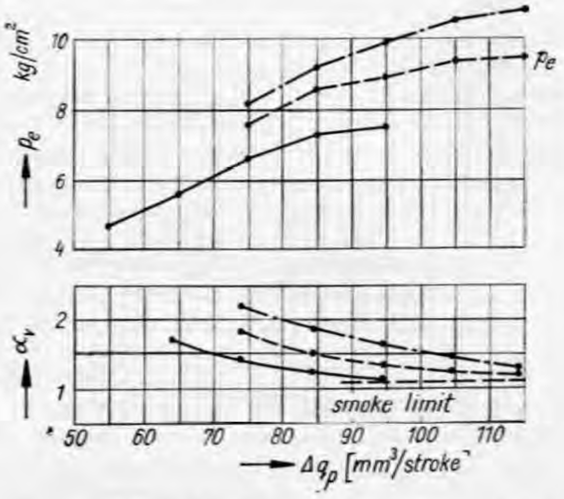


FIG. 43. Load characteristics of the T 928 engine at 1,200 rev/min (continued). p_e – mean effective pressure; α_v – surplus of air; key to lines as in Fig. 42.

consequently also the thermal load of the engine. Thus the quantity of heat removed from the cylinder by cooling is not increased, but more heat is carried off by the exhaust gases. However, the temperature of the exhaust gases is not necessarily increased, because, due to the volume of scavenging air, the quantity is greater.

The above experiment clearly illustrates the fact, that engine performance can be considerably increased by supercharging without the necessity of improved cooling. However, with a very large performance increase cooling must be intensified.

When the engine load is increased by supercharging, the overall quantity of heat removed by cooling increases, but not in direct proportion to the mean effective pressure. Therefore the percentage loss of heat transferred to the walls is lower. More heat is removed by the exhaust gases, but if exhaust gases are used as a fuel for a supercharger turbo-blower, a part of the heat energy is recovered in the form of useful work and the overall thermal efficiency of the engine is improved.

The correlation between heat removed by cooling and engine load is shown in Fig. 52. According to the diagram the quantity of heat transferred to the cylinder head is directly proportional to the engine load, while the heat transferred to the cylinder walls increases only by the 0·8 th power of the load. The overall quantity of heat removed by cooling increases with the 0·95th power of the engine load.

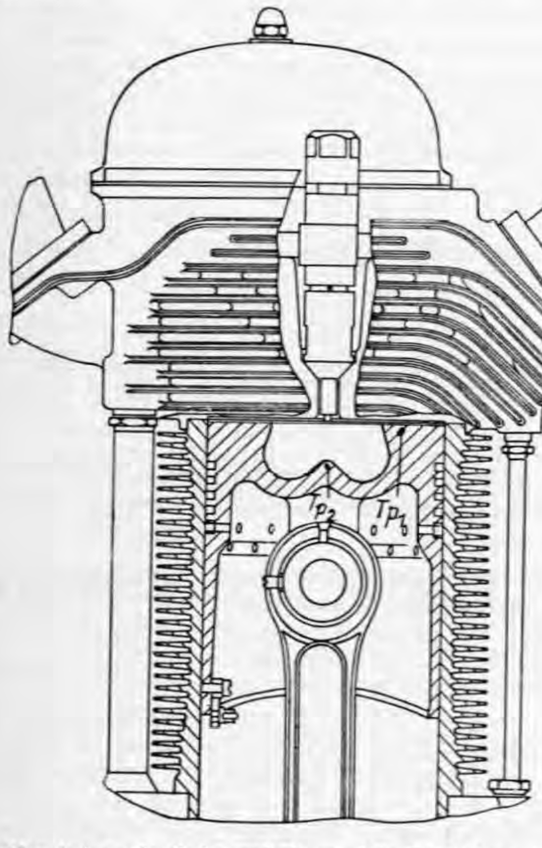


FIG. 44. Sectional view of cylinder and piston of the T 928 four-stroke experimental engine.

The above relation was established by measurements carried out on an air-cooled Ranger cylinder having a bore of 101·6 mm and a stroke 133·4 mm. Displacement was 1·08 litres and compression ratio 6·5 : 1. The cylinder head was made of aluminium and the steel cylinder was fitted with a cast-on aluminium jacket with turned fins. The mean temperature of the cylinder was kept at 146 °C, the temperature of the cooling air at 38 °C.

3. The effect of engine speed on the quantity of heat removed by cooling

Fig. 45 illustrates the dependence of the cylinder head, cylinder, piston, and exhaust gas temperatures on the engine speed at constant fuel quantity

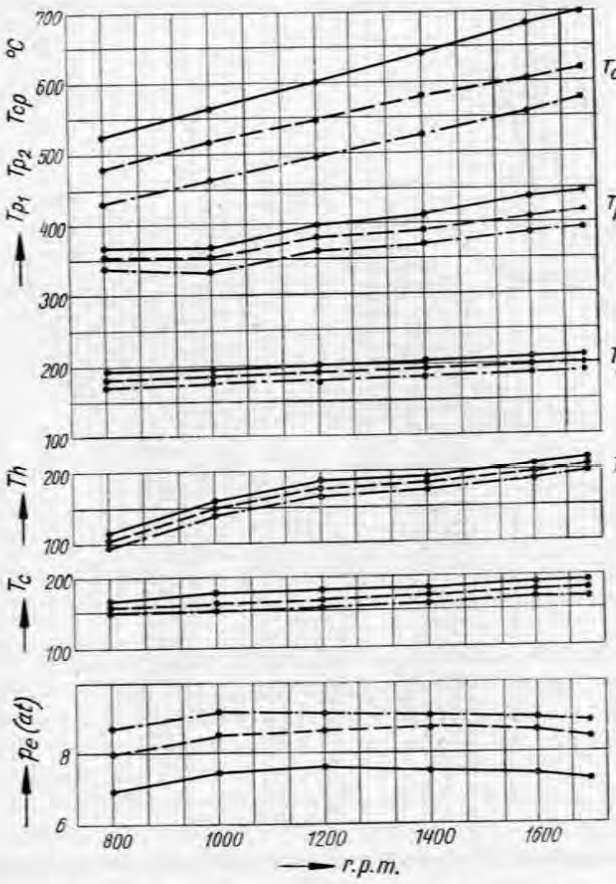


FIG. 45. Speed characteristics of the T 928 engine.
Symbols and lines as in Figs. 42 and 43. Fuel input 88 mm³ per stroke.

and various charging pressures. The temperature of the cylinder head rises more rapidly with an increasing engine speed, than that of the cylinder. Heat removed by the exhaust gases increases with higher speed, while the heat removed by cooling decreases. This fact is explained by the shorter time of contact between the hot gases and cylinder wall at higher speeds. The total quantity of heat removed does not increase in direct proportion with the increasing speed, but approximately in proportion to the 0·65 power of the speed increase only.

The quantity of heat removed by cooling increases with growing speed, but not in a direct proportion. At lower speed ranges the heat transfer is proportional to 0·8—0·7 power of the speed increase; at high speed this exponent is reduced. If the mean effective pressure is reduced due to the increased speed, heat transfer to the walls may remain constant even at an increased speed.

The effect of the speed of a water-cooled SV automobile engine on heat transfer is shown in Fig. 39. [27].

In low speed engines heat transfer to the cylinder walls is calculated from the following equation:

$$dQ = q \cdot A \cdot (T - T_w) \cdot dt$$

where $q = 0{\cdot}0224 \times 228{\cdot}3^n \cdot p^n \cdot T^{1-n} (1 + 1{\cdot}24\, c_m)$ (kcal/m² h °C)

- p denotes the gas pressure (kg/cm²)
- T „ „ absolute gas temperature (°C)
- T_w „ „ absolute wall temperature (°C)
- A „ „ surface area in contact with the hot gases (m²)
- t „ „ time (h)
- c_m „ „ mean piston velocity (m/s)
- n „ an exponent equal according to Nusselt $n = 2/3$, according to Jacklisch $n = 0{\cdot}9$ to $0{\cdot}44$ and varies with the gas temperature which, during the cycle, changes from 273 °K to 3000 °K.

In standard spark-ignition engines without supercharger and mean piston velocity 39 ft/s (12 m/s) heat transfer to the combustion chamber walls amounts to 11,904,960 B.t.u./ft²h (300 000 kcal/m²h). With high efficiency engines with superchargers the corresponding figure is 19,841,600 B.t.u./ft²h (500 000 kcal/m² h) and even more. The respective values for compression-ignition engines are slightly lower.

It is clear from the above considerations that heat transfer to the cylinder wall is not influenced by the cylinder bore. The thermal load of walls of large diameter cylinders does not differ from that prevailing at normal conditions. As far as proper cooling of cylinder head and cylinder is concerned, the cooling of large engines depends on proper direction of the cooling air stream and the mastering of the thermal deformation of the cylinder and cylinder head. The cooling of pistons is a more complex problem. Should the total heat transferred to the piston face be removed via the piston rings to the cylinder wall, the rings would be subjected to a very heavy thermal stress and the danger of sticking and seizure would arise. Therefore, part of the heat from the piston crown must be removed directly by the lubricating oil, the temperature difference between the centre and periphery of the piston face must be reduced, so as to obtain thermal relief of the piston rings.

CHAPTER IV

HEAT TRANSFER THROUGH CYLINDER WALLS

Heat transferred from the combustion products to the cylinder walls is conveyed to the exterior cylinder surface, and then either directly to air or to the cooling liquid. The temperature difference in the cylinder wall depends on the thermal conductivity of the material of which the cylinder or the cylinder head is made.

The propagation of heat in the wall of the cylinder does not take place only in the radial, but also in the axial and tangential, directions. The direction of the heat flow depends on the thickness of the cylinder wall, on the thermal conductivity of the material of the cylinder wall, and on the degree of unevenness of distribution of heat in the cylinder. In a large aluminium cylinder head this tangential propagation may be rather considerable and welcome, since it relieves the thermally overloaded regions. In principle, heat flows from the hottest to the coolest point.

Thermal conductivity is expressed by the coefficient determining heat transmission in a substance as follows:

$$\lambda = \frac{Q}{A \dfrac{\Delta t}{l}} \text{ (kcal/m h°C)}$$

Hence, thermal conductivity indicates the amount of heat which is transmitted in a unit of time through a unit of area at a temperature difference of 1 °C per unit of wall thickness of 1 m. Thermal conductivity and other physical properties of some substances are given in Table 6.

The amount of heat transferred through a simple wall will equal

$$Q = \frac{\lambda}{\delta} \cdot \Delta t \cdot A \text{ (kcal/h)} \tag{13}$$

where A denotes the area of the flat wall (m^2);
δ „ „ wall thickness (m)
$t_1 - t_2 = \Delta t$ — temperature difference in the wall (°C)
t_1, t_2 temperatures of exterior surfaces (°C)

Table 6

Thermal Properties of Some Substances

	t [°C]	γ $\left[\frac{kg}{m^3}\right]$	λ $\left[\frac{kcal}{m\,h\,°C}\right]$	C_p $\left[\frac{kcal}{kg\,°C}\right]$	$a \cdot 10^4$ $\left[\frac{m^2}{h}\right]$
Air	20	1.20	0.0219	0.240	758
Air	100	0.94	0.0267	0.241	1210
Water	80	972	0.575	1.003	5.91
Glycol	80	1070	0.225	0.663	3.32
Glycol	130	1028	0.228	0.710	3.12
Methanol	60	774	0.160	0.645	3.20
Motor oil	80	876	0.120	0.499	2.81
Petrol	50	900	0.095	0.44	2.4
Aluminium		2670	175.0	0.22	3280
Bronze	20	8000	55.0	0.091	750
Copper		8800	330.0	0.091	4120
Silver		10500	394.0	0.056	6700
Steel	20	7900	39.0	0.11	450
Cast iron	20	7220	54.0	0.12	625
Boiler scale	60	—	0.5—1.0	—	—
Coke, pulver.	100	449	0.164	0.29	1.26
Carbon black	40	190	0.027	—	—
Rubber		1200	0.14	0.33	3.53
Mica		290	0.5	0.21	820
Glass	20	2500	0.64	0.16	16

For a cylindrical wall the relationship [42] holds true:

$$Q = \frac{2\pi \cdot \lambda \cdot l}{\ln \dfrac{r_2}{r_1}} \Delta t \text{ (kcal/h)}$$

where l denotes the length of the cylinder,
r_1 „ „ inner radius,
r_2 „ „ outer radius.

Calculations based on this formula are not easy, for the formula contains a logarithmic value; a simplified formula may, therefore, be used:

$$Q = \frac{\lambda}{\delta} \frac{\pi \cdot d_m \cdot l}{\varphi} \Delta t \text{ (kcal/h)} \tag{14}$$

where $d_m = r_1 + r_2 = 2\,r_1 + \delta$
φ = coefficient of shape.

For thin walls of engine cylinders φ can be assumed as being equal to 1, and the heat transfer can be calculated similarly to areas of flat walls.

For a known Q the temperature difference in the wall will equal

$$\Delta t = \frac{Q \,.\, \delta}{\lambda \,.\, A} \; (^\circ\mathrm{C}) \tag{15}$$

In order to calculate a temperature difference, the thermal load of the wall Q/A must be known. For an approximative calculation it may be assumed that all the heat will be transferred through the surfaces of the cylinder head, piston crown and cylinder wall, when the piston is situated in the mid-stroke position. Leaving the height of the compression chamber out of account, the area of the surface is:

$$A = \frac{2\pi D^2}{4} + \pi D \frac{L}{2} \; (\mathrm{cm}^2)$$

If we assume, for example, a petrol engine with a 75 mm bore and 72 mm stroke, with a performance of 12·5 b.h.p. at 4800 rev/min, from which the cooling will transfer 60% of the heat converted into work, this heat will equal:

$$Q = 12{\cdot}5 \times 632 \times 0{\cdot}6 = 4575 \; (\mathrm{kcal})$$

and the area will amount to

$$A = \frac{2 \,.\, \pi \,.\, 7{\cdot}5^2}{4} + \pi \,.\, 7{\cdot}5 \frac{7{\cdot}2}{2} = 173{\cdot}8 \; (\mathrm{cm}^2)$$

The thermal load of the cylinder wall will equal:

$$Q_1 = \frac{Q}{A} = \frac{4575}{0{\cdot}01738} = 263{,}000 \; (\mathrm{kcal/m^2\,h})$$

The temperature difference in the wall of a 5 mm thick cast-iron cylinder will be:

$$\Delta t_c = \frac{Q_1 \delta}{\lambda} = \frac{263{,}000 \times 0{\cdot}005}{54} = 24{\cdot}4 \; (^\circ\mathrm{C})$$

Similarly the temperature difference in the wall of a 15 mm thick aluminium cylinder will be:

$$\Delta t_a = \frac{263{,}000 \times 0{\cdot}015}{175} = 22{\cdot}6 \; (^\circ\mathrm{C})$$

Hence, the temperature difference indicates the magnitude of the heat flow. Knowing the temperature difference, we can, for a known material, calculate the flow of the heat along the temperature gradient. If isotherms were traced on the surface of the piston crown, the direction of maximum heat-flow could be established, and this would be the zone of the densest isotherms. The amount of heat can be established from the temperature difference and thermal conductivity of the respective materials. A similar conclusion could be reached from the course of the isotherms in the cross-section through the cylinder head. From the differences in the temperatures at various points of the surface of the cylinder head we can estimate the magnitude of the tangential heat flow under certain conditions (e.g., even thermal load of the internal surface of the cylinder, etc.).

The conditions governing the heat transfer through cylinder walls are, however, more involved [4].

From Fig. 46 it may be noted that heat must first pass through a film of carbon or oil on the inner surface of the cylinder wall, and be conveyed thereafter through the cylinder wall, and eventually, on the exterior side of a water-cooled engine, it must be transferred through a layer of scale or through a layer of outer varnish in the case of an air-cooled engine, before it is transferred to the coolant.

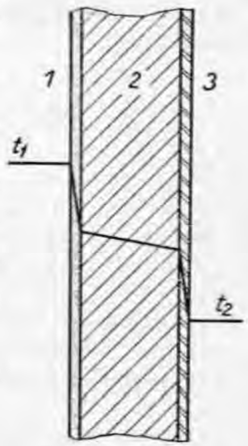


FIG. 46. Heat flow through cylinder wall. *1* — oil film, *2* — metal wall, *3* — outside layer of scale or varnish.

For this reason the transfer of heat is slowed down by the wall and the resulting coefficient of transfer q_x can be calculated from the equation:

$$q_x = \frac{1}{\dfrac{1}{q_g} + \dfrac{\delta_1}{\lambda_1} + \dfrac{\delta_2}{\lambda_2} + \dfrac{\delta_3}{\lambda_3} + \dfrac{1}{q_a}} \; (\mathrm{kcal/m^2\,h\,^\circ C}) \tag{16}$$

The symbols employed in this formula signify:

- q_g — coefficient of heat transfer from gases to wall,
- δ_1 — thickness of film of carbon or oil,
- λ_1 — coefficient of thermal conductivity of carbon or oil,
- δ_2 — thickness of cylinder wall,
- λ_2 — coefficient of thermal conductivity of the material of the cylinder wall,
- δ_3 — thickness of layer of scale (varnish),
- λ_3 — coefficient of thermal conductivity of scale (coating),
- q_a — coefficient of heat transfer from wall to water or air.

From this equation the resultant coefficient of heat transfer through the cylinder wall q_x can be calculated, or according to equation (15) from a known external temperature (temperature of water in the cylinder block) and a known temperature difference the temperature of the inner surface of the cylinder wall can be calculated.

It is evident that the thicker the layers of carbon and of scale the higher will be the temperature on the inner surface of the cylinder wall. As carbon, oil and scale are poor heat conductors, the temperature of the inner surface of the wall will rise rapidly till it reaches a point at which it causes pre-combustion. It will then be necessary to dismantle the engine and remove the carbon layer. Greater difficulty is involved in cleaning the exterior surface of a water-cooled cylinder which, as a rule, is inaccessible. For this reason it is essential to fill the radiator with clean, if possible soft water, so as to reduce the deposits of scale to a minimum. This difficulty does not arise with air-cooled engines, where the exterior surface of the cylinder is readily accessible.

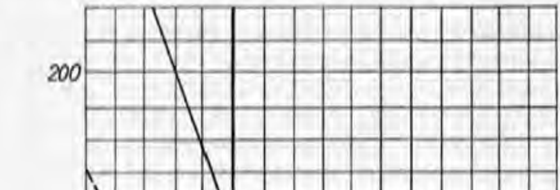


FIG. 47. Temperature gradient in a steel and aluminium rod of 25 mm diameter joined by the Al-Fin process.

For the relatively thick walls of a cylinder block or head thermal conductivity of the material is important. Fig. 47 illustrates the temperature difference in steel and aluminium, bonded by the Al-Fin method, employed for the cylinders of air-cooled engines. It is the result of a test to which a steel and aluminium bar, 25 mm in dia., had been subjected.

During testing one end (steel) of the rod was electrically heated, while the other (aluminium) was immersed into water or, at a lower temperature gradient, into ice. The thermal load of the rod section was about the same as that of the cylinder wall.

The temperature difference in a 20 mm thick aluminium wall is about 9 °C, whereas in a steel wall of equal thickness it reaches 64 °C. The temperature difference in an aluminium cylinder with a 2·5 mm thick wall and a cast-in steel sleeve with a 2 mm thick wall is no more than 5·7 °C.

CHAPTER V

THERMAL BALANCE OF THE ENGINE

1. Heat removed by cooling

From the heat released in the cylinder by the combustion of the air-fuel mixture approx. 25 to 30% is converted into mechanical work, 20 to 25% is carried off by cooling, and the balance of 45 to 50% is removed by exhaust gases. Most of the heat drawn off by cooling is transmitted through the walls of the cylinder and head, and only a minor portion is transferred through the piston to lubricating oil, from which it is either transferred directly to air or through the walls of the crankcase or oil-cooler.

The heat removed by cooling represents a loss, and cannot be converted into mechanical work. Further, in order to transfer this heat to air a portion of the effective performance of the engine is consumed. The heat carried off in exhaust gases also represents a loss, but by fitting adequate equipment at least a part of this thermal energy can be recovered and converted into mechanical work.

The purpose of cooling the cylinder is to secure a sliding surface for a piston moving at an average speed of 49 ft/s (15 m/s) and over. This surface is flushed by hot gases, whose maximum temperature at the outset of the expansion stroke reaches over 2000 °C. The conditions for the formation of a lubricating film on the cylinder walls are adverse, and the temperature of the walls must be maintained at a certain level. The maximum temperature on the interior surface of the cylinder wall should not exceed 180 °C for normal operation.

At elevated temperatures lubrication of the cylinder walls becomes irregular and the piston rings suffer from excessive wear.

The purpose of cooling is to carry off heat with the least possible loss in performance, while maintaining the required temperature of the cylinder walls.

The amount of heat transferred from a petrol engine by cooling can be calculated approximately from the following formula:

$$Q_p = 0{\cdot}0949 \,.\, z \,.\, D^{1{\cdot}73} \,.\, L^{0{\cdot}575} \,.\, n^{0{\cdot}71} \,.$$

$$\left[1 + 1{\cdot}5\,\frac{L}{D}\right](\varepsilon - 1)^{-0{\cdot}286}\ \text{(kcal/h)} \tag{17}$$

For crude oil engines the formula has been modified as follows:

$$Q_o = 0{\cdot}0637 \,.\, z\, D^{1\cdot73} \,.\, L^{0\cdot575} \,.\, n^{0\cdot71}\left[1 + 1{\cdot}5\frac{L}{D}\right] \text{(kcal/h)} \qquad (18)$$

where z denotes the number of cylinders,
D " " bore of cylinder in cm
L " " stroke of piston in cm
n " " rev/min
ε " " compression ratio.

From air-cooled engines with a high mean temperature of cylinder walls less heat will be removed through cooling than from water cooled engines. For aircraft petrol engines it is reckoned that cooling will draw off some 0·5 to 0·6 of the amount of heat converted into useful work. For motor vehicle engines the amount is estimated to be 0·6 to 0·7. The heat transferred by way of cooling will therefore amount to

$$Q = 0{\cdot}6 \times \text{(b.h.p.)} \times 632 \text{ (kcal/h)}$$

since 1 h.p. = 632·43 kcal.

Table 7 gives the thermal balance of the mark R-2800 Pratt and Whitney

Table 7

Thermal Balance of the Pratt & Whitney Aircraft Engine

Experimental Single-Cylinder of the Type R-2800

	Operating Performance	Starting Performance
Fuel: Air ratio	0.06	0.11
Mean effective pressure, kg/cm²	8.75	14.0
Engine speed, rev/min	1,800	2,700
Useful work, %	100	240
Burnt fuel, %	98	46
Unburnt fuel (internal cooling), %	2	54
Thermal efficiency, %	28	16
Distribution of heat produced by burnt fuel, %:		
Useful work	29	35
Cooling air	18	12
Exhaust gases	44	45
Oil, radiation, etc.	9	8
Total	100	100
Output consumed by air-cooling system in % of useful work	63	35

aircraft engine, bore 146·1 mm, stroke 152·4 mm [50]. This illustrates the influence of a rich starting mixture on the cooling of the engine. During the starting period no more than 46% of fuel is burnt in the cylinder, the balance is used for cooling. It is thus possible to keep, even at an increased performance, the temperature of the cylinders within satisfactory limits. Owing to such substantial *cooling through fuel* the quantity of heat removed by cooling to air does not undergo any substantial changes. Upon increasing the performance from 100% to 240% the amount of heat removed by cooling increases from 63 to 84 thermal units (Fig. 48).

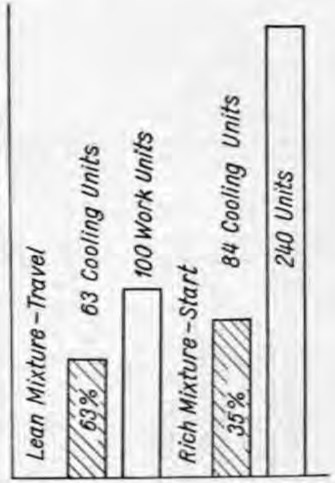


FIG. 48. The thermal balance of an aircraft engine operated with rich and lean mixture respectively.

During travel out of 100 heat units representing the total input 63 units are used for cooling through fuel. The corresponding total input during starting amounts to 240 units out of which 84 are used for cooling through fuel.

For internal cooling during starting water may be injected into the induction manifold. Short of using rich mixtures or other type of internal cooling it would not be possible to cool the large cylinders of modern aircraft engines at their very high mean effective pressures.

The thermal balance of an air-cooled diesel engine with 110 mm bore and 130 mm stroke is shown in Fig. 49. This diagram makes it apparent how the amount of heat removed through cooling varies with the engine speed.

In a water-cooled engine the removal of heat from the cylinder is more intensive, for the water maintains the walls of the cylinder at a temperature of 80 to 90 °C. In view of a greater temperature difference the heat losses will be greater. The relationship between the distribution of heat in a water-cooled diesel engine with 110 mm bore and 140 mm stroke at 1600 rev/min, and the mean effective pressure p_e, is illustrated in Fig. 50 [46].

2. Thermal stress on the interior surface of a cylinder

Thermal stress on the interior surface of a cylinder differs according to whether the latter is exposed to hot gases constantly, or whether it is periodically sheltered by the piston. Measurements have established thermal stresses of the surface of the combustion chamber amounting to about 400,000 kcal/m²h. The walls of the cylinder and the surface of the exhaust ducts are under thermal stresses of 125,000 kcal/m²h. Average thermal stress of the interior surface of the cylinder for various types of engine is indicated in Table 8. The calculation is based on the assumption that 0·6 Q (b.h.p.) is carried off

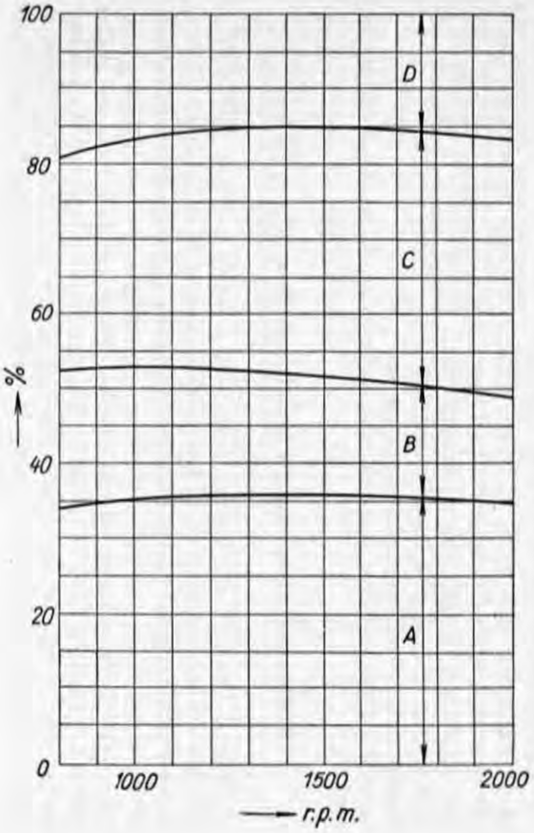


FIG. 49. Thermal balance of an air-cooled Tatra engine of 110 mm bore.

A — heat converted into work, B — heat removed by cooling air, C — heat taken off by exhaust gases, D — residue.

Shown in % of total heat.

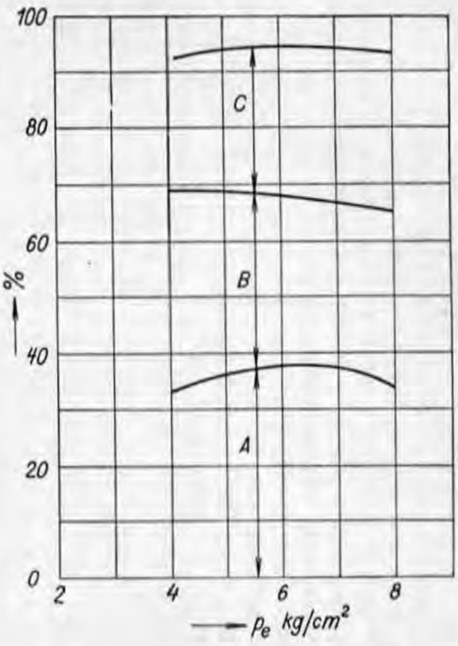


FIG. 50. Thermal balance of an air-cooled compression-ignition engine of 110 mm bore at 1,600 rev/min.

A — useful performance, B — loss by cooling, C — loss through exhaust gases.

Shown in % of total heat.

Characteristic Data of Internal-Combustion Engines

Table 8

	Mark	Bore, mm	Stroke, mm	Volume of 1 cyl.	Perform., b. h. p.	Speed, rev/min	No. of cylinders	Perf. of 1 cyl.	Heat for cooling, kcal/h	Surface area $2\frac{\pi D^2}{4}+\frac{L}{2\pi}\cdot D$, cm²	Thermal stress in cylinder wall $U\left[\frac{\text{kcal}}{\text{m}^2\,\text{h}}\right]$	Specific output (bhp/l)	Mean eff. pressure, kg/cm²	Type of cooling
aircraft	Continental E 185 - 5	101.6	127	1.03	185	2300	6	30.8	11650	365	320 000	30.0	11.7	air
	Franklin 6AB-225-B7	127	114.3	1.45	225	2600	6	37,5	14230	481	296 000	25.8	8.95	air
	Pratt & Whitney Wasp-Major	146.1	152.4	2.56	3250	2700	28	116.0	44000	685	642 000	45.4	15.1	air
	Ranger SGV-770D-4	101.6	130.2	1.05	575	3400	12	48.0	18190	370	491 000	45.8	12.1	air
	Wright Cyclone 836 C 18 CA 1	155.6	160.3	3.05	2700	2900	18	150.0	56880	772	738 000	49.2	15.25	air
	Rolls - Royce Merlin	137.0	152.4	2.25	1760	3000	12	146.5	55600	633	879 000	65.2	19.5	water
	Rolls - Royce Eagle	137.0	130.0	1.92	3440	3500	24	143.5	54350	575	948 000	74.8	21.4	water
railway	Maybach MD 650, oil.	185.0	200.0	5.38	1200	1600	12	100.0	37920	1120	339 000	18.6	10.6	water
	GM, 2 stroke	222.0	267.0	6.45	1600	750	16	100.0	37920	1703	223 000	9.6	5.77	water
	Fairbanks 2 str.	206.0	2×254	5.15	2000	810	2×10	100.0	37920	2310	164 000	11.7	6.35	water
car	Tatra T603	75.0	72.0	0.32	100	4800	8	12·5	4750	173	274 000	39.4	7.4	air
	Tatra T602S	75.0	72.0	0.32	200	7500	8	25.0	9500	173	548 000	80.0	9.6	air
	Ferrari 12V	80.0	74.5	0.38	330	6200	12	27.5	10400	191	546 000	73.5	10.7	water
	Simmering, oil	135.0	160.0	2.30	620	2000	16	38.8	14670	625	235 000	16.9	7.6	air
	Tatra 108, oil	110.0	130.0	1.23	125	2000	8	15.6	5915	414	143 000	12.7	5.7	air

through cooling, and the area of the interior surface corresponds to the mid-stroke position of the piston. Thus

$$A = 2\frac{\pi D^2}{4} + \pi . D\frac{L}{2}$$

From the table it is apparent that the thermal stress of the cylinder wall is a function of the mean effective pressure and of engine speed. For small engines greater values of thermal stress can be allowed. The amount of heat transferred through the interior surface of the cylinder must be transferred to the cooling air. If the heat removed through cooling were not to offset the heat introduced into the cylinder by hot gases, the temperature of the cylinder would continue to increase to the point at which the engine would become overheated.

In a properly designed engine the least possible quantity of heat should be carried off by cooling. Hence, it is possible to evaluate the standard of an engine by the number of calories to be removed per b.h.p. If, at a given performance the interior surface of the cylinder is extensive, as is the case with low speed engines, the heat losses are considerable. Conversely, if high performances are attained from small dimensioned cylinders, as is the case with supercharged engines, the heat losses are low. In the latter instance the thermal stress per unit of the interior surface of the cylinder is, however, great.

3. Distribution of heat between head and cylinder

Factors on which depend the amounts of heat introduced into a head and cylinder were discussed in a preceding section. The distribution of heat depends on the design, chiefly on the shape, of the combustion chamber. The surface of the combustion chamber in the cylinder head is always smaller than the surface of the cylinder and piston combined. Even if we take into account only half the surface of the cylinder, seeing that most of the time a varying portion of this surface is cut off by the piston, this surface, including the surface of the piston crown is still greater than the surface of the combustion chamber. On the other hand substantial amounts of heat reach the cylinder head from the surface of the exhaust duct. This is the basic reason that more heat is transferred to the cylinder head than to the cylinder itself.

If the combustion chamber of a petrol engine is of hemispherical design, the surface area of the combustion chamber is small, and only small amounts of heat are transferred to the head. Wedge-shaped and other type combustion chambers have larger surfaces and are exposed to very heavy thermal loads.

If the heat transfer from the walls of the exhaust duct to the head is to be small, the duct must be short, straight and of small diameter. Sometimes the transfer of heat to the head is reduced through inserting a steel tube into the exhaust duct, a small, thermally insulating air-gap being left between the tube and the wall of the cylinder head. At elevated temperatures this un-

cooled portion of the exhaust tubing may be burnt down. Fire-proof material must therefore be used for high-performance engines.

If the cylinder has been screwed into the head, as it is the case with aircraft engines, a portion of the heat from the upper section of the cylinder is transferred to the head. As this effects the portion of the cylinder on which the piston rings rest, the upper dead centre is the area of greatest heat.

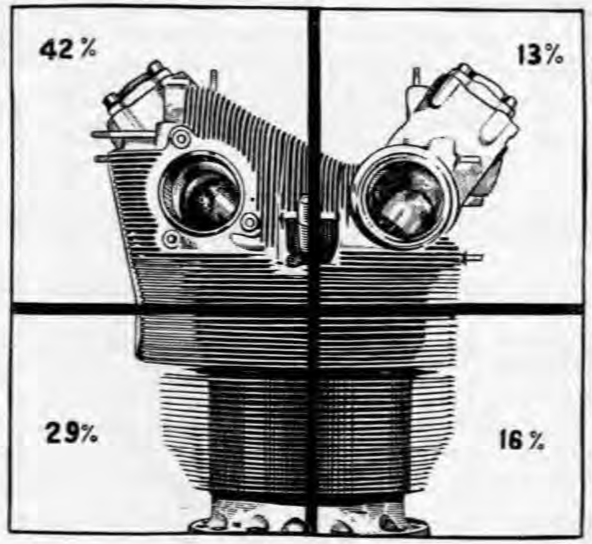


FIG. 51. Distribution of heat shown on a cylinder of a Pratt and Whitney aircraft engine. Note the high percentage of heat removed from the zone of the exhaust valve.

In the course of a test on a cylinder of a Mark R-2800 Pratt Whitney engine the exhaust charge was conveyed from the cylinder through four ducts, and in each of the ducts separate measurements were taken of the amount of the charge and its temperature. It has thus been possible to establish the exact amount of heat removed by cooling. An interesting fact was found: most of the heat was transferred from the quarter of the cylinder housing the exhaust valve. The division of the heat, expressed in percentages of the total, is illustrated in Fig. 51. As 42 % of the total heat is drawn from the area of the exhaust valve, great attention must be devoted to the design of this part of the head.

In a compression-ignition engine with direct fuel injection, where the combustion chamber is formed by the piston crown, the division of the heat between head and cylinder may undergo radical changes. The flat area of the cylinder head will absorb only minimum amounts of heat, and the heat from the piston will be primarily transferred to the cylinder walls. In extreme instances greater amounts of heat may have to be removed from the cylinder than from the head. In engines with a combustion chamber inside the cylinder head, the head of the cylinder is exposed to heavier thermal stresses.

The amount of heat removed through cooling is almost directly proportional to the i.h.p. of the engine. Fig. 52 shows the amount of heat removed by

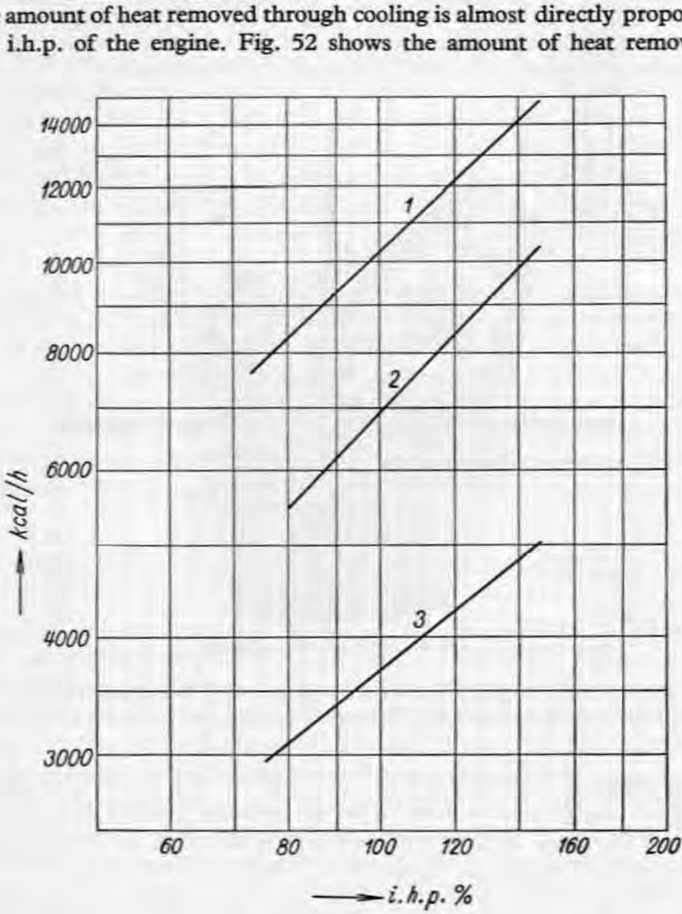


FIG. 52. Distribution of heat in the cylinder and head of an in-line Ranger aircraft engine in dependence on the indicated performance i.h.p. The temperature of the aluminium cylinder was maintained at 145 °C, cylinder head temperature varied between 216—275 °C. Air-fuel ratio and temperature of intake mixture was constant. *1* — total heat, slope 0.95; *2* — cylinder head, slope 1.04; *3* — cylinder, slope 0.8.

cooling from a Ranger series aircraft engine [44]. From this diagram it can be seen that with the thermal load increasing the amount of heat carried off the cylinder increases slowly, but more rapidly from the head.

Total heat transferred is a function of . . . . $(\text{i.h.p.})^{0.95}$
Heat transferred from cylinder head is a function of . . . . $(\text{i.h.p.})^{1.04}$
Heat transferred from cylinder is a function of . . . . $(\text{i.h.p.})^{0.8}$

In the case of this particular engine, most of the heat was removed from the cylinder head. It is however to be taken into consideration that for aircraft engines steel cylinders screwed into the cylinder head are frequently used. The hottest upper part of the cylinder therefore depends on the transfer of heat through the head. For detachable heads of air-cooled motor-vehicle engines the dividing plane is normally located on the level reached by the top ring of the piston at upper dead centre, and for this reason all the heat from the sliding surface of the cylinder must be removed by the finning of the cylinder.

4. Removal of heat from the exterior surface of the cylinder

If thermal equilibrium is to be reached, amounts of heat removed from the exterior surface of the cylinder must equal those introduced into the cylinder through its internal surface. This requirement is easy to comply with for water-cooled systems, where the average coefficient of heat transfer amounts to 5000 to 6000 kcal/m²h °C. For air-cooling, where the coefficient of heat transfer is no more than 100 to 200 kcal/m²h °C, the situation is more difficult.

Assuming that the coefficient of heat transfer is equal on the interior and on the exterior surfaces of the cylinder, the amounts of heat transferred through

Table 9

Internal to External Surface Area Ratio in Air-Cooled Aircraft Engines

Mark	Bore mm	Stroke mm	ε	Surface area, m²		Area ratio
				intern.	extern.	
Wright	155.6	174.6	6.7	0.157	2.39	15.4
Wright	155.6	160.3	6.9	0.149	2.29	15.4
Franklin	117.5	101.6	7.0	0.054	0.695	12.9
Franklin	117.5	101.6	7.0	0.054	0.856	16.0
Franklin	117.5	101.6	7.0	0.054	0.910	16.9
Franklin	108.0	108.0	7.0	0.044	0.573	13.1
Franklin	108.0	108.0	7.0	0.044	0.710	16.2
Jupiter 1939	146.1	190.5	5.3	0.123	0.807	6.6
Napier 1939	88.9	88.9	6.0	0.033	0.293	8.9

these surfaces would depend on the temperature difference between gas and wall, and on the area of the surface. The temperature difference inside the cylinder, assuming an average temperature of the interior surface of the wall of the cylinder of 180 °C and an average temperature of gases at the expansion stroke of 1500 °C, would amount to some 1320 °C. On the exterior surface of the cylinder wall however, assuming a temperature of air equal to 50 °C and that of the hot cylinder equalling 200 °C, the temperature difference would amount to only 150 °C. If equilibrium of heat transferred to and from the cylinder is to be attained also in the latter instance, the area of the exterior surface would have to amount to nine times the area of the interior surface, in order to offset the 9 : 1 ratio between the two temperature differences.

Table 9 gives a comparison of the areas of interior and exterior surfaces of the cylinders of certain aircraft engines. The division of the cooling surface to cylinder heads of different types of engines is given in Table 10.

Table 10

Distribution of Cooling Surface Area in Aircraft Engines

	Bore mm	Stroke mm	Area of Head m^2		Cylinder Area m^2		Total Area m^2	Ratio Head to Cyl.	Ratio Area to Volume cm^2/cm^3	Ratio Area to Output cm^2/bhp
			total	fins	total	fins				
1	117.5	101.6	0.433	0.376	0.260	0.222	0.693	1.66	6.3	280
2	117.5	101.6	0.605	0.543	0.260	0.222	0.865	2.31	7.85	283
3	108.0	88.9	0.367	0.334	0.207	0.174	0.574	1.77	7.05	355
4	108.0	88.9	0.708	0.635	cylinder and head made of one piece		0.708	—	8.55	316
5	117.5	101.6	0.901	0.815			0.901	—	8.25	257
6	155.6	174.6	—	1.550	0.845	0.845	2.390	1.82	7.20	180
7	155.6	160.3	—	—	0.755	0.755	2.290	2.05	7.50	190

CHAPTER VI

THE FINNED SURFACE

1. Heat transfer from fins to air

Removal of heat from the outer cylinder surface is improved by finning. Fins are variously shaped according to the requirements they have to fulfil and their method of production. The simplest fin has a rectangular cross section with a thickness b constant along its whole length h. A fin of this type is illustrated in Fig. 53. The second widely used type is the fin with a trapezoidal cross section. The thickness b_b at the base of the fin is bevelled down to b_t at the top.

The trapezoidal fin approaches most closely the ideal conditions: by the subsequent removal of heat in the base-top direction the heat flow is diminishing, and the fin thickness is gradually reduced with a better utilization of the material. The chamfering of the fin is required also for easier production of cast and machined fins. However, the bevelling is insignificant in both cases and need not to be considered in calculations.

Although a fin of triangular cross section fulfils all thermal requirements, it is difficult to produce. Thickness at the top is zero and the fin is terminated by a sharp edge. The sharp edge may cause injuries during assembling, or it may be easily bent and damaged.

From the thermal and material point of view a parabolic cross section is the best. For the removal of a given quantity of heat this cross section produces the lightest fin but owing to machining difficulties parabolic fins are not used. Rectangular and trapezoidal cross sections are more usual.

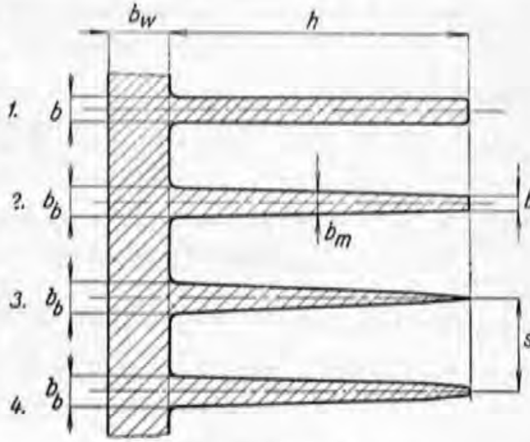


FIG. 53. Basic shapes of fins.
1 — rectangular, 2 — trapeziodal, 3 — triangular, 4 — parabolic.

Conditions of heat transfer from fins to air have been studied by numerous authors. An exact calculation of a trapezoidal circular fin is a very complicated matter. Therefore, for practical purposes, this cross section is calculated as a rectangular one having a thickness equal to the mean thickness b_m of the trapezoidal fin. In calculations of rectangular fins we assume a constant temperature along the whole base of the fin and, further, the temperature at any point of the fin being dependent only on the distance from the base. We assume also the surface coefficient of heat transfer being, on the whole surface area, directly proportional to the temperature difference between the fin and surrounding air, and also to the flow velocity of the air. Direction of air flow is considered along the fin area, i.e. to the fin radius.

The calculation of thermal conditions on the fin surface is very complicated and must be based on the introduction of certain ideal assumptions. However, exact calculations are of small value to the designer, because the prerequisites of the calculations are not complied with anyway. We have seen in the preceding chapters, that the heat transfer coefficient q is not a constant valid for the entire surface area of the fin, because the laminar flow changes, at a certain distance, into a turbulent one, and this distance from the leading edge varies in turn with the air velocity. Air temperature will not be the same in the whole interspace between the fins, because temperature differences at the base and the top of the fin are different, and the quantity of heat transferred to air is determined by the temperature difference.

Despite these circumstances it is sometimes desirable to establish thermal conditions and fin efficiency at least approximately. In such cases Pye's formulas are the most suitable. According to Pye the temperature difference between fin surface and air, at a distance x from the base, can be calculated as

$$t = t_0 \frac{\cosh a\,(x - h')}{\cosh a\,.\,h} \tag{20}$$

and the fin efficiency

$$\eta_f = \frac{Q}{Q_0} = \frac{\tan\mathrm{h}\, a\,.\,h'}{a\,.\,h'} \tag{21}$$

where t_0 denotes the temperature difference between fin base and air (°C)

h „ „ height of the fin (cm)

$$h' = h \frac{b_m}{2} \quad \text{„ the corrected height of the fin} \tag{22}$$

b_m denotes „ mean thickness of the fin (cm)

Q „ „ quantity of heat transferred from the fin in unit time (kcal/s)

Q_0 „ „ quantity of heat transferred from the fin in unit time at a temperature difference of t_0 (kcal/s)

cosh, tanh are hyperbolic functions

$$a = \sqrt{\frac{2\,q}{\lambda\, b_m}}$$

λ denotes the coefficient of thermal conductivity of the material o the fin (kcal/m h °C)

Calculations with the above equations start from the basic equation of thermal equilibrium

$$\frac{d^2t}{dx^2} = \frac{2\,q\,t}{\lambda\, b_m} = a^2\,.\,t$$

and from the conditions that at the base of the fin, i.e. for $x = 0$, $t = t_0$; and at the top of the fin, i.e. for $x = h'$, $dt/dx = 0$.

Thus the fin efficiency is the ratio of heat actually transferred to the heat which would be transferred if temperature on the entire fin surface were equal to O_0, i.e. the temperature at the fin base. The larger the fin, the worse is its efficiency. Excessively long fins have a bad efficiency and their weight is not properly utilized. Greater fin lengths are permissible with materials having a good thermal conductivity. Normally, however, production considerations and not efficiency, are the factors limiting the length of fins. Some typical types of fins used in practice are shown in Table 11. The diagram in Fig. 54 serves as a practical aid for rapid calculations of fin efficiency.

Table 11

Comparison of Different Cooling Fins

Material	Height cm	Thick-ness cm	λ	$\mathbf{a} = \sqrt{\frac{2\cdot \mathbf{q}}{\lambda \cdot \mathbf{b_m}}}$	**h′**	**ah′**	η_f
Aluminium	2.5	0.23	175	0.228	2.61	0.595	0.90
Steel	1.6	0.08	39	0.830	1.64	1.360	0.65
Copper	2.54	0.05	330	0.364	2.56	0.930	0.79

The role of fin thickness can be well demonstrated by the case of the steel fin given in Table 11. By reducing the fin thickness from 0.8 mm to 0.4 mm, efficiency is reduced from 65% to 51%. [See equations (21) and (22)]. If the same weight of metal is used for two fins instead of one, the cooling surface is doubled and the quantity of heat transferred is increased to

$$\frac{51 + 51}{65} \times 100 = 157\ \%,$$

i.e. the same amount of steel removes 57% more heat. The conclusion that thinner fins mean better utilization of the material is thus evident. Fin thickness is limited by production problems rather than by thermal conditions, even in the case of steel which is a material with notoriously bad thermal conductivity. A full utilization of thermal conductivity would, in such metals as aluminium or copper, lead to extremely thin and practically unusable fins. Therefore, fin thickness is chosen as small as permitted by the production process. The number of fins should be the largest possible, but the interspace between the fins must be large enough to ensure a satisfactory flow of cooling air.

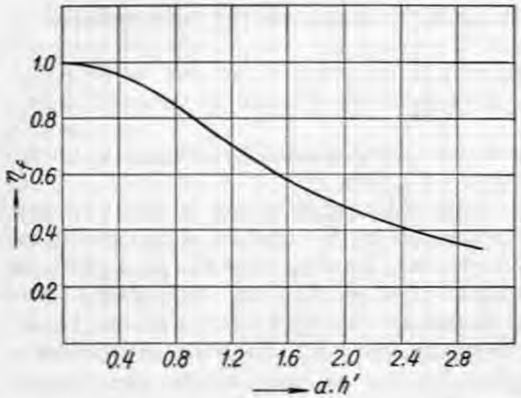


FIG. 54. Efficiency of cooling fins according to equation (21).

2. Spacing of fins

The lower limit of the interstice is determined by the cooling conditions. In the case of a motorcycle engine with a low cooling air velocity, the spacing of fins must ensure an air flow to reach the base of the fins. Distribution of velocities in small and large interstices is shown in Fig. 55. However, it should be borne in mind that the diagram applies to small air velocities and to a freely exposed cylinder having no cowling. In this case a spacing of 8—12mm is recommended.

If a stronger air flow is available for cooling, fins are more densely spaced. Critical conditions depend on the thickness of the boundary layer at the fin surface. The influence of the boundary layer upon fin efficiency was established by tests carried out on straight fins 152 mm long, 25 mm high and 0·5 mm thick. Spacing varied from 12 to 2 mm. The frontal edges were carefully rounded and air flow direction was parallel to the fins. These experiments proved, that a rapid deterioration of fin efficiency started at spacings below 2·5 mm at varying air velocities. Thickness of boundary layers, calculated for the given test velocities and distances of transitions from laminar to turbulent

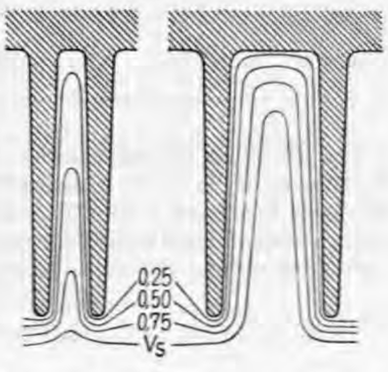


FIG. 55. Velocity of air with a small and a wide interstice between the fins.

flow, are shown in Table 12. The shape of the boundary layer is illustrated in Fig. 56. It can be seen from Table 12 that the critical interstice is smaller than twice the thickness of the turbulent layer at the end of the fin. Therefore, it can be safely assumed that a contact established between two adjacent turbulent layers has no adverse effect.

More important is the thickness of the laminar boundary layer. It must be

Table 12

Thickness of Boundary Layers for Straight Fins, 152.4 mm Long, Exposed to Air Stream of Velocities 140 and 254 km/h.

Transition of laminar into turbulent flow is assumed to take place at Reynolds number = 2×10^5. (See Fig. 56). Values are given in mm.

Air velocity	140 km/h = = 39 m/s	254 km/h = = 70 m/s
Distance between leading edge and point of transition from laminar to turbulent flow	114.4	63.5
Thickness of turbulent boundary layer at the fin root	4.32	4.06
Thickness of laminar boundary layer at the point of transition	1.016	0.508
Thickness of turbulent boundary layer at the point of transition	3.30	1.778

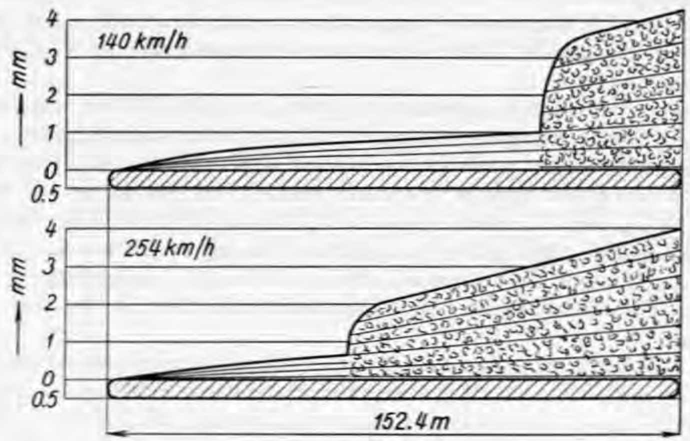


FIG. 56. Shape of boundary layer for speeds of 140 km/h and 254 km/h. Thickness of layer is drawn to a scale 10 times that of the length.

remembered that velocity in the turbulent boundary layer does not drop below $0{\cdot}9\ v_m$ as long as the distance from the surface is not less than half the thickness of the boundary layer.

A rapid deterioration of flow conditions occurs at an interstice smaller than two thicknesses of the laminar boundary layer. At an air speed of 87 m.p.h. (140 km/h) the double thickness of the laminar boundary layer is 2 mm. Fig. 57 illustrates the interdependence between fin efficiency and spacing of fins; efficiency drops rapidly at interstices below 2·5 mm. However, with a decreasing spacing, the cooling surface increases, and the overall quantity of heat removed begins to diminish only at spacings below 2·5 mm. This is clearly shown in Fig. 77.

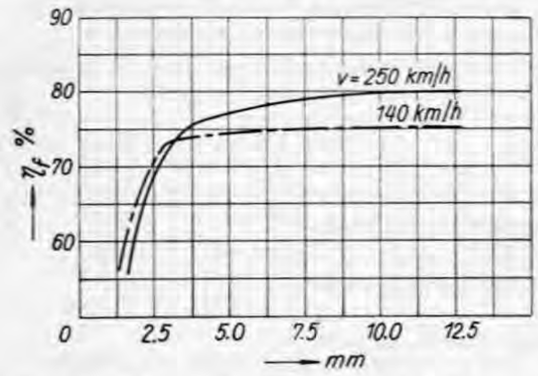


FIG. 57. Influence of interstice upon fin efficiency.

3. The surface coefficient of heat transfer

Equation 11 in Chapter II enables only an approximate calculation of the heat transfer coefficient q. The exponent of the mean air velocity v_m varies with the proportion of the surface area exposed to laminar and turbulent flow respectively. Dimensions of the test cylinder and fins are given in Chapter II. Temperatures of the electrically heated testing unit were measured by 24 iron-constantan thermocouples spot welded to the cylinder and fins.

Results obtained from measurements with one type of finning are shown in Fig. 58. Temperatures were measured on the entire circumference of the cylinder, and transfer coefficients at different spots were thus determined. The cylinder was freely exposed to the air stream. The frontal part of the cylinder facing the air stream is marked 0°, the rear 180°. It can be clearly seen that the highest coefficient of heat transfer belongs to the frontal part with rapid deterioration in the region between 90°—120°, particularly at higher speeds. The worsening is due to insufficient air flow at the rear of the cylinder.

Air flow in the fin interstice was observed on fins coated with a mixture of kerosene and carbon black [7]. It was clearly seen, that a dead space due to the "cylinder shadow" was formed in the rear beginning at 115° which explains the deterioration of the heat transfer coefficient in this region. This shows a bad utilization of fins on the rear of a freely exposed cylinder. See Fig. 59.

In our further considerations we shall use the mean surface coefficient of heat transfer

$$q = \frac{Q}{A \cdot t} \tag{23}$$

where Q denotes the total heat transferred from the cylinder
A „ „ total cooling surface area
t „ „ mean temperature difference between cylinder surface and cooling air.

The value of the mean surface coefficient of heat transfer obtained from measurements on rectangular and trapezoidal fins are given in Fig. 60, where they are plotted against fin spacing s and air velocity v_m respectively. The following conclusions can be drawn from the test results.

The influence of fin height. On a freely exposed finned cylinder with low fins the mean air velocity on the fin surface is higher, and so is the coefficient q. Tests carried out on cylinders with fins of various height and spaced 3·8 to 6.35 mm apart have proved that q deteriorates rapidly up to a height of 10 mm. A further increase of the height of the fins causes only a slight increase in the

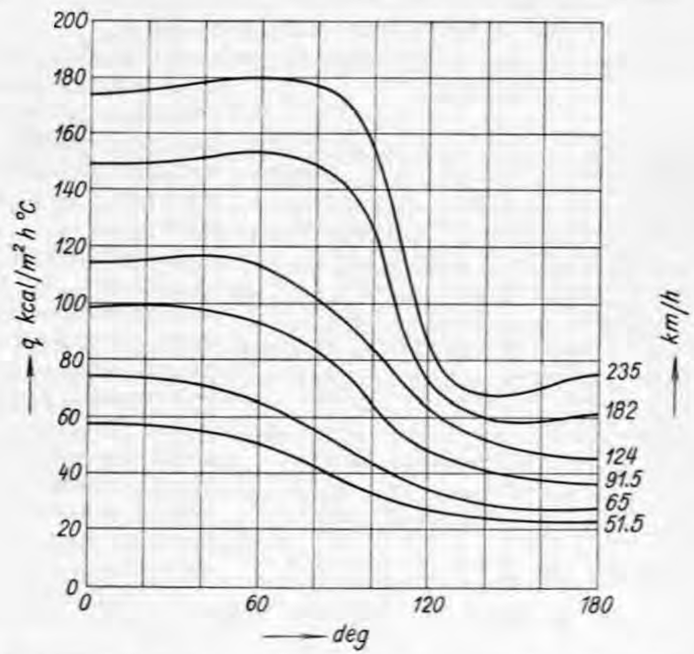


FIG. 58. Values of the coefficient of heat transfer q around the circumference of a cylinder without cowling.

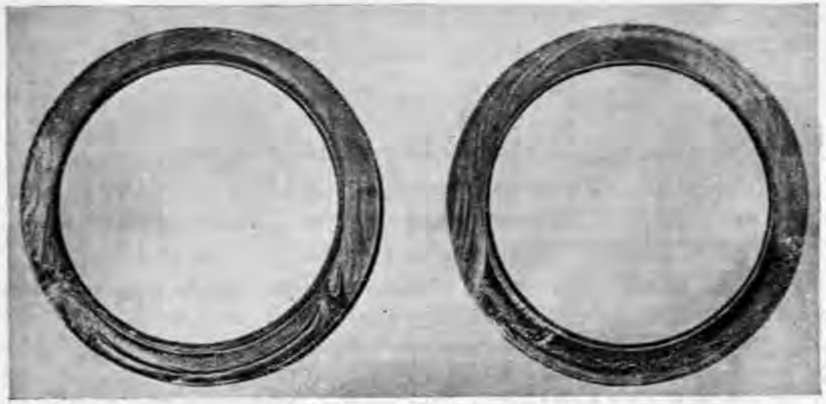


FIG. 59. Traces of air flow in the interstice between fins.

deterioration of q. In cylinders fitted with cowlings fin height had even smaller effects than in freely exposed cylinders. Fins used in practice are invariably higher than 10 mm, and therefore, it can be stated that the height of the fins has no influence upon the surface heat transfer coefficient q.

The influence of fin thickness. At a given fin spacing, fin thickness b has only little influence upon the coefficient q. Therefore, we may conclude with sufficient accuracy that q does not depend on fin thickness.

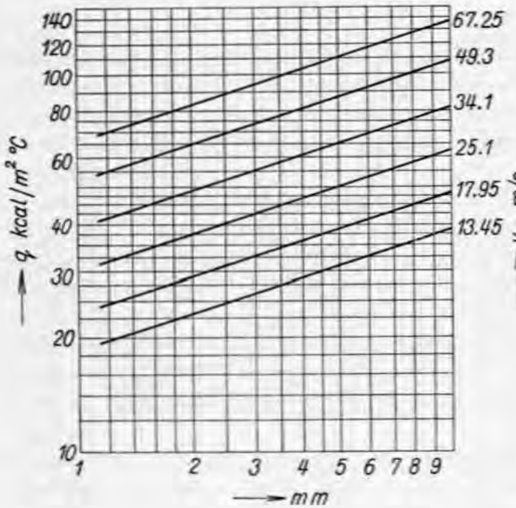


FIG. 60. Dependence of the mean surface coefficient of heat transfer on the width of the interstice and on the velocity of air.

The influence of interstice between fins. If the interstice is too small, air flow between the fins deteriorates owing to interference between the boundary layers. Consequently the surface heat transfer coefficient q deteriorates also. With increased interstice, flow conditions improve and q increases.

Fig. 60 illustrates results obtained from measurements on rectangular and trapezoidal fins of various height and thickness. The straight line showing the correlation between q and fin spacing s represents the mean values of measurements on different fins. Individual values deviated from this line by 8 % at the maximum, thus proving that the influence of height and thickness of the fin upon q is negligible. The coefficient is influenced mainly by the fin spacing. The slope of the lines in this diagram drawn in log-log scale is 0·322 signifying the relation

$$q \propto m^{0 \cdot 322}.$$

In cylinders with cowlings fins are spaced very closely, and it is interesting to examine the influence of spacings below 1 mm. This dependence is illustrated in Fig. 61 where q is plotted against s in a logarithmic scale. It is evident that q deteriorates considerably at spacings below 1·5 mm.

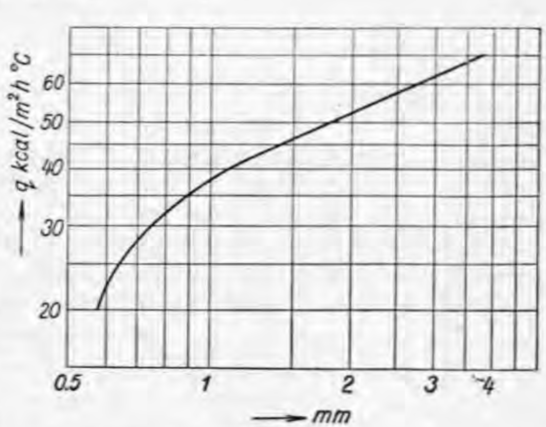


FIG. 61. Effect of interstice upon the surface coefficient of heat transfer.

The influence of air velocity. As explained before, air velocity has a considerable influence on the transfer coefficient q. Fig. 62 illustrates, on a log-log scale, the correlation between q and v_m for one type of finning at a spacing $s = 4 \cdot 45$ mm. The slope of line 4 in the diagram indicates that q is proportional to $v_m^{0 \cdot 796}$.

Line 3 in the same diagram shows the results obtained from measurements on a plain cylinder without fins having the axis normal to the direction of the air flow.

The break of the curve indicates the transition from laminar to turbulent flow occuring at an air velocity of about 28 m.p.h. (45 km/h). For turbulent flow q is proportional to $v_m^{0 \cdot 66}$.

The diagram also shows results of measurements on a square plate 152 × × 152 mm. In this case (line 2) q is proportional to $v_m^{0 \cdot 725}$.

Test carried out on a plain cylinder with cover [35] produced the result $q \propto v_m^{0 \cdot 57}$. The above results prove that the relation between q and v_m varies greatly with the type of air flow.

For cylinders with fins of standard dimensions the correlation from equation 11 can be regarded as a safe average, thus

$$q \propto v_m^{0 \cdot 73}$$

If the cylinder casing bears tightly on the fin edges, the flow in the interstice between the fins proceeds under conditions similar to those prevailing in a bent tube of a rectangular cross section. The equivalent diameter of a round tube is calculated from the formula

$$d_e = \frac{4sh}{2\,(s+h)} = \frac{2sh}{s+h} \tag{24}$$

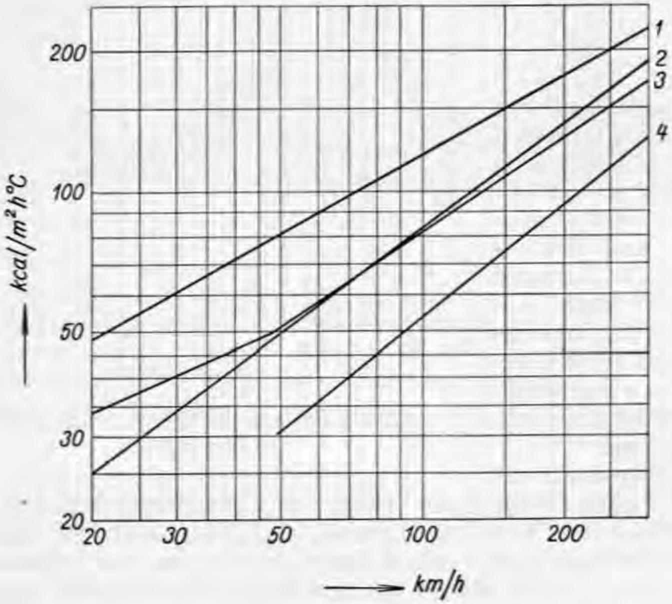


FIG. 62. Effect of air velocity upon the surface coefficient of heat transfer.
1 — measured on a plain cylinder (Löhner) of a diameter 160 mm with air flow normal to the cylinder axis, 2 — measured on an even plate 152.4 × 152.4 mm (Pye), 3 — measured on a plain cylinder to different diameters with air flow normal to the cylinder axis (Biermann), 4 — measured on a finned cylinder with an interstice between fins s = 4.4 mm (Biermann).

For a straight tube of circular cross section the transfer coefficient q is calculated, according to Nusselt, from the equation

$$q = 0.0346\,\frac{\lambda}{d}\left(\frac{x}{d}\right)^{-0{\cdot}05}\left(\frac{v_m\,.\,d}{k}\right)^{0{\cdot}79} \tag{25}$$

A different relation is given by Kraussold as follows:

$$q = 0{\cdot}024\,\frac{\lambda}{d}\left(\frac{v_m\,d}{\nu}\right)^{0{\cdot}8}\left(\frac{\nu}{k}\right)^{0{\cdot}37} \tag{26}$$

According to Jeschke heat transfer coefficients in spirals made of round tubes are calculated from the equation

$$q = \left(0{\cdot}0392 + 0{\cdot}139\,\frac{d}{D}\right)\frac{\lambda}{d}\left(\frac{v_m\,d}{k}\right)^{0{\cdot}76} \tag{27}$$

where d denotes the internal diameter of the tube
D ,, ,, mean diameter of the spiral
$k = \dfrac{\lambda}{c_p\,.\,\gamma}$,, ,, thermal conductivity of air
ν ,, ,, kinematic viscosity of air.

Mass velocity of air. The quantity of air passing between the fins does not depend on the velocity only, but also on the temperature of the air. The quantity of heat removed from the finned surface depends mainly on the weight of air passed. Therefore, all measurements of volume must be calculated to one comparative temperature. It is beneficial to introduce the value of mass velocity, i.e. the quantity of air (kg) which passed the area of 1 m², thus

$$v_w = \frac{G}{A} = \frac{V\,.\,\gamma}{A}\ (\text{kg/m}^2\,\text{s}),$$

where A denotes the flow area (m²)
G ,, ,, flow rate of air (kg/s)
V ,, ,, volume flow rate of air (m³/s)
γ ,, ,, specific weight of air (kg/m³)

Fig. 63 illustrates the influence of mass velocity of air upon the surface coefficient of heat transfer q. Measurements were carried out on various types of finnings and fin dimensions according to table 13 [55]. It is evident, that q changes only very slightly between spacings 1·2—3·3 mm, it deteriorates rapidly at smaller spacings, and increases at larger ones. In this case the transfer coefficient q is calculated from the temperature difference between the *inlet* temperature of the air and the mean temperature of the fins. The resultant correlation is

$$q \propto v_m^{0{\cdot}667}.$$

If we calculate with the mean air temperature, i.e.

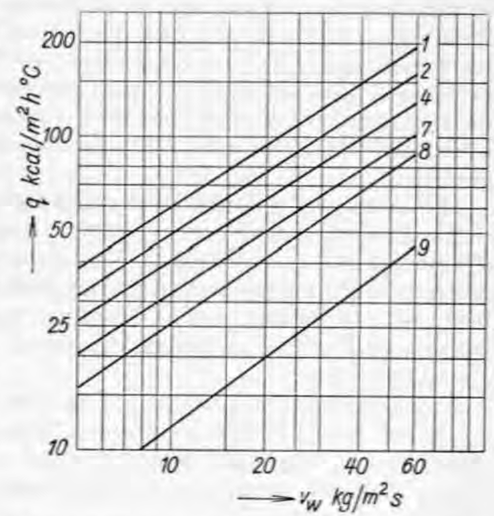


FIG. 63. Influence of "mass velocity" of air upon the surface coefficient of heat transfer q.

the mean value of the inlet and exit temperature of the air, the above relation changes to

$$q \propto v_m^{0.544}.$$

In this case the absolute value of q changes evidently too; it is larger because the same quantity of heat removed is calculated at a smaller temperature difference. For the assessment of heat transfer coefficients it is important to know if they are related to the inlet or mean temperature of the air.

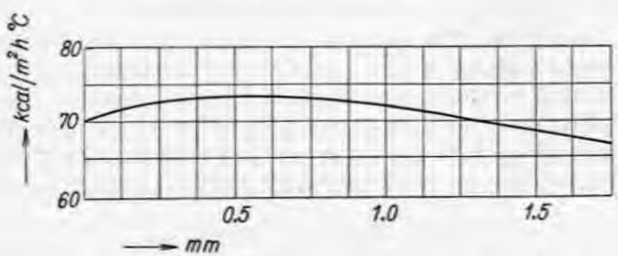


FIG. 64. Influence of the thickness of varnish layer upon the surface coefficient of heat transfer q.

The influence of surface quality. As mentioned before, the heat transfer coefficient is proportional to surface friction. Heat is better removed by a rough surface than by a smooth one. In the case of large projections of the rough surface, increased resistance to air flow is not caused by surface friction, but by pressure head differences between the windward and leeward edges of the projections. This increased resistance impedes heat transfer. Therefore, it is not true, that heat is transmitted better by a roughly cast surface than by a machined one. However, a mat surface has definite advantages over a glossy one.

The influence of a varnish layer on q is illustrated in Fig. 64. It can be seen that a fin surface coated with a varnish layer having a thickness of up to 0·5 mm exhibits an increasing q. This improvement of the coefficient is due probably to the uneven surface projections being levelled by the varnish and thus a smooth surface created. However, at thicker varnish layers the bad thermal conductivity of the varnish makes itself felt, and q deteriorates accordingly [19].

A steel surface transmits 5—10 % more heat than aluminium.

The influence of the cylinder diameter. Values of surface transfer coefficients determined for a cylinder do not hold for cylinders of different diameters. This is due to the varying ratio of surface areas exposed to laminar and turbulent flow respectively.

To date only few data are available on the influence of cylinder diameter on q. By measuring the heat removed from two similarly finned cylinders

having diameters of 93 and 161 mm respectively, the following correlation was established:

$$q \propto \frac{1}{D^{0.25}}.$$

When measurement results obtained from one cylinder are to be applied to another cylinder, first we must establish the scale factor

$$m = \frac{D_x}{D_t},$$

where D_x denotes the diameter of the calculated cylinder and
D_t „ „ diameter of the test cylinder.

All test results are then recalculated for the new cylinder by multiplication by the scale factor, i.e.

$$\begin{aligned} D_x &= D_t \,.\, m, \\ b_x &= b_t \,.\, m, \\ h_x &= h_t \,.\, m, \\ s_x &= s_t \,.\, m, \\ v_x &= v_t \,.\, m, \end{aligned}$$

$$\text{and } q_x = \frac{q_t}{m}$$

The influence of the air stream direction. If the air flow is not parallel to the plane of fins, the continuity of the air flow between the fins is disrupted. The investigation of air traces on fin surfaces coated with a mixture of kerosene and lamp black produced evidence of an intensive turbulence in the fin interstice [19]. Heat transfer is considerably augmented by this turbulence. Fig. 65 illustrates the possibility of reducing the temperature difference between fin and air (in %) for a given cylinder and a fixed quantity of heat transferred in the case of an air stream inclined by 45° from the plane of the fins. The denser the finning, the smaller is the influence of the inclined air stream.

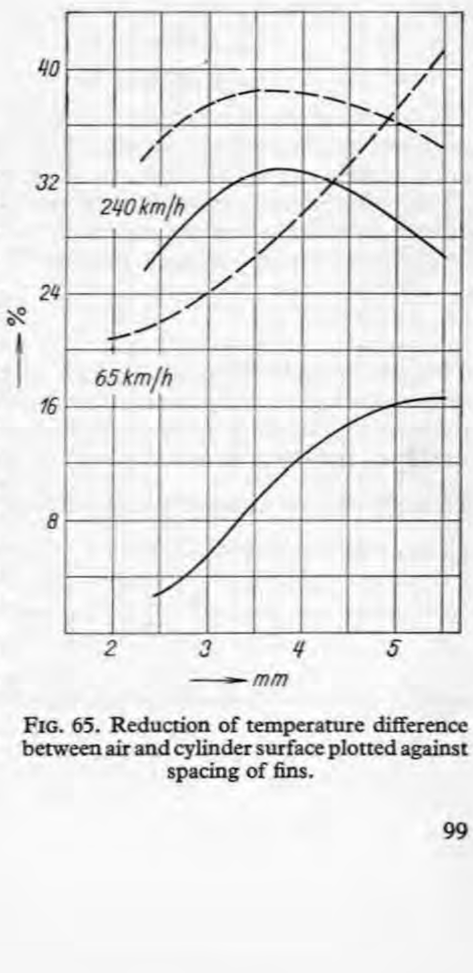


FIG. 65. Reduction of temperature difference between air and cylinder surface plotted against spacing of fins.

4. The coefficient of heat transfer from the fin base surface area

If large quantities of heat have to be removed from a limited cylinder surface area, a maximum total cooling surface must be obtained by a suitable grouping of the fins with due consideration of the high heat transfer coefficient q. Calculations are simplified and better orientation is secured by using transfer coefficients related to the fin base surface area.

The fin base surface area of a finned cylinder is

$$A_b = 2\pi \,.\, r_b \,.\, f \,.\, S \tag{29}$$

where r_b denotes the radius of the cylindrical surface of the fin base
f „ „ number of fins
S „ „ overall spacing; $S = s_m + b$.

The transition area of cooling air between the fins A_t is calculated from the equation

$$A_t = 2 \,.\, f \,.\, s \,.\, h \tag{30}$$

where s denotes the spacing between fins
h „ „ height of fins
f „ „ number of fins

In accurate calculations of the cross section of the area between the fins we must take into account the bevelling of the fin and the chamfering at the base and top. However, equation 30 is sufficient for calculations.

Some simplifying suppositions are introduced in calculations of the base transfer coefficient. Fins are only slightly bevelled and, therefore, they are considered as having a constant thickness

$$b = \frac{b_b + b_t}{2}$$

Further we shall suppose a constant coefficient of heat transfer q for the whole surface of the fin. Temperature difference Δt is distributed symmetrically to the axial plane of the cylinder. Heat losses at the top edge of the fin are not considered individually, and the error thus created is compensated for by a correction of the calculated fin height $h' = h + \frac{b_t}{2}$. Subject to these simplifying suppositions the quantity of heat removed from a circular fin, is

$$Q_1 = \frac{4\pi\, q \left(r_b + \frac{1}{2} h\right)}{a} \,.\, \Delta t_b \,.\, \tanh a \,.\, h',$$

where Δt_b denotes the mean temperature difference at the fin base, and

$$a = \sqrt{\frac{2\, q}{\lambda\, b_m}}$$

The quantity of heat Q_2 removed from the cylinder surface between two fins is calculated from the equation

$$Q_2 = 2\pi \,.\, r_b \,.\, s_b \,.\, q \,.\, \Delta t_b$$

provided, that q has the same value for both the cylinder and fin surface.

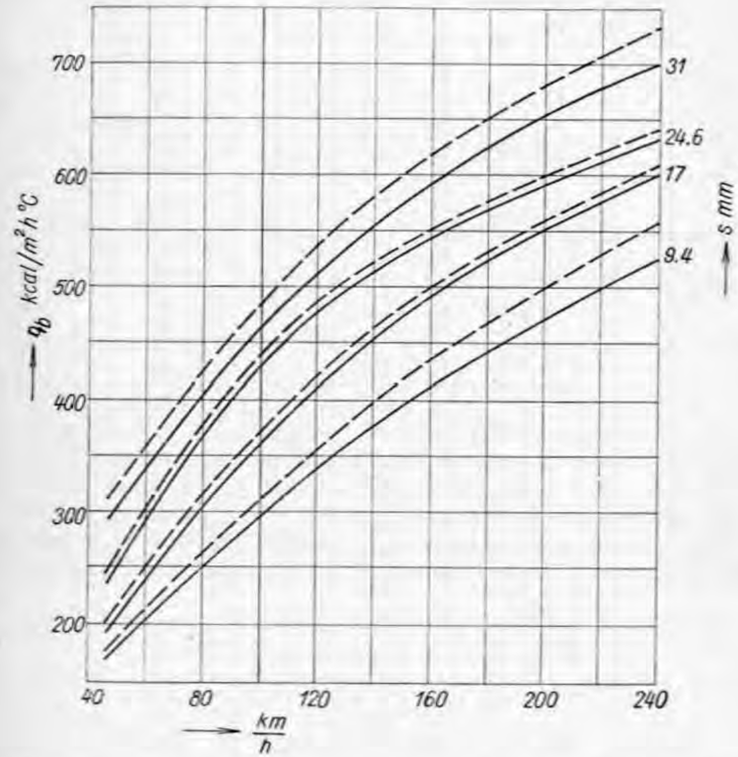


FIG. 66. Basic coefficient of heat transfer measured on a freely exposed cylinder without cowling. Values calculated according to equation (31) are represented by dashed lines.

The quantity of heat removed from a unit area of the cylinder surface at a temperature difference of 1 °C between air and cylinder surface in unit time is given by the equation (applied by Biermann):

$$q_b = \frac{Q_1 + Q_2}{2\pi \,.\, r_b \,.\, b \,.\, \Delta t_b} = \frac{q}{b}\left[\frac{2}{a}\left(1 + \frac{h}{2r_b}\right) \tanh a \,.\, h' + s_b\right] \tag{31}$$

Fig. 66 illustrates base transfer coefficients q_b measured on a finned cylinder without cowling having fins spaced at 6·35 mm and of various heights. For comparison calculated values of q_b are entered in the diagram also. It is evident that results obtained from equation 31 correspond very well with actual measurements [55].

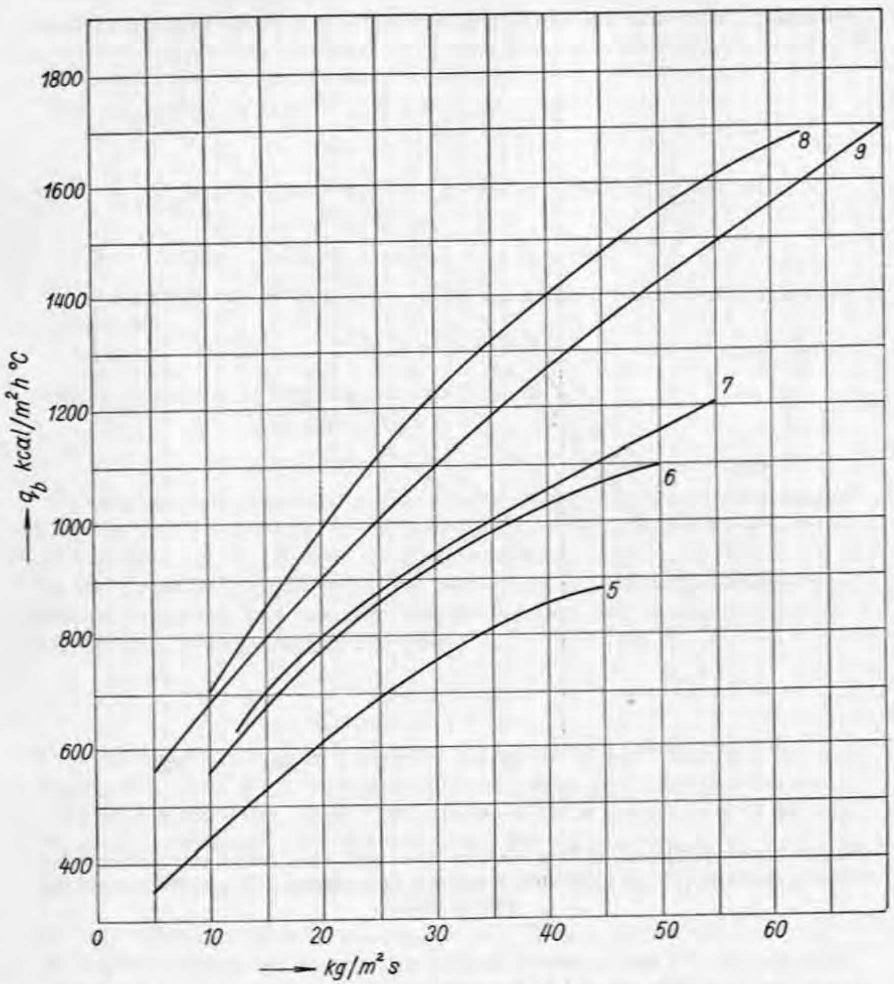


FIG. 67. Relationship between basic coefficient of heat transfer and "mass velocity" of air in a cylinder with cowling.

The correlation between coefficients q_b and mass velocity of air are illustrated in Fig. 67 for the case of a cylinder with cowling where fin spacing varies and fin height remains constant. Dimensions of fins used for the

measurements are given in Table 13. Calculated and measured values are also in agreement.

Table 13

Fin Dimensions of the NACA 587 Test Cylinder (in mm)

Mark	Spacing	Height	Thickness	Interstice
1	6.35	31.0	1.0	5.35
2	6.35	24.6	1.0	5.35
3	6.35	17.0	1.0	5.35
4	3.81	24.6	1.0	2.81
5	4.20	31.0	0.89	3.31
6	3.49	31.0	0.89	2.60
7	2.84	31.0	0.89	1.95
8	2.10	31.0	0.89	1.21
9	1.45	31.0	0.89	0.56

The influence of fin spacing on the coefficient q_b is shown in Fig. 68. Results of measurements carried out on cylinders with five different types of finning are entered in the diagram. The fins employed had a thickness $b = 0{\cdot}89$ mm and a height $h = 31$ mm. The smaller the spacing, the higher is the value of q_b; with the exception of spacings below 1·45 mm, where heat transfer deteriorates due to the interference between the boundary layers.

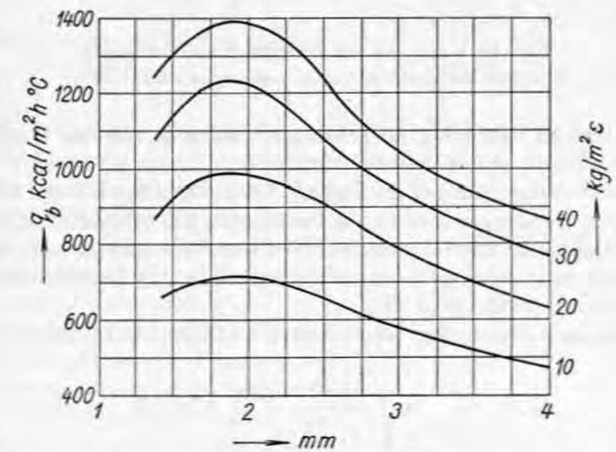


FIG. 68. Influence of fin spacing upon the basic coefficient of heat transfer at a given "mass velocity" with finning according to Table 13, h = 31 mm, b = 0.89 mm.

These results are in full agreement with conclusions arrived at in the preceding discussions. Dimensions of fins used in tests are again given in Table 13. In this case the base transfer coefficient q_b is calculated to the inlet temperature of the cooling air.

Fig. 68 illustrates the correlation between fin spacing and q_b at a particular mass velocity v_w. Maximum q_b is obtained at an optimal spacing of about 2 mm. The optimum distance between fins depends on the fin thickness.

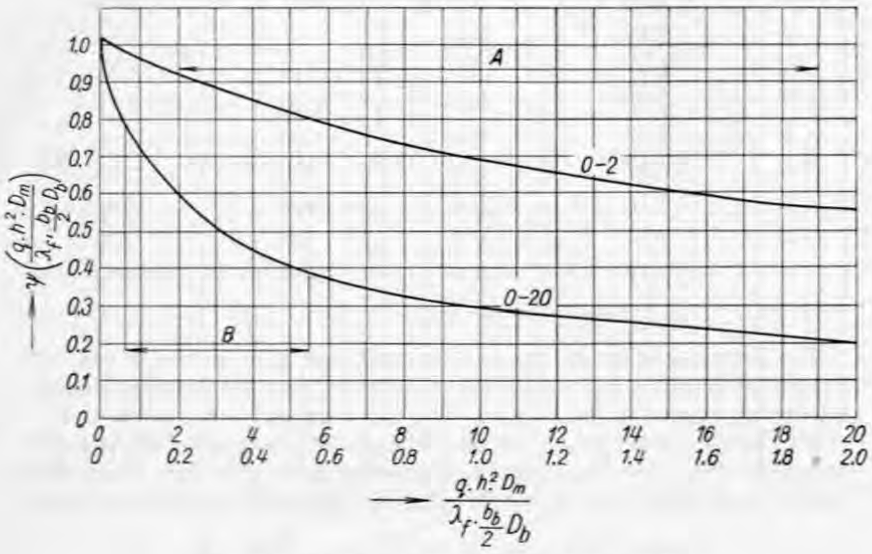


FIG. 69. Values of the dimensionless function ψ.
B: conventional cast iron fins. A: conventional aluminium fins

However, with an increasing fin thickness spacing is reduced and so is the number of fins per unit of cylinder length.

If the mean temperature of cooling air, i.e. the arithmetic mean of the inlet and exit temperatures, is used in the calculation, the calculated value of q_b is higher than obtained from calculations based on the inlet temperature. At low mass velocity v_w, where air is warmed up considerably between the fins, the increase of q_b amounts up to 55%.

According to Löhner, the base transfer coefficient is calculated from the formula:

$$q_b = q \frac{A}{A_b} \cdot \psi \left(\frac{q \cdot h^2 \cdot D_m}{\lambda_f \dfrac{b_b}{2} D_b} \right) + \pi \cdot D_b \cdot s \cdot q \qquad (32)$$

where A denotes the surface of one fin

A_b denotes the cross section area of a fin base, $A_b = \pi \cdot D_b \cdot b_b$
D_m „ „ mean fin diameter; $D_m = D_b + h$
D_b „ „ diameter of the fin base
λ_f „ „ coefficient of thermal conductivity of the fin

The function ψ of the dimensionless term $\left[\dfrac{q \cdot h^2 \cdot D_m}{\lambda_f \dfrac{b_b}{2} D_b} \right]$ is plotted in Fig. 69.

This function expresses the fin efficiency η_f, by which we must multiply all results obtained from calculations at which the assumption of equal temperature of base and fin surface was made. A small thermal conductivy λ_f will suffice in cases of small values of q and h.

The second term in equation 32 represents the quantity of heat transferred by the cylinder surface in the fin interspace. With densely spaced, high fins this term may be omitted without a grave error being committed. For the calculation of the transfer coefficient q Löhner uses the temperature difference between the mean temperature of the fin and the mean temperature of cooling air.

Values of the base transfer coefficient q_b measured on a cylinder with cover are plotted against air velocities v_m in Fig. 70. Fins employed for this test were triangular, i.e. with a sharp edge at the top. Dimensions and markings of the fins employed are given in Table 14. The cylinder diameter at the fin base was $D_b = 160$ mm.

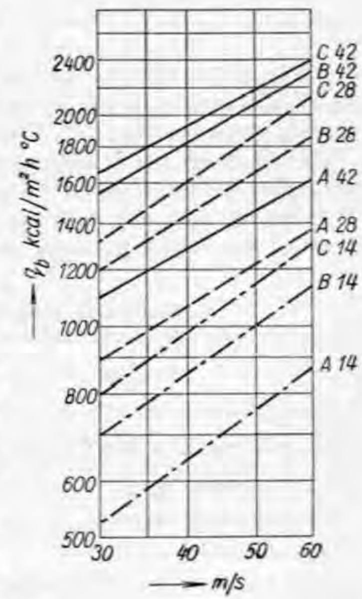


FIG. 70. Influence of air velocity and fin height upon the coefficient of heat transfer in a cylinder with cowling.

5. Power consumed by the cooling system

In a well designed engine only a small portion of the engine output is used up by the cooling system. Factors determining this power consumption are the quantity and pressure of cooling air required, efficiency of the cooling fan, and the conditions of conveying the cooling air.

Heat transfer from a solid body to air always demands a certain amount of power, at least that required to overcome surface friction. In the exceptional case of transmitting heat to air from the surface of the car body, friction becomes an unavoidable part of the air resistance of the vehicle.

Table 14

Fin Dimensions of the Löhner Test Cylinder (in mm)

Mark	Spacing	Height	Thickness at the root
A 42	8	42	4
A 28	8	28	4
A 14	8	14	4
A 7	8	7	4
B 42	5	42	4
B 28	5	28	4
B 14	5	14	4
B 7	5	7	4
C 42	4	42	2
C 28	4	28	2
C 14	4	14	2
C 7	4	7	2

However, this cooling method is complex and unpractical and it was never used in automobiles; it was applied in certain racing aircraft with water-cooled engines, where the wing surface was used as a radiator.

The minimum power consumed by the cooling system is calculated from the surface friction required for the transfer of heat to air. Table 15 includes characteristic data of conditions prevailing in cooling a 100 b.h.p. engine mounted in a vehicle travelling at 50 m.p.h. (80 km/h).

Table 15

Minimum Cooling Input for a 100 b.h.p. Engine

Coolant	Units	Air	Water
Heat removed by cooling 0.6 b.h.p.	kcal/h	38,000	38,000
q at a speed of 80 km/h	kcal/m² h °C	100	100
Temperature gradient	°C	160	60
Required cooling surface area	m²	2.38	6.33
Friction loss for 1 m² of surface area at 80 km/h	kg/m²	0.1	0.1
Loss by friction of air	b.h.p.	0.74	1.97
Minimum input for cooling	%	0.74	1.97

Supposing the value of the surface transfer coefficient q being 100 kcal/m²h °C at a travelling speed of 50 m.p.h. (80 km/h), the part of engine performance required to overcome surface friction at a temperature difference of 60 °C amounts to 1·97% of the b.h.p.; at a temperature difference of 120 °C the figure would be reduced to 0·74 % of the b.h.p. However, a power consumption as low as the latter is never attained in vehicles.

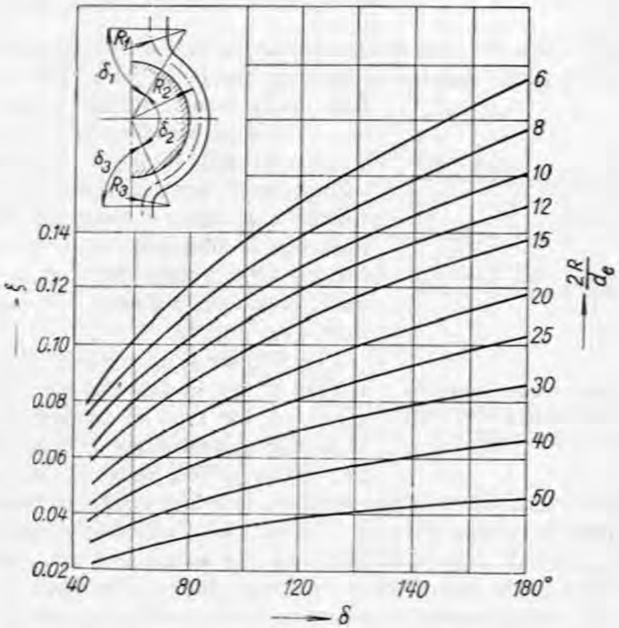


FIG. 71. Chart for the determination of the pressure drop factor.

In practice the cooling surface is formed by fins and the cooling air flows through the narrow space between them. When investigating only the loss on pressure head caused by the flow around the cylinder, the fin interspace may be considered to be a bent tube of rectangular cross section. Pressure loss in the bent tube is composed by the following components:

a) the component Δp_1 calculated according to equation 8, i.e.

$$\Delta p_1 = \frac{\lambda \, . \, l}{d_e} \, \frac{\varrho}{2} \, v_m^2,$$

where d_e is the equivalent diameter calculated from equation 24,

b) the component Δp_2 representing the head loss in the bend:

$$\Delta p_2 = \gamma \xi \frac{v_m^2}{2g} \tag{33}$$

where v_m denotes the mean air velocity in the duct (m/s)

ξ „ „ factor of pressure drop in the bend, plotted in Fig. 71. The respective angles and radii must be established in each individual case [5].

The air flow in a closed duct between the fins is best investigated by coating the fin surface with a mixture of kerosene and lamp black. Traces of the air flow on the coated surface supply evidence of a centripetal streaming on the duct surface. This is explained by the fact that the air layer adjacent to the surface moves with a smaller velocity than layers in the inner region of the duct, and is subjected thus to a smaller centrifugal force during the flow in a circular duct. Therefore, a double flow occurs in the duct – see Fig. 72.

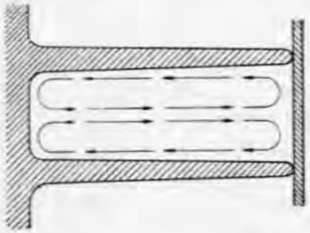


FIG. 72. Double vortex in the fin interstice due to friction between air and fin surface.

The velocity head required for the propagation of the air flow between the fins depends on the type of finning and spacing. It is proportional to the 1·75th power of the air velocity, 0·75th power of the mass of the air, and the 1·25th power of the circular equivalent of the rectangular duct.

In automobile engines the type of finning selected ensures optimal cooling conditions. Aircraft engines without fans rely on the dynamic pressure head corresponding to the speed of the aircraft (during climbing). Fin spacing and type of finning must be determined in accordance with the dynamic pressure available and without regard to the resultant weight of fins. Were an aircraft engine equipped with a fan, an optimal fin spacing could be applied independently of the aircraft speed.

The correlation between the surface transfer coefficient q and the velocity head in the fin interspace Δp is shown in Fig. 73. Minor losses occuring at the inlet and exit of air are disregarded in this diagram. A rapid deterioration of cooling conditions in interspaces below 0·5 mm is evident. Measurements were carried out on a cylinder having a fin base diameter of 118 mm and equipped with a cowling in close contact with the fin edges [6].

In Fig. 74 mass velocity of air is plotted against velocity head. It is evident that at a given velocity head the quantity of air passed between the fins decreases with diminished spacing.

Small fin thickness and small interspace are prerequisites of economical cooling. This means a smaller quantity of cooling air but higher velocity heads. Therefore, it is imperative to prevent air losses through leakages in the cowling.

With a flat plate exposed to turbulent flow the friction factor is proportional to the 1·8th power of the air velocity [35], [55]. For flows through a finned surface the value of the exponent varies from 1·75 to 1·96 according to the spacing and height of the fins. The friction factor is directly proportional to

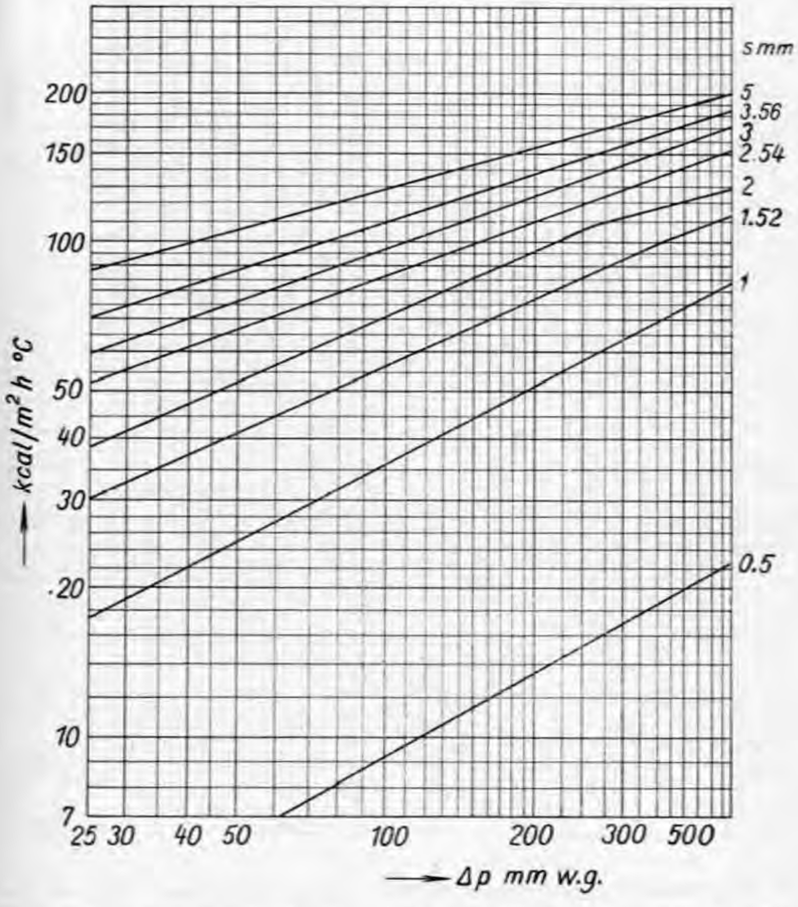


FIG. 73. Surface coefficient of heat transfer q plotted against pressure head Δp in the interstice.

the velocity head and subject to the same physical relations. Power consumed by the cooling system is the product of velocity head and quantity of air passed, i.e. theoretically it varies with the 2·8th power of the air velocity.

Tests carried out on finnings listed in Table 13 have shown, that the power consumed by the cooling system is proportional to 2·69 power of the mass velocity of air v_w. In Fig. 75 the transfer coefficient q_b is plotted against power consumed for cooling. The latter is expressed in h.p. per 100 mm cylinder length. At a given power consumption the highest value of q_b is

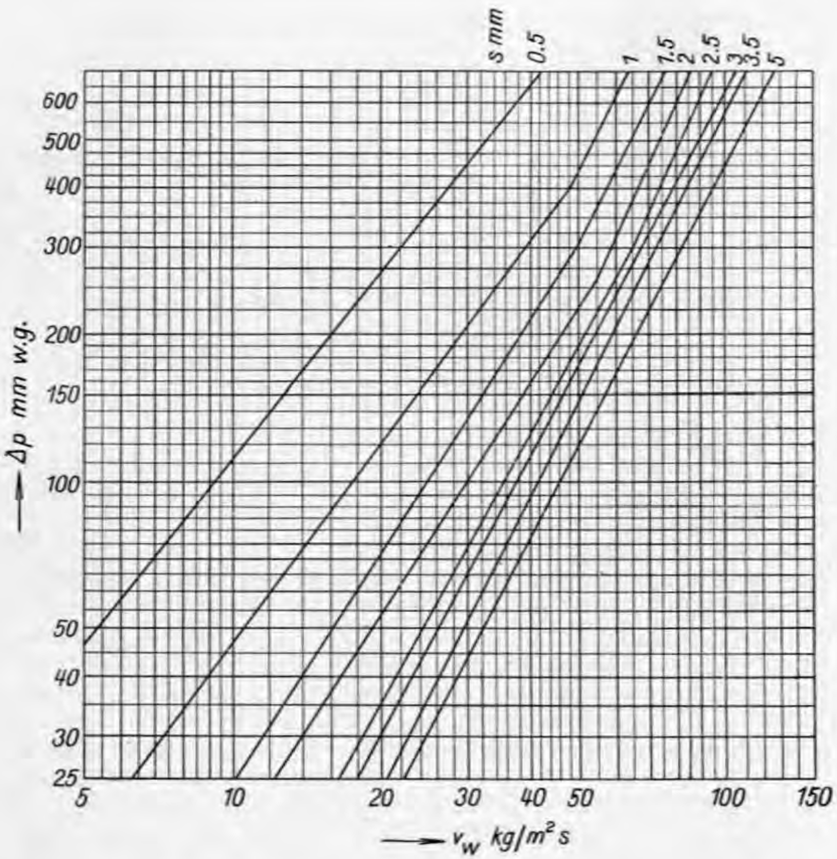


FIG. 74. Relationship between quantity of cooling air in kg flowing through a transition area of 1 sq. m per second and the pressure head Δp for different fin spacings s.

obtained in finnings with an optimal spacing, i.e. q_b gradually increases with diminishing spacing up to an optimal value. Beyond this limit a further reduction of the fin spacing does not result in an increase of the coefficient q_b, on the contrary, q_b decreases. Optimal values were obtained with finnings listed in Table 13. The optimal spacing was 2·1 mm, and by its reduction to 1·45 mm a considerable deterioration of q_b took place. A comparison of Fig. 67

and 75 shows that the correlation between optimal fin spacing and highest q_b corresponds to the correlation between optimal mass velocity v_w and smallest power consumed by the cooling system.

Small fin spacing means increased friction and higher air temperatures. In the case of automobile engines the output consumed by the cooling system must be considered in the first place, and fin weight remains a secondary factor.

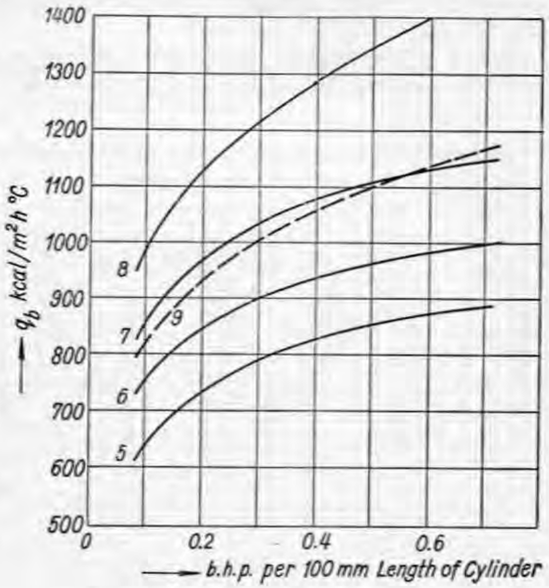


FIG. 75. Cooling input in relation to heat transfer coefficient. Curves are numbered according to fin dimensions in Table 13.

In the case of a given engine and type of finning it is of some interest to find out the relation between the power consumed by the cooling system and the indicated engine output (i.h.p.) at a certain engine speed. Results of measurements carried out on an in-line aircraft engine are shown in Fig. 93. The diagram shows that the quantity of cooling air is proportional to 1·12 power of i.h.p. Thus, for constant engine speeds

$$W \propto \text{i.h.p.}^{1\cdot 12} \qquad (34)$$

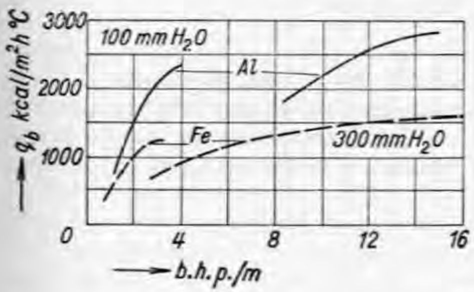


FIG. 76. Coefficient of heat transfer related to cooling input b. h. p. per 1 metre of finned cylinder length for pressure heads 100 mm w.c. (left) and 300 mm w.c. (right) for aluminium (Al) and steel (Fe) fins.

The portion of engine performance consumed by the cooling system does not increase with the third power of the indicated output, but is proportional approximately to 3·35 power of i.h.p. Thus, the conclusion is that it is uneconomical to improve cooling at higher engine outputs by increasing the quantity of cooling air.

From this it is evident, that

minimum power consumption by the cooling system is attained with a finning which, at a given velocity head, presents the highest base transfer coefficient q_b. Optimal conditions can be determined from the diagrams illustrated in Fig. 78. and 79.

Values of base transfer coefficients q_b are plotted in Fig. 76 against values of power consumed by the cooling system per 1 metre of finned cylinder length.

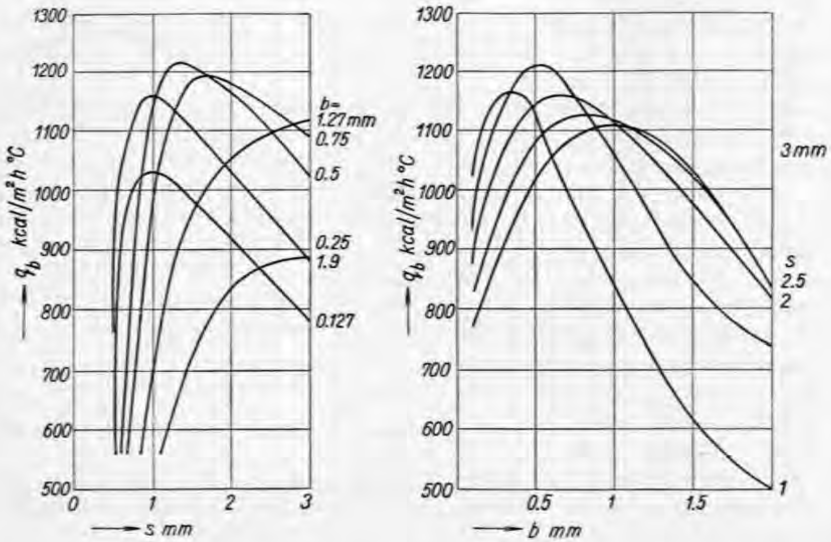


FIG. 77. Dependence of the coefficient of heat transfer upon fin spacing s and fin thickness b for different types of finning with the same unit weight of fins $m = 0.595$ kg/dm².

Two velocity heads, 100 and 300 mm of water column, are considered in the diagram. Standard types of finnings are denoted by dots. It can be seen that for a 100 mm velocity head almost optimum conditions are attained, whereas for 300 mm the performance of aluminium finning is about 33 % below optimum.

The corresponding figure for steel fins is about 36%.

6. The utilization of fin weight

It is of great importance to designers of aircraft engines to remove a given quantity of heat by using the smallest possible weight of material. It is useful, therefore, to introduce a new factor m representing fin weight in kg per

1 dm² of fin base surface area. This weight is calculated from the equation

$$m = \gamma \frac{hb}{S}\left(1 + \frac{h}{2r_b}\right) \text{ (kg/dm}^2\text{)} \tag{35}$$

where γ denotes the specific weight of the fin material in kg/dm³, and
h, b, δ, r_b are dimensions of the finning in dm.

A comparison of different finning types having the same m leads to interesting conclusions. Fig. 77 depicts the correlation between the base transfer coef-

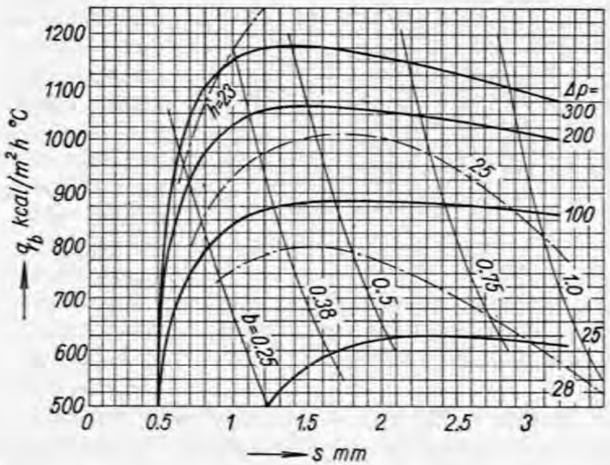


FIG. 78. Heat transfer coefficient related to the specific properties of finning for steel fins of a constant unit weight = 0.595 kg/dm².

ficient q_b and fin spacing s for steel fins having $m = 0{\cdot}595$ kg/dm². At a velocity head of 300 mm water column maximum q_b is attained at a spacing of about 1·5 mm and a fin thickness of 0·5 mm. However, this type of finning approaches the limits of production possibilities. The diagram clearly illustrates how rapidly q_b deteriorates with increasing fin thickness at constant m.

Fig. 78 illustrates how the coefficient q_b for steel fins is influenced by various factors, i.e. fin height h, fin thickness b, velocity head Δp, and fin spacing s. The same correlations for aluminium fins are plotted in Fig. 79. From these diagrams it is evident that highest values of q_b at lower velocity heads Δp are attained with larger spacings s and larger thickness b. At a given fin thickness the q_b curve has a distinct peak and declines rapidly with increased spacing s.

The comparison of both diagrams shows the superiority of thin aluminium fins over steel. Unfortunately this advantage cannot be fully exploited in practice, because thin aluminium fins are very difficult to produce. The same holds for copper fins which, produced in paper thickness, could well replace aluminium fins. However, thicker copper fins have no marked advantages over aluminium.

Optimum dimensions of aluminium fins plotted against q_b are shown in Fig. 80. We can see that e.g. the best dimensions for $q_b = 1000$ kcal/m² h °C are $b = 0{\cdot}21$ mm and $h = 16$ mm. Fins of these dimensions are impossible for production reasons; practical dimensions are approximately ten times larger. The same applies to fin spacings. Fig. 81 shows similar values for steel fins. Both diagrams are constructed for a velocity head of 100 mm water column which is about the normal head used in automobile engines.

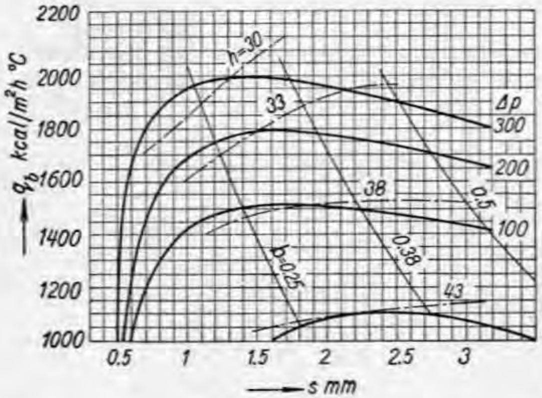


FIG. 79. Heat transfer coefficient related to the characteristic properties of aluminium fins with a constant unit weight 0.232 kg/dm².

Fig. 82 shows the relation between q_b and the weight of aluminium and steel fins. This correlation is of interest to designers of aircraft engines. However, it must be borne in mind that the illustrated correlation applies to ideal fins. Steel fins approach this theoretical diagram quite closely, but aluminium fins corresponding to the diagram are too thin for practical purposes. Aluminium fins could transfer 23% more heat, or could be 50% lighter for a given heat transfer, if their weight would be fully utilized for the transfer of heat. Some further interesting conclusions can be drawn from Fig. 82. At the same unit weight m and a velocity head of 100 mm, aluminium fins can transfer 2·25 times more heat than steel fins. However,

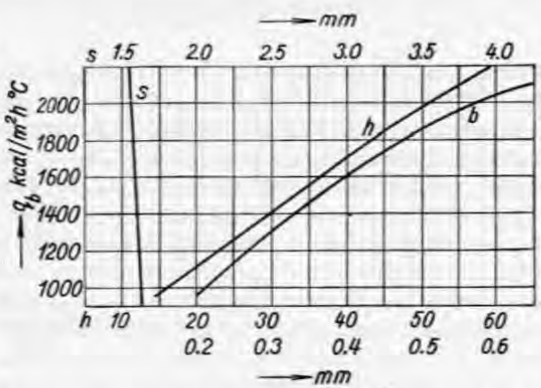


FIG. 80. Optimum aluminium fin dimensions for a desired coefficient of heat transfer.

in practice the difference between aluminium and steel fins is far smaller, because optimum spacing and thickness of aluminium fins is not practicable.

Fig. 82 shows also that the quantity of heat removed can be considerably increased by increasing the unit weight of fins. For aluminium fins at $q_b = 1600$ a double weight of the fin means heat transfer increased by 24%. For steel fins at $q_b = 600$ an increase of m by 100% results in a 21% increase of the quantity of heat removed. Better cooling achieved by an increased m is paid for by an overall increase of the cylinder weight. Therefore, it must be calculated, wether the use of a fan, i.e. improvement of cooling by increasing the air velocity, is not more economical.

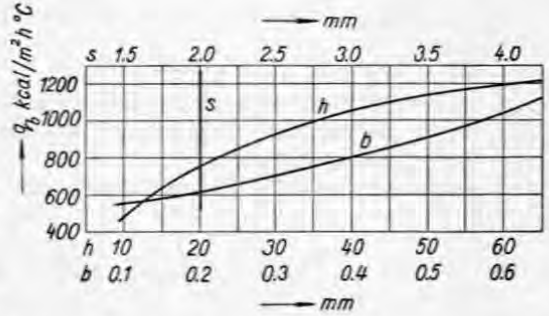


FIG. 81. Optimum steel fins for a desired coefficient of heat transfer.

7. Fin production methods

Before making the final decision on the type of finning to be used, the designer must consider the methods by which fins are produced. Individually mounted cylinders without excessive thermal stresses can be fitted with cast fins such as those for heads of automobile engines. At a mean thickness of 2 mm and height of 40 mm fin spacing should never be less than 6 mm. Castings of smaller spacings present great difficulties.

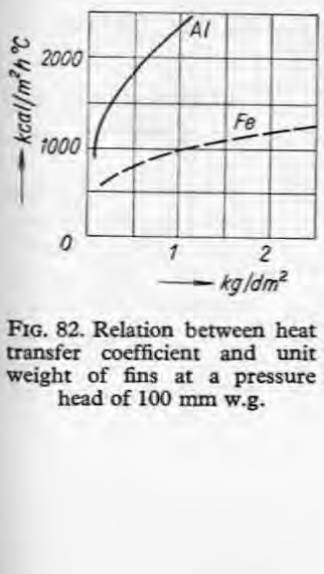


FIG. 82. Relation between heat transfer coefficient and unit weight of fins at a pressure head of 100 mm w.g.

Machined fins are used for cylinders subjected to high thermal stresses. In these cases cylinders made of cast iron can have spacings of 3·5—4 mm, thickness of 1 mm and heights of 15—20 mm. Fin dimensions for steel cylinders can be further reduced to $s = 1{\cdot}5$ mm, $b = 0{\cdot}5$ mm, $h = 25$ mm. However, these figures should be regarded as limits of production possibilities closely approaching theoretical optimum values.

Theoretically derived fin dimensions cannot be realized with materials having good thermal conductivities (aluminium, copper). When using machined fins it is necessary to calculate if the increased production cost is compensated for by the cooling system. In cast iron cylinders with cast-on fins the high percentage of waste must be considered.

Fig. 83 shows the correlation between the base transfer coefficient q_b and quantity of cooling air for various types of fins. The black dots on the curves indicate positions with a velocity head of $\Delta p = 200$ mm w.c. The following types of fins are represented in the diagram:

1. steel fins, $s = 3{\cdot}6$ mm and $h = 17$ mm, turned from compact steel blocks;
2. aluminium fins, $s = 3$ mm, $b = 1$ mm, $h = 20$ mm mounted in the machined grooves of a steel cylinder by rolling;
3. copper fins cut from 0·5 mm thick sheet metal, $s = 2$ mm, $h = 22$ mm, brazed to a steel cylinder;
4. copper fins as in 3, but intermittent and off set.

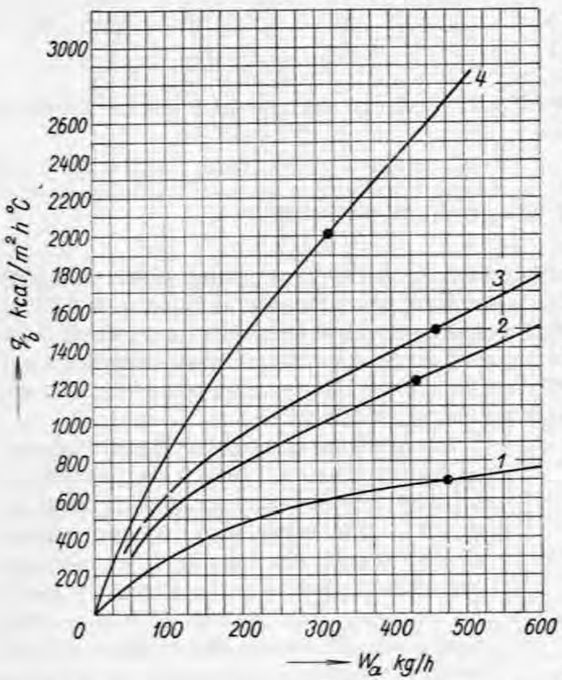


FIG. 83. Coefficient of heat transfer plotted against flow rate of cooling air.

1 — machined steel fins, $h = 17$ mm, spacing $s = 3.6$ mm. *2* — aluminium fins rolled into machined grooves of a steel cylinder, $h = 20$ mm, $s = 3$ mm, $b = 1$ mm. *3* — copper fins, brazed to steel cylinder, $h = 22$ m, $s = 2$ mm, $b = 0.5$. *4* — copper fins as above, but intermittent and off-set. Marked points represent values for a pressure head of 200 mm w.g.

It can be clearly seen from the diagram how cooling can be improved for a given quantity of air by the application of suitably produced fins. The intermittent copper fins are the best, because at each intermittance similar conditions exist as at the inlet edge, i.e. a small thickness of the boundary layer. Cooling conditions are improved on account of a higher overall resistance to air flow and this means an increased velocity head.

8. Distortion of the cylinder

The different temperatures on the circumference of the cylinder are the cause of the distortion of the circular shape of the cylinder surface. In such cases piston rings have no reg-

ular contact with the inner surface of the cylinder and blow-by occurs, the engine performance is reduced, oil consumption increased, the piston is overheated and even a seizure of piston rings may occur.

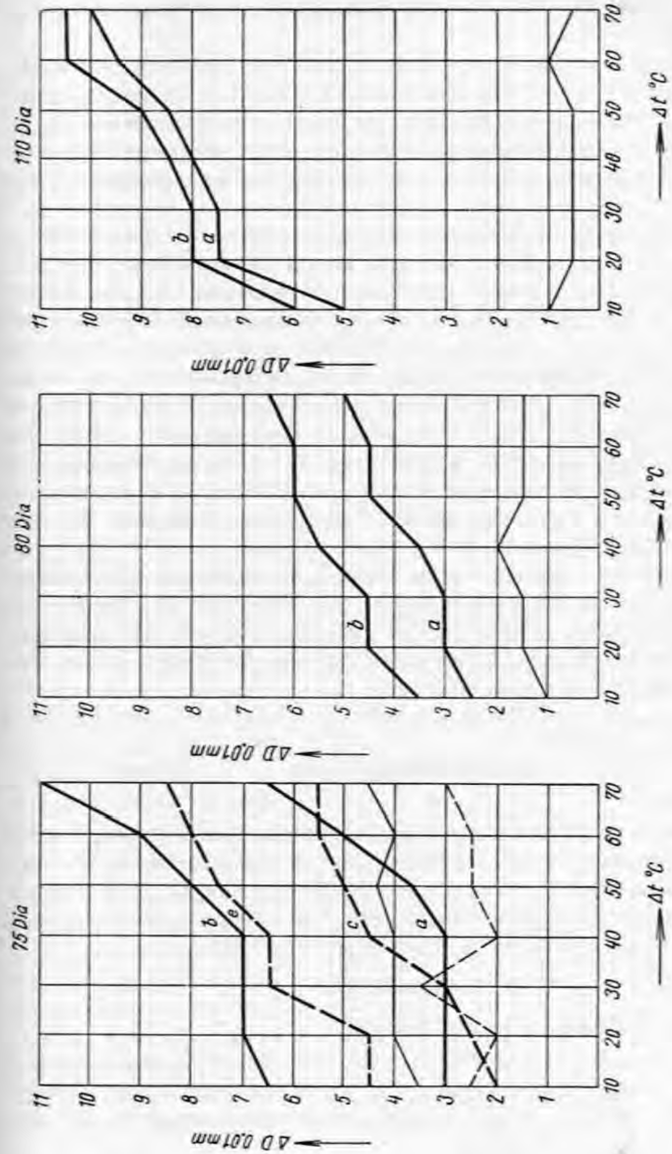


FIG. 84. Change of cylinder dimensions due to uneven temperatures, for cylinder diameters 75, 80 and 110 mm.

Arrows signify direction of heating in a plane normal to the axis of the crankshaft. Temperatures in t °C were measured at the front and rear of the cylinder. Change in dimensions at the 75 mm cylinder measured also for heating in the direction of the plane of the crankshaft axis (dashed c, e lines). Deviation is given by the thin curve $b - a$.

The causes of uneven temperatures in the cylinder may be either an increased heat transmission at certain zones of the inner surface or a reduced heat removal from some regions of the outer surface of the cylinder.

The same deficiencies can cause a thermal distortion of the cylinder head with subsequent leakages of valve seats or sealing troubles between cylinder head and cylinder.

Cylinder head and cylinder are sometimes distorted by the uneven initial stresses of fastening bolts. The distortion can be reduced by strengthening the cylinder wall near the flanges and by a proper adjustment of the tightening. If the distortion is not removed entirely in this way, valve seats must be ground in the cylinder head when fixed in a jig exerting the same pressure as the fastening bolts.

The extent of distortion due to uneven temperatures was established by direct measurements on a cylinder internally heated by a blow torch. Temperature differences were measured on the cylinder circumference. Results of these tests are shown in Fig. 84. The diagram shows that distortion is larger in small cylinders. Further it can be seen that the diameter of the heated spot remains almost unchanged but the diameter normal to that of the heated spot is expanded.

The basic temperature of the cylinder was maintained at 40 °C and temperature differences above this value were entered in the diagram. The distortion can be explained as in bimetallic machine parts. By heating a certain spot of the inner wall surface an expansion of the material takes place at that spot, while the expansion of the comparatively cooler fin tops is smaller. The resultant stresses cause a flattening of the cylinder wall.

The distortion of the cylinder can be reduced by cutting the fins through the cylinder wall or by intermittent finning (see Fig. 252). The permissible temperature difference in normally finned cylinders is 40—50 °C maximum. The permissible temperature difference in cylinder heads depends on the design and dimensions of the cylinder head.

9. The warming of air

Air passing between the fins absorbs heat from the walls and its temperature is increased. The warmer the air becomes, the more heat is carried off and less air is needed for cooling. However, the permissible warming of air is limited.

Let us consider one interstice between two fins. The cross section of this interstice is A_t and the quantity of air passing through it, is

$$V = A_t \cdot v_m \; (\mathrm{m^3/s})$$

where v_m denotes the air velocity in m/s.

Heat is absorbed by this quantity of air, and we can calculate the temperature increase during the passage of the air through a given length of interstice. Let us calculate the temperature increase during the passage of a 100 mm wide and 100 mm long fin surface.

Thus

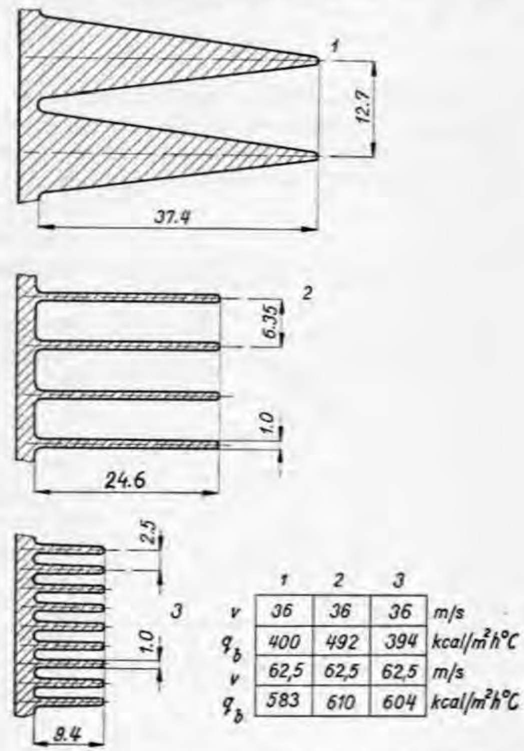
$$t_{100} = \frac{q_b \cdot \Delta t}{100 \cdot c_p \cdot A_{100} \cdot v_m \cdot 3600} \; (^\circ\mathrm{C})$$

where q_b denotes the base transfer coefficient (kcal/m² h °C)
Δt „ „ temperature difference between cylinder and air (°C)
A_{100} „ „ transition area between fins 100 mm wide (m²)
v_m „ „ air velocity (m/s)
cp „ „ specific heat of air at eonstant pressure (kcal/kg °C)

Calculated temperature increases of air are shown in Table 16 for three types of fins illustrated in Fig. 85 at a temperature difference of $\Delta t = 120$ °C. The table clearly indicates the influence of fin types on the warming of air. The quantity of heat removed by types 1 and 2 is approximately the same, but for type 1 we require four times more air and the warming of the air is only $^1/_4$ of that in type 2.

Heat transfer from a finned surface at a given air velocity depends on the temperature difference between the finned surface and air. If the same quantity of heat is to be removed from each cm² of the finned surface, the temperature difference must be also the same along the entire circumference of the cylinder. As air temperature increases during the passage of air between the fins, fin temperature at the air inlet is smaller than at the exit provided that the temperature difference is constant and the entire cylinder surface is cooled with an equal intensity.

This conclusion was confirmed by the results of measurements carried out on an electrically heated cylinder. Thermocouples measuring temperatures of the cylinder wall were placed along the whole circumference of



	1	2	3	
v	36	36	36	m/s
q_b	400	492	394	kcal/m²h°C
v	62,5	62,5	62,5	m/s
q_b	583	610	604	kcal/m²h°C

FIG. 85. Three types of finning with equal coefficients of heat transfer q_b at given air velocity v.

Table 16

Comparison of Three Types of Finning

Type of Finning			1	2	3
Transition area of a 100 mm wide strip of the cylinder surface, (cm^2)			21.63	20.72	5.64
Air flow at a velocity of 36 m/s, (m^3/s)			0.07788	0.07459	0.02034
Air flow at a velocity of 62.5 m/s, (m^3/s)			0.13519	0.12950	0.03525
Heat removed from 1 dm^2 of base area at a temperature gradient of 120 °C, (kcal/s)	at an air velocity of	36 m/s	0.13333	0.16400	0.13140
		62.5 m/s	0.1949	0.20333	0.20134
Warming of air, (°C)	at a velocity of 36 m/s		5.92	7.60	22.36
	at a velocity of 62.5 m/s		4.96	5.42	19.74

the cylinder. Measurement results are shown in Fig. 86. In spite of a regular and uniform heating of the cylinder, temperatures along the circumference differed considerably. Temperature of the cylinder at the air inlet was 76 °C, at the exit 124 °C. Air temperatures were measured at the inlet and exit only, because measuring in the interspace was rendered impossible by the radiation from the fin surface. The rate of temperature increase is, therefore, shown by a straight line in the diagram. Temperature difference remained almost constant which is in full agreement with our theoretical considerations. The rear wall of the cylinder was 48 °C warmer than the front wall, and the total warming of the air was also about 48 °C.

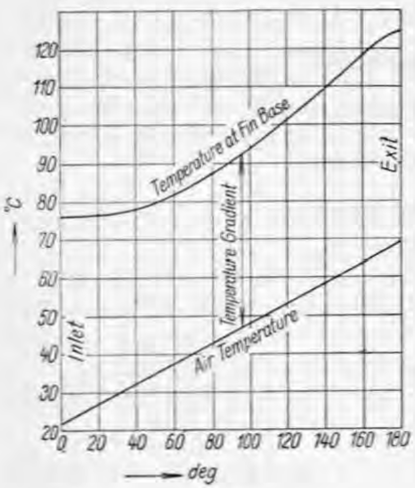


FIG. 86. Distribution of temperatures on the circumference of a cylinder with cowlings.

Temperatures measured on an air-cooled compression-ignition Tatra engine are shown in Fig. 87. The engine had a 110 mm bore and 130 mm stroke.

Measurements were taken on a fully loaded engine running at a speed of 2000 rev/min. Temperature of the cylinder wall closely below the cylinder head was 158 °C on the

air inlet side and 184 °C at the exit, i.e. a temperature difference of 26 °C.

The warming of air may thus greatly influence temperatures on the cylinder circumference and consequently the cylinder distortion. The danger of large

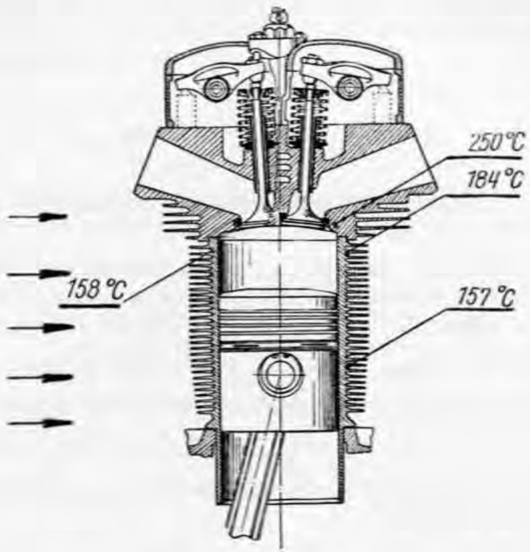


FIG. 87. Cylinder temperature of an air-cooled Tatra oil engine.

temperature fluctuations on the circumference arises particularly with dense finning where a long travel of the air flow between the fins produces a considerable warming of the air.

CHAPTER VII

THE QUANTITY OF COOLING AIR

1. Calculation of quantity of cooling air

The portion of the total heat which must be transferred to air by cooling is determined from the thermal balance of the engine. For a preliminary calculation it may be assumed, that the heat Q removed by cooling equals

$$Q = 0{\cdot}6\text{—}0{\cdot}8\ (\text{b.h.p.}) \times A \times 75 \times 3600 = 0{\cdot}6\text{—}0.8\ (\text{b.h.p.}) \times 632 \quad (36)$$

where $A = 1/427$ kcal/kg m denotes the reciprocal mechanical equivalent

$$1 \text{ h.p.} = 1/427 \times 75 \times 3600 = 632 \text{ (kcal/h)}$$

The quantity of air required for the transfer of the above quantity of heat is calculated by the equation

$$Q = V_w \,.\, c_p \,.\, \Delta t \,.\, 3600 \quad (37)$$

where V_w denotes the mass velocity of air (kg/s)

c_p „ „ specific heat of air at constant pressure (kcal/kg °C)

Δt „ „ temperature increase of air (°C)

The increase of air temperature is larger in air-cooled engines than in water-cooled ones; it depends on the temperature difference between cylinder and air, the air velocity, the size and shape of the space between the fins, and the length of air travel between fins.

In well designed air-cooled automotive engines the temperature increase of air amounts to 60—80 °C; therefore the quantity of cooling air is about half of that required in water-cooled engines of the same performance.

In properly finned engines the quantity of cooling air is 52—104 lb/b.h.p.-h. (20—40 kg/b.h.p.-h). With too widely spaced fins and large heat transfer from the combustion products to the cylinder the quantity of cooling air may rise up to 200 lb/b.h.p.-h (75 kg/b.h.p.-h.) For rough estimates we may assume an air consumption of 2·6 lb/min b.h.p. (1 kg/min b.h.p.)

The larger the increase of air temperature during its flow along the cylinder, the smaller the quantity of cooling air required. Large temperature increase of air is beneficial as each unit of air removes a large quantity of calories. However, we must consider also the fact that the quantity of air removed is

proportional to the temperature difference Δt and coefficient of heat transfer q. Thus

$$Q \propto q \,.\, \Delta t \quad (38)$$

If air temperature is increased during its passage between the fins, and if equal amounts of heat are to be removed from the unit area of the finned surface in the front and rear of the cylinder, the temperature at the rear of the cylinder is necessarily increased.

2. The influence of air velocity

The surface coefficient of heat transmission q depends largely on the air velocity. Higher air velocity means better heat transfer and better cooling.

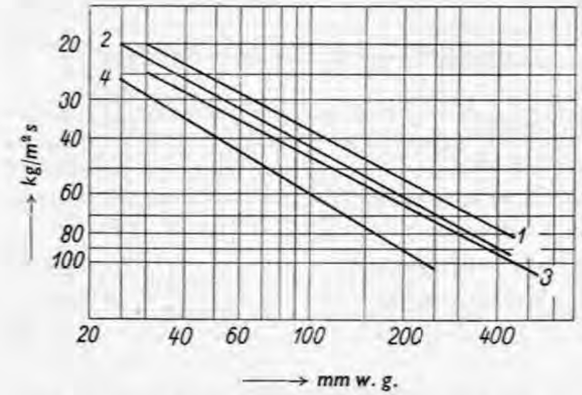


FIG. 88. Air flow in kg related to pressure head Δp.

1. Fin spacing 3 mm, bore 75 mm, without securing bolts. *2.* Fin spacing 5.5 mm, bore 110 mm, cylinder with cowling. *3.* Fin spacing 5 mm, bore 75 mm. *4.* Fin spacing 5.5 mm, bore 110 mm, cylinder without cowling and bolts.

However air velocity must be suitable to the type of finning used. With cylinders exposed to small thermal stress, widely spaced fins and low air velocity should be satisfactory. By increasing the thermal stress, cooling must not be improved by an increased air velocity only; in such case the finning must be changed as well.

The coefficient of heat transfer q increases with 0·73 power of the air velocity. Thus a double air velocity means a coefficient q improved by 66%. If 1 m³ of air removed a heat quantity of 100% Q, 2 m³ of air will remove 166% Q. Thus by increasing air velocity to its double value the quantity of heat removed by 1 m³ of air is reduced to 83% of the former value. Consequently the utilization of air becomes worse and the exit temperature is also lowered. Thus, specific air consumption is increased.

Table 17

Variation of Cooling Input for a Water- and Air-Cooled Engine, if Cooling is Improved only by a Changing Air Flow at an Increase of Ambient Temperature from 0 to 40 °C

Coolant	Water	Air
Temperature of ambient air, °C	0	0
Temperature of cooled surface, °C	80	180
Temperature gradient, °C	80	180
Temperature of ambient air, °C	40	40
Temp. gradient at this temperature, °C	40	140
Reduction of temp. gradient, %	50	22
Mean absolute temperature at 0 °C, °K	313	363
Mean abs. temperature at 40 °C, °K	333	383
Improved heat transfer, %, (11)	4.5	4
Required improvement, %	100	28.5
Must be improved by %	95.5	24.5
Air velocity must be increased by %	100	30
Air supply will increase by %	147	32.8
Velocity head will be increased (39)	5.3 times	1.725 times
Fan input will be increased (40)	13.25 times	2.33 times

From Fig. 74 and 88 we can find that the velocity head of air Δp increases with the 1·3 and 1·96 power of the air quantity according to the type of finning employed. Let us, therefore, assume that, with finnings normally used in automobile engines, the velocity head is proportional to 1·82 power of the air quantity. Table 17 illustrates the increase of power consumed by the cooling system, with unchanged type of finning, if cooling is being improved by an increase of air velocity only. For simplification we suppose that air velocity is directly proportional to the weight of air and

$$q \propto v^{0\cdot73} \qquad \Delta p \propto Q^{1\cdot82} \qquad Q \propto \Delta p^{0\cdot55} \qquad (39)$$

From these correlations we can establish the quantity of air, velocity head and the power consumed by the cooling system at an increased engine performance and constant finning.

If, for instance, engine output is to be increased by 33%, the quantity of cooling air must be increased in accordance with a transfer coefficient q increased also by 33%:

$$133\% = v^{0\cdot73}, \text{ thus } v = 147.7\%.$$

Air velocity must be increased by 47·7 %, and the quantity of air is increased in the same proportion. The velocity head is then calculated from the relation

$$p = 147\cdot7^{1\cdot82} = 300\%$$

The velocity head is increased by 103%, and the fan input, depending on the increased air quantity and velocity head, is calculated as

$$\text{fan h.p.} = 147\cdot7 \times 203 = 300\%.$$

If the original engine required 50 kg/b.h.p. h of cooling air, the specific air consumption of the engine with a performance increased by 33% is

$$50 \times \frac{147\cdot7}{133} = 55 \text{ (kg/b.h.p.-h)}$$

3. Transition area and cylinder spacing

The transition area of the cooling medium has a considerable bearing on the spacing of cylinders. Let us consider the difference between conditions prevailing with cylinders cooled by air and those cooled by water. Differences between the physical values of both cooling mediums are shown in Table 18.

Table 18

Coolant	Specific gravity kg/m³	Specific heat kcal/kg °C	Specific heat kcal/m³ °C
Air at 80 °C	1.00	0.241	0.241
Water at 80 °C	972.00	1.003	974.92

One m³ of water heated by 1 °C removes 4000 times more heat than 1 m³ of air. Therefore, a far greater volume of the cooling medium passes the cylinder of an air-cooled engine than that of a water-cooled engine.

For comparison let us determine the required flow area for both types of cooling related to a cylinder of internal combustion engines having the following parameters:

D = cylinder bore . . . . 100 mm
L = piston stroke . . . . 100 mm
p_e = mean effective pressure . . . . 6 kg/cm² (85 lb/in²)
n = engine speed . . . . 3000 rev/min

The output of the cylinder is

$$\text{b.h.p.} = \frac{V \cdot p_e \cdot n}{900} = 15{\cdot}7 \text{ (b.h.p.)}$$

where V denotes the piston displacement in litres. Let us convert the result to thermal units

$$15{\cdot}7 \text{ b.h.p.} = 15{\cdot}7 \times 632 = 9{,}920 \text{ (kcal/h)}$$

For further calculation let us assume that 70 % of this heat is removed by cooling, i.e.

$$Q = 9{,}920 \times 0{\cdot}7 = 6{,}944 \text{ (kcal/h)}$$

We shall now calculate the thermal load of a unit area of the outer cylinder surface. For simplicity we shall consider the outer cylinder surface as that limited by the bold line in Fig. 89, i.e. regardless of the valve ports and with a supposed cylindrical combustion chamber. Subject to these assumptions the area of the outer cooling surface is calculated from the equation

$$A_c = \pi (D + 2b) \cdot \left[L + \frac{L}{\varepsilon - 1} + b\right] + \frac{\pi (D + 2b)^2}{4}.$$

The thermal load is calculated by dividing the removed heat by the above area, i.e.

$$U = \frac{Q_c}{A_c}$$

b denotes the thickness of the cylinder wall (6 mm)

ε „ „ compression ratio = 16

For the cylinder discussed $A_c = 494{\cdot}5$ cm², and $U = 14$ kcal/cm²h.

During its passage through the engine cooling water is warmed by 5—6 °C. The quantity of cooling water required for the removal of the heat Q_c is

$$V_1 = \frac{Q_c}{c'_p \cdot t_1} = 1.15 \text{ (m}^3\text{/h), where } t_1 = 6 \text{ °C}$$

The quantity of cooling air required is calculated in a similar way under the assumption of the temperature increase of air being $t_2 = 60$ °C

$$V_2 = \frac{Q_c}{c'_p \cdot t_2} = 482 \text{ (kg/h)}$$

which, at a mean air temperature of 50 °C and specific weight $\gamma = 1{\cdot}09$ kg/m³ amounts to

$$V_2 = 482 : 1{\cdot}09 = 442 \text{ (m}^3\text{/h)}$$

At a water velocity of 1 m/s the transition area required is

$$A_1 = \frac{1.15}{1 \times 3600} = 3{\cdot}2 \text{ (cm}^2\text{)}$$

and at an air velocity of 25 m/s the transition area required is

$$A_2 = \frac{442}{25 \times 3600} = 49 \text{ (cm}^2\text{)}$$

The transition area around the cylinder of an air-cooled engine is reduced by the cross section of the cooling fins. The area occupied by the fins amounts

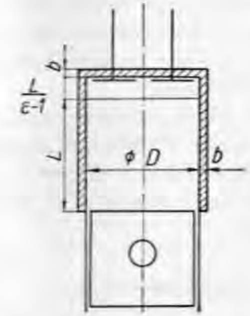


FIG. 89. The outer surface of a cylinder serving for the removal of heat.

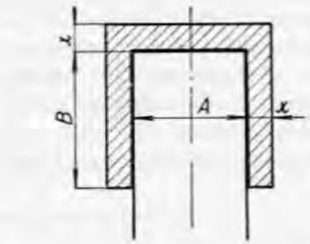


FIG. 90. Surface area around the cylinder for heat transfer to the cooling medium.

to 20—45% of the total area. Therefore the calculated transition area in an air-cooled engine must be increased by an average value of 30 % to

$$A_2 = \frac{49 \times 100}{70} = 70 \text{ (cm}^2\text{)}$$

Supposing that this transition area is uniformly distributed around the circumference of the cylinder according to Fig. 90 and limited by the bottom plane of the piston skirt at the bottom dead centre, the following relation holds:

$$A_f = 2Nx + Mx + 2x^2$$

$$2x^2 + x(M + 2N) - A_f = 0$$

$$x = \frac{-(M + 2N) + \sqrt{(M + 2N)^2 + 8A_f}}{4}$$

According to Fig. 90 $M = D + 2b = 11{\cdot}2$ cm

$$N = L + \frac{L}{\varepsilon - 1} + b = 11{\cdot}27 \text{ cm}$$

From the above quadratic equation we can calculate the width of the transition area x_1 of the water-cooled and x_2 of the air-cooled engine. Thus

$$x_1 = 0{\cdot}09425 \text{ cm, and } x_2 = 1{\cdot}868 \text{ cm}$$

The following conclusion may be drawn from the preceding analysis:

At the conditions given the minimum spacing of cylinders of a water-cooled engine is

$$S_1 = D + 2b + 2x_1 = 113{\cdot}9 \text{ mm} = 1{\cdot}138\ D,$$

and that of an air-cooled engine

$$S_2 = D + 2b + 2x_2 = 149{\cdot}4 \text{ mm} = 1{\cdot}494\ D.$$

It can be seen from this example that the minimum cylinder spacings of water- and air-cooled engines are 1·14 D and 1·5 D respectively. At air velocities higher than 25 m/s the required transition area is smaller and so is the minimum cylinder spacing. However a higher velocity head is then required and power required for the fan must also be considered.

The calculated minimum spacing is difficult to attain in water-cooled engines because of the proper dimensioning of the bearings of the crank mechanism. In air-cooled engines, particularly of the in-line type, cylinder spacing is determined mainly by cooling conditions. Cylinder spacings of some well known air-cooled engines are listed in Table 19.

Table 19

Cylinder Spacings of Air-Cooled Engines

Make	Fuel	Bore **D** mm	Spacing **T** mm	Ratio **T/D**
Tatra T 600	petrol	85	143	1,685+
Tatra T 87	petrol	75	108	1.44
Tatra T 603	petrol	75	100	1.34
Steyer	petrol	78	118	1.51
KdF - VW	petrol	75	110	1.47
Tatra T 111	oil	110	155	1.41
Tatra T 116	oil	120	200	1.67+
Tatra 928	oil	120	155	1·29
Deutz F4L	oil	110	150	1.36
Deutz F8L	oil	110	180	1,64+
Simmering	oil	135	215	1.59+
ČKD	oil	145	260	1,79+
Škoda	oil	140	205	1.46
SLM	oil	110	150	1.36
Robur NVD 12.5 SRL	oil	90	150	1.66
Robur NVD 12.5 SRL	oil	100	150	1.50
NATI DV 30	oil	95	140	1.47
Continental AVDS-1790	oil	146.1	228.6	1.56

Engines marked+ have a cylinder spacing determined by the crankshaft bearings

The condition of minimum cylinder spacing in air-cooled engines requires a design which fully respects the elimination of an undesirable increase of engine dimensions and weight. Air-cooled radial, horizontally opposed and V engines are being designed with complete regard to the condition of minimum cylinder spacing. The crank mechanism of these engines permits a cylinder spacing beneficial from the standpoint of air-cooling. The ratio between the transition areas of cylinder head and cylinder depends on the quantity of heat to be removed. However, the type of finning employed is also important.

4. The economical velocity head of cooling air

In the design of engines a compromise must be found between cylinder spacing, engine weight and power consumed by the cooling system. The latter is given by the formula

$$\text{fan h.p.} = \frac{V_a \,.\, \Delta p}{75 \,.\, \eta_f} \tag{40}$$

where V_a denotes the quantity of air (m³/s)

Δp „ „ velocity head (kg/m²)

η_f „ „ overall fan efficiency.

It is evident that small air quantity and small velocity head are requirements of first importance. Small quantities of air usually entail large velocity heads. In such cases air leakage must be prevented by carefully produced ducting.

In high performance engines with large cylinders dense finning ensuring a high transfer coefficient q_b should be used. The flow area between the fins is small, and, in spite of the small specific air consumption the absolute quantity of cooling air is large. Therefore velocity heads in these engines amount to 200—400 mm of water column. The dynamic pressure head in aircraft engines is often insufficient, and fans must be used.

Vehicle engines have smaller cylinders and the ratio of cooling surface area to displacement is also favourable. Velocity heads in these engines do not exceed 100—150 mm. If the calculated velocity head is higher than 200 mm, redesigning of the engine is usually indicated. The cooling system of a properly designed air-cooled engine requires 6—8% of the b.h.p. as a maximum.

Power consumed by the cooling systems of various engines is shown in Table 20.

Motor-cycle engines with single cylinders have sufficient space for finning. The dynamic pressure head produced by the moving vehicle ensures good cooling. Small engines mounted on bicycles exhibit a particularly beneficial ratio of cooling surface to displacement, and, therefore, a velocity head of 1.5 mm of water column is sufficient. This velocity head corresponds to a travelling speed of only 11 m.p.h. (18 km/h).

Table 20

Cooling Power Required by Some Motor Engines

Make	bhp	rpm	fan hp	Ratio bhp: fan hp	Coolant
GAZ AAA [1]	50	2800	4	8 %	water
GAZ M1 [1]	50	2800	2.7	5.4 %	water
GAZ A [1]	40	2200	1.9	4.75%	water
Ford	103	3800	7	6.8 %	water
Deutz [31]	90	2300	7	7.8 %	water
V 2 in tank [56]	500	1800	42	8.4 %	water
Maybach in tank [56]	700	3200	56	8.0 %	water
Tatra T 301	160	1600	4.8	3.0 %	air
Tatra T 111	200	2000	7.6	3.8 %	air
Tatraplan T 600	52	4000	3.5	6.7 %	air
Deutz [31]	90	2300	7.0	7.8 %	air
Arsenal USA D 117 [3]	32	3200	1.6	5.0 %	air
Arsenal USA D 146 [3]	68	3000	4.1	6.0 %	air

Small power consumption for cooling results usually from a small velocity head. It increases with 2·75—2·9 power of the air velocity. However, the choice of a small velocity head is limited by the conditions of cylinder spacing and by the possibility of mounting large fin surface areas per unit of the outer cylinder surface. Owing to the small increase in temperature, use of cylinders without covers is inefficient — large quantities of air are required and, in spite of the small velocity head, power consumed by the cooling system is large.

At a low heat transmission per unit area from combustion chambers and with dense finnings only small quantities of cooling air are required. Results of measurements carried out by Löhner [35] on a supercharged air-cooled engine are shown in Fig. 91. At high specific performances a cylinder can be cooled at a velocity head of 150 mm w.c. and 36 lb/b.h.p.-h (14 kg/b.h.p.-h) air consumption corresponding to a 1% b.h.p. power consumption for cooling purposes.

Power consumed by cooling depends very much on the cylinder temperature. A 10% reduction of the cylinder temperature (say from 200 to 180 °C) means an increase of power for cooling by 150%. The rate of increase is higher with rising cylinder temperatures. The use of a properly designed finning, made

of a material having a good thermal conductivity, enables a considerable reduction of power required for cooling. Tests carried out on a cylinder of the Ranger type aircraft engine are most instructive in this respect [44]. The engine concerned is a small bore in-line engine, and, therefore, the test results should be of interest to designers of automobile engines.

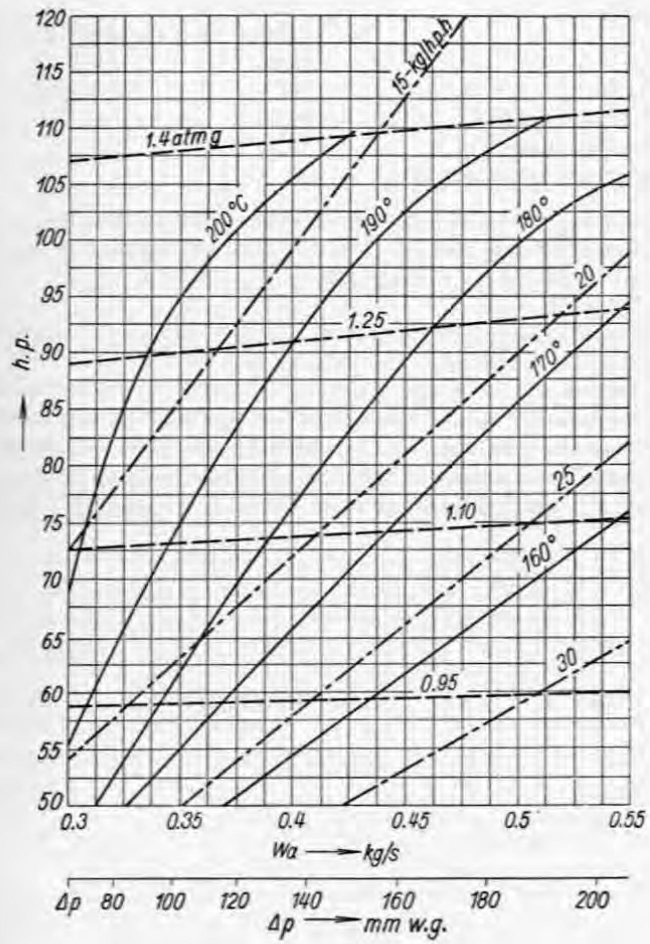


FIG. 91. Characteristics of an air-cooled supercharged engine.
Vertical scale — corrected performance of one cylinder, b.h.p. Horizontal scale — weight of cooling air, kg/s. Lower horizontal scale — static pressure head of cooling air, mm w.g. Full line — temperature of cylinder head, °C. Dash line — intake pressure from supercharger, atm. g. Dash and dot line — unit weight of air, kg/b.h.p. h. Bore 115.5 mm, stroke 162 mm, 2,400 rev/min, $\varepsilon = 6.5$, $\lambda = 1$, temperature of intake air 80 °C.

The Ranger experimental single-cylinder was identical with Mark V-770-12 of the following parameters:

bore and stroke	101·6 and 133·4 mm
displacement	1080 cm^3
compression ratio	1:6.5
valve timing	Inlet valve opens 15° before dead centre Exhaust valve closes 30° past dead centre
ignition advance	30°
indicated performance	49 i.h.p.
fuel-air mixture ratio	1:10
inlet temperature of cooling air	38 °C

Tests were carried out with the original steel cylinder and with two cylinders having machined aluminium fins (Fig. 92). The aluminium fins were cast on by the Al-Fin method. Technical data of the cylinders and test results are given in Table 21.

It is interesting that heat transferred per unit surface area increased with a declining velocity head Δp. The cylinder and cylinder head were exposed to the same velocity head Δp. Whenever heat transfer from the head was reduced, the quantity of heat transmitted from the cylinder was increased. In this way the mean temperature of the unit was maintained at a constant level.

Temperatures maintained in the steel and aluminium cylinders, at $\Delta p =$ $= 483$ mm w.g. and conditions given in Table 21, are listed in Table 22.

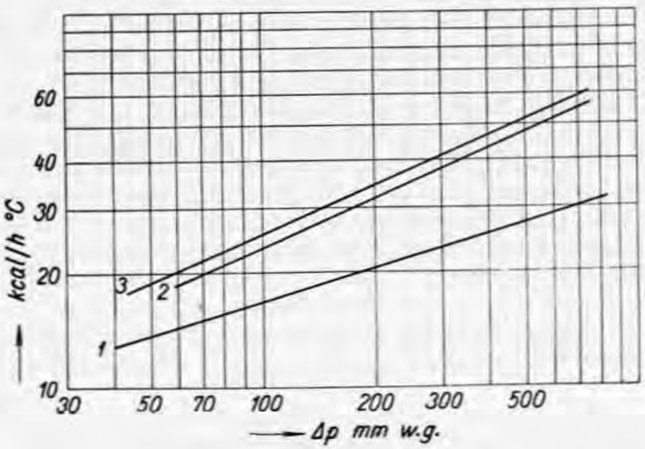


Fig. 92. Relationship between heat removed from the cylinder and pressure head Δp
1 — series cylinder with steel fins, 2 — cylinder with aluminium fins, 3 — aluminium cylinder with special fins.

Table 21

Cooling Tests on the Ranger V-770-12 Aircraft Engine

Type of cylinder	Steel	Alumi-nium	Testing
Number of fins	28	33	43
Interstice between fins, mm	2.95	2.28	1.57
Height of fins, mm	13.45	12.7	23 (13.3)
Fin spacing, mm	3.45	2.95	2.24
Diameter of fin base, mm	105.5	111.0	110.0
Cooling surface area, cm^2	3,160	3,610	8,220
Heat transfer area, cm^2	22.2	19.3	18.0
Thickness of steel cylinder wall, mm	2.03	2.15	2.41
Thickness of aluminium wall, mm	—	2.66	1.94
Basic coefficient of heat transfer at a velocity head of 254 mm w. g., kcal/m^2 h °C	810	1,090	—
Mean cylinder temperature, °C	148	146	145
Mean spark plug temperature, °C	221	254	271
Corrected static head, mm w. g.	483	193	170
Temperature of cooling air, °C	38	38	38
Corrected gradient of static head, mm w. g.	456	175	152
Total flow of cooling air, kg/h	1,090	659	559
Air flow to cylinder, kg/h	477	240	204
Air flow to head, kg/h	613	417	354
Total heat removed, kcal/h	10,130	10,100	9,830
Heat removed by the cylinder, kcal/h	2,970	3,200	3,530
Unit heat removed by cylinder, kcal/h °C	26.8	29.9	33.6
Heat removed by head, kcal/h	7,160	6,900	6,300
Unit heat removed by head, kcal/h °C	38.5	31.5	26.7
Cooling input, hp_f	1.60	0.41	0.31

In the case of the aluminium cylinder the cooling required at a given temperature could be achieved at a velocity head of $\Delta p = 193$ mm w.g. only. At this velocity head the prescribed temperature is attained in a steel cylinder at an engine output of 54% i.h.p. However, this does not mean, that an aluminium cylinder enables an increase of engine performance by 46%. Maximum performance depends, apart from cylinder temperature, also on the temperatures of the cylinder head and piston, and other factors.

Table 22

Comparative Temperatures of Steel and Aluminium Cylinders

	Mean temperature, °C	
	Cylinder	Flange
Steel Cylinder	148	158
Aluminium Cylinder	116	146
Difference	32	12

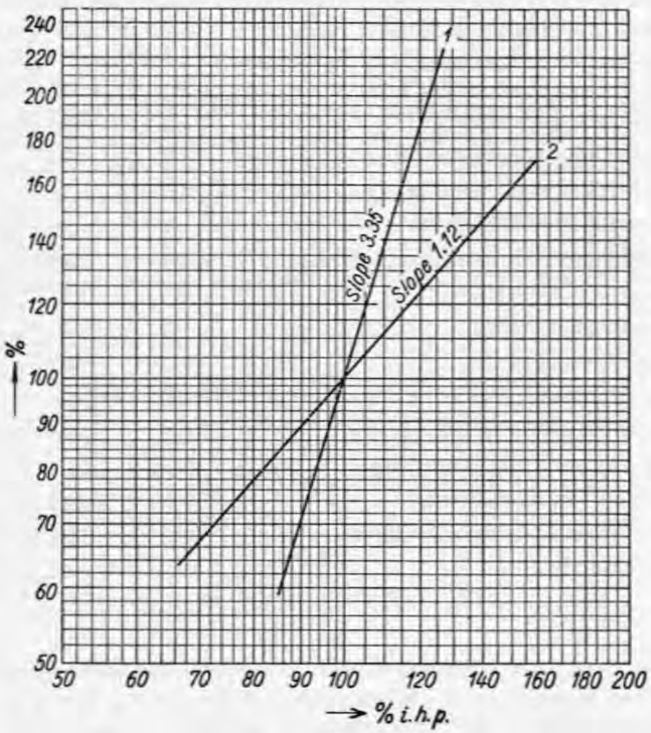


FIG. 93. Cooling input and quantity of cooling air plotted against indicated performance in % (Ranger).

1 — cooling input, 2 — quantity of cooling air. Temperature of the aluminium cylinder 145 °C, cylinder head temperature varying between 255—275 °C, constant air-fuel ratio of the mixture.

The change in the quantity of heat removed from the cylinder and cylinder head due to varying i.h.p. was investigated on an aluminium cylinder (Fig. 52). The temperature of the cylinder flange was maintained at the constant level of 146 °C. Corresponding figures of mass velocity and power required for cooling are given in Fig. 93. The experiments have shown that the temperature of the finned part of an aluminium cylinder decreases more rapidly, than that of the cylinder flange. Therefore, the conclusion seems to be justified, that flange temperatures in aluminium cylinders may be increased without involving the risk of simultaneously increasing the critical piston temperature. At a given flange temperature the temperature of the cylinder head is higher

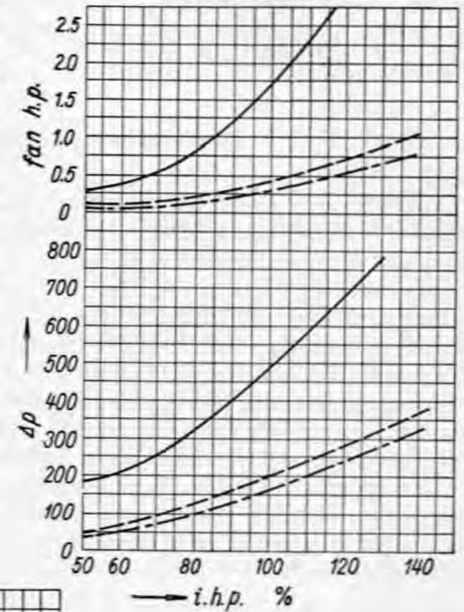


FIG. 94. Reduced cooling input and pressure head for aluminium cylinders of the Ranger aircraft engine plotted against indicated performance in %.

Full line — steel cylinder, dash line — aluminium cylinder dash and dot line — aluminium special cylinder.

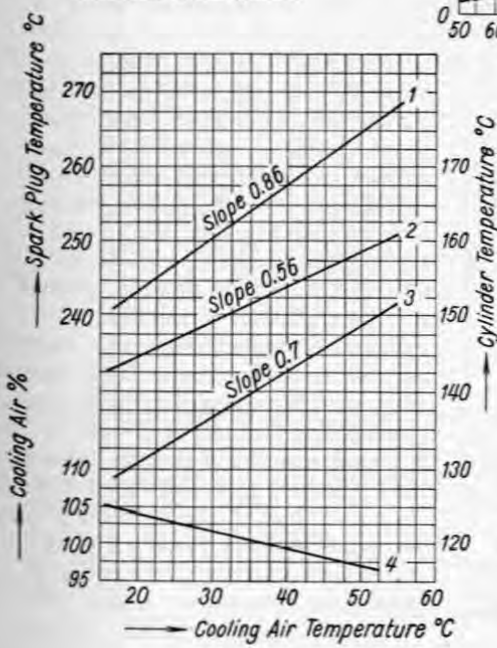


FIG. 95. Effect of inlet temperature of the cooling air upon cylinder temperature at constant indicated performance and pressure head across the cylinder.

1 — temperature of the spark plug, 2 — maximum cylinder temperature, 3 — average cylinder temperature, 4 — quantity of cooling air.

in an aluminium cylinder than in steel. These conditions are evident from Table 21.

In the case of aluminium cylinders the total power for cooling is lower by 25% than for steel cylinders. Final results are given in Fig. 94. The influence of air inlet temperature on cylinder temperatures, at constant velocity head and i.h.p., is illustrated in Fig. 95.

5. The influence of cylinder cowlings

Properly made cowlings reduce the cylinder temperature and increase the transfer coefficient q. Cowlings consisting of air guide sheets mounted closely to the fin tops have proved to be the best.

In cylinders freely exposed to an air stream, cooling of the cylinder rear is unsatisfactory. By covering the cylinder we prevent a disruption of the air stream in the rear, a forced air flow around the entire cylinder circumference is attained, and heat transfer is intensified.

Temperatures of cylinders with and without cowlings are shown in Fig. 96. In covered cylinders a marked drop in temperatures of the rear part can be seen, and temperatures around the circumference are evenly distributed.

Heat transfer in the front is satisfactory, and the use of a casing is not very important. However, cowlings are almost indispensable for the maintenance of a uniform temperature in the rear of the cylinder. The outlet area of the cowling should be 1·6—2·3 times the flow area between the fins. The application of an exit duct forming a diffuser about 75 mm long would be most advantageous. For constructional reasons the mounting of such a duct is, under normal circumstances, impracticable.

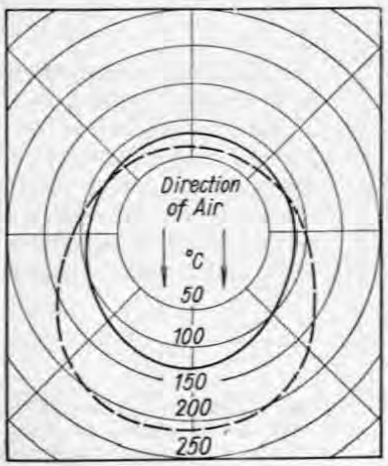


FIG. 96. Temperatures on the circumference of cylinders with cowlings (full line) compared with those of cylinders without cowlings (dash line).

Different types of air flow between fins are shown in Fig. 97. In case a, representing a cylinder without casing, only a portion of the air flows between the fins and solely in the front of the cylinder. A negligible quantity of air flows between the fins and the rear. Instance b illustrates a cylinder with cowling closely fitted to the fin tops; the whole quantity of air flows between the fins leaving at the centre of the cylinder rear. In c the cowling in the front part is mounted slightly off the fin tops, cooling and warming of air in the cylinder front is somewhat reduced. The last case, d, is a turbulent cowling.

At each entry of air a new thin boundary layer is formed, and an increased heat transfer attained. Turbulent cowlings are usually employed at the cylinder rear; their resistance to air is rather high, and, therefore, b type cowling is used for dense finnings.

A suitable cowling is one of the best means to maintain an even distribution of temperatures round the cylinder circumference. If a cowling is applied,

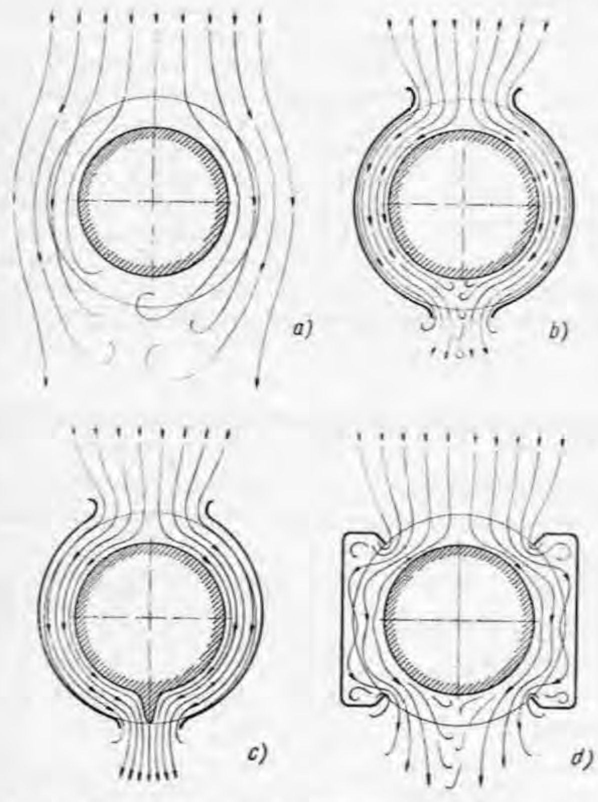


FIG. 97. Air flow in the fin interstice in cylinders with and without cowlings.

according to Fig. 98, for the air exit only, velocity drops between the fins of the cylinder front, heat transfer is worse, and temperature increases in this region of the cylinder surface. The rear casing can be declined at the entry so that velocity increases in the exit direction and cooling of the cylinder rear is improved.

The problem of uneven temperatures becomes more difficult with large bore cylinders. This is one of the reasons for the reduced application of air-cooling in large engines. Today, air-cooling is practical for cylinders up to a 160 mm bore also for engines running at heavy loads (aircraft engines). Air temperature is greatly increased in the long fin interstice particularly in finnings having a high efficiency. A good method of equalizing temperatures is illustrated in Fig. 99, where a double air stream is introduced with air flowing round 1/4 of the cylinder circumference instead of 1/2. The same principle can be applied to the cooling of cylinder heads.

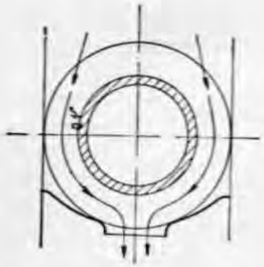


FIG. 98. Cowling at the rear end of the cylinder.

Table 23 contains temperature increases of air, velocity heads and power required for the flow of air through a 100 mm cylinder of a 200 mm bore and finned according to Table 24. By the introduction of a double flow, air warming, with unchanged finning, is reduced to one half with a slight increase of power consumption. If the same warming of air is permitted in the second case, the finning is altered and power consumption reduced. Power consumed by the cooling system is calculated from the air quantity and velocity head and

Table 23

Comparison of Single and Double Flow Around the Cylinder

Type of air stream		simple	double	double
Type of finning (Table 24)		I	I	II
Heat flow, kcal/h		8,800	8,008	8,800
Base coefficient, kcal/m² h °C		1,080	1,080	1,080
Unit air flow, kg/m² s		47	2 × 47	2 × 29.5
Air flow, kg/s		0.212	0.424	0.231
Warming of air, °C		48	24	44
Loss in velocity head mm w.g.	by friction	242	115	91.6
	bend-entrance	8.32	8.32	2.66
	bend-exit	8.53	8.53	2.41
	bend-centre	3.85	2.30	0.20
	total	262.70	134.15	97.17
Cooling input for 100% efficiency, hp_f		0.683	0.700	0.276

assuming the same fan efficiency in all cases. Power consumed by a single flow is 2·5 times larger than that by a double flow, with the same warming of air. Air resistances were calculated according to Berndorfer whose method is in full agreement with results obtained in practice.

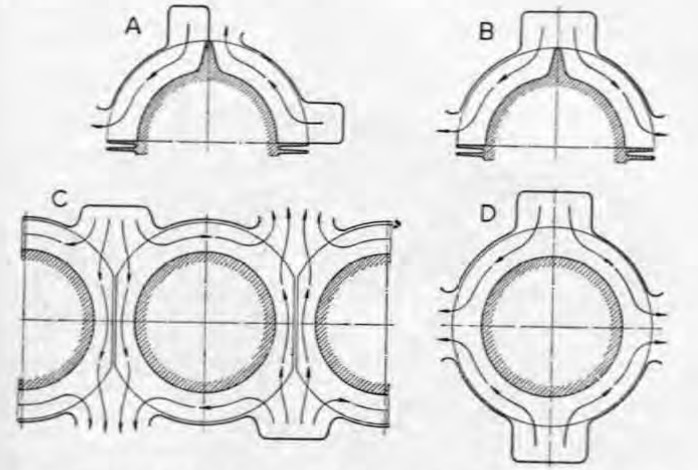


FIG. 99. Illustration of the double air ducting to the cylinder and cylinder head.

Another method of equalizing temperatures is the counter-current cooling system developed by the Tatra works. A stream of cold air is introduced countercurrently to the normal flow of cooling air. This method has the advantage of cooling with air quantities larger than those corresponding to the transition area between the fins.

Table 24

Type of finning	I	II
Spacing of fins, mm	3.49	2.47
Thickness of fins, mm	0.89	0.89
Height of fins, mm	31	31

Thus small fin spacing can be used. The countercurrent cold air is best introduced at the thermally most stressed region underneath the exhaust valve.

Fig. 100 illustrates this method applied in the Tatra T 603 car. A stream of cold air is led through a separate duct directly underneath the exhaust valve.

One part of the current passes through the fins to both sides of the cylinder, the other part flows to the finning of the cylinder head beneath the exhaust duct. The irregularity of temperatures around the cylinder circumference

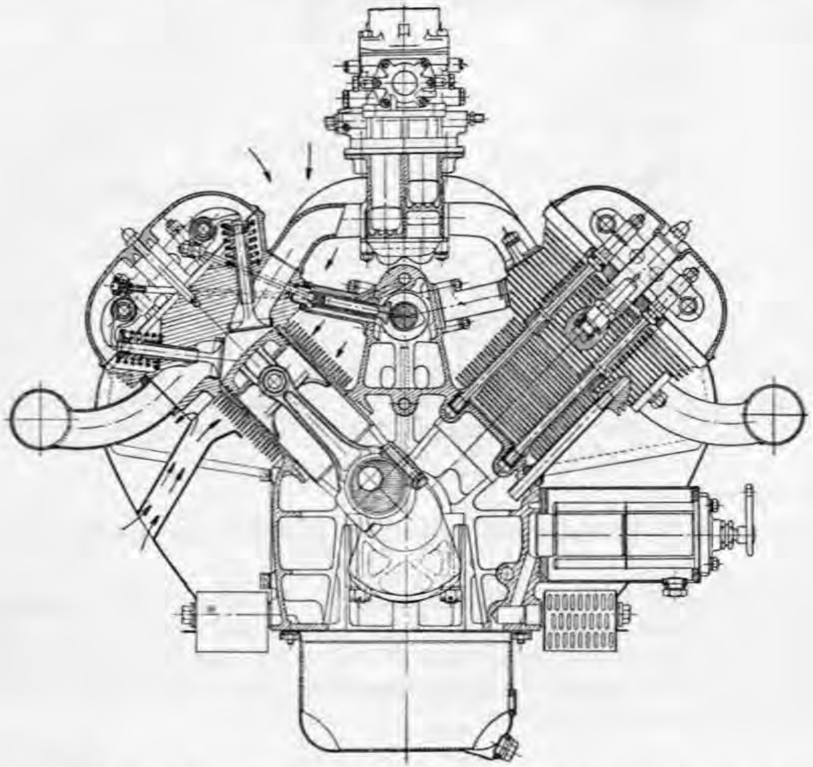


FIG. 100. On the Tatra 603 engine an auxiliary stream of cooling air is led directly underneath the exhaust valve. Arrows show direction of air flow.

was reduced to 5 °C. This cooling system is easy to assemble; the suction fan is located directly in the exit stream and the mouth of the supplementary air duct lies in the free atmosphere.

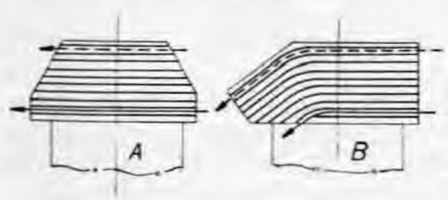


FIG. 101. Good and bad design for utilization of cooling air.

A — Air path through cold fins (dash line) is short, through hot fins (full line) long. Uneven heating of air. *B* — Air path through cold fins is long, through hot fins short. Uniform heating of the cooling air.

Cooling air should be fully utilized and should never pass any part of the cylinder without absorbing the maximum amount of heat. Air flowing along a cooler wall should be led by a longer path to have time for warming and absorbing the desired quantity of heat. Correct and incorrect directions of air-flow between fins are illustrated in Fig. 101. The application of correct principles of air flow is shown in Fig. 102 on the example of the cylinder head of a Tatra diesel engine having a 120 mm bore and 130 mm stroke.

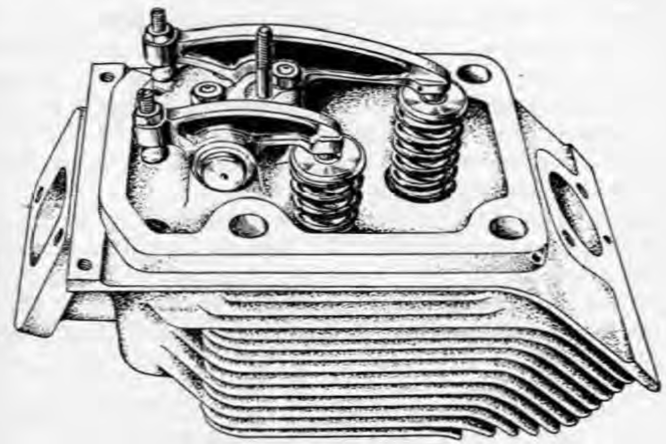


FIG. 102. Cylinder head of a compression-ignition engine. Air is directed by the curved fins to give better cooling in the region of the exhaust valve.

6. Internal cooling

This is the cooling effect exerted by the air passing through the cylinder upon cylinder walls, valves or inner walls of manifolds. The quantity of air passing through the cylinder is far smaller than that of the cooling air. Therefore its cooling capacity is also far smaller. Nevertheless, internal cooling is very effective, because the air comes into direct contact with wall surfaces which, shortly before, have been exposed to the glowing combustion products.

Internal cooling is brought about either by the combustion air solely or by surplus air. Outer cooling can never be replaced by inner cooling. Quantities of air larger than those corresponding to the rated displacement can be conveyed through the cylinders of engines without superchargers only by utilizing to some extent the pressure oscillations in the intake and exhaust manifolds. Quantities of air considerably exceeding the displacement volume can be put through the cylinder with the aid of blowers. Air is compressed in blowers to pressures considerably higher than the velocity heads produced by cooling fans. However, the compression of a unit weight of air is very expensive and is uneconomical in spite of the high heat absorption that can be achieved.

Apart from the surplus air being beneficially used for scavenging the combustion chamber, it is used with advantage for cooling the exhaust valve. Particularly in supercharged engines internal cooling of the piston crown near the exhaust valve became a necessity. A further advantage of supercharged engines is the reduced temperature of exhaust gases due to their dilution by the surplus scavenging air.

Some engine parts can be advantageously cooled by internal air even in cases where the air quantity is not larger than the displacement volume. Evidently, any heat transferred to the air prior to the closing of the inlet valve is harmful and lowers the volumetric efficiency of the engine. However, heat transfer from the hot cylinder and manifold walls to the intake air cannot be prevented, because of the direct contact between walls and air. Therefore, it is worthwhile making the best use of this unavoidable evil, and utilize this heat transfer in the most suitable way.

The adverse effects of uneven temperature distribution round the cylinder circumference, and of elevated temperatures at the cylinder rear, were discussed in a preceding section. The main flow of the intake air can be advantageously directed around the leeward side of the cylinder and in this way utilized for a better distribution of temperatures. This arrangement is being successfully applied in practice.

7. Flow direction of the cooling air

In air-cooled engines the inlet of cooling air can be arranged either on the side of the inlet valve or on that of the exhaust valve. Either arrangement has its advantages and shortcomings.

Cold air entering the hottest zone near the exhaust valve gives the highest temperature difference between cooling fins and air. Cooling is here very intensive and a maximum lowering of the cylinder head temperature is attained. Consequently, temperatures on the intake and exhaust side of the cylinder are equalized to a large extent, and possible distortions of cylinder head and valve seats are suitably affected. Therefore, this cooling arrangement should be applied in cases of cylinder heads bursting because of excessive thermal loads due to uneven temperatures. The high temperature of the exhaust valve region can sometimes cause a permanent distortion of the cylinder sealing surface with subsequent leakages or even burning of the cylinder head.

By arranging the cooling air inlet at the intake side, an intensive cooling of the inlet valve region is attained, heat transfer to the intake air is small and higher volumetric efficiency results.

Thus engine performances attained with this arrangement are better than those achieved by the cooling air entering the exhaust side. A properly designed cylinder head should ensure a sufficient cooling of the exhaust valve region. In this case the increased engine output is a clear point in favour of the second

arrangement. Also it must be borne in mind that particularly with this arrangement the major portion of the intake air is brought to the leeward side of the cylinder contributing to a better cooling of this region. Therefore, the second arrangement is generally preferred.

Sometimes, the arrangement of cooling air inlet is determined already by the design of the engine. Fig. 103 illustrates this point quite clearly. In the flat

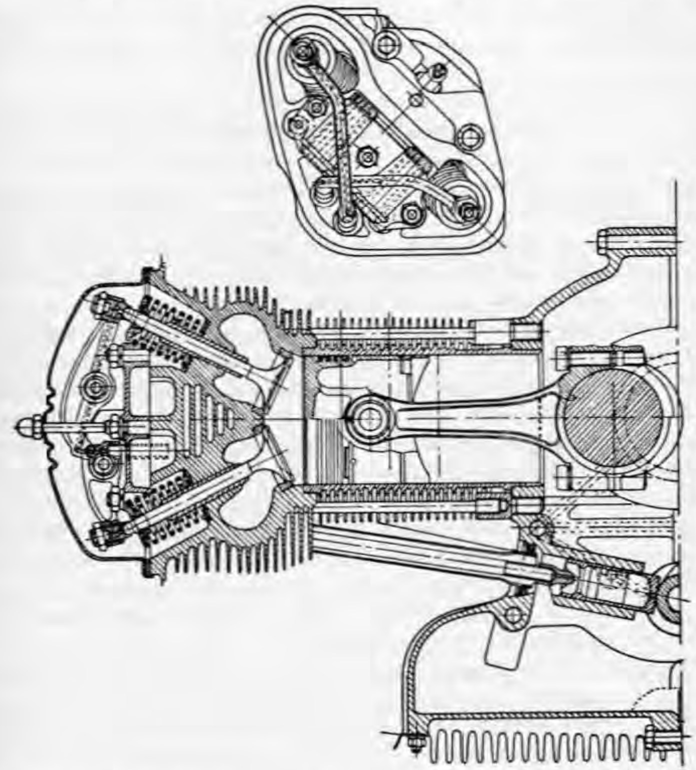


FIG. 103. Cylinder of the air-cooled Tatraplan engine. The inclined plane of the valves improves the passage of cooling air.

four-cylinder Tatraplan engine the obvious arrangement is represented by the exhaust manifold leading downwards and the intake manifold directed upwards. Cooling air inlet from below would be difficult to arrange, the air would be warmed by the exhaust pipe prior to its reaching the cylinder surface, and its cooling capacity would be greatly reduced. In this engine cooling air must be brought to the region of the inlet valve and obviously from above.

In all described arrangements it is supposed that valves are placed roughly in the direction of the air flow so that large flow areas of cooling air are obtained even with small cylinder spacings.

8. Arrangement of the cooling fan

The cooling fan may be arranged either with suction from atmosphere and delivery through the engine, or with suction through the engine and delivery directed to atmosphere.

The first arrangement is the most usual, and from the standpoint of power consumption the more suitable. Care must be taken that the dynamic pressure head should not produce an uneven distribution of cooling air to the individual cylinders. This problem is difficult to solve particularly in the case of several in-line cylinders located in the direction of the air stream. The situation can be improved by a suitable shaping of the cooling air duct. Inserted deflectors and guide sheets in the inlet duct are less satisfactory, because their effect varies with the velocity of air.

Sometimes the second arrangement, with the engine located in the suction zone of the fan, proves to be more satisfactory. This is the case with engines built-in into enclosed spaces where the hot exit air exerts a considerable heating effect, e.g. in armoured cars, etc. A suction fan ensures here the shortest exit of the hot air into the free atmosphere. A further important advantage is the uniform distribution of cooling air to all cylinders. The dynamic component of the pressure head is absent.

Air guide sheets are more simple in a suction arrangement which means a reduction in leakages. Air losses through such leakages are very undesirable particularly in a high pressure cooling system. The elimination of guide sheet troubles in the suction arrangement is a partial compensation for the higher power consumption of this system.

The suction side of the fan is handling hot air and, therefore, larger volumes of air must be conveyed, at the same velocity head, by the second arrangement. This entails a higher consumption of power. Apart from this, the fan conveys more air when the engine is cold, a fact very undesirable from the point of view of regulation. Another advantage is the possibility of locating on the delivery side a diffuser by which a part of the energy of the exit air, otherwise lost, can be recovered.

It is hard to state which of the two systems is the better. According to actual conditions either of them can be the more suitable one. In the T 603 engine the suction arrangement proved to be more satisfactory for the following reasons: movement of air through the engine space was simplified, engine performance was increased due to the flow of cold air along the carburetter to the inlet valve, cooling air was evenly distributed to all cylinders, and the control of the cooling system was facilitated. Apart from this a countercurrent cooling of the exhaust valve region was easy to arrange (see Fig. 100).

CHAPTER VIII

CONTROL OF THE COOLING SYSTEM

1. Importance of cooling control

If the engine is to run properly the temperature of the engine must not fluctuate excessively under differing loads or at varying outside temperatures. The maintenance of an adequate temperature of the cylinder wall influences not only the losses caused by friction, but also the rate of wear of the cylinders.

If the walls of the cylinder are cold, oil on their walls is thick and offers strong resistance to the movement of the piston. This is particularly characteristic of high viscosity oils. The hotter the cylinder wall, the smaller the losses through friction. Fig. 104 [48] illustrates the relationship between friction losses and the temperature of the cooling water, such relationship being expressed by mean pressure p_z in kg/cm^2 of the piston area. With regard to this friction a high temperature of the cylinder-wall is an advantage. The measurements were taken at a constant temperature of intake air.

Of particular significance is the rate of wear of the cylinders at low temperatures of the cylinder walls. If the engine is undercooled, products of the combustion condense on the cylinder walls, etching of the cylinder walls sets in and this, together with poor lubrication and high pressure of the upper rings is responsible for the excessive rate of wear of the cylinders. The highest rate of wear occurs in the upper third of the cylinder, which is an unmistakable proof that this is not an instance of mechanical wear alone, but that also chem-

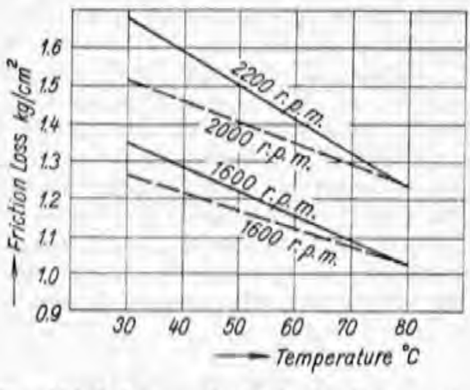


FIG. 104. Influence of cylinder (cooling water) temperature upon the average loss of pressure in a twelve-cylinder engine with 114.3 mm bore, 152.4 mm stroke and a compression ratio of 5.3 : 1 (full line). The dash line represents friction loss in an eight-cylinder engine, bore 140 mm, stroke 150 mm and compression ratio 5.4 : 1.

ical factors are at work. The rate of wear of the cylinders can be substantially lowered if the temperature of the cylinder is kept within an adequate level. Fig. 259 shows, in accordance with tests, the rate of wear as a function of the temperature of the cylinder. At low temperatures of the cylinder walls the increase in the rate of wear is noteworthy.

From daily experience we know that engines of vehicles used for long-distance travel suffer much less from wear than engines of vehicles employed

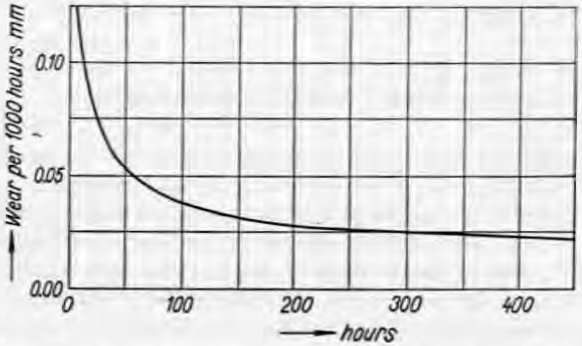


FIG. 105. Wear of cylinders plotted against running time in hours after one start.

for local traffic. In local traffic the engine is started up frequently. If the engine is started up at a low temperature, conditions for a low rate of wear are impaired as the oil film is washed off the walls of the cylinder by a rich starting mixture (Fig. 105) [21].

The control of cooling prevents the fluctuation of temperatures of the cylinder wall in the course of operation and enables a speedy heating of the engine. With an air-cooled engine the changes in the outside air temperature are of not so great importance for the conditions of cooling as with a water-cooled engine. Let us assume, that the engine runs first at an outside temperature of 0 °C, and then at a temperature of 40 °C. In both instances the temperature of the engine will be maintained at the same level only through controlling the amount of cooling air. The coefficient of heat transfer q and the amount of air will be calculated according to the following equations:

$$q = 1{\cdot}18 \,.\, (1 + 0{\cdot}0075\, T_m) \,.\, v_m^{0{\cdot}73} \ (\text{kcal/m}^2\ \text{h}\ ^\circ\text{C}),$$

$$V_a = A \,.\, k \,.\, \Delta p^{0{\cdot}55} \ (\text{kg/h})$$

where T_m denotes the arithmetic mean of the absolute temperatures of fin surface and air (°K)

v_m „ the mean velocity of air in the fin interspace (m/s)

A „ the flow area between the fins of the cylinder (m²)

k „ a constant,

Δp „ velocity head between the fins (kg/m²)

The assumed values will produce conditions given in Table 17. Assuming the temperature of the outside air stands at 40 °C, the temperature difference in a water-cooled engine will be decreased to half its value, hence the amount of cooling-air will have to be increased by 147 % and the input of the fan will be increased 13·25 times. On the other hand, for an air-cooled engine all other conditions being equal, it will do to increase the amount of cooling-air by 32·8%, and the input of the fan, 2·33 times.

Despite these facts it is, however, essential to provide a cooling control also for air-cooled engines. Especially if running constantly undercooled during the wintermonths the engine will suffer from excessive wear. Furthermore the above example demonstrates that the control of cooling will make it possible to achieve substantial economies in the power consumed by the cooling system. Low sensitivity of an air-cooled engine to outside temperature makes these engines suitable for use in tropical regions.

2. Methods of cooling control

The cooling of air-cooled engines can be controlled by one of several methods. The simplest and most frequently applied method consists in throttling the cooling air. The feeding of cooling air can be governed by:

1. Throttling the inlet air to the fan,
2. Throttling the air between fan and engine,
3. Throttling the air leaving the engine.

Ad 1. To throttle the air introduced to the fan is of particular advantage where air leaves the engine at several points. Such is the case with engines whose cylinders are arranged in several rows, i.e. the V-type engine, flat engines and the like. In such instances it would be difficult to throttle the outgoing air.

This type of solution presents constructional problems particularly if the air has to be throttled at the point where it enters the fan. To throttle the air by louvres is not recommended for this arrangement, for part-closed louvres affect the direction of air towards the impeller or guide wheel of the fan. Fig. 106 shows a convenient throttling of air fed to the fan of a VW engine. A ring controlled by a thermostat is inserted into the suction orifice of the fan. If the pressurized air fed to the fan is used for heating, this type of control is not desirable for the air-pressure behind the fan decreases due to the throttling.

Ad 2. Throttling of air between fan and engine is practicable only with certain types of designs.

An example of such a control is shown in Fig. 107. It is the control of the air-cooled Phänomen engine provided with a radial fan, mounted directly on the crankshaft. The air is conveyed from the fan to the engine through a short, bent duct; into this duct, directly behind the fan two throttle valves are fitted, one of them being governed by a thermostat, and the other being mechanically connected with the throttle valve of the carburettor. A bellows-thermostat is located in the stream of air leaving the cylinders. The purpose of the second throttle valve, connected with the throttle valve of the carburettor, is to prevent overcooling of the engine on downhill runs.

Ad 3. Throttling of outgoing air is convenient wherever there is a single point of joint discharge of the air from the engine. This method produces

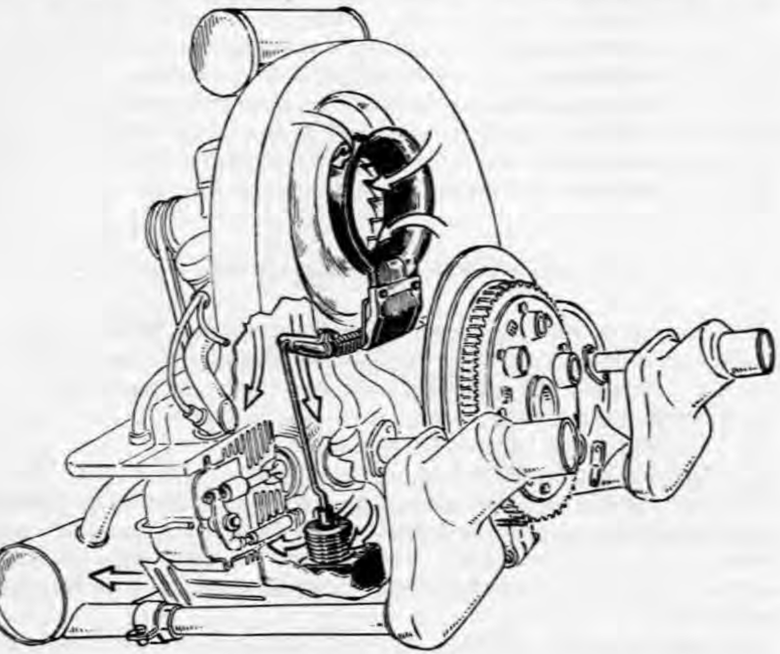


FIG. 106. Cooling control in the VW engine by throttling the air inlet. Automatic control is provided by a thermostat.

air-pressure under the hood of the engine to be used for heating the vehicle and the like. An instance of a control of this type is illustrated in Fig. 108: air leaving a V-type engine is throttled by two flaps connected in parallel, controlled by a thermostat.

Control effected through throttling leaves a very limited scope for saving on the power consumed by the fan; the throttling of air lowers the efficiency of the fan, so that a reduction in the amount of air will give only a slightly reduced input to the fan.

Cooling control through a change in the speed of the fan will result in a substantial economy in input. Taking into consideration that the amount of air flowing through the fan is directly proportionate to the speed while the

air-pressure is proportionate to the second power, and the input of the fan to the third power of the speed, the importance of economies of this type are apparent. If half of the original amount of air is required for the cooling, 1/8 of the initial input will be sufficient for the purpose. This is of particular importance in the case of partial load and a high speed fan, where the cooling input represents a major share in the effective performance of the engine.

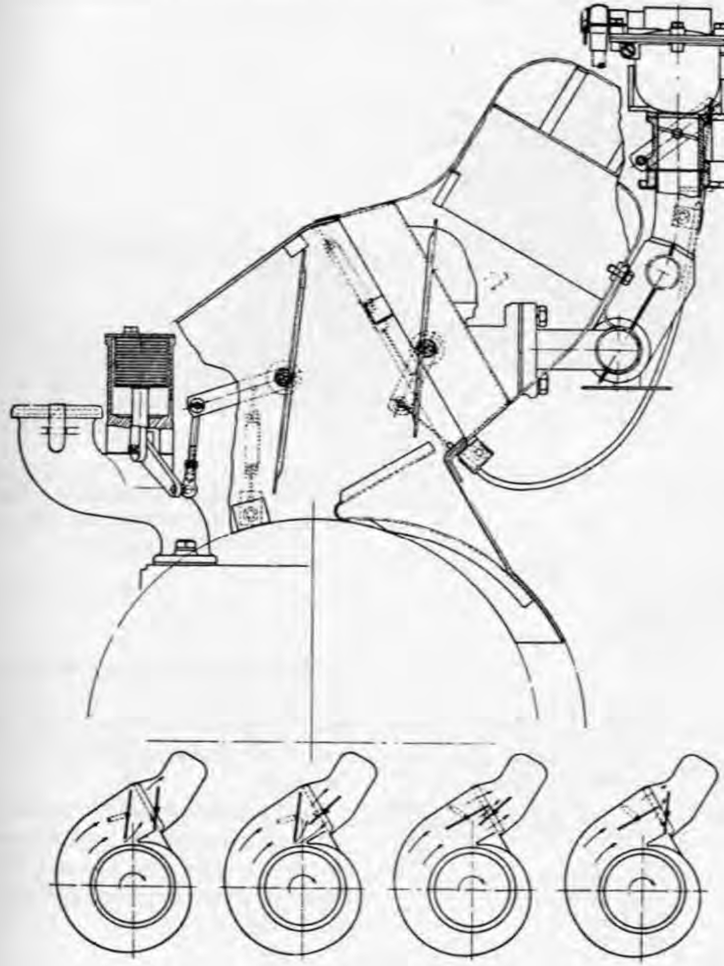


FIG. 107. Cooling control in the Phänomen engine by air throttling between fan and engine.

One flap is actuated by the thermostat, the second by the throttle o f the carburettor. Bottom drawing illustrates function of control flaps during start and down-hill travel.

The speed of the fan can be changed by electrical, mechanical or hydraulic means.

Electrical transmission gear between engine and fan is desirable; it is, however, suited only for larger units. It is used for railway engines, where

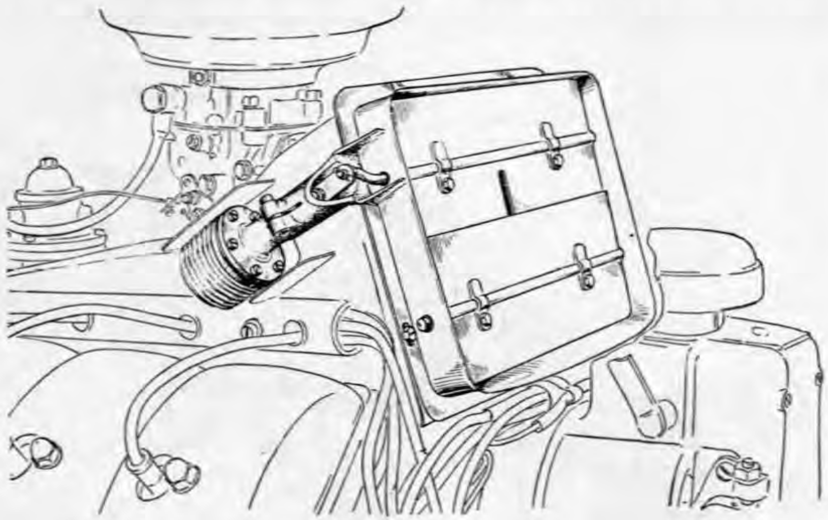


FIG. 108. Cooling control in the Tatra 603 engine by throttling the exit air. Experimental design.

the fans of the coolers are driven by electric motors. The speed of the electric motors can be regulated and thus, a regulation of cooling can be achieved.

The electromagnetic turbulent coupling is much simpler and suited also for smaller size engines. The working of this coupling is illustrated in Fig. 109.

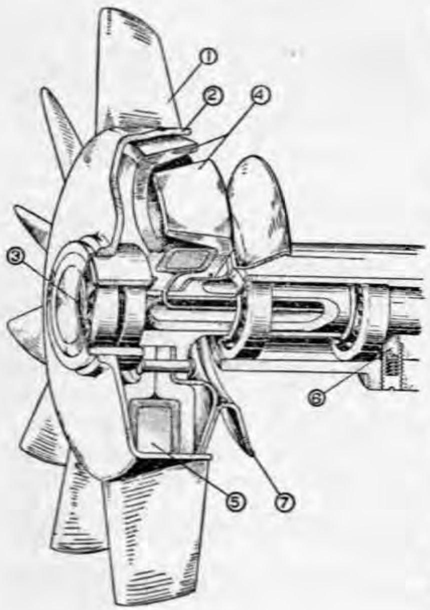


FIG. 109. Principle of the electro- ← magnetic turbulent coupling.

The impeller of fan *1* is fitted to steel drum *2* and by means of ball bearings it is mounted on shaft *3*. An electromagnet with alternate pole shoes *4* is rigidly fastened to this shaft. The electromagnet is excited by coil *5* which is fed through carbon leads *6*. The shaft of the fan is driven by the engine by means of a V-belt and belt-pulley *7*.

If the current is not switched on to the coil of the electromagnet, only the shaft of the fan with the electromagnet will revolve while the impeller of the

FIG. 110. Electro-magnetic turbulent coupling of the Scintilla type as an integral part of the dynamo.

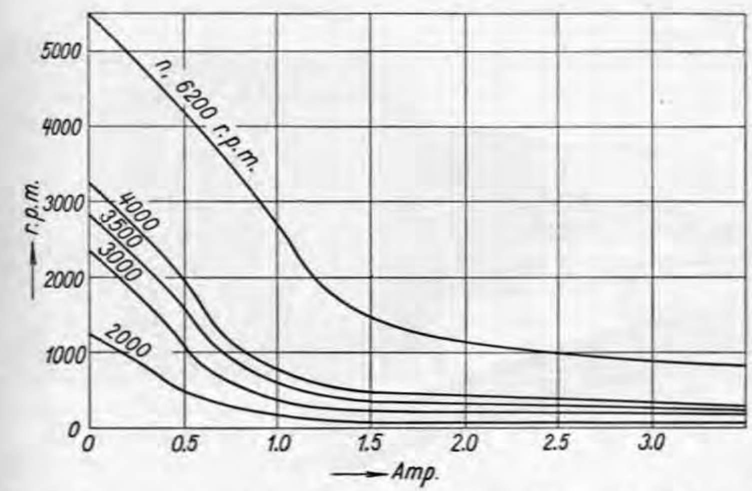


FIG. 111. Characteristics of the Scintilla electro-magnetic coupling. Effect of the exciting current on the magnet in Amp upon the coupling slip $n_1 - n_2$.

fan stands still. When current has been introduced into the coil, a magnetic field is set up between the pole shoes and crosses the steel drum of the impeller. With the intensity of the excitation of the electromagnet, the increasing speed of the steel drum of the impeller approaches more closely the speed of the electromagnet, ultimately giving slip of only about 4%.

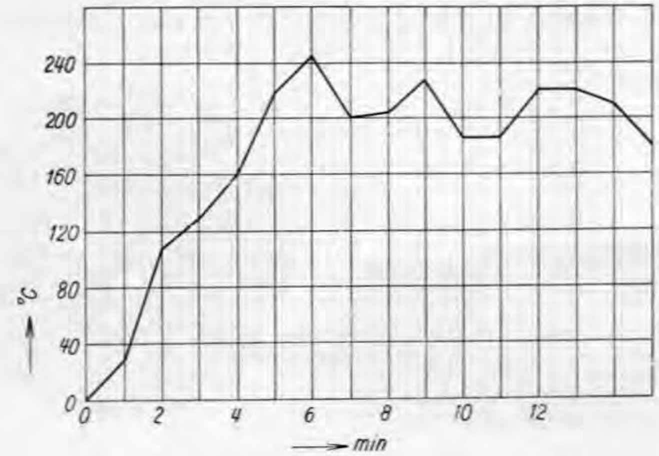


FIG. 112. Cylinder head temperatures after starting the Tatraplan engine with the Scintilla electro-magnetic coupling.

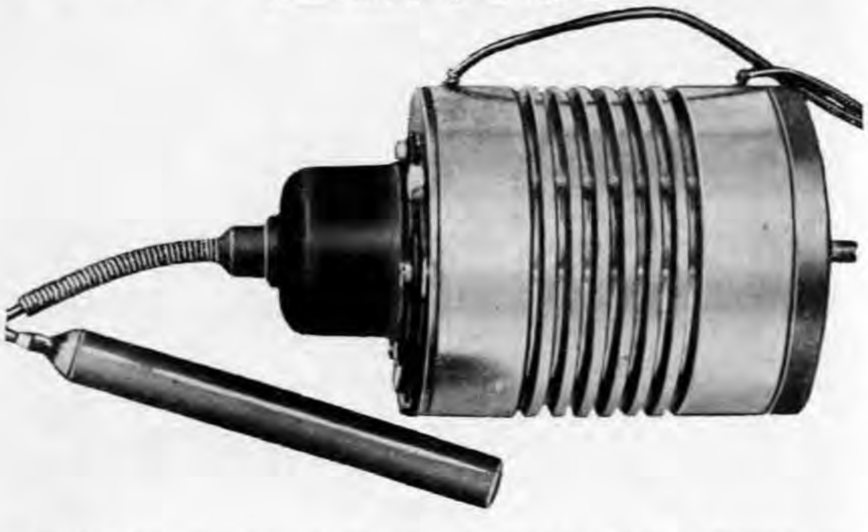


FIG. 113. Thermostatic device for continuous change of speed in the Scintilla electromagnetic coupling.

This type of coupling is the basis of the electromagnetic coupling, forming one unit together with the dynamo and tested on the Tatraplan engine. This is shown in Fig. 110. The alternate pole shoes of the electromagnet can be distinctly seen on the dynamo. The impeller of the fan with the steel drum has been dismounted. The details of this coupling are apparent from

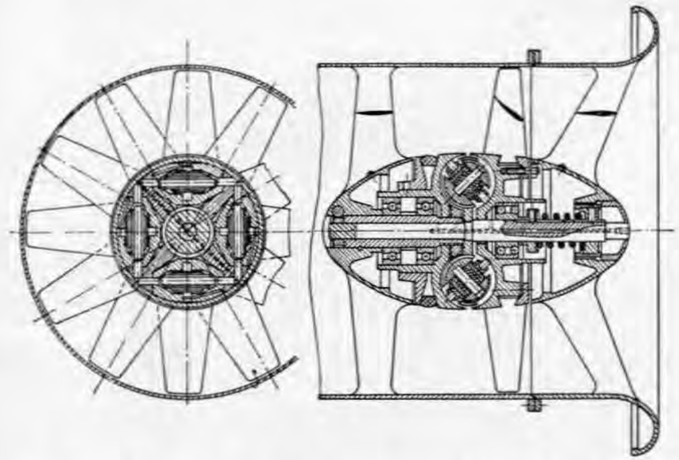


FIG. 114. Friction gear for the continuous change of fan speed.

Fig. 111. The coupling is switched on by a bimetallic thermostat, fitted in the outgoing air stream behind the engine. Immediately after the engine has been started up the impeller of the fan does not revolve, and it is only after the temperature of the engine has reached a predetermined level that the thermostat will switch on current to the turbulent coupling and the fan will begin to supply air. As soon as the temperature of the engine has dropped below the set limit, the fan is automatically switched off. Fig. 112 records the relationship between the temperature of the head of the Tatraplan engine provided with a control of the cooling by means of a turbulent coupling, and time. It shows that the engine is heated very rapidly.

A turbulent coupling provides for gradual changes in speed. This alternative requires the replacement of the switch by a thermostat-controlled

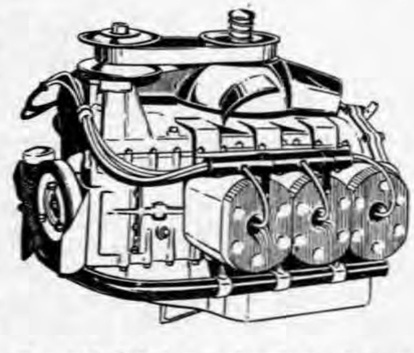


FIG. 115. V-belt gear for the continuous change of fan speed in the Jack and Heinz engine.

resistance for the excitation of the electromagnet. However, such a device is both expensive and delicate. When the speed is changed, the efficiency of the turbulent coupling drops. Hence the costliness of the device is not warranted by adequate saving in input. A thermostatic device for gradual changes in speed is illustrated in Fig. 113.

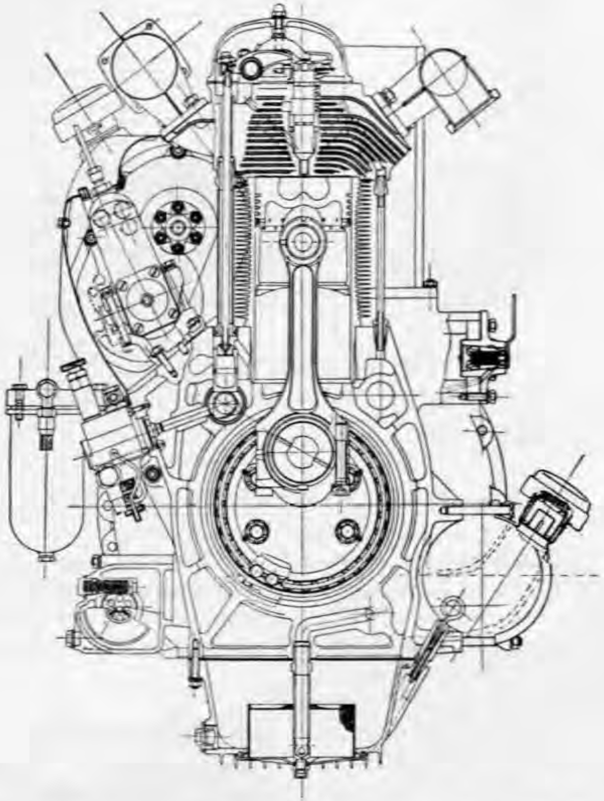


FIG. 116. Transverse section of the four-cylinder Tatra 924 engine.

For switching off the fan an electromagnetically opened friction coupling can be used. As, however, the individual elements of a friction coupling suffer from high wear this device can hardly be recommended.

The design of a continuously controlled mechanical gear operating according to the temperature of the engine has not been satisfactorily solved as yet. Experiments have been made, with a friction gear of the Hyes type, operating by means of friction discs. The drive is controlled by tilting the discs (Fig. 114).

The air-cooled Jack and Heinz slide-valve engine applies a very simple method of changing the speed of the V-belt driven fan. (Fig. 115). The two sections of the driving pulley are moved closer together or apart by the action of a thermostat and the two sections of the driven pulley, by means of a spring. If the two sections are moved closer, the rate of transmission increases and the fan revolves at a higher rate. A spring fitted to the driven pulley ensures

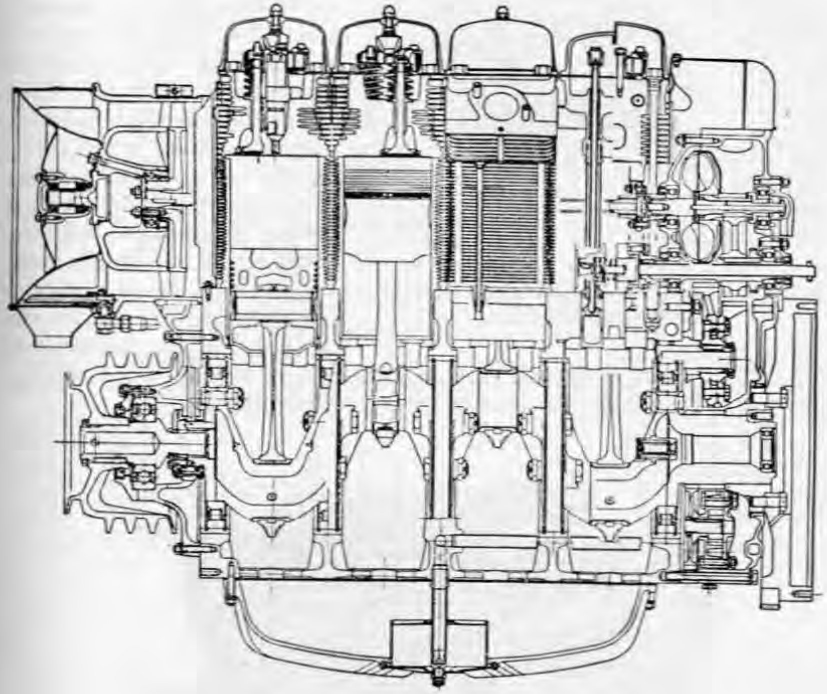


FIG. 117. Longitudinal section of the four-cylinder Tatra 924 engine.

the required tightening of the belt. This arrangement is not suited for large fans driven by several belts.

The properties of the hydraulic coupling are similar to those of the electromagnetic turbulent coupling. No mechanical friction arises and thus the coupling does not suffer from mechanical wear. With the electric coupling only the carbons for feeding the current suffer from wear, in the hydraulic coupling friction occurs only in the oil seal. This sensitive part of the hydraulic coupling can be dispensed with if the whole unit is mounted in the crankcase of the engine and filled with oil.

The manufacture of hydraulic couplings is simple and the couplings are not subject to wear. As the character of the transmitted torque is the same as that of the input to the fan, with a filled hydraulic coupling the slip is constant within the entire speed range. Hence the control of speed and of the slip of the coupling can be altered by a change in the oil. In practice, this objective is attained through regulating the feed of oil to the coupling, the drainage being controlled by means of calibrated orifices. The mounting of the hydraulic coupling directly in the crankcase does away with the problem of the sealing glands, the most delicate element in the hydraulic coupling. The hydraulic coupling possesses the advantage of damping oscillations in the drive, and if the resistance of the fan is increased (through submerging the latter in water, etc.) slippage of the coupling occurs and thus prevents damage to the drive.

FIG. 118. Casting of one half of the hydraulic coupling for fan drive

FIG. 119. View of the four-cylinder Tatra 924 engine.

Because of this advantage, control based on the above principle has been applied to the new series of Tatra diesel engines with 120 mm bore and 130 mm stroke. Figs. 116 and 117 show cross- and longitudinal sections of a four-cylinder T 924 Tatra engine. The longitudinal section illustrates the arrangement of the hydraulic coupling in the driving system of the fan. The power is transmitted to the fan from the rear end of the crankshaft (from the

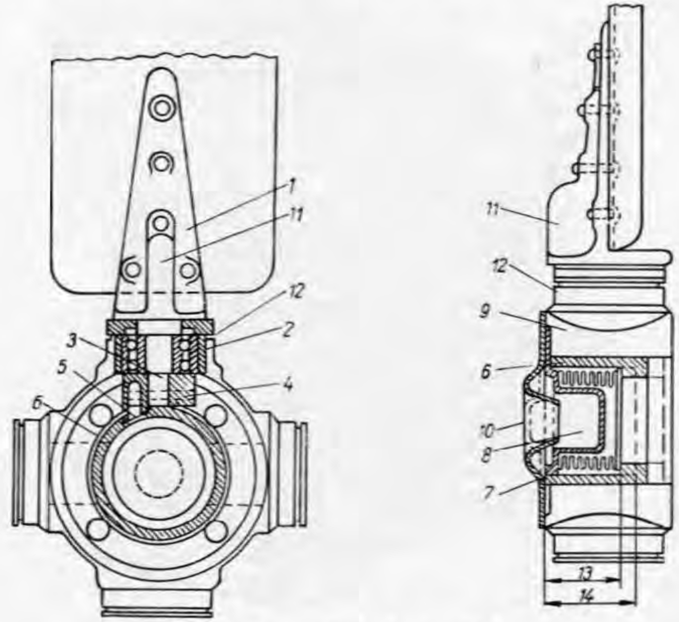


FIG. 120. Fan with feathering blades.

Blade mountings (*1*) are located in ball bearings (*2*). Lower journal (*3*) is connected with disc (*4*). Pin (*5*) is engaged in the opening of disc (*4*) and actuated by its hub (*6*) which is shifted in an axial direction by bellows thermostat (*7*). The liquid heating the thermostat is led into compartment (*8*). Length of the thermostat in a cold engine equals *13*, in a hot engine it increases to *14*.

flywheel) through gears and a long flexible shaft. One section of the hydraulic coupling is connected with a pinion gear, while the other section with the guard is connected directly to the shaft of the fan. The manufacture of the hydraulic coupling is not involved. Both sections are machined from the same type of casting (see Fig. 118), with cast vanes, so that only the surface and the hub of the wheel have to be machined.

As long as the hydraulic coupling is not filled with oil, slippage will occur and the fan remains stationary. The feed of oil to the coupling is controlled by a bellows-thermostat, mounted in the air-stream leaving the cylinders. As soon as the engine (outgoing air) begins to heat, the oil-feed to the hydraulic coupling opens and the fan operates. The perimeter of the outer section of this type of coupling is provided with calibrated openings through which oil returns constantly from the coupling into the crank case. As at constant speed the oil-pressure at the perimeter of the coupling is proportionate to the quantity of oil, equilibrium between oil change and drain is reached at a certain rate of filling. For this reason, through controlling the amount of incoming oil, the rate of filling and thus also the slippage of the coupling can be controlled.

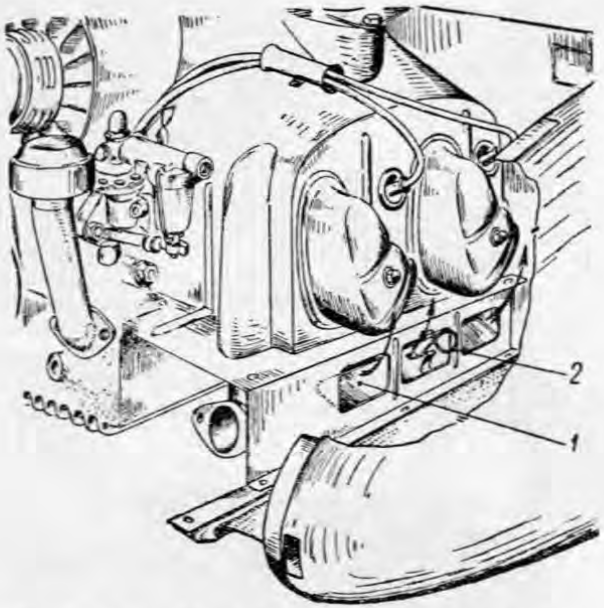


FIG. 121. Control by air by-pass in the Tatraplan engine.

This type of control has proved satisfactory in operation and has been applied to a series of engines. In the four-cylinder T-924 Tatra engine the bellows thermostat is fitted in the stream of the air leaving the last cylinder. In Fig. 119 the thermostat can be seen in front of the oil-cooler.

Control by way of altering the pitch of the blades of the fan is another method enabling considerable economies to be made. As the blades are exposed to a powerful centrifugal force and at the same time must revolve readily, they have to be mounted on roller bearings. Hence, the design of a fan of this type is rather complicated [56].

Fig. 120 illustrates a fan with adjustable blades, mounted in a recent water-cooled engine.

Water from the engine enters through branch *10* into a box in which an alcohol thermostat *7* is fitted. When the water is heated the thermostat begins to expand, it moves hub *6* towards the right and increases the angle of inclination of the blades of the fan. It is thus possible to obtain a change in the amount of cooling air and energy consumed without a change in the transmission from engine to fan.

Control through changing the temperature of cooling air is also very advantageous, since it provides an improved air-fuel mixture for the engine. If the temperature of the air under the guard is maintained at a constant level,

FIG. 122. View of the Tatra 603 engine.

the air drawn into the engine has an even temperature, and the irregularities in combustion caused by the intake of excessively cold air and by the condensation are eliminated.

Such a method is shown in Fig. 121. With this engine the cooling air travels from top to bottom, and is discharged below the car. If flap *1* is opened, steel spring *2* keeps this flap opened in such a position as to reach the stream of outgoing air and deflect the latter to a horizontal direction below the guard of the engine. This hot air "heats" the entire space below the guard, from which also the air for the engine is drawn.

The design of the Tatra 603 car also uses cooling control through changing the temperature of the cooling air. Fig. 122 gives the overall view of the Tatra 603 engine. Distinctly noticeable are the fans drawing air from the proximity of the carburettor and the induction tubing across the cylinders and extruding it by way of the shortest path through the openings in the rear to atmosphere. The control valve is governed by a bellows thermostat. By means of the closed valve the outgoing air from the fans is deflected upwards and through orifices opened by the valve it enters the engine space to be drawn round the carburettor into the engine (see Fig. 123).

FIG. 123. View of the thermostat operating the cooling control flap.

If the valve is part-opened, a portion of the air escapes by the rear opening into the atmospere while the balance is deflected upwards into the engine space. With the engine hot the flap is in its upper position and the openings into the engine space are closed.

In this position air is introduced into the engine space from air traps in the rear mudguards, which utilize the dynamic pressure head. The air passes through the extensive space in the rear mudguard, loses speed and is partially filtered of coarse impurities. It then passes into the engine space through a perforated wall (see Fig. 124). This perforated wall and the space in the rear mudguard act as sound insulation between engine and body.

The control of the cooling of the Fiat 500 engine is solved in a similar manner. In Fig. 125 the radial fan can be seen, from which the air is directed primarily towards the cylinder heads. Air is inducted by a flexible hose from the space below the rear window. A portion of hot air leaving the engine is led by piping likewise into this suction space. However, the hot-air outlet is shut off by a thermostat-controlled flap. As long as the engine is cool the flap is opened and the fan partly returns the hot air for the cooling of the engine. If the temperature of inlet air is 0 °C, the flap is completely opened, while at a temperature of 20 °C it is completely closed. In this latter instance the engine draws cold air only.

The suction box is fitted to the body and it is connected to the engine by flexible hoses. This is why the thermostat mounted in this box is not affected

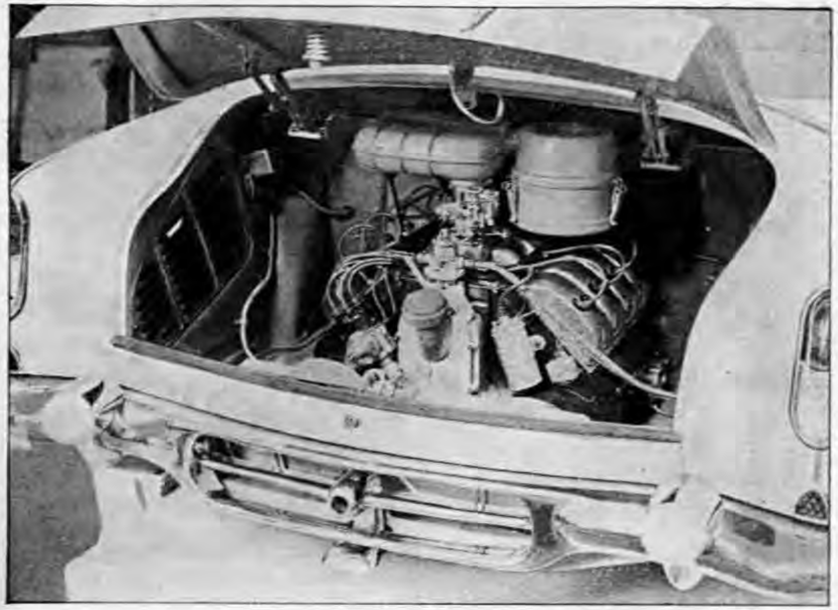


FIG. 124. Cooling air enters the engine compartment of the Tatra 603 through the perforated wall of the rear mud guards and leaves it through an opening in the rear bumper.

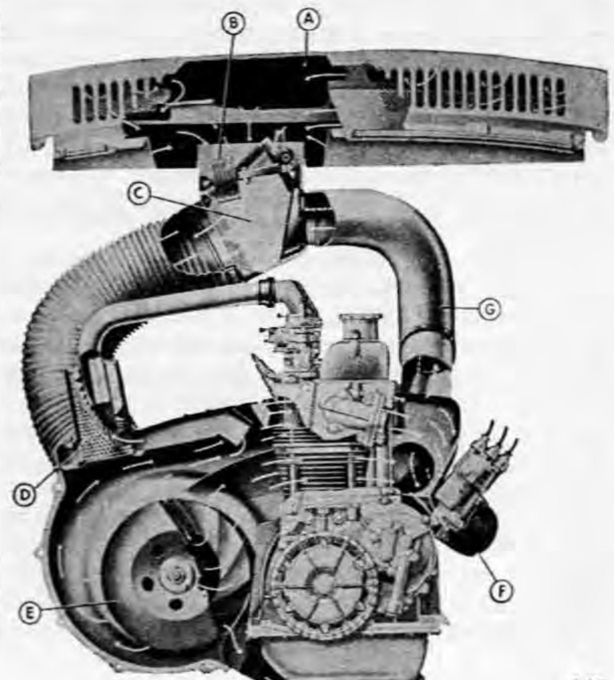


FIG. 125. Cooling control in the Fiat 500 engine.

by vibrations transferred from the engine. Air is brought to the carburettor from the delivery side of the fan and is therefore also heated in winter. A small overpressure of cooling air improves the charging of the engine. As is apparent from the illustration, the whole of the cooling circuit is entirely closed and no polluted air from the vicinity of the engine can enter it. The opening for the incoming air below the rear window is located sufficiently high, so that clean air is inducted.

3. Automatic control

The control should be automatic, i.e. independent of driver's action. To comply with this requirement a thermostat must be employed to control the cooling by the temperature of the engine. Automatic control can be based on either of the following factors:

1. temperature of oil,
2. temperature of air,
3. temperature of cylinder-head.

Temperature of oil. The transfer of heat from the flowing oil into the thermostat takes place under favourable conditions and is not liable to fluctuation. For this reason it is advantageous to base the control of the cooling of the engine on the oil temperature. Care must, however, be taken to ensure that heating of the oil does not proceed at a considerably lower rate than the heating of the cylinders. In the latter case the thermostat would not react sufficiently readily to the changes in the temperature of the engine. Besides, it will be necessary to maintain the temperature of oil in a definite relationship to the temperature of the cylinder head. Assuming, that the oil sump is exposed to an intensive stream of air during running, in winter-time the temperature of oil may remain low while the temperature of the cylinder head has reached a high level.

The temperature of exit air reacts readily to the changes of temperature of the cylinder head. Hence, the temperature of air is most frequently used for the control of the cooling. However, assuming a constant temperature of the cylinder head the temperature of exit air is strongly influenced by the temperature of outside (inlet) air. At a constant setting of the thermostat, the temperature of the head will be higher in summer, and lower in winter. This irregularity is restricted in scope if the thermostat is exposed to radiation heat from the cylinder head, or if the temperature of inlet air is controlled.

The temperature of the cylinder head is the most relevant value for control of cooling. However, no convenient thermostat is available of sufficiently small dimensions to fit it into the cylinder head without impairing the cooling. In aircraft engines thermocouples are frequently fitted by drilling, to signalize on the control-panel the temperatures of individual heads.

To a considerable extent the temperature of the cylinder head depends on

the load of the engine. If, therefore, the load of the engine is used as a criterion for the cooling control, the scope of the thermostat will be narrowed, the latter being called upon only to balance the difference in the outside temperature through changing the speed. Rapid changes in the load of the engine will not strain the thermostat, for such rapid changes in temperature will be automatically prevented. One of the following methods can be used to advantage:

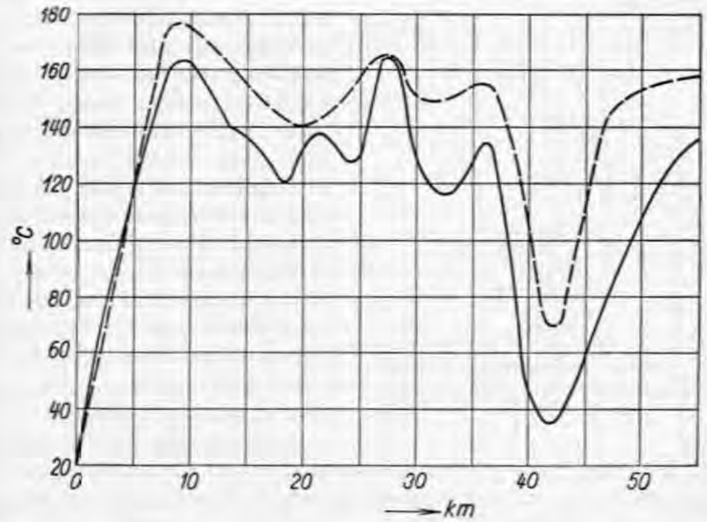


FIG. 126. Dependence of cylinder temperature in the VW engine upon distance. Full line — without thermostat, dash line — with thermostat.

1. mechanical linking of the throttle valve of the carburettor to the control device,
2. utilizing the vacuum in the induction manifold to correct the control of cooling,
3. correcting the cooling by the torque transmitted from the engine.

Experience alone will show which of these alternative methods is preferable.

The purpose of the automatic control is, in the first place, to reduce the fluctuation of the temperature of the engine during the running of the vehicle. If the cooling has been rated and dimensioned for hot summer weather, then, without a control, the engine will be undercooled in the winter-months. If excessive wear of the cylinders is to be avoided, their temperature must not drop below 70 °C. Fig. 126 illustrates the relationship of the temperature of the cylinder of a VW engine and the number of kilometers in a hilly region, at the outside temperature of —5 °C. When the vehicle runs down long slopes, the temperature of the cylinder drops as low as 35 °C. This temperature implies considerable wear of the cylinder, as the combustion products may condense on the walls (sulphur trioxide condenses at 45 °C) [8].

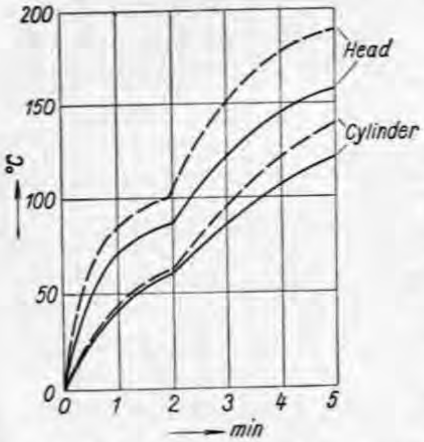


FIG. 127. Rate of heating of a VW engine with cooling control (dash lines) and without (full lines).

The dashed line indicates the temperatures of the cylinder on the same track, with a controlled cooling. In the latter instance, the temperature of the cylinders will not drop below 70 °C and hence no risk of excessive wear is incurred. In the former instance, without control of cooling, the vehicle covered some 3·7 miles (6 km) at a temperature lower than 60 °C, and this becomes apparent in the increased wear of the cylinder.

The curves of increased rate of heating of the air-cooled VW engine are apparent from Fig. 127. Full-line curves apply to an engine not provided with a control, and the dashed to an engine equipped with a cooling control. In this diagram the relationship is plotted between time on the one side, and the temperatures of the cylinder and of the cylinder head on the other. The engine had first been heated for two minutes on idle run, and at the lapse of two minutes it was run on even ground. The temperature of the outside air was 0 °C. After three minutes, the temperature of the cylinder equipped with a control of cooling was 15% and the temperature of the cylinder head was 22% higher than the temperatures registered in engines not provided with a cooling control.

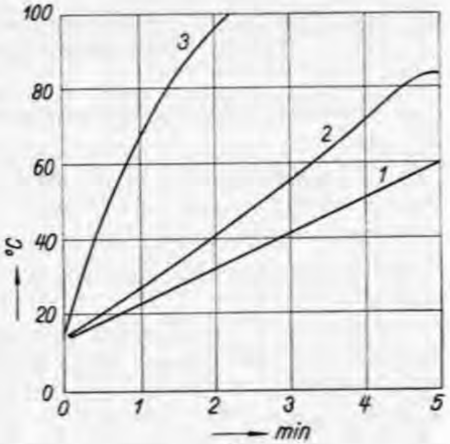


FIG. 128. Rate of heating of Deutz engines cooled by water and air.
1 — water-cooled engine without control, *2* — water-cooled engine with control, *3* — air-cooled engine.

The diagram demonstrates one of the noteworthy advantages of the air-cooled engine, namely speedy warming up. As only a limited quantity of material is concentrated around the cylinder, the cylinder can reach a temperature of 80 °C after only three

minutes, while in a water-cooled engine the heating of the water-contents in the block up to that temperature takes much longer. The difference between the cooling times of an air- and a water-cooled Deutz diesel engine [32] of equal dimensions can be observed in Fig. 128.

4. Thermostats

For an automatic cooling control thermostats are required to control the cooling of the engine according to temperature either directly, or indirectly through a servomotor. For the control of cooling the following thermostats are suitable.

Alcohol thermostats, consist of a metal bellows filled with alcohol, or a liquid having a low boiling point. As the temperature of the casing increases evaporation of the liquid sets in and a pressure arises proportionate to the pressure of the vapours of the liquid employed at that temperature. With the increase in pressure the metal bellows expands in length and governs either directly or indirectly the cooling control.

The metal bellows also serves at the same time as a spring. An example of the bellows thermostat is shown in Fig. 129. In order to increase the stroke a lever-type gear is employed. The operating force of the thermostat is illustrated in Fig. 130.

The beginning and the end of the action of the thermostat is determined by the boiling point of the liquid, the operating force, by the diameter of the bellows, and the control path, by the length of the bellows. According to the required temperature a suitable liquid or a mixture of different liquids must be selected.

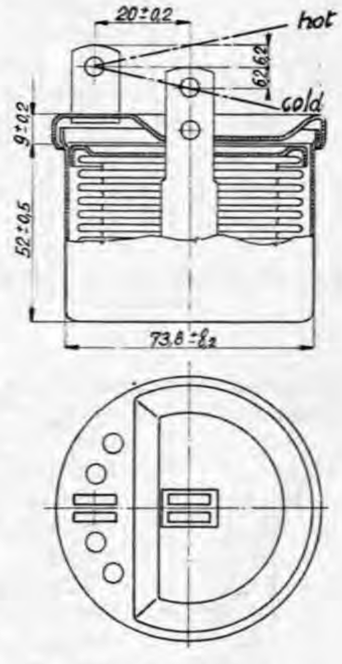


FIG. 129. Alcohol thermostat.

The bellows thermostat is the most universal, and it is easy to manufacture. However, if fitted directly to the engine, the risk is incurred of bursting the bellows through vibrations. The greater the stress imposed on the material of the bellows, the greater the risk of bursting. It is therefore preferable to fit the thermostat separate from the engine as is the case with the Tatra 603 (Fig. 123), or Fiat 500 (Fig. 125). It is also beneficial for the life of the thermostat not to use the full permissible stroke, as the latter would impose a severe stress on the material of the bellows.

Vacuum thermostats can also be used. When cold, the thermostat is contracted by the effect of the vacuum acting contrary to the force of the bellows (spring). When the liquid has evaporated the pressure of the vapours inside

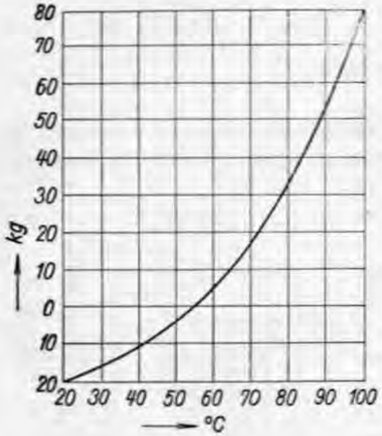


FIG. 130. Actuating force of an alcohol thermostat.

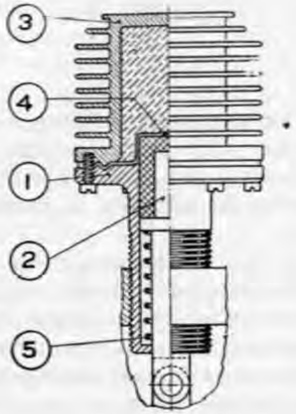
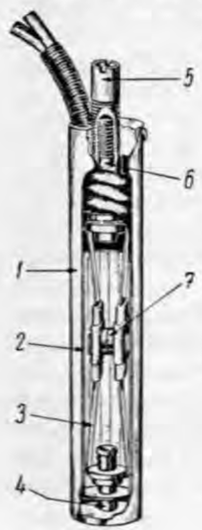


FIG. 131. Paraffin thermostat.
1 — body, *2* — piston, *3* — finned container, *4* — sealing diaphragm, *5* — reverse motion spring.

the bellows is increased to the atmospheric pressure or above it, and the elasticity of the bellows causes it to lengthen. This feature is useful in that in case of a breakdown in operation caused by a leakage in the thermostat atmospheric pressure will enter the bellows, the thermostat will open the cooling system fully and thus prevent overheating and seizing of the engine.



The paraffin thermostat operates on the principle that the contents of paraffin are increased considerably when paraffin passes from the solid into the liquid state. Such change in volume is utilized to operate a small plunger governing the control. A cross-section of a paraffin thermostat is illustrated in Fig. 131. The thermostat consists of body *1*, in which a guide for plunger *2* has been moulded. Vessel *3* containing the paraffin has been fitted to the thermostat body. The paraffin is separated

FIG. 132. Bimetal connector.
1 — cover tube, *2* — internal insulation, *3* — bimetal spring, *4* — pin, *5* — adjusting screw, *6* — fixing screw, *7* — silver terminals.

from the plunger by a neoprene diaphragm *4*, preventing the emptying of the paraffin from the pan. The plunger is held in its extreme position by spring *5*, fitted in the body of the thermostat.

If the thermostat is heated, at a definite temperature the state of the paraffin will change, the increased volume of the liquid paraffin will act through the diaphragm on the plunger of the thermostat. The operating force of the paraffin thermostat is considerable. According to the temperature at which the thermostat is to operate, different substances can be used.

The bimetal thermostat consists of two metals with differing thermal expansions. As a rule strips of two different metals are welded together forming a rod or a spiral. The bimetallic thermostat is simple and dependable. But its operating force and control path are small. The bimetal is frequently used for thermostatic electric switches, operating with a considerable accuracy, up to $\pm$ 1/20 °C. A thermostat of this type is shown in Fig. 132.

For electric switches it is an advantage to have contacts on the bimetallic spring which at a critical temperature will change suddenly from one position to the other and thus prevent burning.

CHAPTER IX

UTILIZATION OF EXHAUST GAS ENERGY

1. The energy of exhaust gases

Modern vehicle engines have reached a high degree of efficiency except for the portion of the energy which leaves the engine, without being utilized, in exhaust gases. Approximately one third of the energy contained in fuel is converted into useful work, one third is removed by cooling, and the balance leaves the engine through exhaust gases.

Through increasing the compression ratio the thermal efficiency of the engine is improved and more fuel heat is utilized. Also by reducing thermal losses in the system the thermal efficiency of the engine is improved. This can be achieved by reducing the surface area of the combustion chamber by using a hemispheric shape and also by using a short stroke design. In air-cooled engines the amount of heat transferred by cooling represents only 0·8 to 0·5 (b.h.p.) which means a certain improvement in performance and less work done by the cooling system. Unfortunately the amount of heat drawn off by exhaust gases is not diminished, and is sometimes increased.

This encourages efforts to recover the energy from exhaust gases and convert it into useful work which can be accomplished in one of several ways.

2. Utilization of heat from exhaust gases

Exhaust gases from stationary engines can be used to heat water although the net gain of this arrangement is not substantial. In a power-station with seven diesel engines and a total output of 4700 kW exhaust gases were utilized to heat water in the boilers; from 1 b.h.p. of engine-output 1 lb/h (0·54 kg/h) of steam under a pressure of 163·5 lb/in^2 (11·5 kg/cm^2) was obtained. This steam was employed to drive two steam-engines whose output totalled 100 kW. Taking into account that the energy contained in exhaust gases roughly equals the performance of the engine, the utilized energy is slight. In latest-type equipment 3·3 lb (1·5 kg) steam/h can be obtained per b.h.p. of the engine performance where supercharged engines with elevated temperatures of exhaust gases are used.

With vehicle engines there are, however, other alternatives. The simplest and oldest method for utilizing heat was heating the vehicle. In buses or

diesel rail cars the exhaust piping is fitted under the seats inside the body and the whole of the car is directly heated. Care must of course be taken to prevent leaking of exhaust gases into the body. This method of using heat involves comparatively low losses.

3. Gas turbine (turbo-charger)

If the losses are to be kept at a low level, efforts must be made to utilize the energy directly, i.e. for the drive of a turbo-charger.

First attemps of this kind met with considerable difficulties with regard to the material to be used for the manufacture of the vanes, in particular with petrol engines with high exhaust temperatures. For this reason the development of turbo-chargers was discontinued and not taken up again until the use of diesel engines had become widespread. The temperature of exhaust gases from the latter engine is no more than 500 to 600 °C, as compared with 700 to 800 °C or more from petrol engines. The lower temperatures of exhaust gases didnot place such high requirements on the material for the turbine blades, and hence several satisfactory designs of exhaust gas driven turbines have become available. The output obtained can be employed for driving a supercharger to give an increase of some 50% in performance, without substantially increasing mechanical and thermal stresses.

A supercharger can now be mounted on any normal engine, as neither mechanical nor thermal stress of the engine are substantially increased. The increase in the performance is not obtained through an increase in peak pressures.

If a blower is employed for the supercharging of the engine, the timing of the valve gear and the exhaust layout of the engine must be modified. Emphasis must be placed on the least possible back-pressure and the greatest possible charging pressure. These requirements can be met through converging exhaust gases from two or three cylinders into one branch of the exhaust manifold. However, the exhausts of individual cylinders must under no circumstances overlap, as this feature would affect adversely the scavenging of the cylinders. Highest pressure, exceeding the charging pressure, is attained in the exhaust piping upon opening the exhaust valve. In the course of the exhaust stroke the pressure decreases, so that towards the conclusion of the exhaust stroke when the inlet valve of the cylinder is opened, the pressure in the exhaust piping is lower than the charging pressure. This provides for the scavenging of the cylinder. A portion of the charging air passes out through the exhaust port directly into the exhaust system. For the working cycle of the engine this air is lost, but it exerts a beneficial influence cooling the walls of the combustion chamber and the exhaust valve.

The scavenging air also lowers the temperature of the exhaust gases. It should be noted that the heat transferred by the scavenging air from the exhaust valve and the walls of the cylinder is not completely lost, as part of it will be converted into work in the turbine.

From the above it is evident that a four or six-cylinder engine must have at least two branches in the exhaust system, while eight- or twelve-cylinder engine must have three to four separate systems. If the thermal losses in the exhaust piping are to be cut down to the least possible amount, the blower must be mounted in such a manner as to reduce the length of the piping to the minimum. Besides, the piping must be thermally insulated. Fig. 133 shows a blower used for vehicle engines.

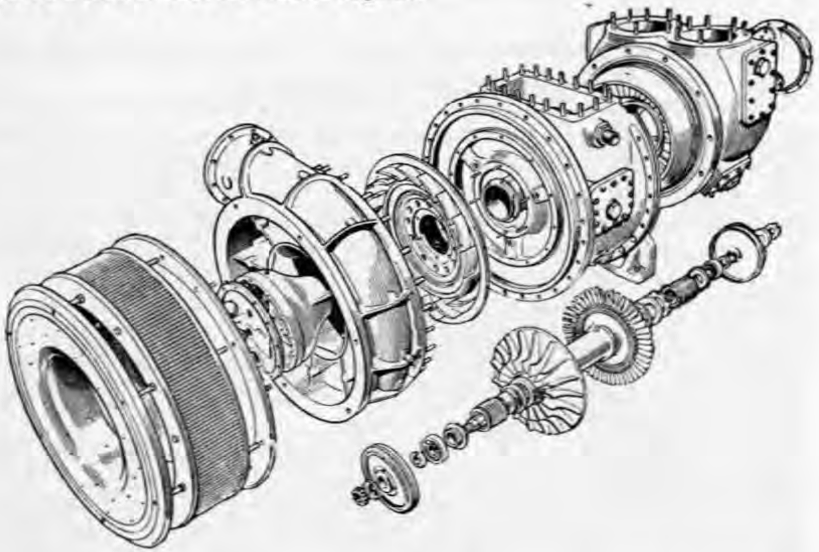


FIG. 133. Exhaust gas turbo-compressor used as a supercharger in compression-ignition engines.

Experiments have been made with much more powerful supercharging. In two-stroke engines with opposed pistons, supercharging by pressures of 2·3 or 6 atm has been tried. When the charging pressure of 6 atm was applied, the output of the engine equalled the input of the blower. Hence, the blower could be mechanically connected with the engine and the useful output was drawn only from the exhaust gas turbine. In the latter instance the reciprocating engine was actually replaced by a combustion turbine in whose design an engine-compressor was substituted for the combustion chamber. The situation is similar in the case of an engine with free pistons combined with an exhaust gas turbine. In that instance also the engine with free pistons is confined to the role of a generator of exhaust gases and the entire output required for the drive of the vehicle is generated by the turbine. By such an arrangement better thermal efficiency is attained than by a turbine provided with a combustion chamber.

For road vehicles the Eberspächer turbo-compressor using a centripetal turbine offers great advantages and permits the design of a small-dimensioned turbo-compressor. The overall layout of the latter is apparent from Fig. 134. This turbo-compressor is particularly suitable for air-cooled engines. Part of the air from the compressor is brought by a special duct to the bearings for cooling purposes and simultaneously prevents the exhaust gases from penetrating into the bearing area. The weight of the turbo-compressor will reach up to 35 lb (16 kg) for engines of up to 150—200 b.h.p. and 17·5 lb (8 kg) for engines of up to 100 b.h.p. Turbo-compressors of this type give a performance increase of some 30 %.

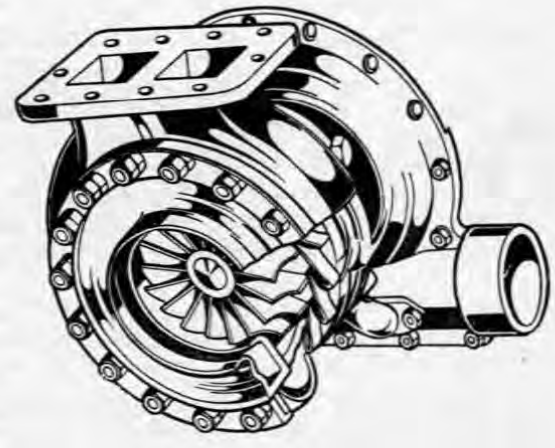


FIG. 134. The Eberspächer turbo-compressor.

Application of the exhaust gas turbine is not confined to compressor driving. It may also be used for driving a fan supplying cooling air. In the latter instance not only is there a gain in work used for the drive of the fan, but the cooling is automatically controlled. When the engine is fully loaded the exhaust gas turbine output is at its highest, so is the speed, and the maximum of cooling air is supplied. With the same speed of the engine and a diminished load the speed of the turbine will be reduced, and the cooling will be proportionate to the lower performance of the engine. The obvious advantages of this system have not been fully exploited because of comparatively expensive equipment.

4. Exhaust gas ejectors

The energy of the exhaust gases can be used directly, without the intermediary of any rotational device, for the suction of cooling air. This can be achieved by the use of an ejector which is a simpler device than an exhaust turbine connected with a fan. There is no lag in operation and it is absolutely reliable.

The ejector uses the kinetic energy of one fluid to draw in another. The layout of an ejector is apparent from Fig. 135. Exhaust gases are brought from the engine by pipes to the mouth of the ejector where the cooling air is entrained and a mixture of this air and exhaust gases leaves the end of the ejector to the atmosphere. The ejector consists of a branch of the exhaust piping (nozzle), a cylindrical mixing chamber, to which a conical extension called the diffuser is sometimes added. The whole equipment is therefore extremely simple, and as it contains no moving parts it does not require servicing.

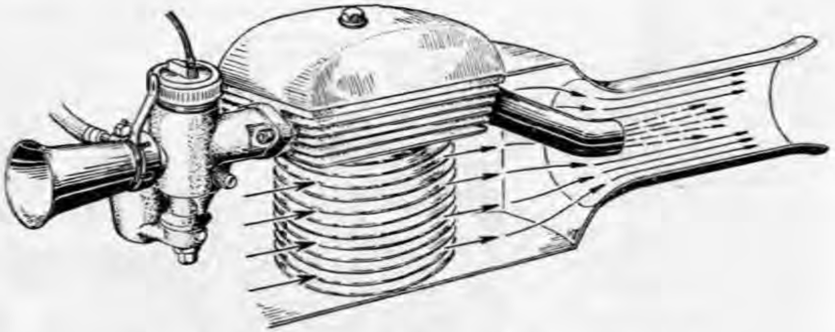


FIG. 135. Diagrammatical illustration of ejector cooling.

The flow of fluids in the ejector proceeds in accordance with Bernoulli's equation. The change in the pressure of the cooling air upon entry into the ejector amounts to:

$$p_0 - p_1 = \frac{1}{2}\gamma_1 \cdot v_1^2;$$

Assuming that the momentums of inlet and exit air are constant, the change in the pressure obtaining in the mixing chamber can be established. This, however, only holds true if the friction of gas against the walls of the mixing chamber is disregarded. The increment in air-pressure will then equal:

$$p_2 - p_1 = \frac{1}{A}\left[\frac{W_e \cdot v_e}{g} + \frac{W_a \cdot v_1}{g} - \frac{W \cdot v_2}{g}\right]$$

The change of pressure in the diffuser can be expressed by the equation:

$$p_3 - p_2 = \frac{1}{2}\gamma_2 \cdot v_2^2 - \frac{1}{2}\gamma_3 \cdot v_3^2$$

The actual conditions prevailing in the ejector are far more involved and cannot be expressed by simple equations. A rough outline of the rating is given in the references [39]. The symbols employed in the foregoing equations signify:

p denotes the pressure of gas,
v „ „ velocity of gas,

v_e denotes the velocity of exhaust gases leaving nozzle,
γ „ „ specific gravity of exhaust gases,
g „ „ gravitational acceleration,
W_e „ „ amount by weight of exhaust gases,
W_a „ „ amount by weight of cooling air,
A „ „ cross-section area of the mixing chamber,
1 „ „ subscript for entry into mixing chamber,
2 „ „ subscript for leaving mixing chamber and entering diffuser,
3 „ „ subscript for leaving diffuser.

Losses arising through friction against the walls of the ejector are not negligible and have to be taken into account for the rating. Every change in the flow entails certain losses, and the flow of gas from exhaust nozzles is not even, but pulsating. All these factors are responsible for the calculation becoming complicated. Empirical coefficients established through testing have, therefore, to be applied.

The action exercised by the ejector on exhaust gases is twofold. Firstly, in the chamber before the ejector vacuum will be formed for the suction of cooling air across the engine, and secondly, the outgoing mixture of exhaust gases and cooling air produces a drag. For a given amount of discharged exhaust gases and a given size of nozzle a greater drag will be obtained with ejectors having a small cross-section area and high velocity of discharge. The dimensions of the ejector are small, but the ejector is more sensitive to maintaining the supposed values.

If the ejector is employed for drawing the cooling air, it is not the drag of the ejector that is of importance, but the amount and the pressure of drawn air. Generally, the ejector can draw either small amounts of air with great pressure heads, or large amounts of air with small pressure heads.

For cooling air-cooled engines ten times the amount of air is required compared to the amount required for the air-fuel mixture. A widely spaced finning requires the supply of large quantities of low-pressure air, whereas a design providing for a dense finning will call for smaller quantities of high-pressure air. Ordinarily, a pressure head 3" to 6" (of 75 to 150 mm) water column is required. The cross-section of the mixing chamber is usually eight to ten times greater than the cross-section of the exhaust nozzle.

When laying down the size of the exhaust nozzle it must be taken into account that if the cross-section of the nozzle is less than the flow area in the exhaust valve at full opening, the back pressure in the exhaust piping reduces the performance of the engine. Therefore, the size of the exhaust nozzle should be somewhat greater than the flow area in a fully opened exhaust valve.

If a separate pipe is fitted from each cylinder to the mouth of the ejector, the pulsating effect of the exhaust will have been made use of, but only for the time of the exhaust of the corresponding cylinder. For the rest of the time the exhaust pipe unnecessarily blocks the useful cross section of the ejector. It is, therefore, preferable to conduct the exhaust from two or three cylinders, with regularly spaced ignition, into one exhaust pipe. A more even flow from the nozzle and a better utilization of the cross section of the ejector will thus be attained. However, the exhaust periods of individual cylinders must not overlap, so that the back pressures in the exhaust system do not obstruct the scavenging of the cylinders.

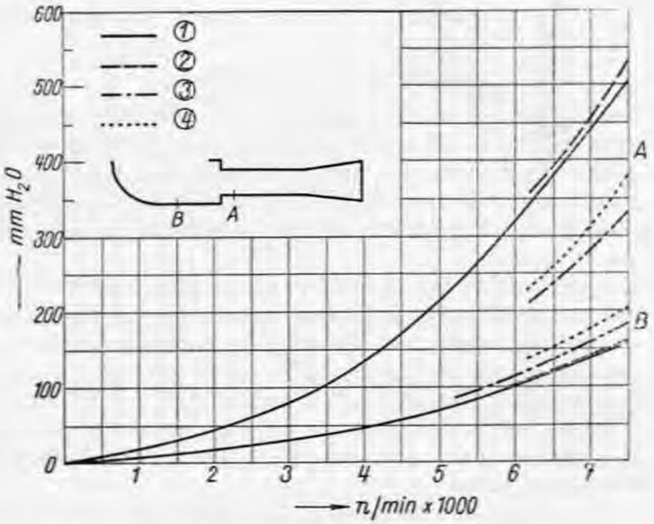


FIG. 136. Measured under pressures in different ejectors.

In a given ejector, joint exhaust piping from three cylinders will yield a greater vacuum than three independent branches each from one of the three cylinders. The branches of exhaust nozzles should open approximately on the level of the ejector orifice. Experiments with even flow nozzles do not reveal an impairing of the performance when the nozzles reach into the ejector branch. However, the useful length of the mixing chamber is diminished. It is of importance to ensure that the nozzles are distributed evenly over the area of the mixing chamber and that the gases are discharged in a direction parallel to the axis of the ejector. If exhaust gases come up against the wall of the mixing chamber, losses are incurred and the efficiency of the ejector is lowered. The diameter of the bevel to the inlet edge of the ejector can reach one sixth of the diameter of the ejector. A greater diameter than this unnecessarily extends the length of the ejector.

The length of the mixing chamber is related to its diameter and to the number of nozzles. At a given diameter of the ejector, the greater its length the

more even is the distribution of the velocity of the gases in the outlet area, and the performance of the ejector improves. But excessively long ejectors imply great losses through friction of the gases against the walls and a drop in the overall performance. The optimum length of the mixing chamber, as established by tests is, for single-nozzle ejectors, four to eight times the diameter. Considering that exhaust gases leave the nozzle at a certain angle,

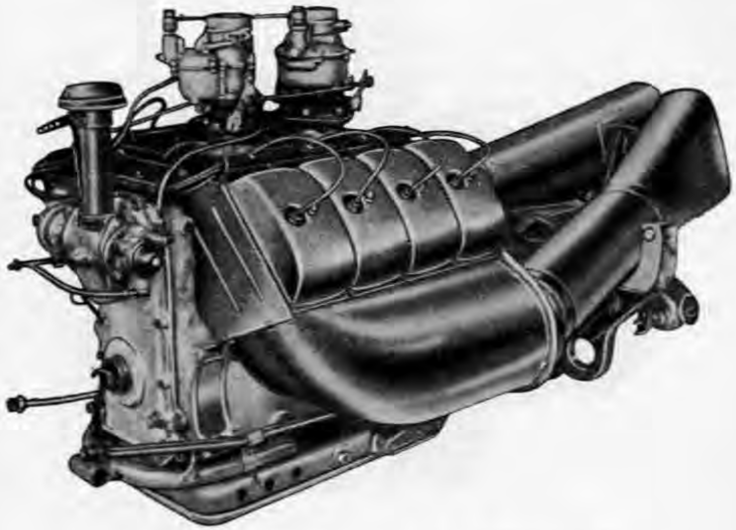


FIG. 137. The T 603 engine with ejector cooling.

the length of the ejector is dependent on the distance or tne edge of the nozzle from the wall of the mixing chamber. Therefore, if several nozzles evenly distributed over the cross-section are used, the optimum length of the ejector will be smaller than in the case of a single nozzle. The length of the ejector can be reduced by forming the mouth of the nozzle in the shape of a star, etc. Similar arrangements are responsible for increasing the contact surface between exhaust gases and air, and the efficiency of the ejector is thus enhanced.

The efficiency of the ejector will not be raised through fitting a greater number of nozzles, such as one nozzle from each cylinder, if the discharge is not regular. For low pressure heads a short mixing chamber will be sufficient. If an expanding diffuser is linked to the mixing chamber, the length of the mixing chamber can be reduced. Tests have proved that a cylindrical shape for the mixing chamber with a constant cross section is entirely satisfactory and that there is no need to apply a Venturi tube.

The efficiency of the ejector is improved by the use of a conical diffuser fitted behind the mixing chamber. An apex angle of 8° has proved the most con venient. In other than circular diffusers the increment in the surface of the cross section of the diffuser should be such as to correspond to a circular cross section with an angle of 8°. A limited length of the ejector combined with a greater angle will produce a greater flow at the expense of the drag. If the

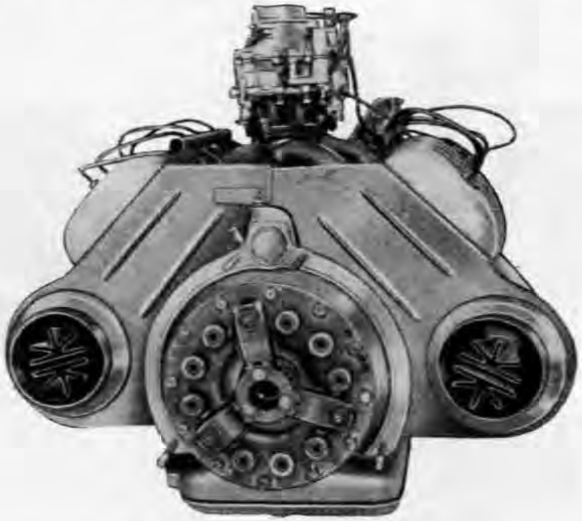


FIG. 138. Star-shaped tail ends of the exhaust pipe.

area at the delivery end of the diffuser is only slightly greater than the area at the intake, the optimum angle of widening will exceed 8°.

If the energy of the exhaust gasses is to be exploited thoroughly, the exhaust piping must be short and not exposed to excessive cooling. This gives difficulties of silencing. The Tatra works experimented with ejector-type cooling for racing cars, where noise is unimportant, while at high speeds the economy in input to be used for the fan is substantial and the dependable operation of the ejector is most welcome.

After the first promising tests with the T 603 engine, where the ejector worked in series with the axial fan, thorough tests have been arranged on a two-cylinder engine with cylinders of 75 mm bore and a stroke of 72 mm. The results of such tests are illustrated in Fig. 136. Different diameters of mixing chambers were experimented with and by means of a telescopic arrangement their length could be modified while the mechanism was in operation. Also different shapes of end-sections have been tried. Based on the results of experiments, ejectors have been constructed both for the two-cylinder T 605

car and for the eight-cylinder T 607 car. These ejectors have proved outstanding in operation and will cool the engine safely even at maximum performance.

The final arrangement of the T 603 ejector-cooled engine is shown in Fig. 137. The engine has a 75 mm bore and 71 mm stroke. At 7500 rev/min and a compression ratio of 12 : 1 the performance of the engine is 200 b.h.p. On each side of the engine one ejector is mounted, with a mixing chamber 140 mm dia. The end-section of the exhaust is double star-shaped as illustrated in Fig. 138. Groups of two cylinders are each connected by one pipe with half the star of an area of 1·5 in^2 (10 cm^2). The length of the mixing chamber including the diffuser is 33·4" (850 mm). The discharge temperature from the ejector is 160 °C at full load. The mixing chamber of the two-cylinder engine is 3·5" (90 mm) dia.

The ejector type of cooling might also be used for stationary engines, providing there is sufficient room for fitting silencers behind the ejector. The saving in fan energy will give lower fuel consumption. The cooling control is to a large extent automatic, for at a full load the greater amount of exhaust gas draws in an increased quantity of air. It is possible that with a convenient type of finning a silencer, possibly of the Burgess type, could be fitted before the ejector, thus making this type of cooling acceptable for road vehicles.

PART TWO

DESIGN

CHAPTER X

GENERAL ENGINE LAYOUT

1. Basic comparative values

In designing an engine of a given power output, the requirements of light weight, compact overall dimensions and economy of operation must be borne in mind. By the latter not only a low specific fuel consumption is understood, but also simple maintenance, fatigue resistance and long engine life. If all these requirements are to be met, they must be first of all thoroughly studied by the designer in order to enable him to select the fundamental engine parts and general layout best suited for given conditions.

In determining the most suitable engine size, use is made of the familiar formula:

$$\text{b.h.p.} = \frac{A\,.\,L\,.\,i\,.\,n\,.\,p_e\,.\,z}{75 \times 60} = \frac{Vn\,p_e}{C} \qquad (41)$$

where C = 450 for two-stroke engines;

C = 900 for four-stroke engines;

$A = \dfrac{\pi D^2}{4}$, i.e. the piston area (cm²)

L denotes the stroke (m)

i „ „ number of cylinders

n „ „ revolutions per minute

p_e „ „ mean effective pressure (kg/cm²)

D „ „ bore (cm)

V „ „ total displacement (dm³)

z „ „ number of expansion strokes per revolution.

By using the mean piston velocity formula

$$c_m = \frac{L\,.\,n}{30}$$

engine performance may be expressed as

$$b.h.p. = \frac{A\,.\,c_m\,.\,i\,.\,p_e\,.\,z}{150} = D^2\,.\,c_m\,.\,i\,.\,p_e\,.\,z\,.\,\frac{\pi}{4 \times 150}$$

This equation makes it clear which values affect the engine performance. By expressing the stroke in terms of stroke: bore ratio, $\xi = \dfrac{L}{D}$ the following formulae are obtained:

$$\text{b.h.p.} = D^3\,.\,\xi\,.\,n\,.\,i\,.\,p_e\,.\,\frac{\pi}{36\,000} \quad \text{for four-stroke engines and}$$

$$\text{b.h.p.} = D^3\,.\,\xi\,.\,n\,.\,i\,.\,p_e\,.\,\frac{\pi}{18\,000} \quad \text{for two-stroke engines} \qquad (42)$$

Before deciding on the size of cylinder to be used, the values of p_e, n, ξ, i must be chosen. Before this is done, relationships derived from the principle of similarity of engines must be considered.

Any two engines with the same basic layout and number of cylinders of differing dimensions may be considered as geometrically similar. For complete similarity some further conditions would have to be met: together with the changes of cylinder bore, proportional changes would have to be made not only in stroke, but also in connecting rod length, the size of the crankshaft etc., and even in the thickness of castings and the dimensions of electrical accessories, carburettors, fuel injection equipment, bolts and so on. These latter conditions cannot be met in practice, but this may be assumed to have been done if the dimensional differences in question are not considerable. In-line engines cannot be compared with V engines etc. The weight: bore ratios of six cylinder in-line engines are shown in Fig. 139, where weights of various

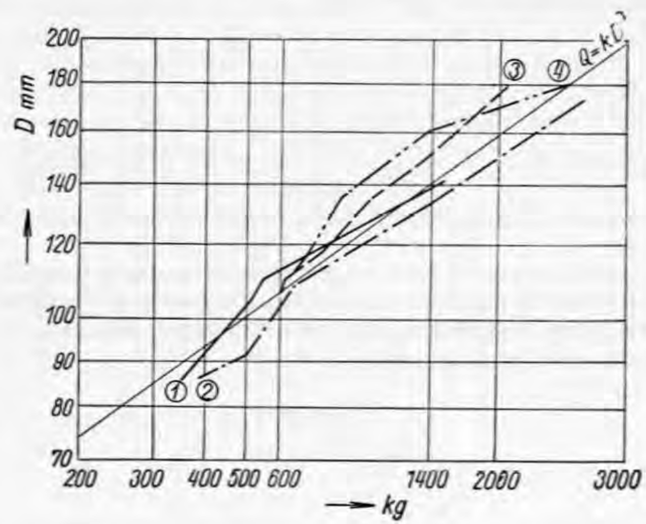


FIG. 139. Weights of six-cylinder in-line engines plotted against bores.

engines of one make are plotted against their bores and connected by one line in the graph. It is apparent that the weight: bore lines of all engines very closely follow the line $Q = k\ D^3$ which expresses the relationship between engine dimensions and weight.

In comparing two geometrically similar engines a means of comparison is provided by the ratio

$$m = \frac{D_0}{D} \tag{43}$$

where D_0 denotes the bore of the derived engine
D „ „ bore of the original engine.

For geometrically similar engines the following relationships hold:

all length dimensions are proportional to m,
all areas (of pistons, valves etc.) are proportional to m^2,
engine displacement is proportional to m^3,
weight of engine and parts is proportional to m^3.

Supposing that the mean effective pressure and mean piston speed of both engines under consideration are equal, further relationships may be expressed:

engine performance	$\propto m^2$,	performance per litre	$\propto m^{-1}$
speed	$\propto m^{-1}$	weight/performance ratio	$\propto m$.

Engine performance should be proportional to D^3 according to (42). Since, however, mean piston velocity is limited and remains constant, n may be expressed in terms of piston velocity as

$$n = \frac{30\,.\,c_m}{D\,.\,\xi} \tag{44}$$

Therefore

$$\text{b.h.p.} = D^3\,.\,\xi\,.\,i\,.\,p_e\cdot\frac{\pi}{600}\cdot\frac{30\,c_m}{D\,.\,\xi} = D^2\,.\,i\,.\,p_e\,.\,c_m\cdot\frac{\pi}{20} \tag{45}$$

The performance of geometrically similar engines therefore varies with D^2 i.e. with m^2.

Engine speed, so long as mean piston speed is constant, therefore varies inversely with D according to (44). Speed will therefore be proportional to m^{-1}.

Performance per litre is obtained by dividing engine performance by displacement. As total displacement is calculated:

$$V = \frac{\pi D^2}{4}\,.\,D\,.\,\xi\,.\,i,$$

performance per litre is

$$\text{b.h.p.}/l = \left(D^2\,.\,i\,.\,p_e\,.\,c_m\,\frac{\pi}{20}\right) : \left(\frac{\pi}{4}\,.\,D^3\,.\,\xi\,.\,i\right) = \frac{p_e\,.\,c_m}{5\,.\,D\,.\,\xi} \tag{46}$$

The performance per litre varies therefore inversely with D and with m^{-1}. The smaller the bore, the larger will be the performance per litre.

The weight of an engine is proportional to the cube of any of its longitudinal dimensions and a given constant C_1. The weight/b.h.p. ratio can be expressed as

$$G_{hp} = \frac{D^3\,.\,C_1}{D^2\,.\,C_2} = D\,.\,C_3 \tag{47}$$

where C_1 C_2 and C_3 are constants.

The weight/b.h.p. ratio of geometrically similar engines therefore varies in direct proportion to their bore D and is directly proportional to the ratio m.

The volumetric efficiency which is affected by various conditions is also directly proportional to the velocity of the mixture (or air) at the inlet valve. Other considerations in respect of volumetric efficiency may be omitted with geometrically similar engines showing only small differences in dimensions. Gas velocity in the inlet valve is proportional to the port area (m^2) and piston displacement (m^3). It follows:

$$v_1 = \frac{m^3\,.\,C_1}{m^2\,.\,C_2} = m\,.\,C_3$$

But keeping the mean piston velocity constant, the amount of gas admitted is not proportional to m^3 but to m^2. Therefore

$$v_1 = \frac{m^2\,.\,C_1}{m^2\,.\,C_2} = C_3 \tag{48}$$

It follows that with constant mean piston velocity the volumetric efficiency does not vary.

Bearing load depends on gas pressure acting on the piston and the accelerating forces of the piston and its accessories. In geometrically similar engines piston area and bearing surface vary concurrently and, therefore, unit bearing pressure remains constant.

Accelerating forces may be expressed as

$$P_a = M\,.\,r\,.\,\omega^2; \quad \omega = \frac{\pi\,.\,n}{30}$$

Unit bearing pressure may be obtained by dividing the force P_a by bearing surface area A_1. The respective values vary with the ratio m as follows:

$$M \propto m^3,\ r \propto m,\ n \propto m^{-1},\ \omega^2 \propto m^{-2},\ A \propto m^2$$

It follows that unit bearing pressure is

$$\frac{P_a}{A} = \frac{m^3\,.\,m\,.\,m^{-2}\,.\,C_1}{m^2\,.\,C_2} = C_3 \tag{49}$$

Unit bearing pressures, therefore, remain constant in geometrically similar engines in which the mean piston velocity is constant.

With greater dimensional differences between geometrically similar engines it has to be remembered that some factors are not governed by engine size; these are, oil and air viscosity, the rate of combustion, pressure oscillations in the fuel system, the induction and exhaust pipes, etc. For this reason the compression ratio is generally reduced with increased cylinder dimensions. This connection between displacement and compression ratio is considerable as is shown by the results obtained from an experimental single cylinder engine with two stroke dimensions of 98·4 mm and 127 mm and bore varying from 70 mm to 95 mm. The experimental engine was water-cooled, its cylinder head was of cast iron, compression ratio variable and its temperature was kept at 100 °C. Spark advance was set for maximum b.h.p. and the fuel air mixture for maximum engine knock. The fuel used was of 80 octane rating. The test was conducted at the engine speed of 600 rev/min and the results are shown in Fig. 140.

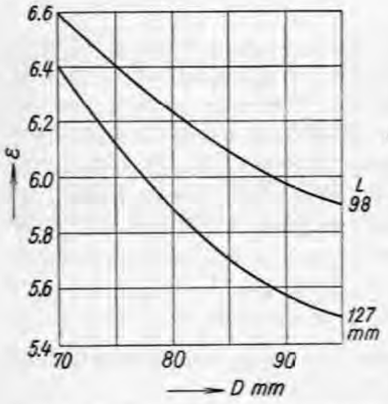


FIG. 140. Effect of bore upon maximum admissible compression ratio.

Another interesting fact is apparent from Fig. 140. Short - stroke engines can work with a higher compression ratio. This is in keeping with the fact that smaller cylinders will operate with higher compression ratios, since the mixture is better cooled during compression, and compression temperatures and pressures are, therefore, lower.

An engine with a stroke of 127 mm and bore 80 mm has a displacement of 638 cm^3. In order to have the same displacement, an engine with a 98 mm stroke must have a 91 mm bore. The curves in Fig. 140 show that with cylinders of equal displacement, a higher compression ratio may be used in the engine with the shorter stroke (in this particular case 5·96 instead of 5·9).

By lowering the compression ratio, both thermal efficiency and engine performance are reduced. The considerable influence of the compression ratio upon thermal efficiency is shown in Fig. 141.

This diagram refers to the thermal efficiency of an ideal cycle following the formula

$$\eta_t = 1 - \left(\frac{1}{\varepsilon}\right)^{\varkappa-1}$$

The thermal efficiency rises rapidly with increasing compression ratio up to 8 : 1, after which the increase becomes comparatively lower. The interests of high thermal efficiency therefore dictate the choice of the highest possible compression ratio.

To complete the picture, the influence of some major factors upon engine performance are tabulated in Table 25. This shows the influence of altering one single factor in order to raise engine performance on the displacement of a single cylinder V_1, total engine displacement V, performance per litre, bore

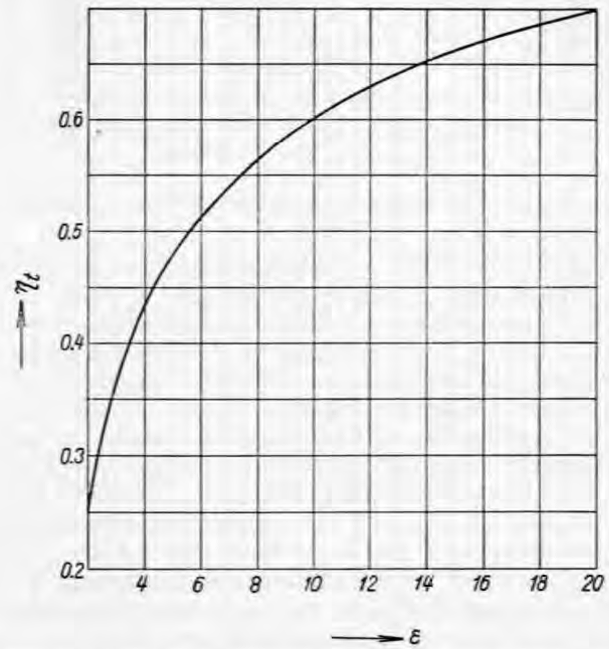


FIG. 141. Effect of compression ratio upon thermal efficiency.

D, number of cylinders i and n. The engines in all cases are geometrically similar.

The upper line of the table contains the fraction expressing in its numerator the factor to be increased x times and in its denominator the factor which remains unchanged.

The first column refers to an engine, the performance of which was increased by increasing cylinder dimensions, and thereby its displacement per cylinder, without altering the mean effective pressure p_e and mean piston velocity c_m. In order to adhere to the same piston velocity with the derived engine, speed must be reduced by $n/\sqrt[3]{x}$. As a result the performance per litre would be reduced and to counteract this a larger number of cylinders $i \,.\, \sqrt[3]{x}$ must be employed. The engine capacity will therefore rise more than x times, the

Table 25

Geometrical Similarity of Engines

Values increased x times	V_1	i	n	z	p_e
Invariables	p_e, c_m	p_e, c_m	$p_e c_m$	p_e, c_m	i, c_m
Total displacement V	$xV . \sqrt[3]{x}$	$x . V$	V	V	V
Bore D	$D . \sqrt[3]{x}$	D	$D/\sqrt[3]{x}$	D	D
Performance b.h.p.	x . (bhp)	x . (bhp)	x . (bhp)	x . (bhp)	x . (bhp)
Number of cylinders i	$i . \sqrt[3]{x}$	$i . x$	$i . x^3$		i
Engine speed n rev/min	$n/\sqrt[3]{x}$	n	$x . n$	n	n
Mean effective pressure p_e	p_e	p_e	p_e	p_e	$x . p_e$
Performance per litre bhp_l	$(bhp_l)/\sqrt[3]{x}$	bhp_l	$x . bhp_l$	$x . bhp_l$	$x . bhp_l$

increase being actually $V . x . \sqrt[3]{x}$. The bore will be altered to $D . \sqrt[3]{x}$ and the same increase applies to all other dimensions except overall engine length, which will increase still more because of the larger number of cylinders. The increase in engine weight is approx. $x . \sqrt[3]{g}$. An increase of performance is, in this instance, accompanied by a considerable increase in the size and weight of the engine.

The second column refers to an increased performance solely by employing a larger number of cylinders, while p_e and c_m remain unchanged. This is a simple case and only the number of cyliders increases x times. The size and weight of the engine are also increased approximately x times.

The third column refers to an increase of performance gained by an increase in speed. In order to keep the mean piston velocity down at its original value, cylinder dimensions have to be decreased $1/\sqrt[3]{x}$ times. Performance per litre rises x times, engine displacement remains unchanged and the number of cylinders will rise to $i . x^3$. Such a solution is advantageous considering engine dimensions and weight only, for the increase in the number of cylinders is impracticable. In automobile prime movers the number of cylinders is limited practically to eight or twelve.

The last two columns are of particular interest. In the fourth column the change concerns the number of expansion strokes per crankshaft revolution, i.e. the comparative values of a four-stroke and two-stroke engine are considered. The performance per litre of a two-stroke engine is double that of the four-stroke assuming that the p_e and c_m remain constant. Other values also remain unchanged. The engine weight however is increased by the addition of a supercharger to supply mixture under pressure or to scavenge the cylinders. The last column demonstrates the effects of changing only p_e with an unaltered number of cylinders i and c_m. Engine performance and performance per litre increase in the same ratio as p_e. The weight of the engine is in this instance increased by the weight of the supercharger.

2. Mean effective pressure

The formula for calculating engine performance is very simple with p_e as the only unknown. The mean effective pressure is dependent on many factors. Its relation to the indicated mean effective pressure p_i is $p_e = p_i . \eta_m$, where η_m is the mechanical efficiency of the engine.

The indicated mean effective pressure is obtained as the average ordinate of the indicator diagram. As there is no indicator diagram available during the various stages of the design of an engine and the construction of such a diagram based on theory is questionable, the designer has to make use of other methods.

The mean effective pressure is influenced in particular by volumetric, thermal and mechanical efficiency.

Volumetric efficiency depends to a great degree on the system of the valve gear and will be mentioned in detail in the chapter on valve gear. It is clear however, that the more air that can be supplied to the combustion chamber, the more fuel can be burned; and with more heat energy liberated by combustion, engine output will be higher for a given thermal efficiency.

Thermal efficiency expresses the percentage of the heat contained in the fuel converted into useful work done by the engine. On any engine, the thermal efficiency may be calculated from

$$\eta_t = \frac{632}{V_p . b}$$

where V_p denotes the calorific value (kcal/kg)

b „ „ specific fuel consumption (kg/b.h.p.-h.)

The theoretical thermal efficiency depends chiefly on the compression ratio and for a petrol engine it is

$$\eta_t = 1 - \left(\frac{1}{\varepsilon}\right)^{\nu-1}$$

This relationship is plotted in Fig. 141, but it must be borne in mind that thermal efficiency is also influenced by engine design. Two engines of equal compression ratio but unequal distances of flame travel (the one having a compact combustion chamber, the other an elongated one) will have different thermal efficiencies.

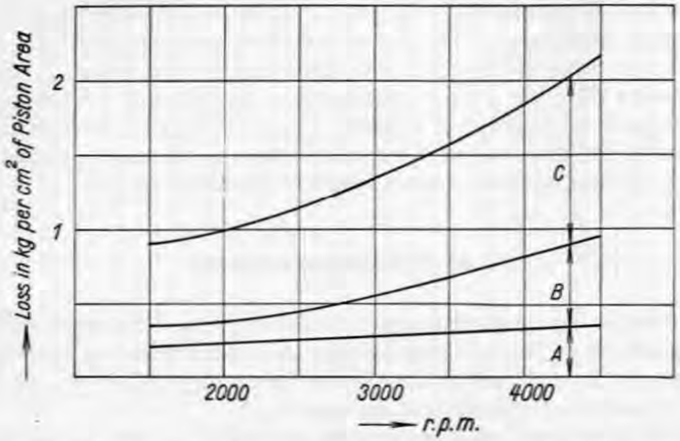


FIG. 142. Distribution of mechanical losses plotted against engine speed.
A — Bearing friction and auxiliaries, *B* — Induction losses, *C* — Losses by piston friction.

Mechanical efficiency is a measure of the losses during the transference of work from the power stroke to the flywheel. With lower frictional losses in the engine, the mechanical efficiency rises and so does the engine performance. The principal mechanical losses in the engine are caused by piston friction. Fig. 142 shows the various losses in relation to the engine speed. It will be seen that frictional losses caused by pistons and by charging show a sharp increase with rising speed thus reducing mechanical efficiency. The accelerating forces acting on the moving mass of the engine increase with higher engine speed, thereby increasing the frictional losses caused by pistons as the square of the speed value.

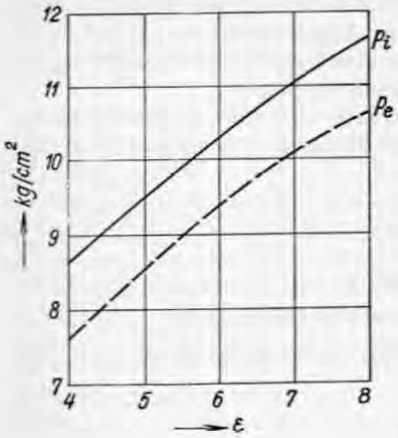


FIG. 143. Influence of the compression ratio upon mean indicated pressure p_i and mean effective pressure p_e.

The use of roller bearings makes for higher mechanical efficiency as compared with plain bearings. Engine accessories, such as the dynamo, supercharger, fan or ventilator, water

pump, etc., all have an adverse effect on mechanical efficiency and lower mean effective pressure.

Fig. 143 illustrates the variation of p_i and p_e with the compression ratio. By increasing the compression ratio of an automobile engine from 6 : 1 to 11½: 1 the results shown on Fig. 144 were obtained. With high compression ratios, iso-octane with 0·8 cm³ of tetra-ethyl-lead added per litre was used for fuel. The improvement on b.h.p. at 3000 rev/min with high compression ratios is relatively small.

During the initial stages of designing an engine the m.e.p. must be approximated, taking into account

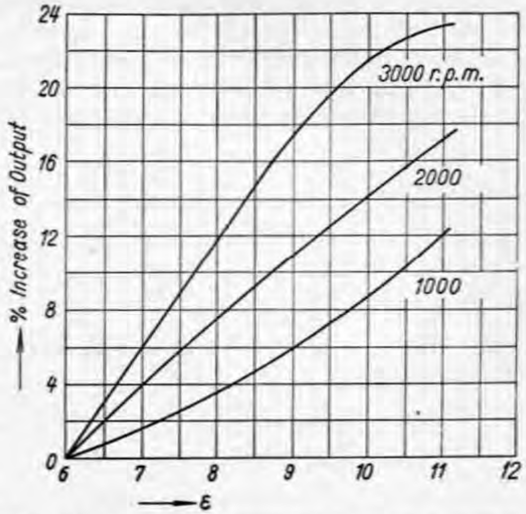


FIG. 144. Effect of compression ratio upon engine performance at various speeds. Note the small increase in output at 3,000 rev/min and $\varepsilon > 10$.

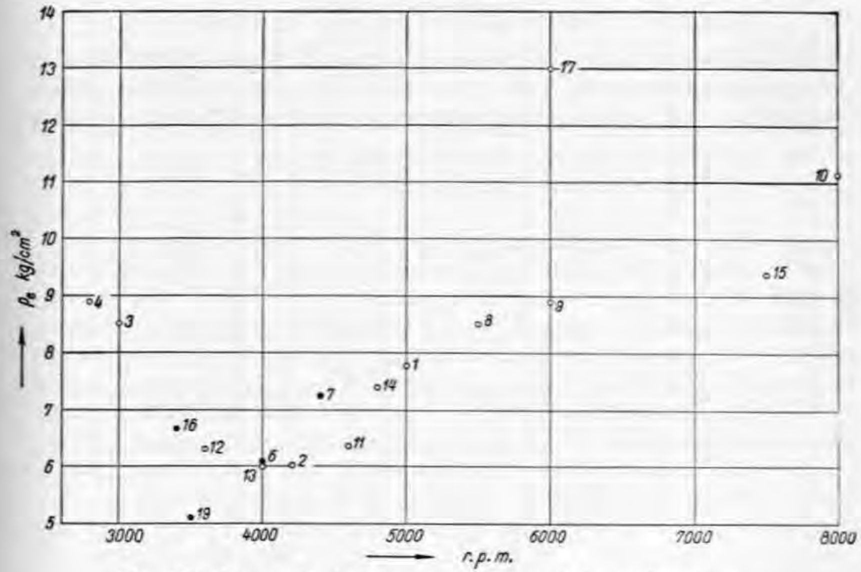


FIG. 145. Mean effective pressures in air-cooled petrol engines.
○ — hemispherical combustion chamber, ● — wedge-shaped combustion chamber.

experience with other engines and allowing for differences caused by the particulars of the valve gear, compression ratio, the octane rating of the fuel

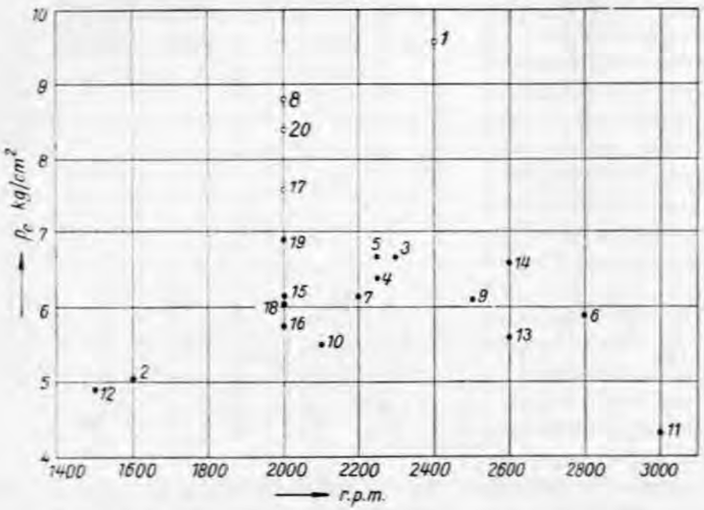


FIG. 146. Mean effective pressures in air-cooled oil engines.
○ — supercharged engines.

to be used, the proposed cooling system and so on. Mean effective pressures of engines mentioned in Table 2 are plotted in Fig. 145. Mean effective pressures of compression-ignition engines referred to in Table 3 are shown

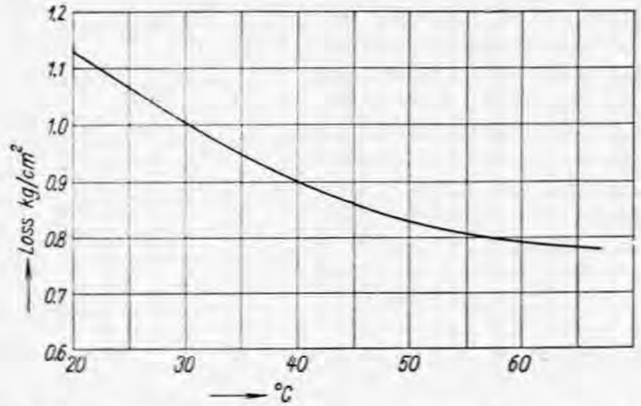


FIG. 147. Effect of cylinder temperature upon friction losses in a four-cylinder lorry engine at 900 rev/min.

in Fig. 146. These values are suitable for reference when estimating the mean effective pressure of a proposed engine design.

Mechanical losses are also influenced by cylinder wall temperature. At low temperatures the viscosity of cylinder lubricants is higher and frictional losses are correspondingly increased. Ricardo [51] observed the variation of mean loss on pressure with cylinder-wall temperature as shown in Fig. 147. One of the

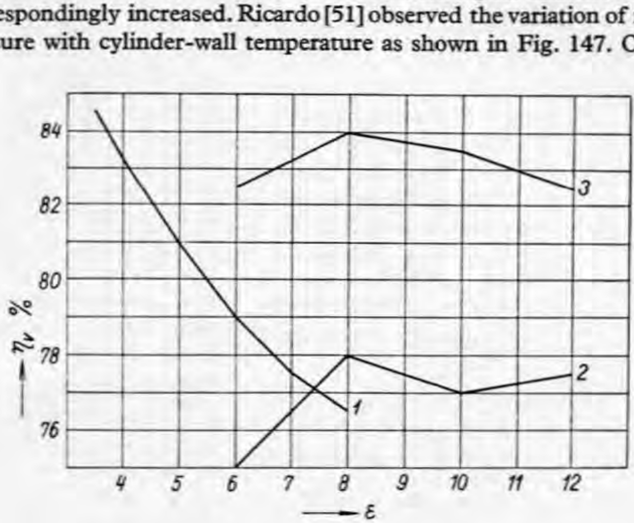


FIG. 148. Effect of compression ratio upon volumetric efficiency.
1 — according to test by Ricardo, *2* — experimental GM engine for a high ε at 3,200 rev/min, *3* — experimental GM engine for high ε at 2,000 rev/min.

apparent advantages of the air-cooled engine lies in the fact that its cylinder wall temperature never, even under light load, drops below a critical point of about 80 °C. In order to reduce loss through friction in winter conditions, low viscosity oils must be used.

The increase in thermal efficiency with compression ratio is to a certain degree cancelled by a drop in volumetric efficiency. The influence of compression ratio upon volumetric efficiency is shown in Fig. 148. Ricardo observed that increased volumetric efficiency in low compression engines is due to the cooling of residual gases in the combustion chamber, which is large in low compression engines. After the exhaust valve closes, hot exhaust gases remain in the combustion chamber. In the following induction stroke these gases are cooled, they contract and the difference in volume makes it possible to bring a correspondingly increased charge of fresh gas into the cylinder.

If a large valve overlap is used the above is no longer valid, as under such conditions cylinder scavenging is good, the exhaust gases being expelled from the cylinder by the fresh charge.

3. Engine speed

In order to keep the engine small in size and light in weight, the best possible use has to be made of crankshaft speed up to the highest permissible mean piston velocity c_m. The mean piston velocity of the modern automobile petrol engine is about 32 to 49 ft/s (10 to 15 m/s) and in the case of the compression-ignition engine about 26 to 39 ft/s (8 to 12 m/s). The mean piston velocity is limited by the accelerating forces of the crank mechanism and the thermal load upon the piston.

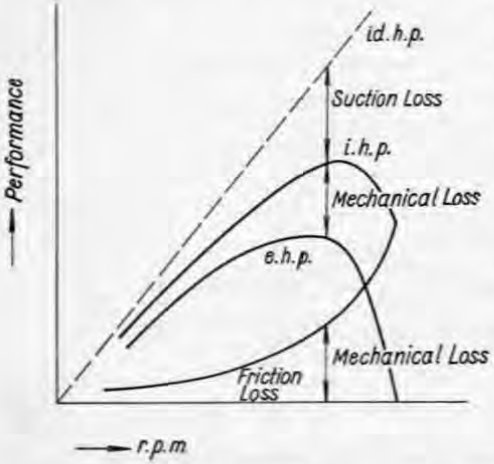


FIG. 149. Dependence of ideal, indicated and effective performance, and losses on engine speed.

It can be seen in Fig. 142 that mechanical efficiency drops with rising crankshaft speed owing to increased frictional losses.

The accelerating force acting on the piston increases as the square of crankshaft speed. With even a small crankshaft speed increase, the drop in efficiency is therefore rapid.

The influence of crankshaft speed on b.h.p. is shown in Fig. 149. Assuming that ideal mechanical and volumetric efficiencies of 100% could be obtained, engine performance would rise in direct proportion to crankshaft speed as indicated by the dashed straight line of ideal engine performance. Under real conditions however, losses in the induction system must occur and therefore, at higher crankshaft speeds, a charge lower than the displacement enters the cylinder, the volumetric efficiency not being 100%, but rapidly decreasing with growing crankshaft speed. That is why inside the cylinder only an i.h.p. reduced by suction losses is produced, as shown by the full curve. This output is then further reduced by mechanical losses to the resulting b.h.p. curve. The aggregate performance loss rises rapidly with rising crankshaft speed and at a certain speed equals i.h.p. This marks the maximum possible crankshaft speed of an engine, reached without load and on a fully open throttle. Under such conditions all i.h.p. is used up to overcome frictional losses and none is left to do useful work. The mechanical efficiency in such a case, expressed as a percentage, equals 0%.

Crankshaft speed may be increased only to about 60% of mechanical efficiency as below this percentage running economy is very low and both

specific consumption and engine wear are high. A large proportion of the i.h.p. is used up by the mutual abrasion of various engine parts. Bearing only this point in mind, the lowest possible crankshaft speed commensurate with high mechanical efficiency and low specific consumption would be of great advantage. This would however produce large and heavy low speed engines.

The modern automobile prime mover must be small and light and hence the apparent effort to increase crankshaft speed. For this trend of development to be successful, good mechanical efficiency must be ensured. Attention must be focused upon the crank assembly and in particular on pistons. A light crank assembly may run at higher speeds without the risk of failure since the accelerating forces in question are small and frictional losses are kept within reasonable limits. Bearing pressures in connecting rods and main bearings can be coped with under such conditions. A short-stroke engine design also makes it possible to make the best use of crankshaft speed.

4. The short-stroke engine

Automobile engine design of recent years shows a quite definite trend toward the short-stroke engine. Modifying equation (41) by using mean piston velocity c_m in place of stroke, the formula for b.h.p. of a four-stroke engine is obtained:

$$\text{b.h.p.} = \frac{A \cdot c_m \cdot i \cdot p_e}{300}$$

Stroke is omitted in this formula and it is thus evident that output depends primarily on piston area (the bore D) and mean piston velocity c_m. Long stroke and lower crankshaft speed or short stroke with high crankshaft speed may be used with equal results.

The use of high crankshaft speed with a short stroke, however, makes it possible to design a smaller and lighter engine, which with its smaller displacement, will give a performance equal to that of an engine with larger displacement.

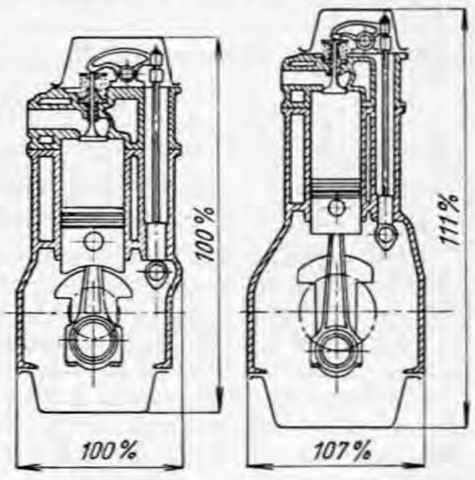


FIG. 150. Comparison of dimensions of engines with stroke: bore ratios of 0.8 and 1.4. The 0.8 ratio is taken as 100 %.

In order to exploit this advantage stroke is being reduced so radically today that in many cases it is smaller than bore and such engines are said to be "over-square". Mean piston velocity generally does not rise, indeed in some cases it is even reduced. As a result of a reduced stroke, connecting rod weight is reduced and often pistons are lighter, thus increasing mechanical efficiency.

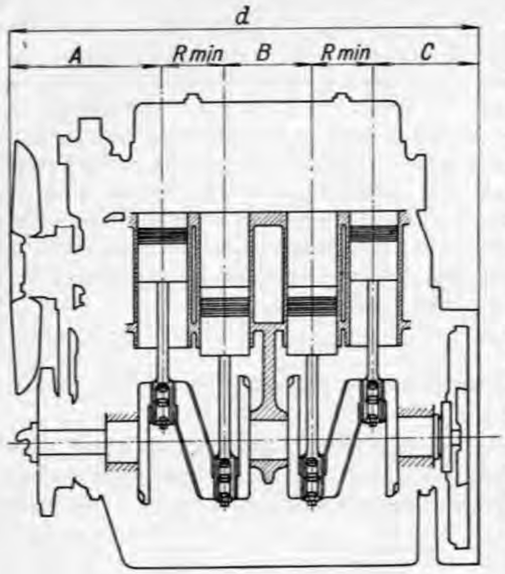


FIG. 151. Longitudinal section of a four-cylinder engine with a three-bearing crankshaft.

Volumetric efficiency of short-stroke engines is good, as with a given c_m it varies with the piston area: port area ratio, which is favourable in this case, since a large diameter cylinder head can house large valves.

Fig. 150 shows the cross sections of two 4-cylinder in-line engines, each having a displacement of 1·4 litres, one with a stroke: bore ratio of $\xi = 1{\cdot}4$, the other 0·8. Taking the latter as a basis for comparison, the overall height of the former is 111% and the width 107%.

Fig. 151 shows a simplified drawing of a four-cylinder engine with a three-bearing crankshaft. It is apparent that an increase in bore D increases only the cylinder spacing $R_{\min}$, the overall increase in length being

$$2x\,(D_1 - D_0),$$

where $R_{\min} = x\,D$ = minimum cylinder spacing

D_0 = bore of the original engine

D_1 = bore of the derived engine.

Overall engine length is therefore reduced from 100% to 94% by increasing the stroke: bore ratio from 0·8 to 1·4.

Linear dimensions A, B, C are given by design and are not directly affected by variations of bore. Indirectly, dimensions A, B, C may be influenced by bearing pressure which chiefly depends on mean piston velocity.

With double-row V engines the increase of the engine length is limited to the increase in length of a single row. But as the cylinder spacing of a V engine is generally set by the crankshaft, being larger than $R_{\min}$, the variation of bore usually does not affect overall length.

Fig. 152 illustrates the influence of the stroke: bore ratio on crankshaft design. In order to reduce engine weight, the removal of metal (bore A) in the

crankpins is worthwhile. Crankshaft weight is reduced by the amount bored out and the counterweight needed to balance the crankpin is correspondingly reduced. Assuming that crankpin diameter varies with bore and not with stroke, then in engines having a shorter stroke than bore (over-square

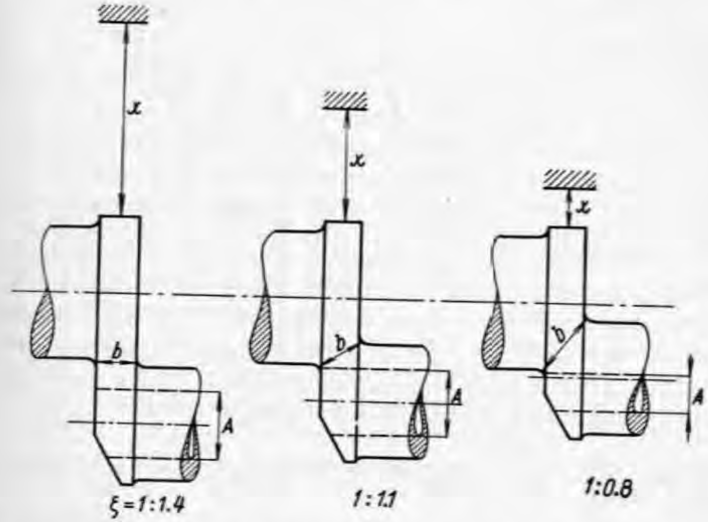


FIG. 152. Shape of crankshaft for engines with stroke: bore ratios 1 : 1.4, 1 : 1.1, and 1 : 0.8.

A — diameter of the crankpin relief bore, b — minimum width of web, x — distance of piston bottom from crankshaft.

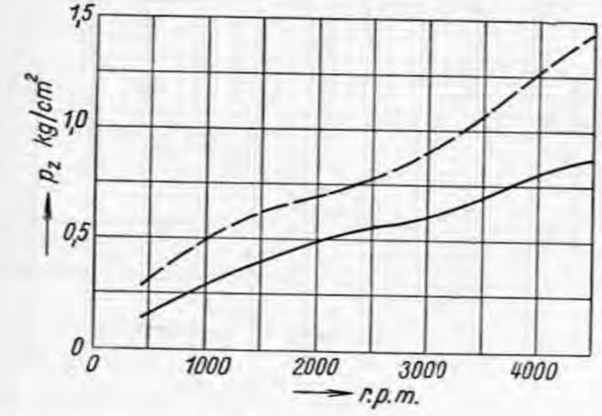


FIG. 153. Mean pressure loss in kg per sq. cm of piston area plotted against engine speed. Dash line — Studebaker engine 1949, full line — 1951 engine.

Table 26

Year of production	Studebaker		Chrysler		
	1949	1951	1950	1951	1954
Number of cylinders	6 I	8 V	8 I	8 V	8 V
Bore, mm	84·1	85·73	82·5	96·8	96·8
Stroke, mm	120·7	82·55	123·8	92·1	92·1
Displacement, cm³	4,020	3,800	5,300	5,425	5,425
Performance, b. h. p.	100	120	135	180	238
Speed, rev/min	3,600	4,000	3,200	4,000	4,400
Perf. per litre, b.h.p./l	25	31·6	25·5	33·4	43·8
p_e, kg/cm²	6·25	7·10	7·2	7·5	8·95
c_m, m/s	14·5	11·0	13·2	12·3	13·5
Stroke/bore ratio	1·43	0·96	1·5	0·95	0·95

I = in-line arrangement of cylinders, V = two banks in V.

engines) the size of the crankpin internal bore is limited by considerations of design.

The minimum web thickness B on the other hand, is increased in short-stroke engines by crankpin overlap. The crankshaft of a short-stroke engine is therefore extremely rigid and thus highly suited for modern high-compression engines.

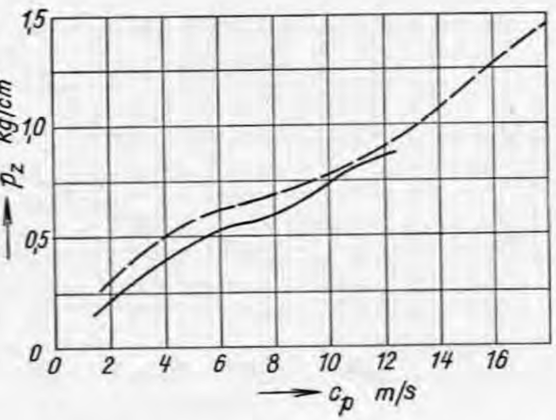


Fig. 154. Mean pressure loss in kg per sq. cm plotted against mean piston velocity. Dash line — Studebaker engine 1949, full line — 1951 engine.

Fig. 153 illustrates the readings obtained on two Studebaker engines [70]. The older, a six-cylinder of 4 litres displacement was replaced by a 3·8 litre eight-cylinder engine. The new short-stroke engine with smaller cylinders has a lower mean piston velocity despite higher crankshaft speed and losses through friction are therefore considerably reduced. Both these engines are compared in Table 26.

Fig. 154 compares the friction losses of both engines at the same piston velocity c_p. The losses were measured by turning the engines with cylinder heads removed. With gas pressure acting on the pistons the frictional losses of the short-stroke engine with its greater bore would be larger than shown.

The connecting rods of the short-stroke engine are considerably shorter and thereby lighter, despite the greater force acting on them (because of the larger bore). Piston weight also does not increase as the cube of D, since for considerations of design, piston length must be reduced and part of the piston skirt must be cut away to provide room for the counterweight.

For this reason the pistons of a modern short-stroke engine are slipper shaped, as shown in Fig. 322. With a stroke: bore ratio below 0·9 the piston and counterweight overlap to such an extent that connecting rod length must be increased above 3·75 r. The advantages of a short-stroke become doubtful at this stage.

The actual weights of the crank mechanism of two Cadillac engines of approximately equal displacement are given in Table 27. This serves to illustrate how the weight of reciprocating parts of the crank mechanism is greatly reduced in a short-stroke engine, whereas piston weight shows only a slight increase.

Table 27

Weights of Sub-Assemblies of two Cadillac V8 Engines

Year of manufacture	1948	1949
Bore, mm	88·9	96.8
Stroke, mm	114·3	92.1
Cylinder arrangement and valve gear	V8 — SV	V8 — OHV
Displacement, cm³	5,675	5,425
b.h.p./r.p.m.	124/3,200	133/3,500
Dry weight of engine, kg	402	318
Wet weight with radiator and cooling water, kg	450	350
Connecting rod with pistons, kg	14·2	12·2
Connecting rod, complete, kg	8·45	6·05
Piston with gudgeon pin, kg	5·75	6·15
Crankshaft, kg	41·9	27·9

The example given in Fig. 155 is still more instructive. It concerns the mechanical efficiency of two eight cylinder Chrysler engines [15]. The older engine was fitted with side valves, the other with overhead valves. Notwithstanding the fact that both these engines have eight cylinders and that the

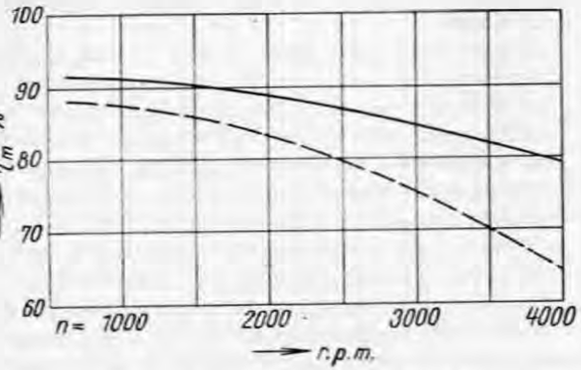


FIG. 155. Mechanical efficiency plotted against speed. Dash line — Chrysler engine made in 1950, full line — made in 1951. Dimensions of both engines are given in Table 26.

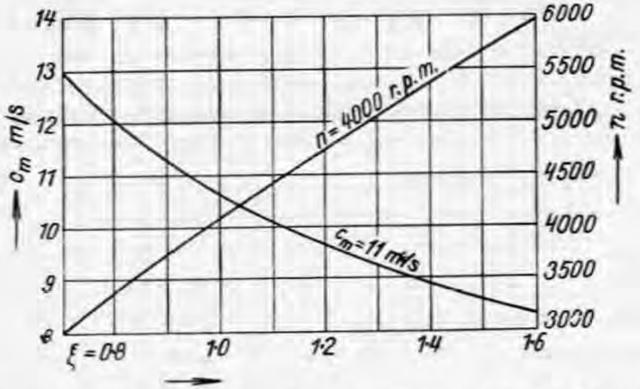


FIG. 156. Dependence of mean piston velocity c_m on the stroke: bore ratio at a constant speed of 4,000 rev/min and dependence of engine speed on the stroke: bore ratio at a constant mean piston velocity of $c_m = 11$ m/s.

second has a somewhat increased displacement, the mechanical efficiency of the new engine is above that of the old despite the higher crankshaft speed employed. The deciding factors in this case are the increased thermal and volumetric efficiencies and the choice of a short-stroke engine layout, keeping the mean piston velocity below the original value despite the higher speed.

Under the heading of mechanical efficiency, pumping losses are included; they are lower in the case of the overhead valve engine with hemispherical combustion chambers than in the case of the side-valve engine.

The short-stroke engine attains a higher crankshaft speed with reasonable mechanical efficiency. High crankshaft speed makes for a high performance per litre and for a good weight: power ratio.

Fig. 156 shows the variation of speed n with the stroke: bore ratio ξ with $c_m = 11$ m/s constant and conversely the variation of c_m with ξ and constant $n = 4000$ rev/min and the same displacement (in this case 350 cm³). It follows from the diagram that, for instance, with the change of the stroke: bore ratio of a 1·4 litre four-cylinder engine from 1·4 to 0·9 crankshaft speed may be increased from 3450 to 4650 rev/min without affecting the mean piston velocity. But by raising crankshaft speed only to 4200 rev/min c_m is reduced to 10 m/s and an improvement of mechanical efficiency can therefore be expected.

The influence of the stroke: bore ratio on cylinder surface area and heat loss is of some interest. For the sake of simplicity the combustion chamber will be considered to be cylindrical in shape and of the same diameter as the cylinder bore. Calculation will be based on a cylinder having a displacement of 350 cm³ and the variations in surface area of the combustion chamber itself and of the whole inside surface area will be observed. The results are given in Table 28. The basis for comparison (100%) is provided by a cylinder with a stroke: bore ratio of $\xi = 0·8$. It is apparent that with an increase in stroke the surface area of the combustion chamber is reduced, whereas cylinder surface area is increased.

Table 28

Effect of Stroke/Bore Ratio ξ Upon Change of Dimensions and Internal Surface of a 350 cm³ Cylinder

ξ	D mm	L mm	Surface area of comb. chamber		Surface area of cylinder		Total surface area	
			cm²	%	cm²	%	cm²	%
0.7	86·025	60·218	90·736	104	162·743	95·1	253·479	98·5
0.8	82·280	65·824	87·290	100	170·150	100	257·440	100·0
0.9	79·112	71·201	84·453	97·0	176·962	104	261·415	101·5
1.0	76·382	76·382	82·300	94·3	183·289	107·8	265·589	103·2
1.1	73·994	81·393	80·669	93·0	189·205	111·0	269·874	104·8
1.2	71·878	86·255	79·530	91·1	194·773	114·7	274·303	106·9
1.3	69·986	90·982	78·476	89·9	200·040	117·8	278·525	108·2
1.4	68·278	95·590	77·581	89·0	205·049	120·5	282·624	110·1
1.5	66·726	100·09	76·934	88·1	209·813	123·0	286·747	111·6
1.6	65·306	104·49	76·371	87·5	214·376	126·0	290·747	113·0

Heat losses to combustion chamber walls will therefore vary inversely as the stroke: bore ratio.

Heat loss to cylinder walls, on the other hand, becomes greater with an increased stroke: bore ratio, as this also increases the cylinder wall area. The cylinder wall however is screened by the piston skirt during piston travel and the uncovered part is contacted by exhaust gas which has already lost some heat. Experiments with a supercharged water-cooled aircraft engine have shown the coefficient of heat transfer to combustion chamber walls to be 400 000 kcal/m^2 h and to the cylinder wall surface only 125 000 kcal/m^2 h.

With an automobile engine (as opposed to an aircraft engine), working at a lower load, the coefficient of heat transfer to the cylinder head may be taken to be 300 000 kcal/m^2 h and that of heat transfer to the cylinder wall 100 000 kcal/m^2 h [22] [42]. Total heat losses are compared in Table 29. It will be noted that they are approximately equal in both instances, but since the short-stroke engine gives a higher performance per litre owing to higher speed, its b.h.p. is higher. The heat carried off by the cooling medium expressed as a percentage of total heat loss is therefore smaller for a fast running short-stroke engine, assuming the cooling area to be equal. This would seem to endow the short-stroke engine with the double advantage of a smaller radiator and a smaller portion of engine power used for operating the cooling system. In practice, conditions may be even more favourable than assumed as shown by actual engine measurement in Table 30.

Table 29

Stroke: Bore ratio ξ =	0.8	1.4
Surface area of combustion chamber, cm^2	87·29	77·58
Surface area of cylinder, cm^2	170·15	205·04
Heat from comb. chamber, kcal/h	2620	2320
Heat removed by cylinder, kcal/h	1701	2050
Total heat removed, kcal/h	4321	4370

Table 30

Chrysler engine made in	1951	1950
Number of cylinders	8V	8 in-line
Displacement, cm^3	5425	5300
Heat removed by cooling, kcal/min	1385	1510

5. The number and arrangement of cylinders

Multi-cylinder engines are desirable because of small engine weight; their torque being more even, they may be designed with smaller and lighter flywheels. An excessively large number of cylinders, however, makes the engine too costly and complicates operation. The driver of a vehicle fitted with an engine of more than eight cylinders needs considerable experience to be able to detect and locate a cylinder which is not firing. Engines of more than eight cylinders are therefore seldom used for passenger cars.

When forced by circumstances to produce a high-output engine having a low weight, as is the case with aircraft or armoured vehicle engines, designers resort to using engines with a large number of cylinders. 16, 18 and even 24 cylinder engines have proved reliable under operating conditions and engines having as many as 28 cylinders are used in aircraft, while others with a still greater number are undergoing tests. A sixteen-cylinder engine is not unusual for tanks and twelve-cylinder prime movers are used in railway practice.

Long development has shown a 160 mm bore to be the maximum for aircraft engines, if cooling is to remain adequate. The overall engine output therefore depends on the number of cylinders which can be grouped into a serviceable unit.

With the reduction of engine weight in view, the arrangement with a common crankpin for the largest possible number of cylinders is most desirable. This is best met by radial engines with their short and rigid build and small size of crankcase per cylinder. With air-cooled engines this arrangement has the further advantage of individual well separated cylinders making possible the use of extensive finning.

The radial arrangement offers fewer advantages for automobile use. Such an engine is rather high, cannot easily be accommodated under the bonnet and accessibility of some cylinders is impaired. The inlet manifold of radial engines presents difficulties, which are absent in supercharged aircraft engines. But owing to the small weight of the radial engine and the ease of cooling, the possibility of developing a good automobile engine of this type should not be ruled out. In aircraft engine practice several radial rows (up to four) of cylinders are arranged in line making a compact and powerful engine.

The cylinders or banks of multi-cylinder engines (e.g. 16-cylinder tank engines) are often arranged in X form or with one set of horizontally opposed banks superimposed on another. These types are low and easily accessible. The latter have two crankshafts. Twin V (or W) type engines are rare, although they are low and accessible.

For use in vehicles the largest V engines are eight- or twelve-cylinder types. This layout is convenient as all engine parts are accessible and the engine is light and short. The crankshaft is simple and well balanced. Eight cylinder V engines are progressively gaining more and more adherents.

The V arrangement is very suitable for air-cooled engines as, owing to considerations of crankshaft housing, the cylinder spacing is large enough for the required size of cooling fins to be used. There is therefore no difference in the lengths of water- or air-cooled V engines if plain bearings are used and if pairs of connecting rods share common crank pins. The induction manifold presents no problems and cooling air may be directed where it is needed without difficulty.

Opposed cylinder engines retain the first mentioned advantage but the induction manifold, if only one carburettor is to be used, is long and it is more difficult to direct cooling air flow. Despite this difficulty horizontally opposed engines are better suited for vehicles than in-line types and are very popular in conjunction with air-cooling.

In-line petrol engines are the most numerous because they are the simplest from the production point of view. Accessibility is good and the inlet manifold simple. The in-line arrangement is used in four- or six-cylinder engines, eight-cylinder in-line engines being too long and, with present day high running speeds, presenting great difficulties caused by torsional vibration of the crankshaft. There are no recent types of in-line eight-cylinder engines in production.

An air-cooled in-line four-cylinder engine is too long, the same applying, but to a greater degree, in the case of a six-cylinder engine.

Motor cycle and stationary engines have a smaller number of cylinders. Twin cylinder engines are arranged as in-line parallel twins, V engines or horizontally opposed (flat twins). Air cooling for twin cylinder engines offers every advantage, whereas water cooled twins and flat-twins in particular, are too complicated.

The opposed twin is unusually well balanced, for which reason it is also used to propel small passenger cars. The size of cooling fins is not limited by any considerations, but the induction manifold is too long and two carburettors may therefore be used with advantage. A short stroke is desirable.

The parallel twin has been popular with motor cycle designers in recent years. Its balance is poor, but firing intervals are regular. The V engine strikes a compromise between these two types. The smallest type of unit uses only a single cylinder, such an engine being simple and cheap to produce. Most engines under 250 cm^3 displacement are singles.

As the positioning of the cylinders of an air-cooled engine is influenced in no way by water jackets, unions or radiators, engine design may be adapted to suit the vehicle layout. Some interesting examples of this are shown in Fig. 157.

6. Balancing of engines

Engine balance of the most common multi-cylinder four-stroke air-cooled engines is dealt with in this section.

Twin cylinder engines are usually designed as opposed twins. (Tatra 12 Fig. 158, VW Fig. 212, Citroen 2 CV Fig. 372, Panhard Dyna Fig. 372, etc.)

The revolving masses are balanced by crankshaft counterweights. Primary and secondary couples remain unbalanced, but owing to the minute offset of opposing cylinder axes (comprising the centre web width and the width of one connecting rod), they are small.

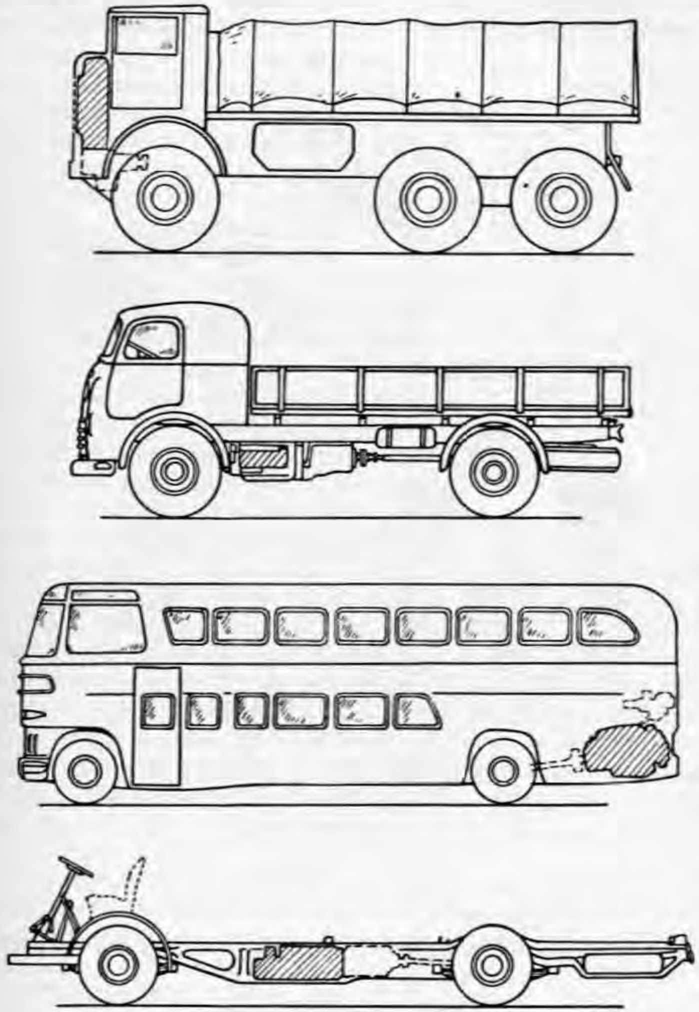


FIG. 157. Characteristic vehicle layouts.

The GM-T 51 truck for off-the-road operation with a vertically mounted Continental AO-536-1 engine. Second – The Tatra 116 lorry with a flat six-cylinder engine mounted underneath the floor. Third — Greyhound bus with two six-cylinder engines installed in the rear. Bottom — The Chassis of a Movag bus with a flat SLM eight-cylinder mounted underneath the floor.

In designing the crankshaft it has to be borne in mind that any saving in crankpin weight will also reduce the weight of the counterweights by the same amount, and, therefore, will be doubly valuable in terms of overall weight reduction.

With a short-stroke engine, weight reduction of the crankpin may be effected only to a limited degree. Under such circumstances a cast crankshaft

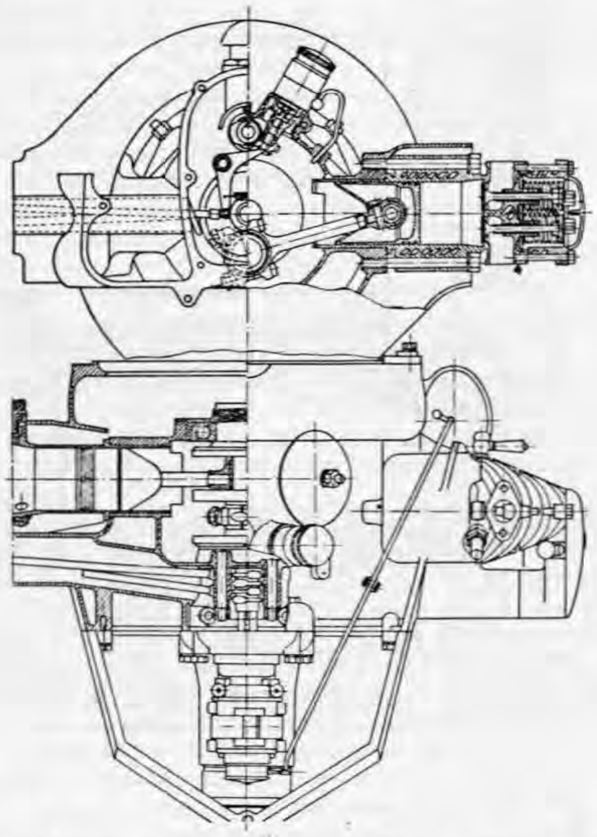


FIG. 158. Sectional view of the Tatra 12 (82×100 mm) twin cylinder.

is convenient as weight may be removed in the best manner (see Fig. 159). The parallel twin cylinder engine is as imperfectly balanced as a single cylinder engine since both pistons travel up and down simultaneously. The total revolving masses are balanced together with about 50 to 70 % of the reciprocating masses (depending on the cylinder position and engine mount-

ing). An example of a parallel twin Sunbeam motor-cycle engine is given in Fig. 382. With larger high speed engines there is a risk of transverse vibrations of the crankshaft and a third bearing is therefore desirable. The parallel twin arrangement permits the design of a compact engine with no difficulties with the inlet manifold and lubrication system.

The in-line four-cylinder engine in its accepted form is rather long when air cooling is applied. It is used quite often, especially with compression-ignition engines where overall length is not a primary consideration. For air-cooled petrol engines, the flat-four arrangement is more popular.

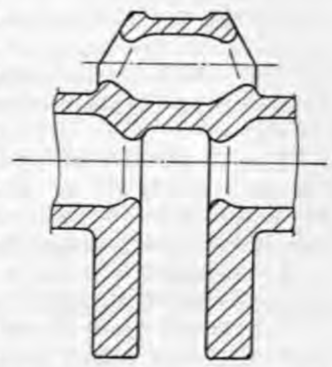


FIG. 159. Relieved casting of a crankpin.

A flat-four engine is short, rigid and well balanced. Its chief disadvantage lies in the long induction manifold, if only one carburettor is employed, and in its great width. In order to provide easy access to the cylinder heads, the demands on engine compartment width are further increased.

A four-cylinder four-stroke 90° V engine can be balanced, but its firing intervals are very irregular. Despite this, V-four engines of such a layout are being produced.

The six-cylinder air-cooled in-line engine is extremely long, but taking into account the space saved by the absence of a radiator, its overall space requirements in a motor vehicle do not exceed those of a similar water-cooled engine. A six-cylinder V engine is much shorter, but primary and secondary reciprocating mass couples cannot be fully balanced. It is used on account of its compact construction in conjunction with air cooling. (Tatra 926, Fig. 414.)

The flat six-cylinder engine with a six-throw crankshaft is well balanced, short and low. It is particularly suited for underfloor mounting. On a petrol engine, however, the induction manifold presents problems.

For eight-cylinder engines the in-line arrangement does not come into consideration, possible layout being confined to the V or horizontally opposed types.

A 90° V-eight cylinder engine with a four-throw crankshaft with throws 90° apart, can be well balanced by crankshaft counter-weights. With any other angle than 90° a symmetrically arranged crankshaft (with throws 180° apart) in a single plane is more suitable. There is no need for counterweights with this type of crankshaft, the engine is lighter, but unbalanced secondary couples remain, although they are relatively small and cause no disturbance (Tatra 928, Fig. 413, etc.).

The case of the opposed eight cylinder engine proves interesting. With the choice of a symmetric four-throw single plane crankshaft the engine is well balanced and cylinder spacings are small, all this at the expense of two cylinders always firing concurrently. In order to arrange the firing intervals regularly and 90° apart, an eight-throw crankshaft must be used with a resulting increase of cylinder spacings (the minimum possible increase being that of crank web thickness).

In another variation of the opposed eight-cylinder engine, the crankshaft has four throws at 90° intervals. In this case firing intervals are regular but reciprocating primary couples remain unbalanced. As an example of this arrangement the SLM engine may be quoted.

Twelve cylinder engines may be designed either as V types (Tatra 111, Fig. 407) or with opposed banks of cylinders. Balancing is satisfactory in both types. Engines with the number of cylinders exceeding 12 are not used in road vehicles.

Up to 16 cylinders are used in prime movers for armoured vehicles, the Simmering and MAN engines being two examples. The first was about to go into large scale production at the end of World War II. It is an X engine with four banks of four cylinders each disposed at 45° and 135°.

The crankshaft is of four-throw design with throws 90° apart and one master connecting rod with three connecting rods acting on each crankpin.

The cylinders fire at regular intervals but crankshaft counterweights have to be employed to balance couples. Primary couples cannot be fully balanced.

The MAN design is an H type with two crankshafts. From the point of view of balancing this engine may be regarded as two opposed eight-cylinder engines with a common crankcase.

Both these sixteen-cylinder engines combine short, low and light build with relatively good accessibility. Beside the arrangements described, a great number of others exist, such as in-line engines with an odd number of cylinders, radial engines etc. In order to be of the greatest possible use, an engine must be fully adapted for the type of vehicle in which it is to be installed. Some examples of air-cooled engines designed for a particular vehicle type are shown in Fig. 157.

Common types of automobile engines have been treated so far. Aero-engines, from which considerably larger performances are required, are built with a greater number of cylinders.

Air-cooled aircraft power units are in the majority of cases arranged as radial engines. It is feasible to balance this type fully even when the number of cylinders is small. As all cylinders are in one plane, primary and secondary couples are absent and primary and secondary forces can be balanced by crankshaft counterweights. Balancing is calculated on the assumption that all connecting rods revolve round the crankpin centre. With one master rod and other connecting rods acting upon it the conditions of balancing are somewhat different.

By grouping balanced radial engines in tandem, an engine with a multiple number of cylinders is obtained, the balancing of which is also perfect. 14, 18, 28 and even 36-cylinder engines (4 × 9 cylinders) may thus be arranged, whereas a grouping of in-line engines produce 12, 24 and even 48 cylinder

units. These high-output engines are, however, today being replaced by the gas turbine.

In order to complete this section, mention must be made of the balancing of in-line engines with an odd number of cylinders. Although such an arrangement is not common in automobile practice, several five-cylinder engines exist (Lancia, Gardner etc.). Resort to such an arrangement is usually only made when necessity dictates the production of an engine to bridge the gap between a four-and a six-cylinder type and standardized engine parts must be used.

It was found, that primary and secondary couples are unbalanced in these engines. With a larger number of cylinders, these couples are small and may be disregarded. A similar arrangement can be used for two-stroke engines, balancing problems remaining the same, but the firing intervals being halved and the order of firing changed.

So far it has always been assumed that the respective weights of pistons, connecting rods etc. are equal for all cylinders. Special care has to be taken during manufacture to ensure that this is really so and the weight of reciprocating parts must be kept within close limits. If, for example, we replace one piston in a six-cylinder engine by a heavier piston, the whole engine balancing is upset. When replacing only some pistons during engine repairs, particular care has to be taken to fit only pistons of matching weight.

For engine balancing to be perfect, all moving masses must be taken into account. The flywheel, which is the heaviest moving part, must be balanced and attention must be paid to all gear wheel teeth, etc.. In very highly stressed engines, even camshafts must be balanced, as the cams are not symmetrically disposed and produce unbalanced couples.

7. The influence of compression ratio on engine economy

Economy of operation is one of the foremost requirements with modern engines. With a petrol engine, running economy is chiefly dependent on the fuel: air ratio and the compression ratio.

The influence of the fuel: air ratio upon m.e.p. and upon thermal efficiency is shown by Fig. 160 which makes it clear that the use of a higher percentage of air than in the ideal fuel-air mixture leads to an improving thermal efficiency

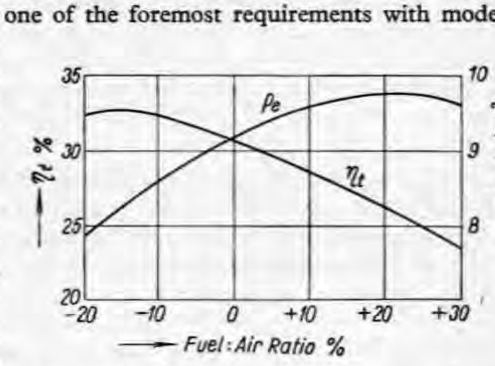


FIG. 160. Influence of fuel: air ratio upon thermal efficiency and mean effective pressure.

of the engine. But as a leaner mixture burns more slowly, spark advance has to be increased in order to provide optimum conditions. With correct spark advance, the highest thermal efficiency of an engine is attained with about 15% of air surplus. With thermal efficiency at its peak, specific consumption is lowest and running economy will be at its best.

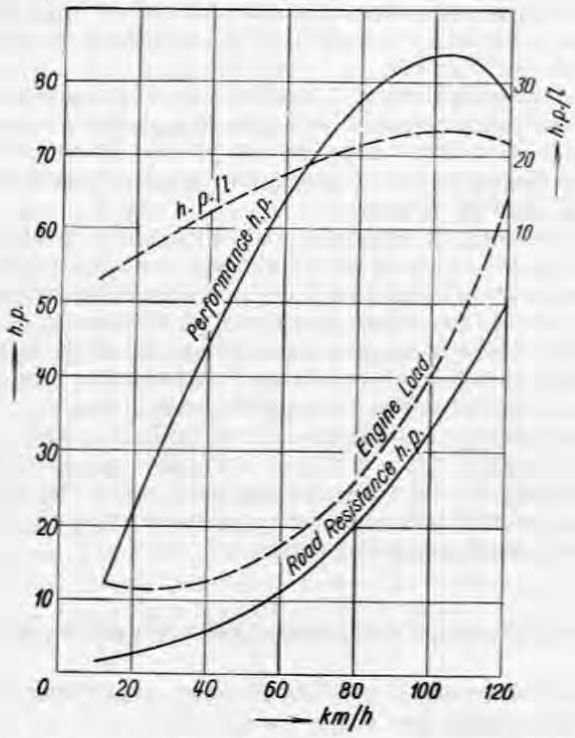


FIG. 161. Engine performance required for travel on level roads, road resistance and percentage engine load.

By observing the influence of the fuel: air ratio on mean effective pressure and therefore on engine performance it will be noted that the latter is reduced with a lean mixture. This is in direct opposition to the requirements of maximum performance with minimum engine weight. Maximum output is reached with a rich mixture having about 20% excess fuel.

In order to attain a high standard of running economy, the engine load under operating conditions must be observed in detail. Fig. 161 shows the maximum engine performance at various road speeds and the output required to propel the vehicle on a level road. It is apparent that with the vehicle in

uniform motion on a level road and at speeds which are usual today, the engine load is relatively low. Extra performance is required for overcoming gradients and for acceleration, but for most of the time the engine is operating under only partial load. This condition is of great interest to the engineer whose aim it is to produce an economical engine. The need for a lean mixture supply to the engine under part-load conditions is evident.

The carburettor of a modern petrol engine must be set to supply a lean mixture on small throttle openings and a rich mixture for a full throttle opening when maximum power is to be provided by the engine. This is effected in two ways in modern carburettors, either by thottling the flow of fuel to the main jet (Solex) or by bleeding air into the fuel jet by a separate valve (Zenith). The fact that an engine may overheat on full throttle with a lean mixture is recognised as is the possibility of a breakdown caused by excessive heat. With part throttle running the amount of heat produced by combustion is proportional to engine output and therefore small. Engine cooling is then efficient despite the lean mixture used and there is no danger of overheating.

In order to provide optimum operating conditions for the engine with both rich and lean mixtures, it is necessary to vary ignition timing with mixture strength, independently of engine speed. This cannot be done by using the common type of centrifugal ignition advance control which accounts only for engine speed.

With the throttle fully open a rich mixture is required, and for a partial throttle opening the mixture must be lean. The vacuum in the inlet manifold being dependent on throttle opening (at constant engine speed), it may be utilized to modify the setting of the centrifugal governor. A vacuum controlled ignition advance regulator unit is shown in Fig. 162. With the throttle fully open, pressure in the inlet manifold rises and the ignition setting is correct for maximum output. With the throttle partly closed, pressure in the inlet manifold is reduced and draws in the diaphragm which overcomes the force of a spring and rotates the whole distributor head against the direction of distributor arm rotation, thereby increasing ignition advance. A greater degree of advance compensates for the slower rate of burning and maximum gas pressure inside the cylinder is again attained at the most convenient moment, i.e. 10° after T.D.C.

The compression ratio should be chosen as high as circumstances permit, for thermal efficiency depends on it to a large extent. The upper limit for the compression ratio used is given by the danger of detonation occuring. If it is intended to use a fuel with a given octane rating, the upper limit of the compression ratio is given by the geometry of the combustion chamber. In conjunction with this, the "mechanical octane number" of an engine is sometimes quoted.

If, by altering the shape of the combustion chamber or by the use of other means, the compression ratio is raised above the original limit without detonation occuring, the engine octane number has also been raised.

A high compression ratio is desirable for running economy under partial load conditions. The shape of the combustion chamber should therefore be chosen with the object of making the highest compression ratio possible. As an automobile engine is mostly working only under partial load, the compression ratio is also chosen for partial load conditions. With partial load operation the entry of air into cylinders is throttled, whereby the real compression ratio, pressure or temperature at the end of the compression stroke are reduced. With part throttle conditions prevailing, the compression ratio

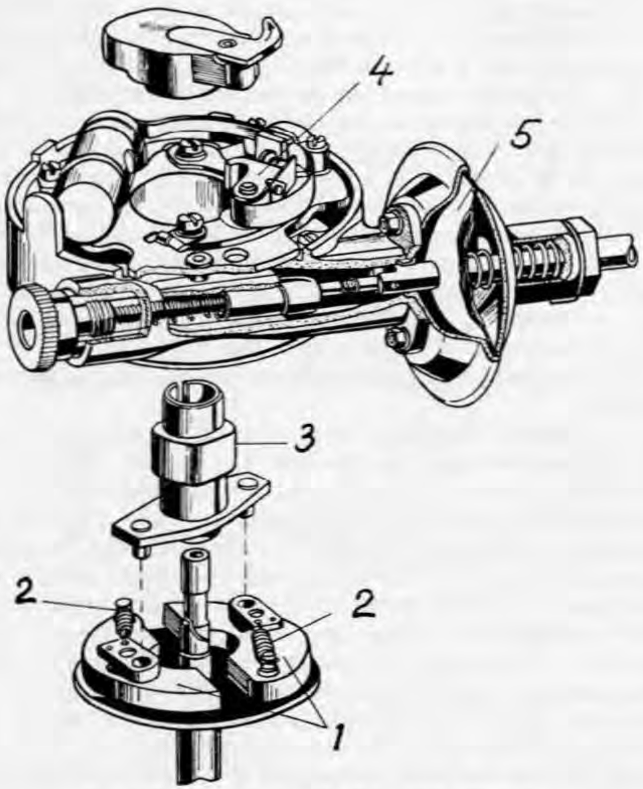


FIG. 162. Vacuum control of ignition advance.
1 — centrifugal weights, *2* — governor springs, *3* — distributor cam, *4* — moving contact, *5* — vacuum control.

may therefore be raised without causing detonation. At full throttle, measures have to be taken to prevent detonation. This may be done in the following ways:

1. enriching the mixture,
2. internal cooling,
3. raising the fuel octane number,
4. altering ignition advance.

The tendency to detonate may be radically curbed by enriching the mixture. Fig. 163 shows the dependence of the maximum useful compression ratio on the point of detonation in a laboratory engine upon the fuel: air mixture ratio. It will be seen that the resistance to detonation of a fuel rises rapidly with a richer mixture. With a lean mixture the resistance to detonation is increased

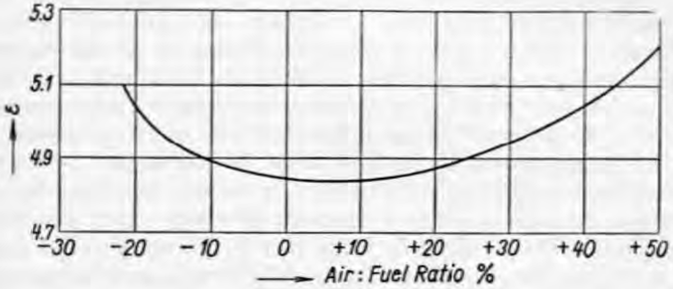


FIG. 163. Effect of air: fuel ratio upon the maximum admissible compression ratio.

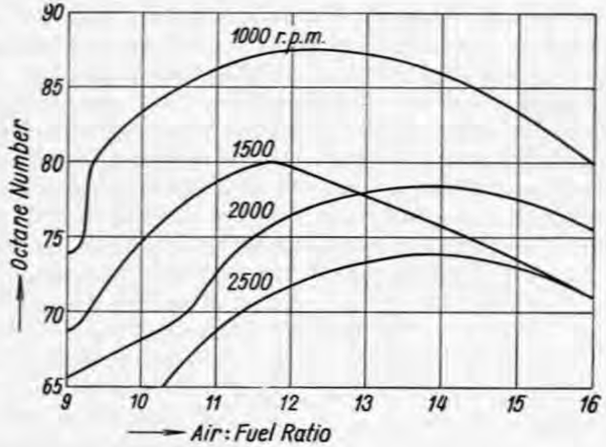


FIG. 164. Effect of air: fuel ratio upon desired octane number in a single-cylinder experimental OHV engine.

by the slower rate of combustion, but this is of no use when maximum power is aimed at.

Efforts have been made to exploit this phenomenon by supplying low grade fuel to the engine at uniform road speed and high grade fuel whenever the need of acceleration or of overcoming gradients called for high performance. These experiments did, however, not meet with success.

The fuel: air mixture ratio influences the velocity of flame travel and combustion, internal cooling etc. and therefore it also influences the octane requirement of an engine. Fig. 164 shows the variation of the octane requirement of a single cylinder overhead valve engine with engine speed. By setting the mixture extremely rich at more than 1 : 11, the beneficial effects of internal cooling through excess fuel evaporation begin to be apparent and the octane number requirement shows a reduction. Use of this effect of an over-rich mixture is made, for instance, for take-off boost in aircraft engines.

Internal cooling reduces temperature at the end of the compression stroke, thereby reducing the tendency to detonate. Enriching the mixture is a means of internal cooling, since the temperature at the end of the compression stroke is lowered by heat loss through evaporation. Fuel being expensive, water is sometimes used to perform this function. In aircraft engines, water may be injected into the inlet manifold for take-off conditions. It evaporates during the compression stroke, using up much heat in the process. The prolonged use of water injection has, however, proved harmful, especially to sparking plugs.

For the maximum engine output to be reached, a mixture about 20 % rich has been found most satisfactory (Fig. 160), and further enriching does not increase resistance to detonation. Increasing the fuel octane number for full throttle conditions is a difficult proposition and is practised only on aircraft engines which are supplied with special high-octane fuel for take-off.

An instrument called the "Vitameter" has been tested on automobile engines. This is basically a simple carburettor with a single jet, float chamber and a valve, which is actuated by inlet manifold vacuum acting on a diaphragm. The instrument was interposed between carburettor and inlet manifold and set to supply high-octane fuel whenever inlet manifold pressure rose owing to full throttle opening, thus preventing detonation. Since the engine works under full load only for a fraction of the total running time, auxiliary fuel consumption is low. The fuel consists of 85% alcohol and 15% water and is effective in suppressing detonation.

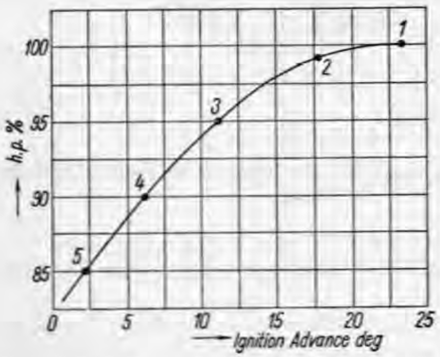


FIG. 165. Influence of octane number and ignition advance upon the detonation point.

Altering the ignition setting is another way of combating detonation. Not even with fuel of a low octane number will detonation occur if the ignition setting is retarded. The results obtained by laboratory test on a single-cylinder engine with a compression ratio of 7·25 : 1, running at 1000 rev/min on full load are shown in Fig. 165. For 100% output to be reached, the engine must be run on 98 octane fuel and the ignition setting must be 23° advance. If fuel of only 93 octane rating is used, 11° is the optimum ignition advance and the output will be 95% (point 3). At point 5 only 86 octane rating is required. Low grade fuel may therefore be used at the expense of output. This requirement can be met when plotting the ignition advance curve for the governor. Failing this, the basic ignition setting may be modified to enable the engine to be run on lower grade fuel, though naturally at the expense of reduced performance.

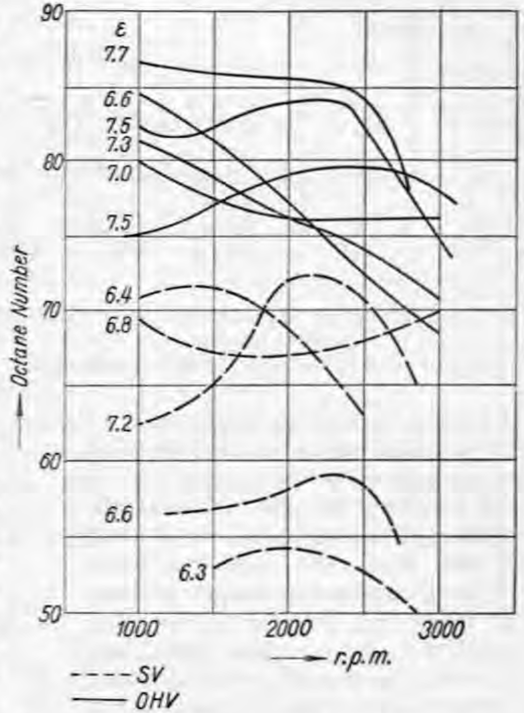


FIG. 166. Required octane numbers of several typical SV and OHV engines plotted against engine speed. Compression ratio of the engine is given at each curve.

The engine octane requirement differs with differing load conditions. Put in another way, this means that under all conditions the engine should ideally be run on such a mixture of the basic fuels, heptane and iso-octane, which will keep it just short of detonating. The compression ratio remains unaltered during the test. Another method uses a variable ignition setting to keep the engine just short of detonation occuring, while the fuel used is the same throughout. The engine octane requirement may therefore be expressed either in octane numbers or in degrees of ignition advance.

Fig. 166 shows the octane number requirement variation with engine speed [68]. The typical curves for overhead valve and side valve engines are of interest. Overhead valve engines require a high-octane number fuel at low engine speeds, whereas at higher speeds the octane number requirement decreases. Side-valve engines do not require high-octane number fuels at low speeds, their octane requirement growing with rising engine speed up to a point below maximum engine speed, after which it drops. The probable cause of the higher octane number requirement of overhead valve engines is their higher volumetric efficiency and their resultant higher mean effective pressure.

Octane requirement values are influenced to a great extent by combustion chamber geometry and by volumetric efficiency. Their curve is also influenced by variation in valve size (Fig. 167).

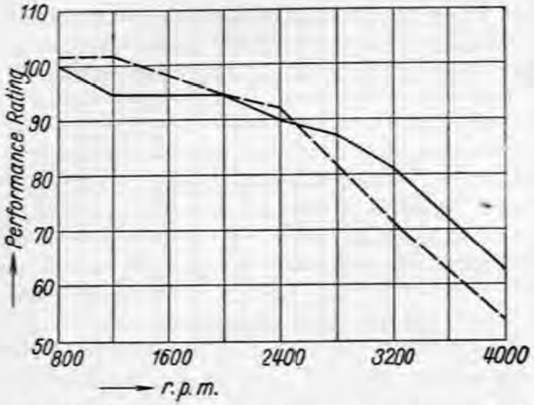


FIG. 167. Effect of varying valve size upon the desired octane number.

Full curve holds for large valves, dash line for small valves.

The effect of throttle opening on the octane requirement of an engine is also considerable. With the throttle only partly open, less air enters the cylinder and compression is low. The effect is the same as when the compression ratio of the engine is reduced.

Fuel with considerably lower detonation resistance is therefore satisfactory under partial-load conditions, as is apparent in Fig. 168, which shows the dependence of the engine octane number requirement at full throttle running (full line) and at a uniform road speed on a level surface (dashed line). It will be noted that with a full throttle opening corresponding to a road speed of 49·7 m.p.h. (80 km/h) (e.g. when climbing a gradient), the engine requires 72 octane fuel, whereas for the same speed to be held on level roads, only 14 octane fuel would be required. Every driver is aware of this, as experience has taught him to close the throttle slightly whenever the engine begins to knock, upon which detonations cease.

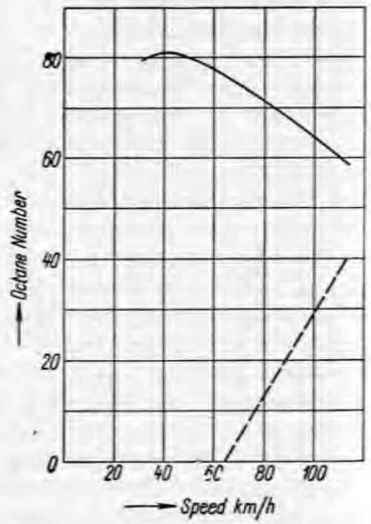


FIG. 168. Influence of engine load upon the desired octane number.

Full line holds good for full load (acceleration, up-hill travel, open throttle), dash, line for partial loads (uniform travel on level roads).

The engine octane number requirement is naturally influenced to a major degree by the compression ratio (Fig. 169). As already stated, the highest possible compression ratio with a given fuel should be used in order to maintain high thermal efficiency and fuel economy.

The compression ratio best suited is determined in the following manner. First of all the octane number requirement of the engine in dependence on

power output and ignition advance is ascertained. The result, found by varying the ignition setting during a dynamometer test, is then entered in a graph. For each octane number, a corresponding performance and ignition setting exists for the detonation point. To quote actual figures, in this instance the corresponding ignition advance for octane number 85 is 20° (in circle) at detonation point and the engine output 99·8 b.h.p. (See Fig. 170.)

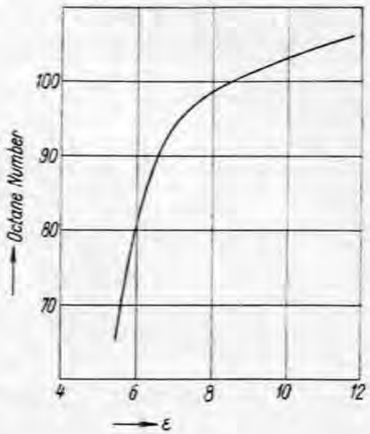


FIG. 169. Influence of compression ratio upon the desired octane number.

By increasing the compression ratio a similar curve, transferred to a higher output and octane requirement range will be obtained. This is repeated two or three times. By marking the detonation points of the fuel tested on these curves (crosses) , it will be found that a fuel behaving under compression of $\varepsilon = 7{\cdot}3$ in the same manner as a primary fuel with an 85 octane rating, will behave as if its octane number were 86 at $\varepsilon = 8{\cdot}1$ compression, and as if it were 88 at $\varepsilon = 8{\cdot}7$ compression. The same engine will have an output of 99·8 b.h.p. at the point of detonation with $\varepsilon = 7{\cdot}3$; an output of 105·5 b.h.p. with $\varepsilon = 8{\cdot}1$; and at $\varepsilon = 8{\cdot}7$ the output will drop to 103 b.h.p., which represents 99·8% of the

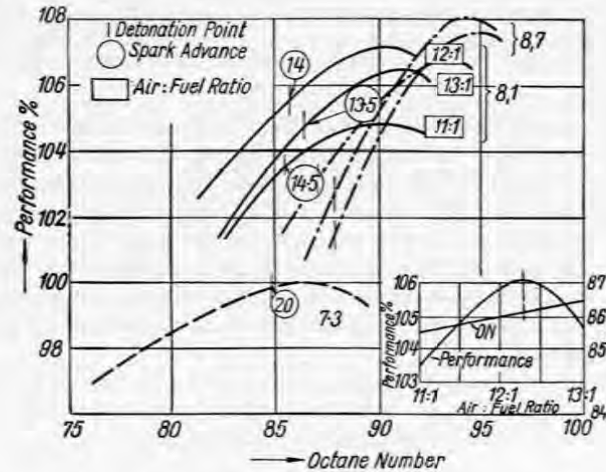


FIG. 170. Dependence of engine performance on octane number at varying compression ratio, ignition advance and fuel: air ratio.

maximum output at $\varepsilon = 7{\cdot}3$; 98% b.h.p. max. at $\varepsilon = 8{\cdot}1$ and only 94% b.h.p. max. at $\varepsilon = 8{\cdot}7$.

It will therefore be found convenient to choose $\varepsilon = 8{\cdot}1$ and to prevent detonation at maximum output by reducing ignition advance to 14° by the vacuum control. At part throttle openings the vacuum controlled regulator advances ignition, thus ensuring running economy.

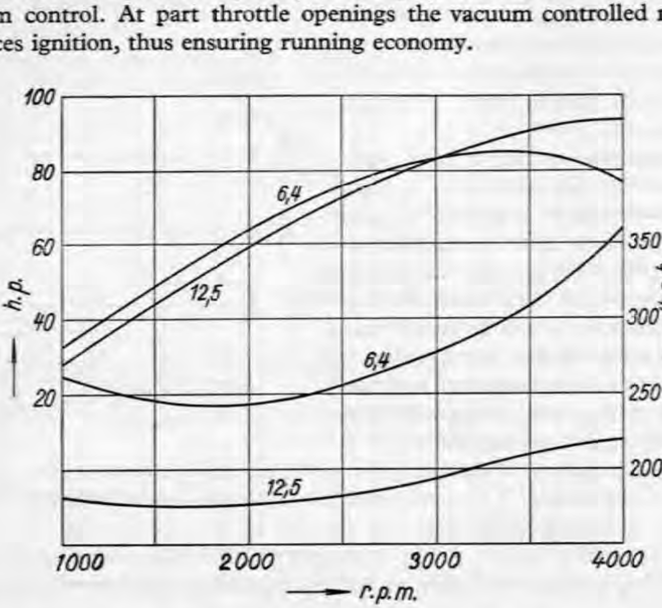


FIG. 171. Effect of compression ratio on specific fuel consumption.
Both engines represented have approximately the same output. Displacement of the first (compression ratio 12.5) is 2.96 litres, of the second (compression ratio 6.4) 3.9 litres. (Kettering).

If high octane number fuel is available, the engine must be adapted by raising the compression ratio if greatest economy is to be maintained. The development of better fuels is accompanied by great possibilities of reducing fuel consumption. Fig. 171 compares a series engine having a compression ratio of 6·4 : 1 with an experimental engine of the same output with a compression ratio of 12·5 : 1. The difference in fuel consumption is remarkable. Although a large increase in the octane number of fuels cannot be reckoned with, new engines should be designed with the possibility of a compression ratio rise in mind.

Octane requirement can be somewhat reduced by enriching the mixture at maximum output conditions. Temperature is also reduced in this manner and the result is particularly beneficial to air-cooled engines.

Properties of Engine Fuels Table 31

Type of fuel	H_2 content, %	C content, %	Boiling point, °C	Solidification point, °C	Calorific value, kcal/kg	Heat of evaporation, kcal/kg	Density kg/litre	Anti-knock value performance rating: without T.E.L.		with 1.06 cm³ T.E.L. per litre	
								lean mix.	rich mix.	lean mix.	rich. mix.
Pentane	16·8	83·2	36	–130	10 720	85·6	0·626	41	41	63	63
2-methyl-butane	16·8	83·2	27.8	–160	10 720	81·1	0·620	76	—	120	130
2,2-dim.-butane	16·4	83·6	50	–100	10 670	73·9	0·649	78	—	130	130
Heptane	16·1	83·9	98	–90	10 670	76·7	0·684	22	22	36	36
2,4-dimethyl-pentane	16·1	83·9	80.5	–120	10 620	71·1	0·673	62	62	95	95
Triptane	16·1	83·9	81	–25	10 620	69·5	0·690	140	200	200	300
Iso-octane	15·9	84·1	100	–108	10 620	65·0	0·692	100	100	153	153
Cyclopentane	14·4	85·6	49.5	–93	10 440	88·9	0·745	65	100>	100	160>
Benzene	7·6	92·4	80	+5.4	9 560	93·9	0·879	68	160>	68	160>
Toluene	8·8	91·2	110.5	–95	9 670	86·7	0·867	93	160>	95	160>
Isopropyl-benzene	10·1	89·9	150	–96	9 840	74·5	0·862	78	160>	93	160>
Isobutylene	14·4	85·6	–6.8	–140	10 780	93·9	0·594	—	—	—	—
Di-isobutylene	14·4	85·6	102	–92	10 500	—	0·715	64	—	85	160>
Aniline	7·6	77·4	185	–6	8 340	104·0	1·022	—	—	—	—
Monomethyl-aniline	8·5	78·5	196	+22	8 670	95·5	0·986	—	—	—	—
Tetra-ethyl lead	6·2	29·7	182	–137	—	40·6	1·653	—	—	—	—
Ethylene dibromide	2·1	12·8	132	+10	—	45·6	2·181	—	—	—	—
Methanol	12·6	37·5	64.7	–94	4 660	263·0	0·793	75	— (3)	75>	— (3)
Ethyl alcohol	13·1	52·1	78.3	–113	6 460	206	0·789	75	—	75>	—

In rich alcoholic mixtures the anti-knock value cannot be measured by the usual method (performance rating). However, they have a high anti-knock value and great resistance to self-ignition.

8. Fuels

The quality of fuel determines to a large extent the output which can be obtained from an engine. The suitability of fuels is judged neither by their calorific value nor by their specific gravity, but chiefly by their resistance to detonation.

The properties of some fuels are given in Table 31. If a fuel of high calorific value is used, the vehicle will travel a long distance per unit of fuel. The calorific value has only a small effect on engine performance, as the engine burns a fixed mixture of air and petrol and it is the calorific value of this mixture which influences engine output. For complete combustion to take place, various fuels need different amounts of air. Heptane, for example, needs 15·2 units of air for the complete combustion of 1 unit of fuel, benzole 13·2 units of air and methyl alcohol only 6·44 units. The calorific values of these fuels on the other hand, are: heptane – 10 720 kcal/kg, benzole – 9630 kcal/kg, methyl alcohol only 5340 kcal/kg.

The energy liberated by the combustion of respective fuel/air mixtures will be: heptane – 410·7 mkg/litre, benzole – 401·5 mkg/litre, methyl alcohol – 406·5 mkg/litre. The energy content per litre of chemically correct fuel/air mixture is therefore very nearly the same for all hydrocarbon fuels.

The resistance of fuels to detonation is generally expressed to-day by their octane number, giving the percentage of iso-octane in a fuel used as a basis for comparison, composed of heptane and iso-octane. The octane number is found by mixing the two fuels in such proportions as to obtain equal detonation or "anti-knock" resistance as that of the tested fuel in a laboratory test engine. Nowadays the so called "motor method" is employed, by which the fuel is tested in a single cylinder CFR test engine with a variable compression ratio. The other method used, referred to as the "research method", varies from the above in engine speed and inlet air temperature (Table 32). The research method gives a higher octane number and approximates real running conditions more closely. If various brands of fuel are tested by both methods, the octane

Table 32

Comparison of the "Motor" and "Research" Test Methods

Test Method	ASTM Motor	CFR Research
Engine speed, rev/min	900	600
Temperature of cooling water, °C	100	100
Temperature of intake mixture, °C	149	52
Spark advance	variable with compression ratio	13 deg

numbers obtained will differ widely in some cases, in others they will be practically the same. This proves that some fuels are highly sensitive to compression temperatures and engine speed, while others remain unaffected by variation of these conditions. If the difference in octane numbers gained by the two methods is less than 4, the fuel is said to be "non-sensitive" and if it exceeds 8, the fuel is said to be "sensitive". By adding the same proportion of knock inhibitor or "dope" to both fuels, the octane number of the "sensitive" fuel may be raised considerably, whereas the octane number of the "non-sensitive" fuel will be improved only slightly.

It follows that the octane number as determined by the "motor method" does not classify fuels in a very reliable manner. This led to a search for other means of classification.

Dopes

The octane number of a fuel can be increased and its qualities improved by the addition of a small amount of chemical compounds known as dopes, which raise the resistance to detonation of the said fuel.

The best known and most extensively used dope to-day is tetra-ethyl lead (TEL). The lead deposits produced by combustion of the mixture cover the surface of the combustion chamber and also tend to soil spark plugs. By the addition of ethylene dibromide the lead oxides are converted into gaseous bromides which leave the combustion chamber with the exhaust gases. A mixture of these chemical compounds is marketed under the name of "ethyl-fluid". Even when this is used, considerable deposit forms and for this reason the maximum tetra-ethyl lead addition is 0·8 cm³ per litre of fuel. Even in such small quantity, tetra-ethyl lead is a very efficient dope. It is also strongly poisonous and fuel containing it must, therefore, be handled with care.

A less known and also less extensively used dope is iron carbonyl, which deposits a reddish powder (iron oxide) inside the cylinder. This, mixed with oil, forms a grinding paste, for which reason its use is not recommended. Of the other substances raising the resistance to detonation, monomethyl aniline may be named as an example. It must be added to fuel in much larger quantities.

The most effective dope is tetra-ethyl lead, the addition of a diminutive

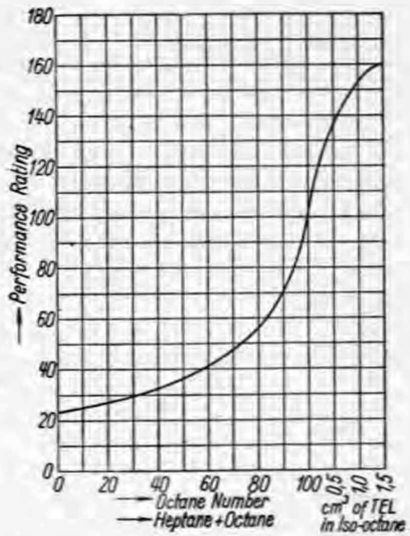


FIG. 172. Relationship between octane performance rating and contents of tetra-ethyl lead in one litre of iso-octane.

percentage of which considerably raises the octane number of a fuel. By adding tetra-ethyl lead to high grade fuel or to iso-octane, fuels with octane numbers exceeding 100 are produced. In such cases the resulting fuel cannot be compared with a mixture of iso-octane and heptane; a mixture of iso-octane and tetra-ethyl lead is used as a basis for comparison. For such comparisons, the octane number has been replaced by the "performance rating". This scale was set as a result of experiments in order to provide a means of classifying fuels of over 100 octane rating by numbers which are directly proportional to the gain in output obtained by using the fuel concerned. The "performance rating" is sometimes used also for fuels of octane numbers below 100. The relative values of octane and performance numbers are given in Fig. 172.

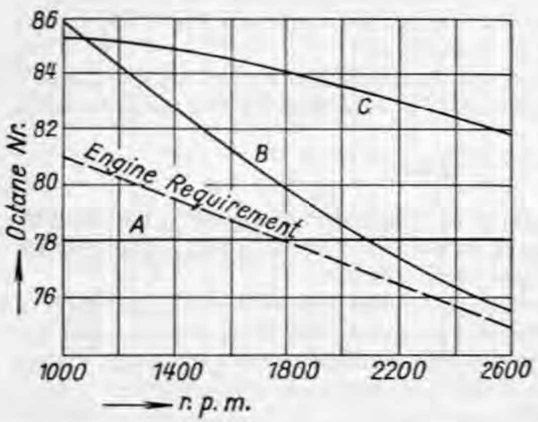


FIG. 173. Comparison of three types of fuel of the same octane number according to the Motor method. In the case of a sensitive fuel the octane numbers varies substantially with speed. Requirements of a typical OHV engine are shown by the dash line. In the case of fuel A octane number does not vary with engine speed.

The anti-detonation value of a fuel

As already stated, the methods used today to determine the octane ratings of fuels do not fully express the value of the fuel. The motor engineer's interest lies chiefly in the behaviour of the fuel in an engine. Fig. 173 shows the properties of three brands of fuel – *A, B, C,* the octane number (motor method) of all of them being 80. Two of them, *B, C,* have the same number of 90 by the research method. From their behaviour at various engine speeds it is apparent how different the properties of these fuels are as seen by the motor engineer despite their common octane number (Table 33).

Fuel A in Fig. 173 behaves as a 78 octane fuel despite its octane number of 80, because mixture temperature at the inlet manifold and the shape of the compression space are different from those in the CFR test engine.

Table 33

Properties of Fuels According to Fig. 173

Octane number according to the method	Motor	Research
Petrol A	80·0	80·0
Petrol B	80.0	90·0
Petrol C	80·0	90·0

A so called "road method" has been developed for fuel testing and this classifies fuels according to their real value. By this method the curve of detonation points can be determined at different ignition settings and varying engine speeds while driving the vehicle on the road (the automatic ignition control having been put out of action).

Fig. 174 illustrates the results obtained by such a test with a typically saturated fuel bearing a chemical resemblance to straight chain hydrocarbon paraffins. Fig. 174 together with Table 34 illustrates the variation of the octane number of this fuel with the addition of tetra-ethyl lead. The dashed line marks the engine octane requirement with normal ignition advance and correct carburettor settings. Fuel *A* would detonate up to 2900 rev/min and only at higher speeds would knock be absent. After the addition of 0·4 cm³ per litre of tetra-ethyl lead, the fuel is nearly satisfactory, the desired properties being fully gained by the addition of only 0·8 cm³ tetra-ethyl lead per litre of fuel. It follows that with the addition of tetra-ethyl lead the octane number of the fuel rises rapidly at high engine speeds, whereas improvement at lower engine speeds is much slower. This fuel is "non-sensitive" and an improvement in octane numbers with both methods does not neccessarily mean an improvement in actual road performance.

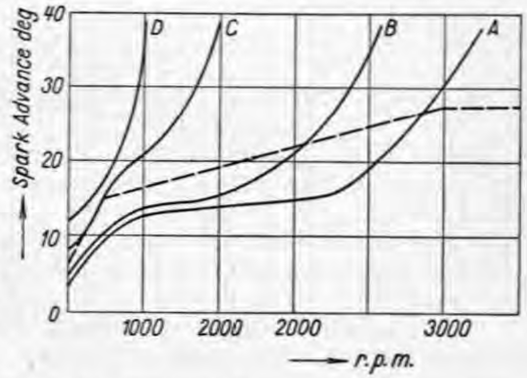


FIG. 174. Relationship between the maximum permissible spark advance at the detonation point and engine speed for a non-sensitive paraffinic fuel *A*. By the addition of tetraethyl lead according to Table 34 the required advance varies in accordance with curves *B, C* and *D*. The dash line shows the requirement on ignition advance of a normal engine, according to which ignition advance was set.

Table 34

Properties of Fuels According to Fig. 174

Fuel	T.E.L. cm³/litre	Octane number	
		Motor	Research
A	0·0	80·0	80·5
B	0·13	85·0	86·0
C	0·4	89·5	90·5
D	0·8	93·5	98·0

A similar diagram, obtained with a non-saturated fuel is shown in Fig. 175. Bearing in mind the difference in octane numbers obtained by the two methods this fuel is obviously "sensitive". It is almost satisfactory without doping except in a small range between 2800 and 3000 rev/min. After a small modification to the automatic ignition control graph this fuel would be perfectly satisfactory. The addition of tetra-ethyl lead raises the octane number of the fuel almost consistently over the entire engine speed range. According to Table 35, the addition of 0·8 cm³ tetra-ethyl lead will raise the octane number (motor method) by 3 as compared with the increase of 13·5 with the fuel considered before. This small rise means a great improvement in terms of road performance as the compression ratio of the engine can be raised (Fig. 175).

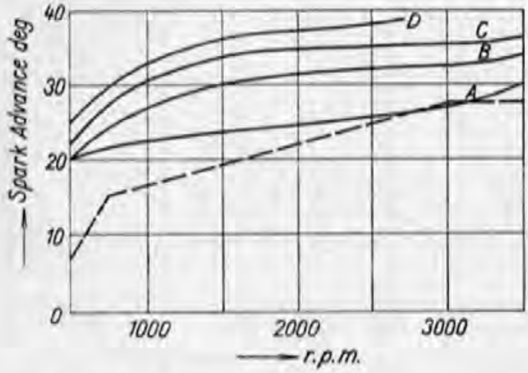


FIG. 175. Relationship between the maximum permissible spark advance at the detonation point and engine speed for a sensitive fuel.

By the addition of tetra-ethyl lead according to Table 35 the requirement varies in accordance with curves *B*, *C*, and *D*. The dash line represent the ignition advance given by the disributor.

These two examples illustrate how the octane number does not classify fuels according to their real value in the engine and how modern 80 octane fuel, produced by catalytic cracking is of much higher quality than fuel with the same octane number produced 20 years ago and composed mainly of paraffin hydrocarbons.

Table 35

Properties of Fuels According to Fig. 175.

Fuel	T.E.L. cm³/litre	Octane number	
		Motor	Research
A	0·0	82·0	94·5
B	0·13	82·5	96·5
C	0·4	84·0	98·0
D	0·8	85·0	99·0

CHAPTER XI

THE CYLINDER HEAD

1. The shape of the combustion space in petrol engines

The cylinder head is the most important part of an air-cooled engine and it is also the most difficult one to manufacture. The correct solution of the problems facing the designer in this respect decides to a large degree the qualities of the whole unit. A well designed air-cooled cylinder head must ensure: 1. high volumetric efficiency up to high engine speeds, 2. small heat losses, 3. an even temperature of the combustion chamber, 4. light weight, 5. small dimensions, 6. interchangeability, 7. easy manufacture.

It is far from easy to fulfil all these requirements and the design of a good cylinder head for an air-cooled engine is one of the most exacting tasks facing the designer.

The first decision to be made concerns the shape of the combustion chamber to be used. This shape must not only aid most complete combustion but it must also ensure high volumetric efficiency and effective exhaust valve cooling.

In order to attain a large mean effective pressure and low specific fuel consumption, especially with the engine running on part throttle openings, the use of a high compression ratio is of great importance.

The upper limit to compression ratio is given by the boundary above which detonation occurs. The example shown in Fig. 176 explains how detonation originates.

At the end of the compression stroke the cylinder contains a compressed mixture of air and fuel. This is ignited by the sparking plug under the inlet valve. We shall not be far from the truth in assuming that flame travels on spherical planes. Although it is true that the piston is still in upward motion when the mixture ignites, tests conducted [61] together with thorough mathematical analysis have shown that this

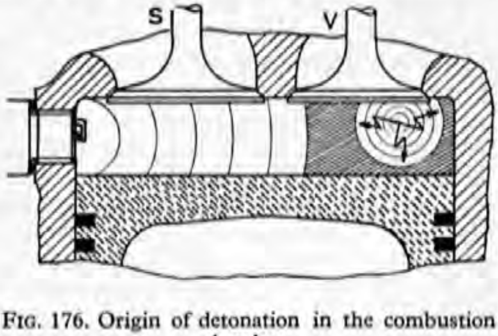


FIG. 176. Origin of detonation in the combustion chamber.

movement may be disregarded. It may be assumed that the piston is at rest at top dead centre during the whole period of combustion.

Between the spark appearing at the sparking plug and the beginning of combustion, which is accompanied by a rise in pressure, a certain period, known as the delay period or "ignition lag", elapses. The duration of this period depends chiefly on the temperature and pressure of the mixture to be

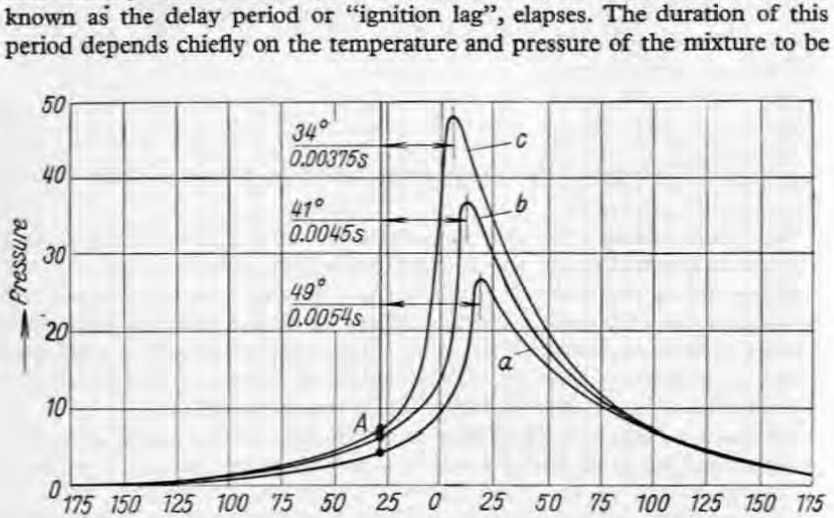


FIG. 177. Effect of compression ratio upon ignition lag at 1,500 rev/min, and constant ignition advance 30 degrees before top dead centre (Table 36).

ignited. The average ignition lag is approximately 0·004 sec. The variation of ignition lag with compression ratio is shown in Fig. 177.

After this delay the mixture begins to burn, the flame front spreading as the surface of a sphere of increasing size. After the burning of the part of the mixture adjacent to the sparking plug, temperature rises expanding the products of combustion. These products of combustion expand until a balance of pressure between them and the remaining unburnt charge in the combustion chamber is attained. The rise in pressure of the unburnt charge must naturally also raise its temperature. The same process takes place as further amounts of the charge burn. As the remaining unburnt charge decreases, its pressure and temperature continue to rise.

Table 36

	Compression ratio	p_i kg/cm²
a	4 : 1	8·4
b	5 : 1	9·6
c	6 : 1	10·3

If the temperature and pressure of the remaining unburnt charge rises above a critical point, self-ignition and explosive combustion of the charge residue takes place. This explosive combustion which does not spread from the sparking plug but from its own nucleus

(shown in Fig. 176) is known as detonation. The velocity of flame travel depends on the crankshaft speed, pressure, the fuel: air ratio, [62] etc. See Fig. 178. The velocity of a detonating wave is many times greater, reaching up to 9840 ft/s (3000 m/s). This extremely rapid combustion-detonation is accompanied by a rapid rise in pressure and is outwardly evident by its unpleasant, high pitched "pinking" or knocking. The very rapid pressure rise is detrimental to long engine life and measures must therefore be taken to eliminate knock.

If the "end gases", which are the last to be ignited, are compressed against the hottest part of the combustion chamber in the vicinity of the exhaust valve, temperature will rise further and the conditions for detonation will arise sooner.

One of the principles of cylinder head design therefore is to place the sparking plug near the exhaust valve in order to compress the end gases into the cool corner around the inlet valve.

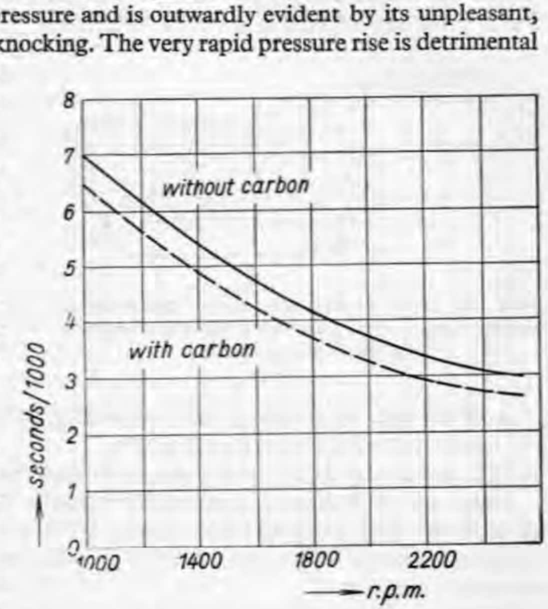


FIG. 178. Influence of engine speed upon rate of combustion in an engine with and without carbon deposits.

The highest engine output can be reached if gas pressure above the piston is at its highest at about 10° passed top dead centre. With the ignition lag and the velocity of flame travel known, the ignition advance must be greater with a longer distance over which the flame front must travel. This period extends to the compression stroke, during which it brings no gain, but a loss in output. A long combustion period also causes a greater heat loss to the cylinder walls.

For these reasons the design of the combustion chamber must be such as to limit to the smallest possible length the distance of flame travel from the sparking plug to the remotest corner of the cylinder head. This commends the use of compact combustion chambers which have the further advantage of a small volume: surface ratio with its accompanying low heat loss.

These deliberations led Ricardo to concentrate the combustion chamber of side valve engines in a pocket above the valves with the plug above the exhaust valve. The piston crown very closely approaches the cylinder head at top dead centre in this engine and the whole charge is displaced and compressed into the combustion chamber above the valves.

The expulsion of the charge before top dead centre produces great turbulence which is typical for a Ricardo type combustion chamber. This turbulence does not permit a sudden local temperature rise which would endanger regular combustion and speed the rate of combustion, thereby making possible a higher compression ratio without the risk of detonation. Side valve engines have the following inherent disadvantages:

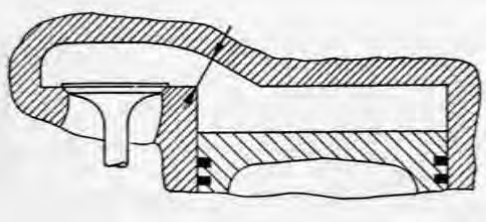


FIG. 179. In SV engines with a high compression ratio a danger of air throttling exists at the point marked on the drawing.

1. The combustion chamber surface is large thus causing large heat losses.
2. A flat combustion chamber is unsuitable from the point of view of a detonation free combustion.
3. Volumetric efficiency is low owing to bends in the mixture flow and the heating of air.
4. With high compression ratios throttling effects occur between cylinder and combustion chamber. Fig. 179.
5. Poor valve access for valve clearance adjustment.

Overhead valve engines are at an advantage as they give higher output and their specific fuel consumption is lower. For this reason side valve layouts are being abandoned. The overhead valve arrangement is particularly suited for air-cooled engines.

Overhead valves may be arranged in the cylinder head in various positions making possible the formation of combustion chambers which are 1. wedge-shaped, 2. hemispherical.

1. With a wedge-shaped combustion chamber the valve operating mechanism can be designed with greater ease. In principle it is beneficial to concentrate maximum volume beneath the exhaust valve and to make the section remote from the plug as small and as well cooled as possible.

Fig. 180 shows an up to date combustion chamber which may be divided into three sections. The first section, adjacent to the sparking plug, exercises the greatest influence on heat losses. The surface of this section is exposed to the longest period of combustion and should therefore not present undue risks for heat loss. For that reason the exhaust valve is placed in this section.

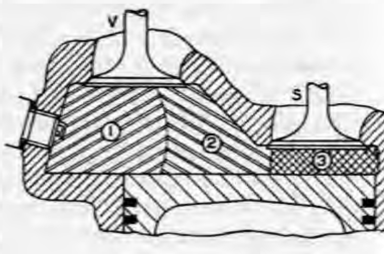


FIG. 180. Illustration of a wedge-shaped combustion chamber.

1 — the first section has a bearing on thermal losses, *2* — the second upon combustion roughness, *3* — and the third upon detonation.

The second section is most important for combustion roughness as when combustion spreads through it, the piston is at top dead centre. A sharp pressure rise must therefore be avoided and the section should be kept as small in volume as possible.

The third section is noted for its influence on detonation. It must therefore be small in volume and well cooled. This part of the compression space is usually formed by the gap between the piston and cylinder head which should not be larger than 8 mm. The cold disc of the inlet valve with its beneficial cooling effect may be positioned here.

During combustion the unburnt charge is compressed and fuel concentration is increased within it. It follows that with the flame spread through one half of the combustion chamber, less than a half of the fuel is burnt.

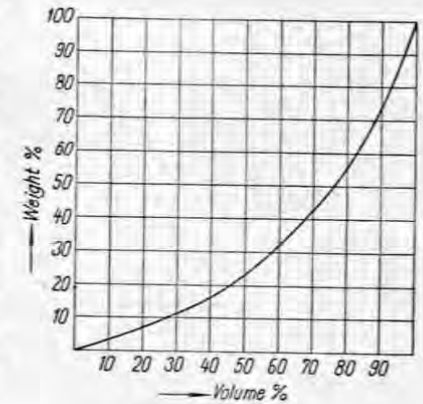


FIG. 181. The relationship between the volume and weight of the burnt mixture is in agreement with different measurements.

The ratio of the weight of this burnt part of the charge to its volume has been the subject of studies by numerous authors [49] and the results obtained by them given in Fig. 181 are in agreement. According to the diagram in Fig. 181 it will be noted that with the burnt gases taking up 50% of combustion chamber volume, only 24% of the charge by weight has been burnt.

The elimination of detonation in the combustion chamber is aided by the quench area extending between the cylinder head and pistons. The charge displaced from this space during the compression stroke causes high turbulence inside the combustion chamber thus favourably affecting the process of combustion.

The effect of air swirl caused by air entering the cylinder tangentially during the induction stroke and of squish during the compression stroke is illustrated in Fig. 182. The curves *A* to *D* represent the properties of similar combustion chambers with parallel vertical overhead valves. The basic shape of all these spaces is flat and their compression ratio is 9 : 1. With spaces *A* and *B* the inlet port is at a tangent to the cylinder circumference, thus causing the air entering the cylinder to swirl rapidly. The inlet port of *C* and *D* points towards the axis of the cylinder and no swirl occurs. In the combustion chambers marked *B* and *D*, 30% of the piston crown forms a quench area between piston and cylinder head, the aperture being about 1·8 mm high. The plug is located approximately in the axis of the cylinder.

Taking the performance of an engine with a combustion chamber *A* as a basis (100%), the output of engines with the other combustion chambers are approximately the same. The octane requirements on the other hand, differ considerably, the lowest being in chamber *D* without swirl and with a quench area.

Comparing the properties of these types on the verge of detonation with current fuel we obtain octane number requirements shown in Fig. 182 (dashed

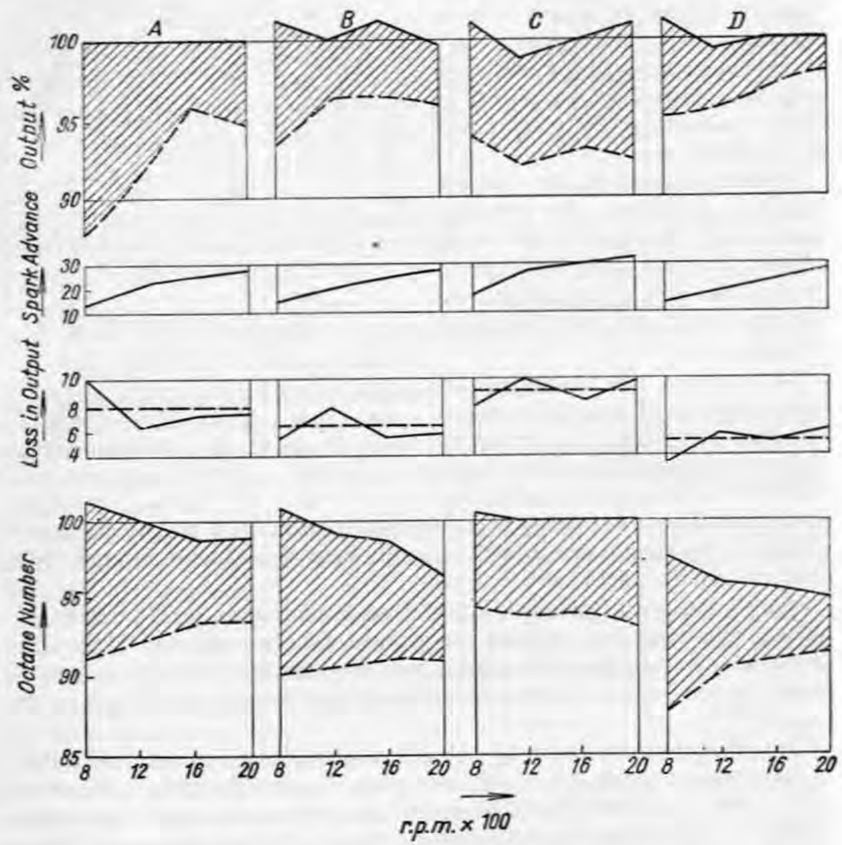


FIG. 182. Comparison of similar combustion chambers of petrol engines.

line), from which it is apparent that types *A* and *C*, without quench area, show the greatest drop in performance and their octane requirement is high. Types *B* and *D* with a quench area show a higher power output and a lower octane number requirement. In this instance, induction swirl does not enhance their properties.

The results given here may not apply in general, but they underline the importance of quench area and the small influence of induction swirl. With other types of combustion chambers, however, swirl does effect a considerable improvement. It is important to direct the stream of mixture entering the cylinder towards the sparking plug.

In another experiment using the same fuel, the compression ratio could be raised from 7·25 to 9 : 1 following and increase in quench area from 15.8% to 30%. This brought about not only an increase in engine output but also a reduction of specific fuel consumption which is particularly important at reduced load running.

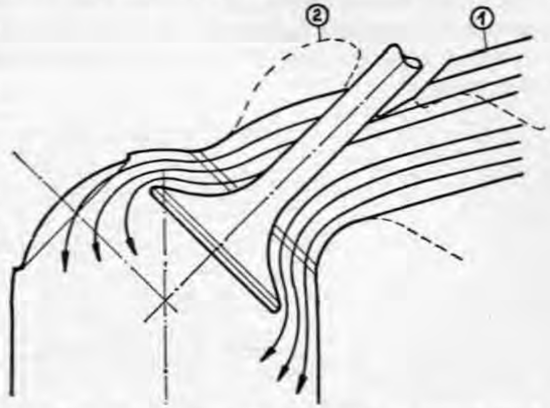


FIG. 183. Air flow round the valve in a hemispherical combustion chamber.

1 – shape preferred in motor cycle engines, *2* – in aircraft engines.

The temperature of the internal surface of the combustion chamber is very important. A lower temperature results in a lower octane requirement and increased volumetric efficiency. This must be realised all the more by the designer of an air-cooled engine who must pay attention to the best possible heat transfer from regions most remote from the sparking plug.

FIG. 184. Experimental piston with well shaped crown for a hemispherical combustion chamber.

In wedge-shaped combustion chambers, good cooling of valve seats between the two ports is important and a sufficient cooling air stream must be directed to this region. With cylinder heads of small bore engines this is difficult to arrange. With the choice of a short-stroke layout, more room for the valves becomes available and the flow area between the ports is also larger.

Local temperatures of combustion chamber walls can be regulated by proper finning and the direction of cooling air to places which need the most intensive cooling.

2. *Hemispherical combustion chambers* hold a great advantage in their low surface area: volume ratio and further advantages lie in the fact that there is sufficient space for large valves to be fitted and high volumetric efficiency is thus ensured.

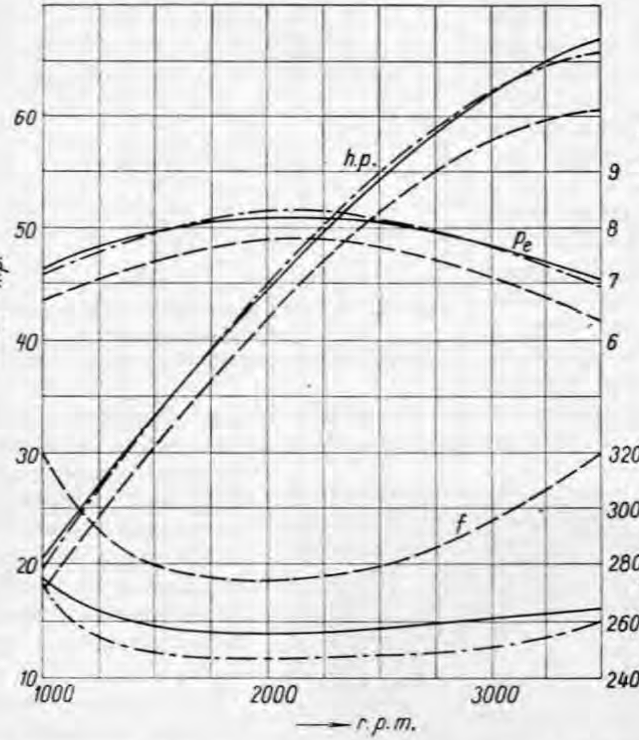


FIG. 185. Comparison of performance and fuel consumption for a hemispherical combustion chamber with a flat piston crown and $\varepsilon = 6$ (dashed line), shaped piston crown and $\varepsilon = 7.2$ (full curve), and shaped piston crown $\varepsilon = 7.2$ and the spark plug shifted towards the centre of the combustion chamber (dash and dot line). In the latter case ignition advance had to be reduced from 46 to 36 degrees.

A convenient feature of the air-cooled engine is its unobstructed exhaust valve seat periphery, permitting the use of extensive finning on the outside. The cooling of the exhaust valve seat is of great importance.

The direction of air flow past the inlet valve is good and on entering the cylinder, the air stream is continually deflected about the inside wall of the

cylinder. Fig. 183 shows the air flow past the inlet valve into a hemispherical cylinder head. The change of flow area is gradual and there are no abrupt bends.

It is owing to these desirable features that engines with hemispherical combustion chambers are known to give highest outputs. By the correct shaping of the piston crown the combustion process may be regulated, as shown in Figs. 184 and 185.

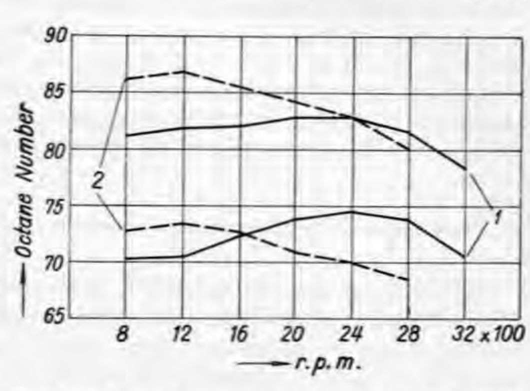


FIG. 186. Octane number requirements of an OHV (1) and SV (2) engine without (bottom lines) and with (top lines) carbon deposit.

The distribution of temperatures in the combustion chamber is of great importance. More can often be gained by correct cooling than by the shape of the combustion chamber itself. The combustion of the mixture must proceed toward the coolest part of the combustion chamber.

Cooling efficiency is considerably reduced and the octane requirement of an engine raised when carbon deposit forms on the combustion chamber walls. A hemispherical space is not so sensitive to carbon deposit and the loss of performance is therefore smaller. Fig. 186 shows the loss of performance caused by carbon deposit in some types of cylinder heads; with a hemispherical head it amounts to only about 2%.

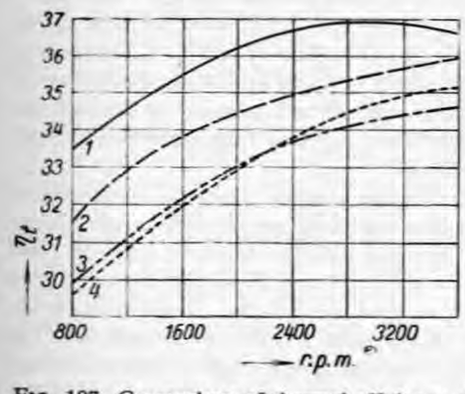


FIG. 187. Comparison of thermal efficiency of different types of combustion chambers plotted against engine speed.

The test was carried out on an experimental single-cylinder engine having a displacement of 770 cu. cm, a stroke: bore ratio 0·94 and $\varepsilon = 7$. Octane number requirements for all four chambers tested were approximately equal. *1* — inclined valves in the cylinder head, hemispherical combustion chamber, *2* — parallel valves in the cylinder head, *3* — SV valves, *4* — F head.

Owing to small heat loss to walls, the thermal efficiency of a hemispherical combustion chamber is high. Fig. 187 compares the thermal efficiencies of several shapes of combustion chambers as tested on an experimental single cylinder engine with a stroke: bore ratio of 0·94 and a compression ratio of 7 : 1. The octane requirement of all these types of combustion chambers, at full throttle and uniform engine speed, was identical. The highest indicated thermal efficiency was attained by the hemispherical combustion chamber [15].

A compromise between side valve and overhead valve types is the F head with a side exhaust valve and an overhead inlet valve. The size of the valves, the inlet valve in particular, is not dictated by limitations of space and the combustion chamber is of convenient shape and permits smooth engine running. The large surface area of the combustion chamber, however, promotes heat loss to cooling, which manifests itself in the lower thermal efficiency of the engine. For air-cooled engines this arrangement shows a definite drawback in that the cooling of the exhaust valve presents difficulties.

2. The shape of the combustion space in compression-ignition engines

Cylinder head temperature must be as low as possible in order to maintain high volumetric efficiency and with it high engine performance. When considering air-cooled light alloy cylinder heads, it must be remembered that the strength of the alloys shows a rapid drop above 200 °C. Fig. 188 shows the variation of strength of aluminium alloys used for cylinder head construction with temperature [28].

In petrol engines, local overheating of the combustion chamber leading to detonation, must be prevented. Special attention must be paid to the region adjacent to the exhaust valve seat and the sparking plug. According to recent experience, the temperature of the inner cylinder head surface in the combustion chamber should not exceed 235 to 250 °C. Temporary, but by no means continuous, local temperatures may be as high as 270 °C. Under normal running conditions, temperatures over 220 °C are to be avoided in view of the possibility of a temperature rise due to lean mixture running and the accompanying risk of detonation. In aircraft engines this temperature may reach 300 °C for a short period during take-off.

The requirements governing combustion chamber shape in compression-ignition engines differ widely from those affecting petrol engines, the diverse means of preparing the air-fuel mixture being the cause of this. In compression-ignition engines the mixture is formed only an instant before ignition commences by the injection and dispersal of the fuel into the heated air at the end of the compression stroke. The period available for the formation of the mixture is extremely short, for the injected fuel ignites and as a rule also burns before injection is over. For this reason the formation of a homogeneous mixture of air and fuel, and the complete utilization of the amount of air present as in the petrol engine, is impossible and larger amounts of air than needed for the chemically correct mixture must be used.

An overloaded compression ignition engine emits black exhaust smoke which is a sign of insufficient air available for combustion; the oxygen content of exhaust gases may be ascertained by analysis. The trouble is caused by local

air shortages. All the oxygen in the neighbourhood of the fuel jets is used for the combustion of fuel and the remaining fuel in the jet centre cannot find sufficient oxygen for complete combustion. Oxygen is contained in the space separating the jets, but it is too remote from the droplets of atomized fuel. A good combustion chamber must, therefore, ensure the best possible mixing of air and fuel in order to utilize the highest possible percentage of the oxygen present in the cylinder for combustion.

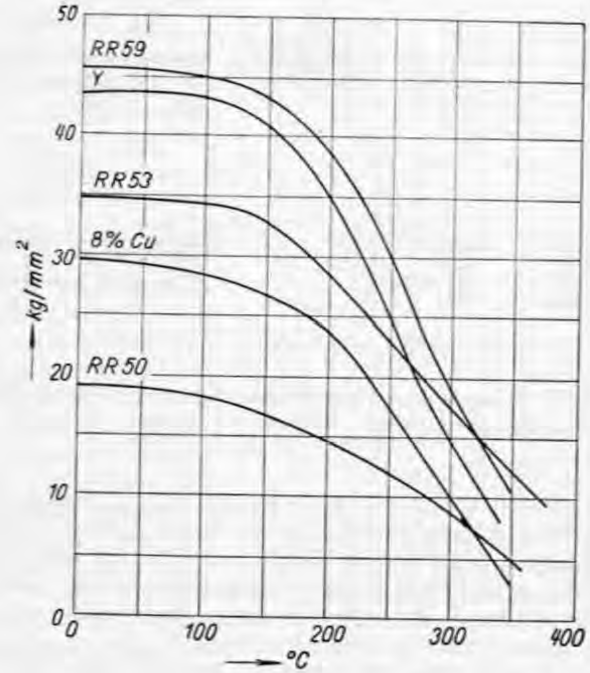


FIG. 188. Effect of temperature upon the strength of aluminium alloys used for cylinder heads.

It has been stated in preceding sections that the performance of a petrol engine depends on the amount of air by weight, which is drawn into the cylinder within a given time.

The same applies to oil engine performance with one difference: the amount of air which takes part in the process of combustion is the deciding factor; much therefore depends on the need for excess air being as small as possible. A compression-ignition engine will, therefore, have a lower output than a petrol engine having the same volumetric efficiency. Lower performance results in a smaller liberation of heat and therefore the heat stress per unit of the inner surface area is lower. From the point of view of cooling, the compression-ignition engine scores over the petrol engine.

The oil engine, however, works with high compression ratios. For this reason pressures at the end of the compression stroke are high, and so are the peak pressure and combustion temperature. Compared with peak pressures of 491 to 568 lb/in^2 (35 to 40 kg/cm^2) which are common in petrol engines,

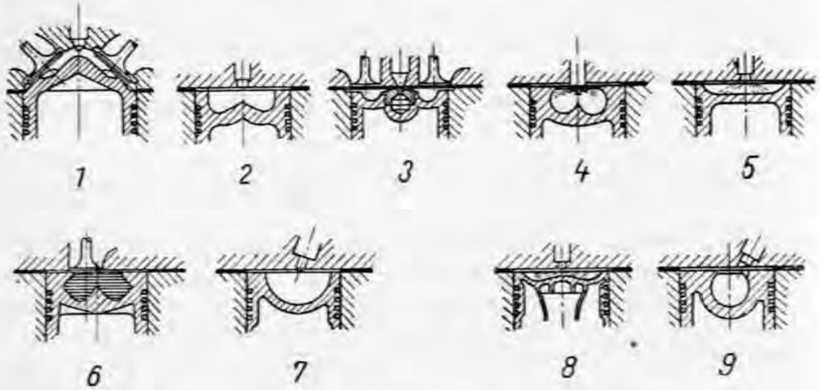


FIG. 189. Different shapes of combustion chambers for fuel injection.
1 — Tatra (obsolete shape), 2 — A.E.C., 3 — Cummins, 4 — Saurer, 5 — Soviet D 1, 6 — Crossley, 7 — Gardner, 8 — JAZ 204, 9 — MAN.

compression-ignition engine peak pressures are twice as high at 853 to 1137 lb/in^2 (60 to 80 kg/cm^2) and peak temperature during combustion must therefore also be higher. Since heat transfer is dependent on temperature difference, the heat transfer of an oil engine during combustion exceeds that of a petrol engine. During the expansion stroke, however, the temperature of burnt gases in the cylinder is reduced and exhaust gas temperature is lower than in petrol engines owing to the increased expansion ratio. The heat stress on exhaust port walls is, therefore, lower than that common in the petrol engine.

The mixing of air and fuel can be performed in two ways in an oil engine:

1. direct fuel injection
2. indirect fuel injection.

Direct injection, as the name implies, aims a jet of fuel directly into the combustion chamber and a good mixture is obtained by shaping and arranging the fuel jets in the most suitable way and by creating the required turbulence inside the cylinder. The surface area of the combustion chamber is made as small as possible and thus has the advantage of small heat loss. The following advantages are derived from this:

a) A small amount of heat is lost by cooling. To transfer this heat into the air, only a small share of the effective engine performance is needed.

b) Thermal efficiency is high. Specific fuel consumption is therefore low.

c) Heat losses are low even during the compression stroke. Air temperature at the end of the compression stroke is high and the starting of a cold engine is thus facilitated without the need for auxiliary sources of heat.

Some shapes of direct injection combustion chambers which have found favour are shown in Fig. 189. Turbulence here depends to a large extent on the cylinder bore: piston crown cavity diameter ratio. The orifice of the cavity is sometimes much smaller than the largest diameter of the cavity – in such cases the engine is really an injection chamber type with the chamber or cell within the piston (e.g. MAN).

For the sake of maintaining a high thermal efficiency, the instantaneous combustion of all fuel with the piston at t.d.c. would be desirable. The heat thus liberated could work for the whole duration of the expansion stroke. If part of the charge burns later, i.e. when the piston is already moving away from t.d.c., the cylinder surface is exposed to hot gases and effective piston travel is reduced. Heat liberated in this way would, therefore, be utilized in a less efficient manner. Fig. 190 shows the dependence of thermal eficiency on the percentage of fuel burnt at a constant pressure and constant volume. The thermal efficiency shows a rapid drop if less than 60% of fuel is burnt at constant volume.

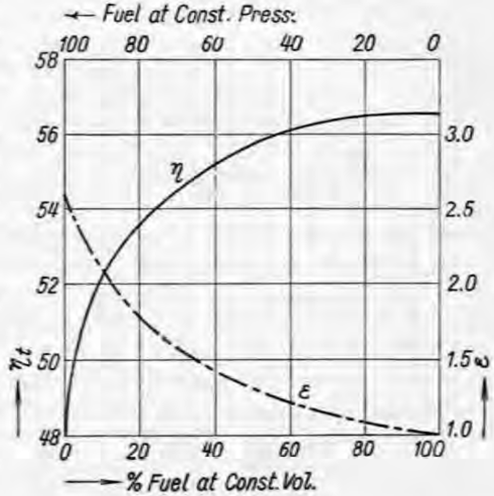


FIG. 190. Comparison of thermal efficiency according to fuel combustion at constant volume or constant pressure. The full curve represents efficiency, the dashed the expansion ratio ε_c during combustion.

The instantaneous combustion of fuel at top dead centre ensures high thermal efficiency, but difficulties arise when trying to achieve silent running. The silent operation of air-cooled engines is of particular importance, since, with no water jackets surrounding the cylinders, it becomes more difficult to subdue the noises of combustion and piston slap. For this reason the geometry of the shapes in question and the means of mixture formation must be chosen in such a way as to ensure as quiet combustion as possible.

High turbulence shortens ignition delay and thereby reduces engine noise. Compact toroidal chambers in the piston promote squish during the compression stroke and aid mixture formation. Swirl, caused by the air entering the cylinder at a tangent during the induction stroke is also desirable. The higher the degree of swirl, the fewer jets of fuel are needed for mixing fuel and air. This is a welcome feature especially in small capacity cylinders, for the holes in the injection nozzle may be comparatively few and large with a subsequent reduced likelihood of blockage by carbon or by fuel impurities.

Better mixing of the fuel with air takes place if the axis of the fuel is normal to the airstream. If the jet is exposed to a side blast the distance which the droplets must cover is shorter.

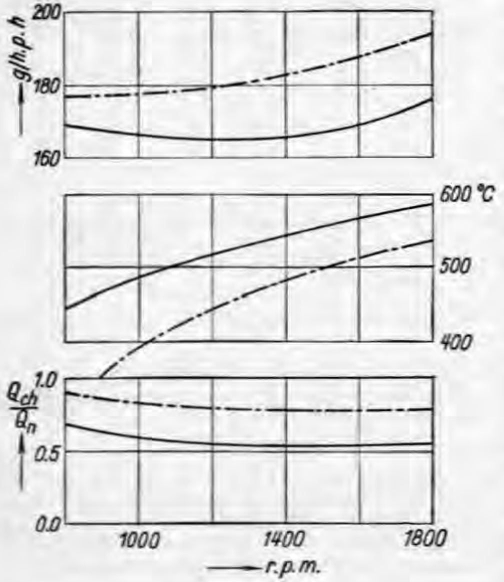


FIG. 191. Characteristic of oil engine with direct and indirect fuel injection. Full curve — direct injection, dashed curve — indirect injection.
Q_{ch} – heat removed by cooling, Q_n – heat converted to performance.

The clearance between the piston crown and cylinder head should be as small as possible (usually 0·75 to 1 mm). The designer must bear in mind the reduction of bearing clearances caused by inertia forces acting upon the piston assembly at top dead centre, and tolerances in the manufacture of connecting rods, crankshaft webs, cylinder length and the distance of the cylinder base from the crankshaft axis.

The clearance chosen must take into account inaccuracies in production, carbon deposit on the cylinder crown and also piston expansion through heat.

The cavity orifice diameter has an influence not only on turbulence, but also on the temperature of the periphery of the orifice. With an orifice which is too small, the edges are insufficiently cooled and may crack owing to overheating. With the cavity mouth too wide, the circumference will be too cold for drops of fuel in contact with it to evaporate fast enough and carbon will be deposited more rapidly. Both the conditions of turbulence and those of temperature must therefore be met. In extreme cases recourse may be made to cooling the underside of the piston crown by a jet of oil directed against it from below.

It is important to keep the cavity walls at the correct temperature especially where the surface is required to evaporate fuel colliding with it by letting it spread into a continuous film and by preventing droplets from bouncing off

after the collision. By high turbulence above the surface the fuel and air may be well mixed under these conditions and the amount of surplus air needed is small.

The great advantage afforded by direct injection to air-cooled engines is the simple design of the cylinder head, the cooling of which presents no great problems and can be carried out without high temperature differences and the risk of failure owing to overheating. The problem of placing the glow-plug (which is necessary for easy starting from cold) does not arise either.

In the case of direct injection, much of the usual heat load upon the cylinder head is transferred to the cylinder barrel and it is not uncommon to carry away as much heat from the cylinder barrel as from the head. Effective cooling for the upper part of the cylinder barrel, especially near the end of piston ring travel, must therefore be provided. For this reason the fitting of cylinder heads by flanges is unsuitable as much cooling fin area has to be sacrificed.

A comparison of two engines, one with direct and the other with indirect fuel injection, is given in Fig. 191 [13]. The heat removed by cooling is much lower in the case of the direct injection engine and generally is just above 0·5 b.h.p. Both engines under consideration are, however, water-cooled; in an air-cooled engine the portion of heat removed by the cooling medium is still smaller expressed in terms of e.h.p. Specific fuel consumption is considerably lower in the direct injection engine owing to its higher thermal efficiency. Not without interest is the fact that exhaust gas temperature is also much lower in the case of the direct injection engine despite the fact that its compression ratio is generally lower than that of the indirect injection engine. This is caused by the rapid ignition and efficient combustion of fuel with the piston at t.d.c. In indirect injection heads the combustion process is slower owing to the different manner of mixture formation; part of the fuel therefore is burnt less effectively during the expansion stroke and the exhaust gas temperature is thus raised. Higher combustion temperature in an injection chamber engine may also be caused by a smaller air surplus.

Indirect fuel injection is used in engines with injection chambers, in which case the fuel is injected into a chamber containing only a part of the air of the combustion chamber. Hot burning gas streaming from the injection chamber at high velocity into the combustion chamber carries with it droplets of remaining unburnt fuel into the air in the combustion chamber. Peak pressure is developed inside the injection chamber and does not act upon the piston. Pressure in the combustion chamber is lower and engine running is therefore smoother. At the high velocity with which burning gas sweeps through the connecting duct between the injection chamber and the combustion chamber, much heat is transferred to the cylinder head walls (the connecting duct). This increased heat transfer occurs also during the compression stroke when hot air entering the injection chamber is cooled down by passing the connecting duct. High compression ratios must therefore be used with this type of engine in order to ensure high temperature in the injection chamber at the end of the compression stroke for easy starting from cold.

Injections chambers are of three basic types:

1. the pre-combustion chamber
2. the swirl-chamber
3. the air cell

The air cell type cannot be applied to air-cooled engines owing to its large demands on space and its highly unfavourable effect on engine cooling.

Injection chambers may be formed either directly in the cylinder head, cast-in to aluminium heads or they may be screwed in. Cast-in chambers are

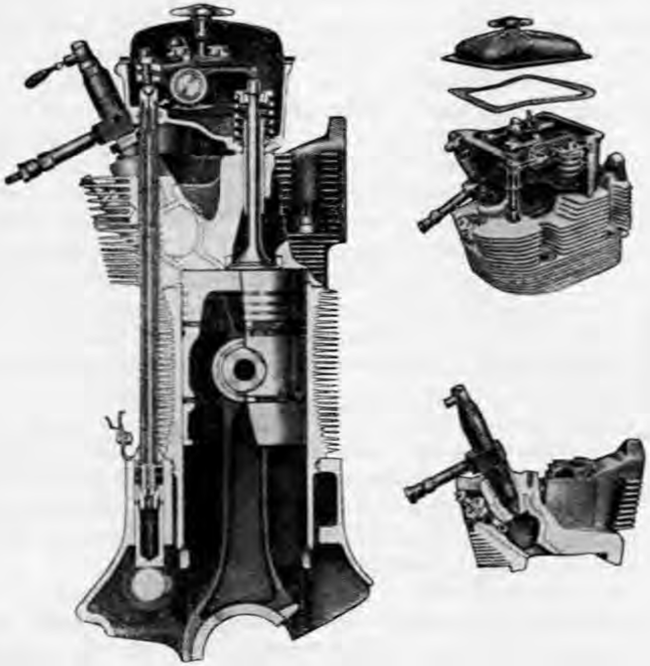


FIG. 192. Cylinder and injection chamber of an air-cooled Deutz engine.

used by Deutz (see Fig. 192) and screwed-in chambers by Caterpillar (see Fig. 193).

Injection chambers with a fairly large orifice in the chamber axis have become popular in recent years. The wide orifice reduces the velocity of the gas flowing into or out of the chamber thereby reducing compression work. As a result of the lower speed of the burning gases, heat transfer to the cylinder head is also reduced, thus lowering heat losses.

The MWM cylinder head shown in Fig. 194 may be demonstrated as a successful example of an air-cooled cylinder head with an injection chamber. The chamber is formed by a separate casting which is screwed on to the

cylinder head. The area of contact with the cylinder head is small and little heat is transferred from the chamber to the cylinder head, thus preventing overheating at this point with its resultant effects which often include cracking of cylinder heads. Surrounding the main orifice of the chamber, a number of smaller apertures are drilled into the chamber parallel to the orifice axis.

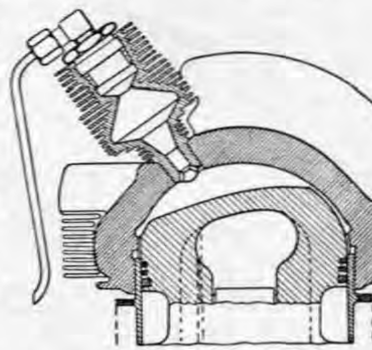


FIG. 193. Injection chamber of an air-cooled oil engine.

The MWM engine is noted for its specific fuel consumption which does not exceed that of a direct injection engine, but which at the same time permits relatively low peak pressures inside the cylinder. The rise in pressure per 1° of crankshaft rotation is also low, the engine being smooth and silent in operation, which is particularly desirable in air-cooled units. Another welcome advantage of this engine is its readiness to run on various fuels. Without modification to the engine, fuel oil may be replaced by petrol, paraffin, jet propulsion fuels, etc. For use with petrol, the pressure of the petrol supply pump must be raised to about 28·5

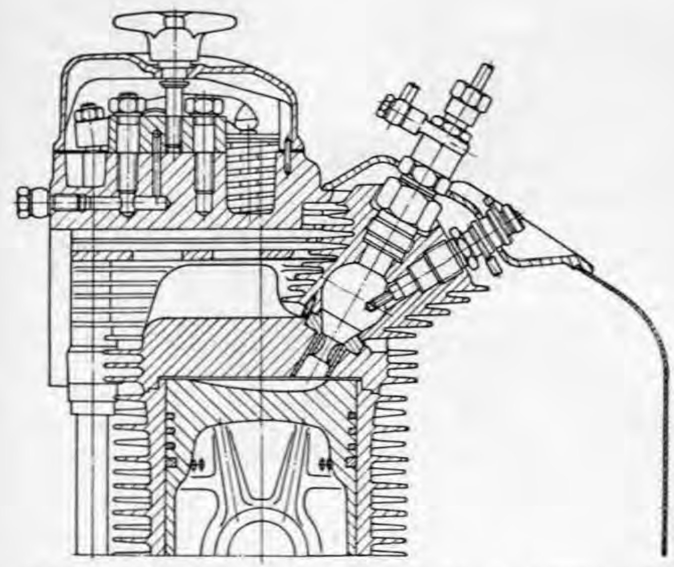


FIG. 194. Sectional view of the cylinder head of an MWM engine.

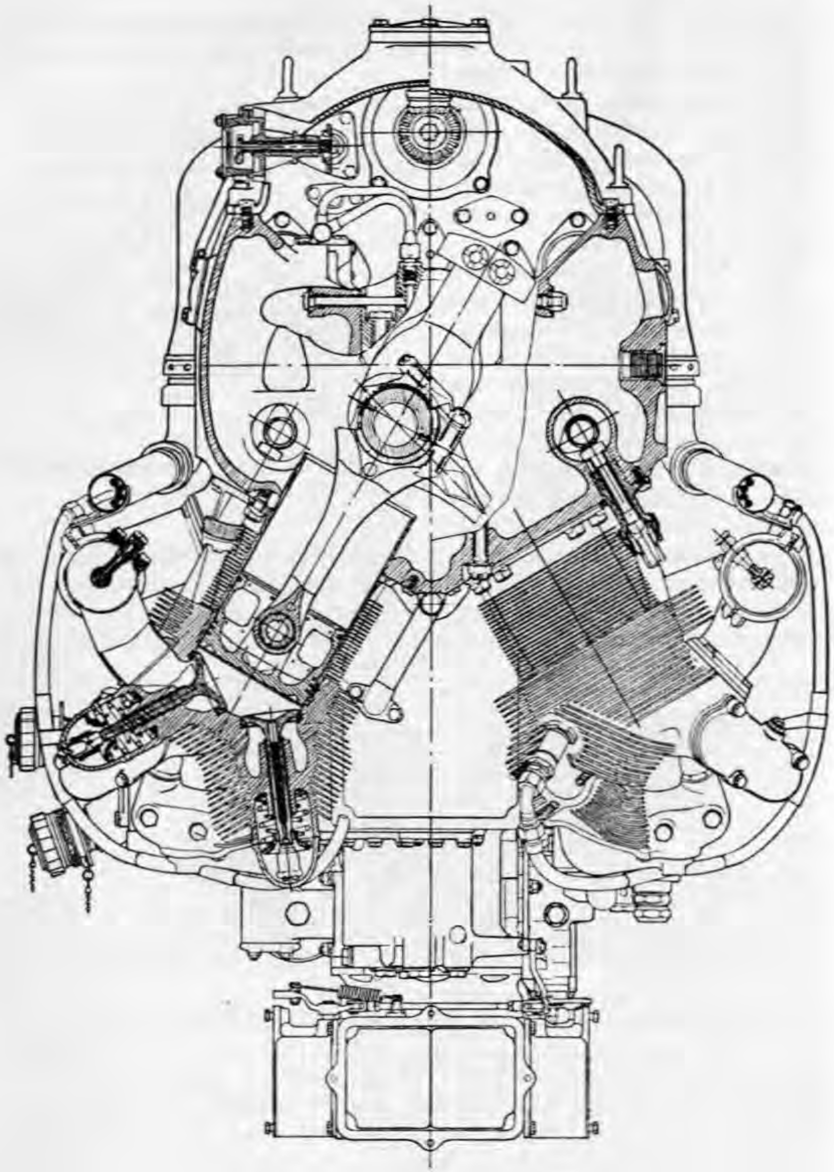


FIG. 195. Sectional view of the Argus As 410 aircraft engine with a well finned cylinder head.

lb/in² (2 kg/cm²) and the rate of flow must be increased. The excess fuel passes through the injection pump thereby flushing and cooling it in order to prevent the formation of gas locks when using volatile fuels such as petrol. This engine may be started at a temperature of 10 °C below zero without a glow plug, for the fuel spray passes through the orifice of the injection chamber into the cylinder, where the air is sufficiently hot to ignite it.

3. Detailed design

Cylinder head temperature is limited by considerations of volumetric efficiency, the requirements of correct combustion (freedom from detonation) and the strength of the material of the cylinder head. With a view to the

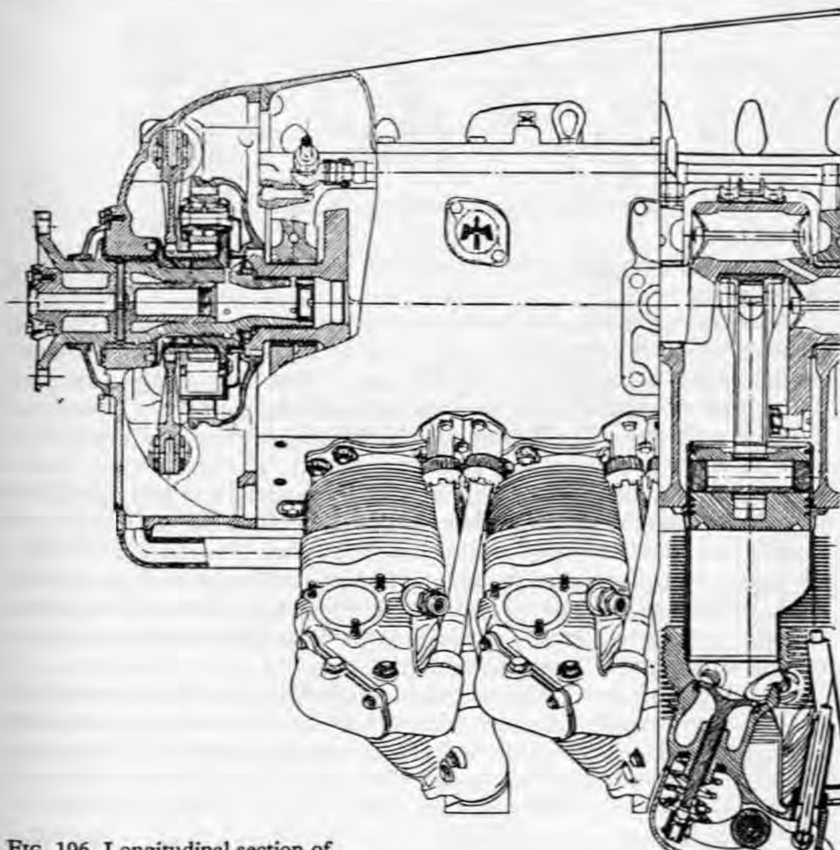


FIG. 196. Longitudinal section of the Argus As 410 aircraft engine.

highest possible volumetric efficiency, cylinder head temperature must be kept as low as possible. The temperature of the inlet valve and its surroundings has considerable influence on the degree of heating of the incoming air and thereby on the volumetric efficiency of the engine. Because of this, it is beneficial to direct cold cooling air to the inlet valve region in the first place. With such an arrangement there is however the possibility of a very uneven

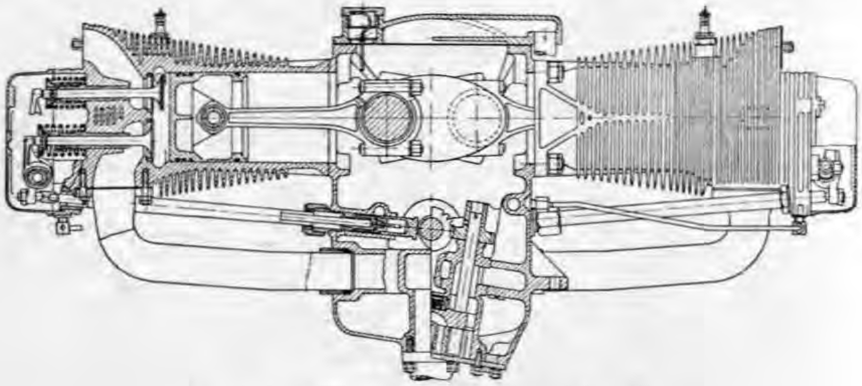


FIG. 197. Sectional view of the Franklin engine.

distribution of temperature over the cylinder head with the subsequent danger of distortion.

The occurence of detonation is aided by hot parts in the combustion chamber such as, for example, the exhaust valve. It is the danger of exhaust valve overheating which prevents the use of higher compression ratios. An even temperature distribution over the wall surface of the combustion chamber is therefore neccessary. The hottest part is the exhaust valve and every effort must be made to remove heat from the exhaust valve seat to air by the shortest route. A convenient design feature in this respect is a free periphery of the exhaust valve seat on which finning may be arranged. This is best provided for by a hemispherical combustion chamber. Fig. 195 and 196 show the Argus As 410 – a good example of finning in the region of the exhaust valve. In order to assist heat flow from the valve disc, a hollow valve stem is used in this instance, partly filled with sodium. With parallel valves, conditions are less favourable as illustrated by Fig. 197.

In the case of parallel valves the cooling of the partition between inlet and exhaust valve seats calls for special attention. Too much metal is concentrated here, the distance to the nearest fin is long and, furthermore, this area is strongly heated and at the same time shows a large temperature difference which may even cause cracking of the head. This is the place at which air-cooled cylinder heads are most liable to crack. The difficulty can be overcome by making the bridge between valve seats as large as possible, keeping the

distance for heat transfer short (thin walls) and by directing a sufficient stream of cooling air to the area (Figs. 198 and 199).

If exhaust valve temperature is to be kept low, cold air should be brought direct to the exhaust valve area. With this arrangement the exhaust valve is the coolest part, whereas the inlet valve region is somewhat hotter, for the heated air coming from exhaust-side fins, has lost some of its cooling ability;

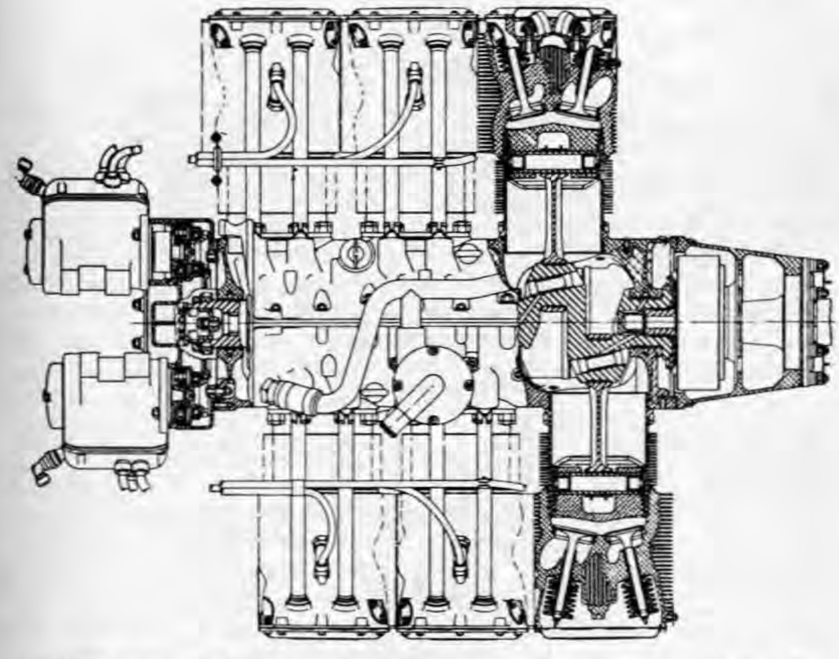


FIG. 198. Longitudinal section of the Lycoming GO – 435 aircraft engine. Bore 123.8 mm, stroke 98.4 mm, 260 b.h.p. at 3,400 rev/min.

but the temperature of the cylinder head is uniform. This arrangement is usually preferred, but much depends on details of design and conditions.

With a view to the economy of performance for cooling and the amount of heat removed by cooling, it would be advisable to choose the highest possible cylinder head temperature. In preceding chapters the point has been made that in order to reduce cylinder head temperature by 10°, fan input must be multiplied by 2·3. This calls for the exercise of extreme caution in deciding the correct cylinder head temperature. The highest admissible temperature for heads cast of aluminium alloys is 235 °C with individual hot spots of 250 °C at the most. Higher temperatures cannot be tolerated. Fig. 188 shows the variation of strength of several aluminium alloys used for cylinder heads with temperature. A considerable reduction of strength of the material is evident at temperatures above 200 °C. The composition of these alloys is given in Table 37. The reduction of strength at high temperatures need not manifest itself through cracks only, but may result in permanent deformation, especially at the pressure gas seal between the cylinder head and cylinder.

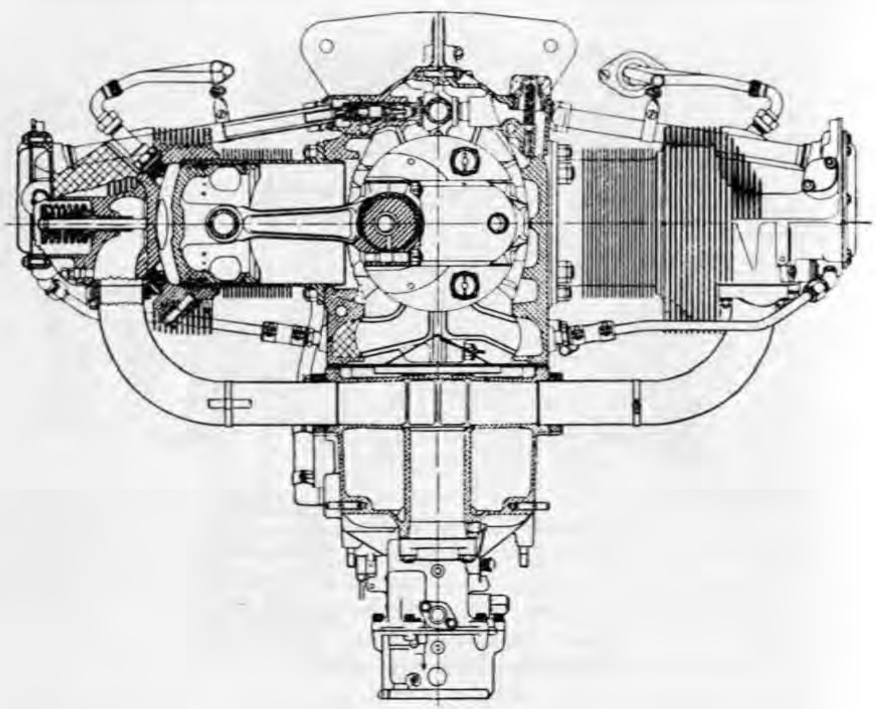


FIG. 199. Transverse section of the Lycoming GO-435 engine weighing 181 kg.

A well designed cylinder head must possess:

1. large and well cooled valves
2. large flow area for cooling air
3. large cooling area with properly located fins
4. easily accessible and oil tight valve mechanism.

The valves in the cylinder head should always be as large as the design permits, since large valves promote high volumetric efficiency and high output even at high engine speeds. Fig. 200 shows four overhead valve layouts. With two parallel valves, the flow area of the inlet port amounts to 18·5% of piston area and that of the exhaust valve to 13·5%. A slight increase in inlet valve size is still possible at the expense of the exhaust valve. The partition

Table 37

Compositions of Aluminium Alloys for Cylinder Heads

	Cu	Mg	Si	Ni	Fe	Ti	Zn	Brinell
RR 59	2·0	1·7	0·3	1·3	1·3	0·05	—	130
Y	4·0	1·5	0·5	2·0	0·4	—	—	110
R R 53	2·0	1·6	0·9	1·3	1·3	0·10	—	135
R R 50	1·4	0·12	2·2	1·1	1·1	0·23	—	76

between the ports is already very small and may cause fractures of the head at this point. Exhaust valve cooling is poor in this case.

With a four-valve layout with parallel valves the flow area of two valves is increased to 21·6%. The small weights of the two valves are convenient for high engine speeds. The cooling of such a four-valve cylinder head by air is, however, a difficult proposition; it may be effected by forming a trough between the pairs of inlet and exhaust valves with vertical fins rising direct from the cylinder head. The sidewalls of the channels may be finned horizontally. An example of this design is shown in Fig. 201.

More advantages are offered by the use of a hemispherical combustion chamber with inclined valves. The exhaust valve of such a layout can be efficiently cooled on the whole of its circumference. The maximum flow area is about 32% at the inlet valve seat and 22.3% at the exhaust valve. Small changes can still be made by altering the respective sizes of the valves and by constructional details. The conditions prevailing with aluminium alloy heads, in which inserted valve seats are used of necessity, are probably less favourable.

By the use of a "pent roof" four-valve cylinder head layout, the flow area at the exhaust valve seats may also be increased to about 32% of the piston crown area. The coefficient of flow through the ports is however less favourable than in the foregoing case (hemispherical combustion chamber), for flow is restricted by the close proximity of two valves and on the outer side also by the vertical wall.

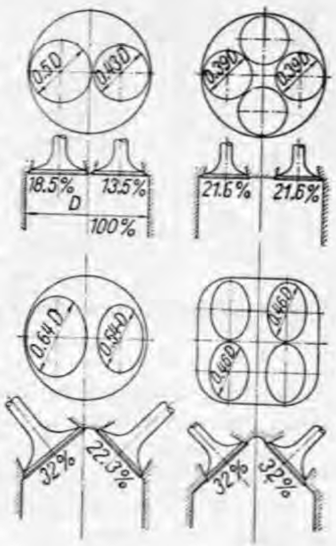


FIG. 200. Comparison of valve flow areas expressed in percentages of the piston crown area for various types of valve gear.

In plan view, some wedge-shaped combustion chambers project outside the cylinder. With such an arrangement much more favourable conditions are created even if parallel valves are used. Sharp and insufficiently cooled edges must however be avoided and good scavenging of the remote corners must be ensured. The inlet valve should not project outside the plan view of the cylinder in order to keep the inlet air stream as straight and as cool as possible.

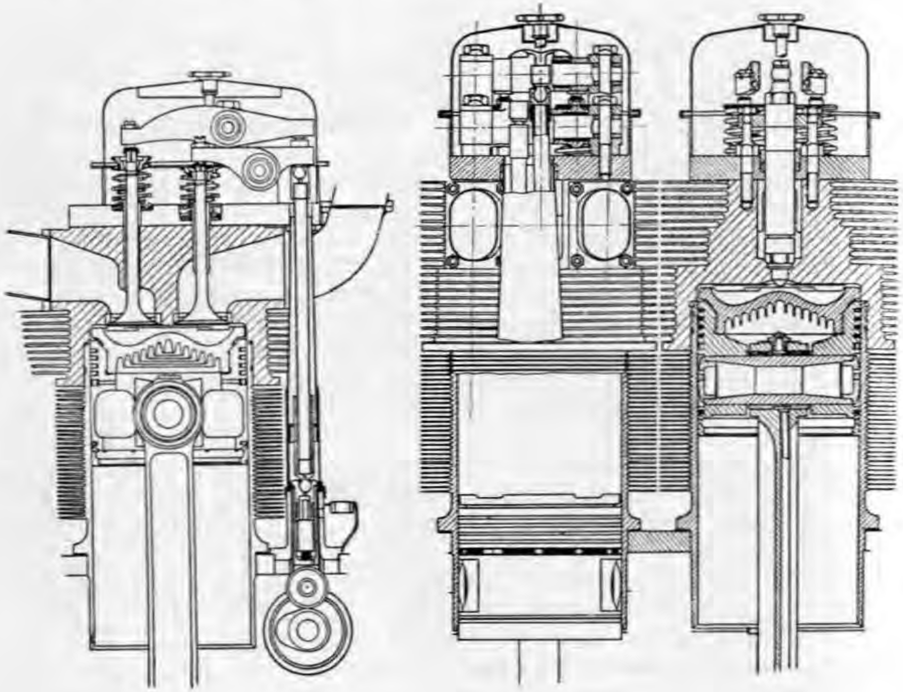


FIG. 201. Four-valve cylinder head of an air-cooled compression-ignition engine of a 140 mm bore.

Wedge-shaped combustion chambers permit a lateral mounting of the sparking plug, thus aiding design and placing the plug in an accessible position.

It is important to watch not only the flow area at the inlet port but also the coefficient of flow which depends on the shape of the inlet manifold.

The shape of the inlet manifold with a hemispherical combustion chamber is very favourable and the flow coefficient is high. Fig. 183 shows the promotion of good air flow characteristics by the hemispherical head layout as the induction elbow behind the valve is well shaped. The cross section varies continuously and bends are slight. This shape of inlet port is favoured by many motor cycle engine designs and makes high mean effective pressures

possible. The dotted line 2 denotes another widely used shape, found in many aircraft engines.

An example of a hemispherical air-cooled head is shown in Fig. 202. Another desirable feature of the hemispherical head is its low rate of carbon deposit resulting in a small performance loss. The valve actuating mechanism is however more complicated and a great deal depends on design details.

The layout shown in Fig. 203 has found widespread application in aircraft engines. The valve rockers are at an angle to the cylinder axis in the plane of the valves, whereas push rods are outside this plane and are disposed radially to the crankshaft axis. The rocker bearings must therefore withstand considerable axial thrust. This disadvantage may be lessened by constructional modifications. The valves are in a plane at right angles to the air stream and their cooling is very good.

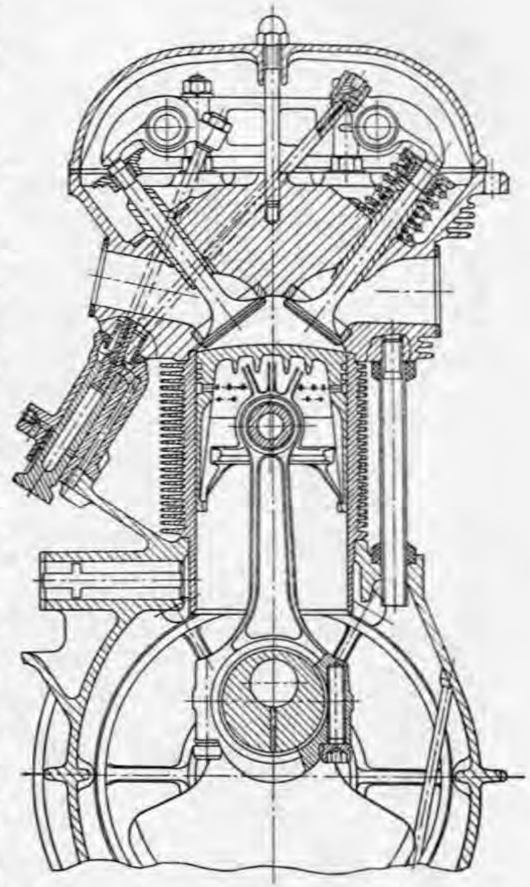


FIG. 202. Hemispherical combustion chamber of the Tatra 910 petrol engine.

Cylinder heads of radial aircraft engines have reached a high degree of perfection and provide excellent cooling for exhaust valves. The finning of the head between both valves is parallel to the cylinder axis, all other fins being at right angles to it. The evolution of an aircraft engine cylinder is shown in Fig. 204. By the constant increase in cylinder fin area the specific output per cylinder has also increased. In the latest Pratt and Whitney 28-cylinder engine the valve gear is arranged in a different manner. (Fig. 205). The massive cylinder head walls and the thin and close finning are worth noting. The removal of heat from cylinder heads of large aircraft engines is becoming difficult and for considerations of cooling, bore is restricted to 150 to 160 mm. Higher power outputs may therefore be obtained only by increasing the number of cylinders. The Pratt and Whitney Company's 28-cylinder engine supplies 3650 b.h.p. at 2800 rev/min. [50]

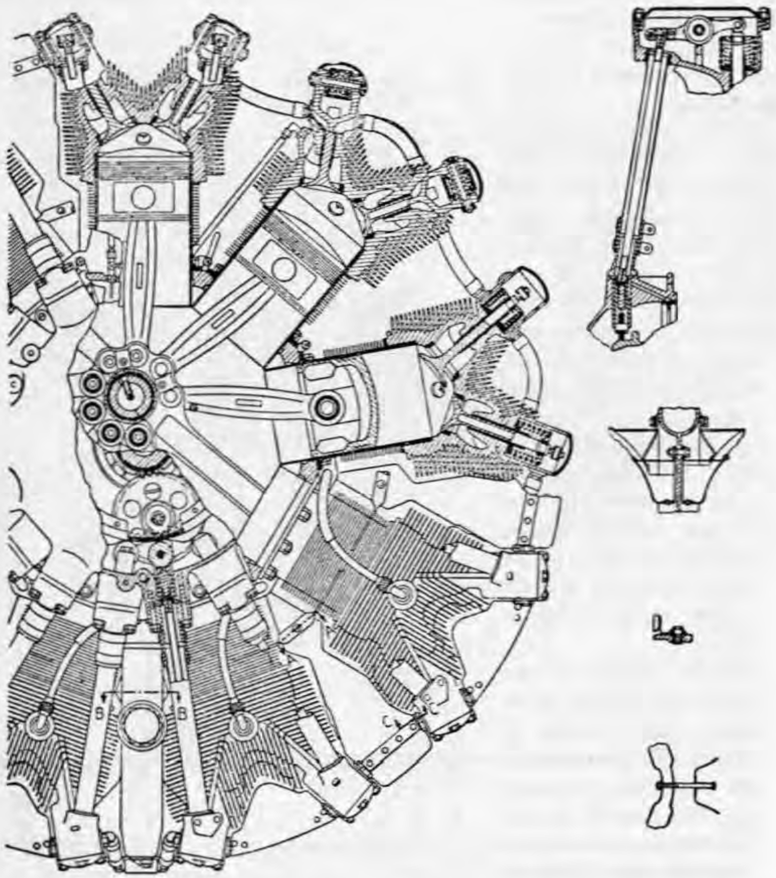


FIG. 203. Transverse section of a radial aircraft engine (Lycoming). Note the valve gear.

The development of an aircraft engine takes from 5 to 8 years; the development of an automobile engine from 3 to 5 years and not even the most painstaking calculations can reduce this period. Various engine parts are exposed to the most complex stresses under operating conditions, e.g. vibrations which

FIG. 204. Development of the cylinder of the Pratt and Whitney engine, bore 146 mm. See Table 38.

cannot be anticipated with any degree of certainly, long-term tests being necessary to show their influence.

In order to reduce the transfer of heat from exhaust gases to the cylinder head, the exhaust port is always kept as short and as straight as possible. Heat transferred from exhaust gases places an unnecessary burden on the cylinder head finning.

Table 38

	Year of manufacture	Performance b.h.p.	Cooling surface area, m²
1	1927	45	0·774
2	1932	67	0·968
3	1940	100	2·000
4	1946	125	2·770

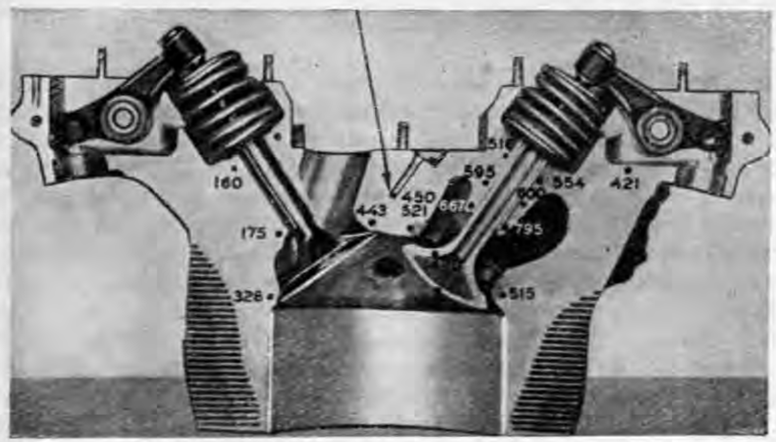


FIG. 205. The new cylinder head of the Pratt and Whitney aircraft engine. Note the wall thickness of the cylinder head. Temperatures in °F.

Experiments have been conducted with stainless steel sheet lining of the exhaust port in such a manner as to leave an air space between the lining and the light alloy head. The temperature of the cylinder head was reduced considerably (as much as 100 °C) by this and the valve guide also operated at a lower temperature. Exhaust valve temperature though, was increased as dissipation to cool port walls was absent.

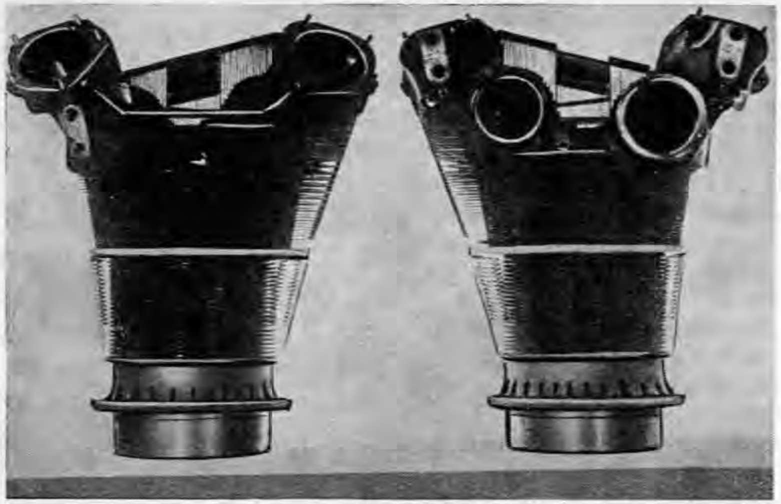


FIG. 206. Cylinder of the Wright aircraft engine. Note the dense finning of the forged cylinder head.

Another example of extensive development is given by the Wright aircraft engine cylinder. With the need of speedy step-up of engine output cast fins no longer filled the bill and forged heads with machined fins went into production. The cooling area was thus greatly increased by using thinner fins and smaller spacing. In the neighbourhood of the exhaust valve the finning was extended in depth without regard to obvious production difficulties. The size of the fins facing forward was reduced in order to reduce air resistance and to prevent unwanted heating of the air before it reaches the hotter parts. The ingenious finning of a forged head is shown in Fig. 206. Note the unequal depth of fins on the inlet and exhaust side of the cylinder head and cylinder. Very thin fins are linked at their extremities in order to prevent fracture through vibration. The outline of the cylinder is without "steps" in order to fit well under a cowling. These modifications brought about a lower power consumption by cooling despite increased performance, the reduced pressure drop over the cylinder resulting in reduced air flow. The power expenditure for cooling of this Wright C 18 engine with a take-off output of 2700 b.h.p.

at 2900 rev/min is shown in Fig. 207 [65] – the values relevant to the cast head being marked in dotted line, those for the forged head and machined fins in full line. The reduction in power consumption for cooling purposes is substantial in this case and proves the importance of finning. The cylinder bore of the engine is 155·6 mm, stroke 160·3 mm, number of cylinders 18.

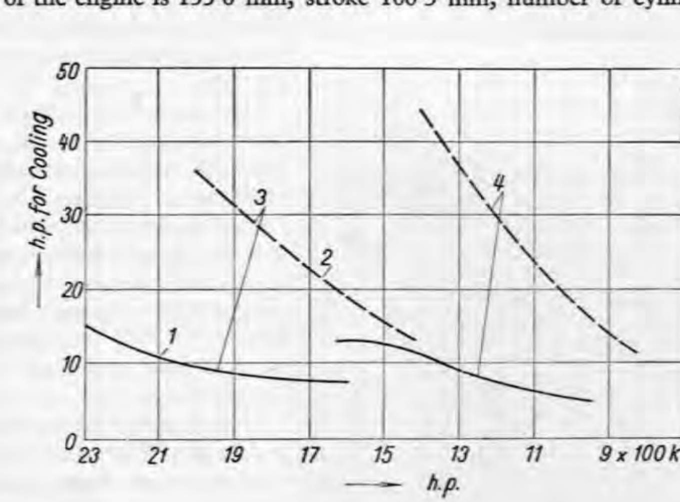


FIG. 207. Cooling input of the Wright aircraft engine with forged (*1*) and cast (*2*) cylinder head, operated with rich (*3*) and lean (*4*) mixture. Note the considerable reduction of input for cooling the forged cylinder head.

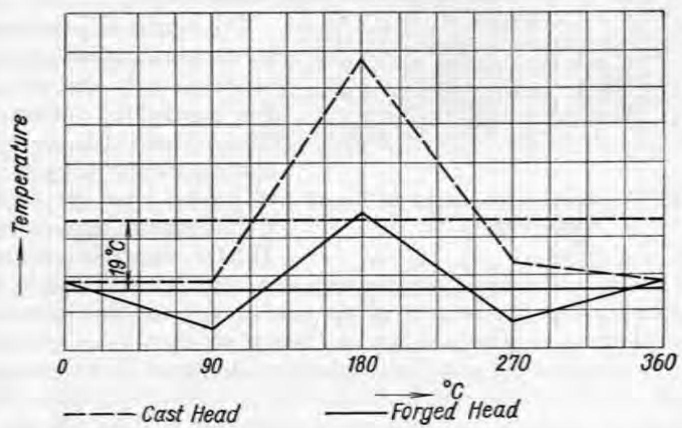


FIG. 208. Reduction of temperature on the circumference of the valve seat in a forged (full line) and cast (dash line) cylinder head of the Wright aircraft engine.

Great difficulty was experienced with the cast cylinder head with exhaust valve leakage caused by uneven temperature distribution over the circumference of its seat. A more even temperature distribution was reached by the use of closer spaced finning on the forged head and by raising the exhaust stub which made possible an increase in the number of fins in the critical position. The resultant reduction of mean valve seat temperature and its better distribution is shown in Fig. 208.

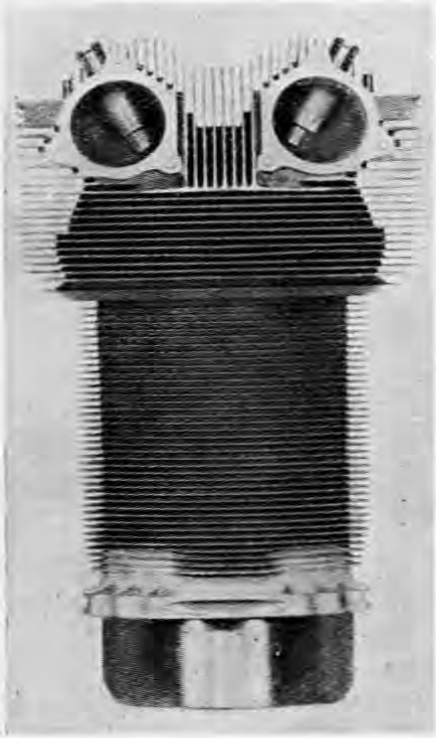


FIG. 209. The four-valve cylinder of the Bristol aircraft engine.

With a pent-roof compression space and four valves, the valve seat flow area amounts to 32% of piston crown area. The only aircraft engines to use this layout are the poppet valve types of the Bristol Aeroplane Co., designated Mercury and Pegasus (Fig. 209). The valve arrangement of this head is shown in Fig. 210.

Among motor cycle engines the Rudge Whitworth 500 c.c. engine uses the pent roof head with four valves. The actuating gear used for the four radially disposed valves of the 250 c.c. engine of Rudge manufacture is interesting; six rockers are needed (Fig. 211).

The second requirement on a good air-cooled engine cylinder head design is that of a large flow area for the cooling air. To remove heat from the cylinder head, a certain quantity of air is needed and this must be forced through between the fins. The flow area must therefore be such as to enable the air to pass through at a high enough velocity. It goes without saying that the velocity of air passing over the fins should be as nearly constant as possible and that the flow of air should not be hindered by narrow passages. At such hindrances the velocity of the air stream must be increased considerably and to do this a large pressure difference is necessary. Every change in flow area and thereby also in flow velocity is the cause of increased pressure loss. The requirement of minimum interstice variation between fins is well met in radial aircraft engines, where all cylinders stand unobstructed and the height of fins is constant over the entire circumference. Conditions are much less favourable in the case of in-line engines where the small centre-to-centre distance between adjacent cylinders limits the height of fins in one direction. The height of lateral finning on both sides of the combustion chamber is particularly restricted.

With parallel valves in a common plane with the crankshaft axis, flow area through the head finning is severely restricted. If exhaust port channels point outwards and a common inlet manifold is used for two adjacent cylinders, only a narrow slot remains for the passage of air between the valves and past the sides of the combustion chamber (Fig. 212).

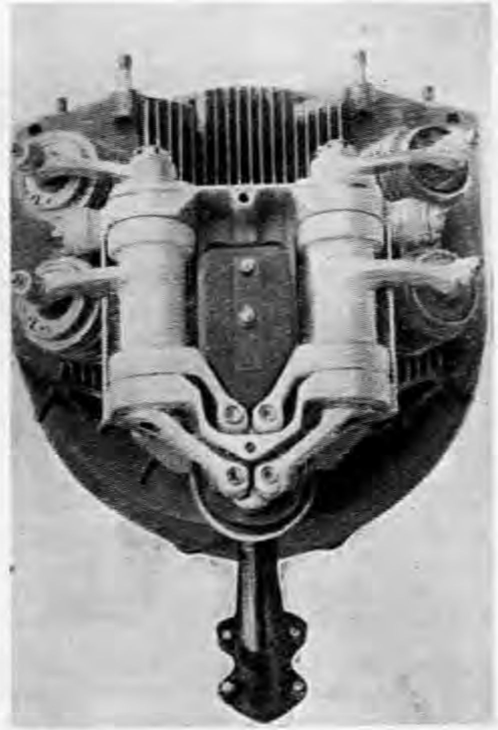


FIG. 210. Valve gear of the Bristol aircraft engine.

A substantial improvement to the VW head is effected by the modification illustrated in Fig. 213 showing the arrangement of the cylinder heads on the Porsche sports car engine. By inclination of valves not only was a more favourable combustion chamber geometry obtained and valve size increased, but the cooling air flow area between both valves was also increased. By abandoning the common inlet manifold, further free area was gained between the two cylinders. Improved cooling made possible a considerable improvement in output per litre. Cooling is satisfactory, despite increased bore, with the cylinder spacing unchanged.

Flow area is increased by turning the ports to be parallel to the direction of air flow. An example of this is given by the Argus aircraft engine cylinder head shown in Fig. 214. Notable are its short induction pipes in the cylinder head.

The single vertical rib connecting fins assists in distributing heat into the horizontal fins and also transmits the load of the rocker pivot bearing spindle direct to the cylinder head. The outline of the cylinder head is unsymmetric and the exhaust side fins are higher. Every second fin in the space between the valves is omitted in order to strengthen the sand core. Fig. 215 shows a similar arrangement of the Walter – Minor aero engine head.

Spacing of cylinders in flat engines is determined by the design of the crankshaft and is therefore relatively large. Nevertheless, it has been found desirable to incline the plane of the valves in relation to the crankshaft centre plane in order to obtain better cooling air flow and shorter ports. This arrangement has been applied to the Tatra-plan engine, shown in section in Fig. 103. Both valves are, furthermore, inclined making the use of a hemispherical combustion chamber possible. Owing to special rocker layout, the valves may be actuated by push rods leading from a camshaft located in the crankcase. The camshaft being beneath the crankshaft, push rod tunnels can be used as oil return pipes from the far side of the cylinder heads. For the lubrication of rockers, oil is delivered through the hollow push rods. Cylinder heads are fastened direct to the crankcase by four long bolts.

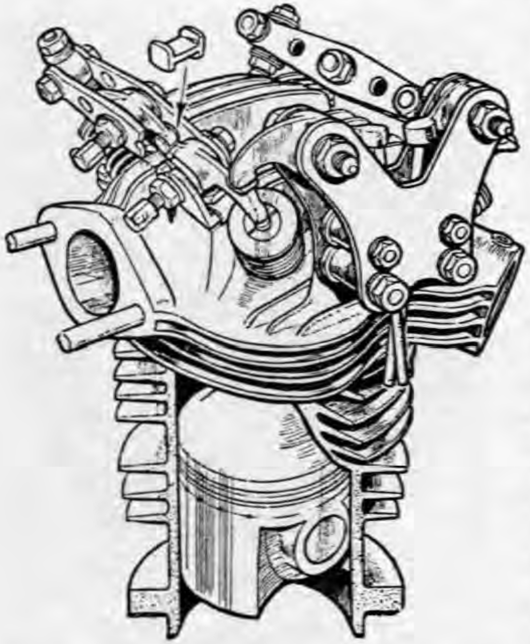


FIG. 211. Actuation of four radial valves in the cylinder head with hemispherical combustion chamber of a motor-cycle engine.

A different finning pattern is used for the in-line Deutz compression-ignition engine with parallel valves (Figs. 216, 217). In order to enable cooling air to penetrate to the centre of the head between valves, both ducts turn to the outside of the head immeditely behind the ports thus leaving the middle section free for vertical fins. Both ducts have a vertically placed rectangular cross section in order to provide enough space between manifolds for the passage of air. The opposite side of the head houses an injection chamber so that a space between vertical fins would in fact form a blind passage preventing the flow of air. Vertical finning is therefore retained only for the middle section of the head, the side fins being horizontal (Fig. 192). Fig. 192 is more instructive, as the ingenious solution of the problem of cooling this head can be seen.

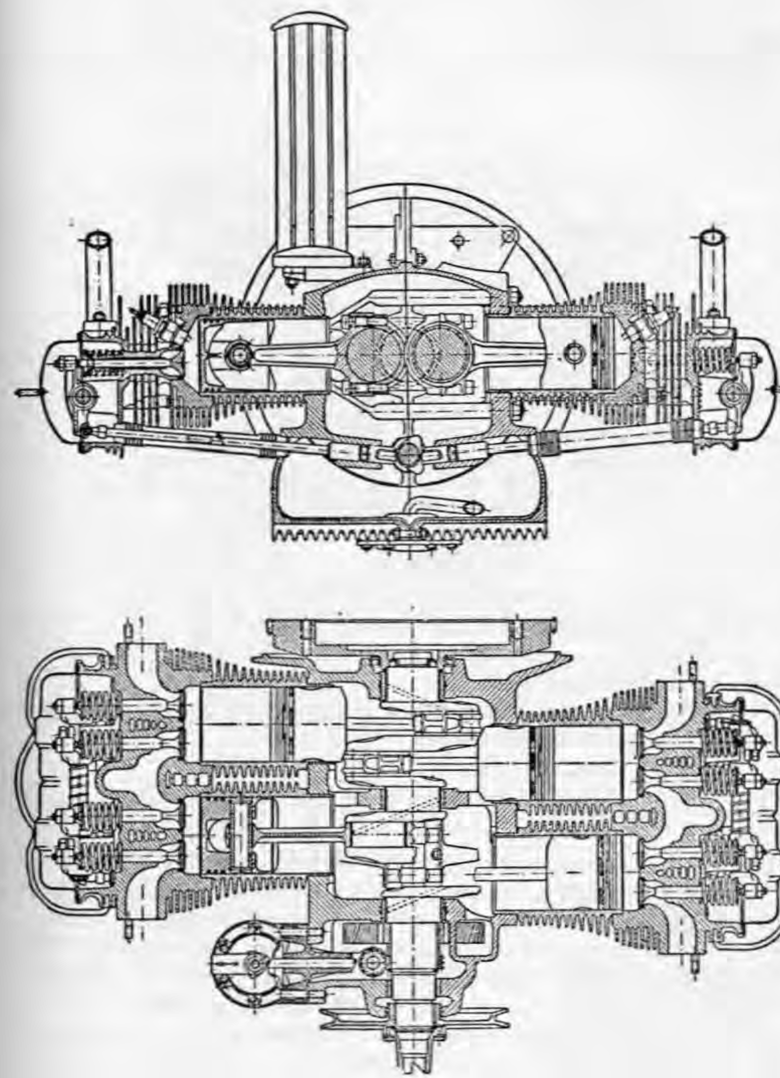


FIG. 212. Transverse and longitudinal section of the VW engine. Note the small air transition area shown in the longitudinal section.

Cold air reaches the head from the injection chamber side and enters between the horizontal fins, between which it streams over both sides of the head past the outer sides of the inlet and exhaust ports to the other side of the engine. Air enters the area between both ports above the injection chamber around the injector boss formed in the bottom of the rocker box. This cold air stream first enters between the vertical fins projecting from the crown of the com-

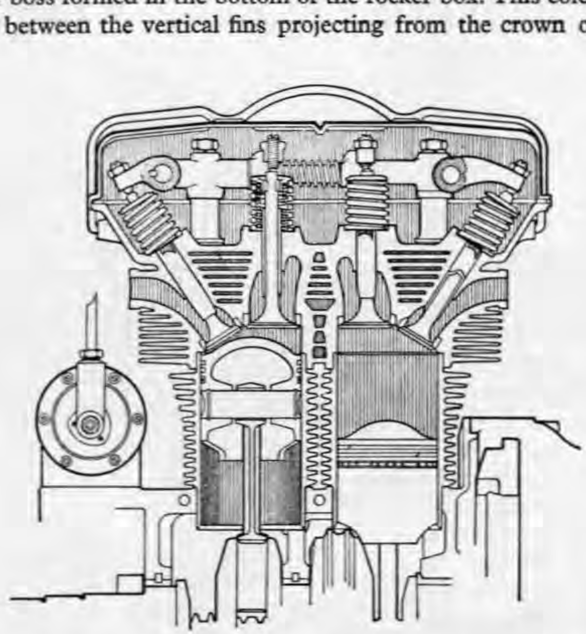


FIG. 213. Cylinder head of the air-cooled Porsche engine. Note the enlarged cooling air passage and larger intake valve.

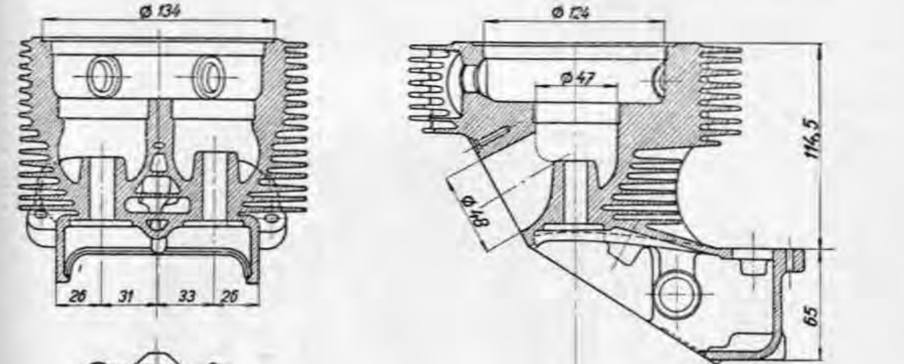
bustion chamber and flows on between the horizontal fins separating both ports.

Vertical fins are convenient for cooling for they transfer heat direct to the air stream. When vertical fins are used, a thick port wall or vertical rib has to be used to assist heat flow from the combustion chamber walls to fins. Figs. 218, 219 show two arrangements used for the Tatraplan engine finning. With vertical finning, not only do the fins project from the combustion chamber crown itself, but cooling air is also made to flow between the valves.

The largest flow area may be attained with the plane of both valves normal to the plane of the crankshaft axis. Fins may then be formed either horizontally (Fig. 220), or obliquely (Fig. 221).

With horizontal fins it is beneficial to assist heat flow to all fins by a thick vertical rib in the plane of the ports. This arrangement is very convenient for moulding and such heads can easily be cast into steel moulds. Casting of

heads into steel moulds is highly advantageous, for precise castings with a level and straight surface of the fins are produced. Furthermore casting time including preparation is greatly reduced and compact castings are obtained.



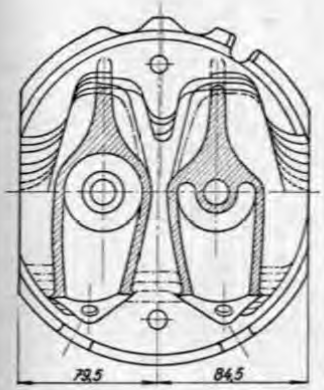


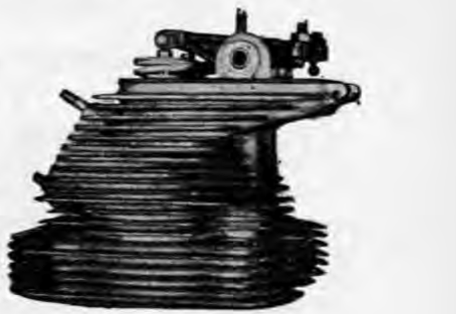
FIG. 214. Cylinder head of the Argus As 10 C aircraft engine.

A diagrammatic drawing of the casting method is shown in Fig. 222.

The case of oblique fins is similar, but more fins project from the combustion chamber wall. Compared with the horizontal disposition of fins previously described, oblique finning has one disadvantage as the uppermost fin cannot be used to conduct the air and a separate sheet metal casing must be provided for the upper part of the cylinder head. (The fins must not form any blind passages).

The evolution of the Tatra 111 oil engine cylinder head is shown in Figs. 223 and 224. On the first type, the lower portion of the head was finned hori-

FIG. 215. Cylinder head of the Walter Minor aircraft engine. Note the thick fin for transmitting forces from the rocker arm to the cylinder head. →



zontally, the upper obliquely. The cylinder head studs are screwed into threads formed in the lower part of the cylinder head. A large area of fins had to be cut away above the stud holes for access during assembly. The second type (upper right) shows modifications to the upper part of the head.

The third type, shown (upper left) eventually went into production with all fins oblique and cylinder head bolts bearing upon the upper machined surface

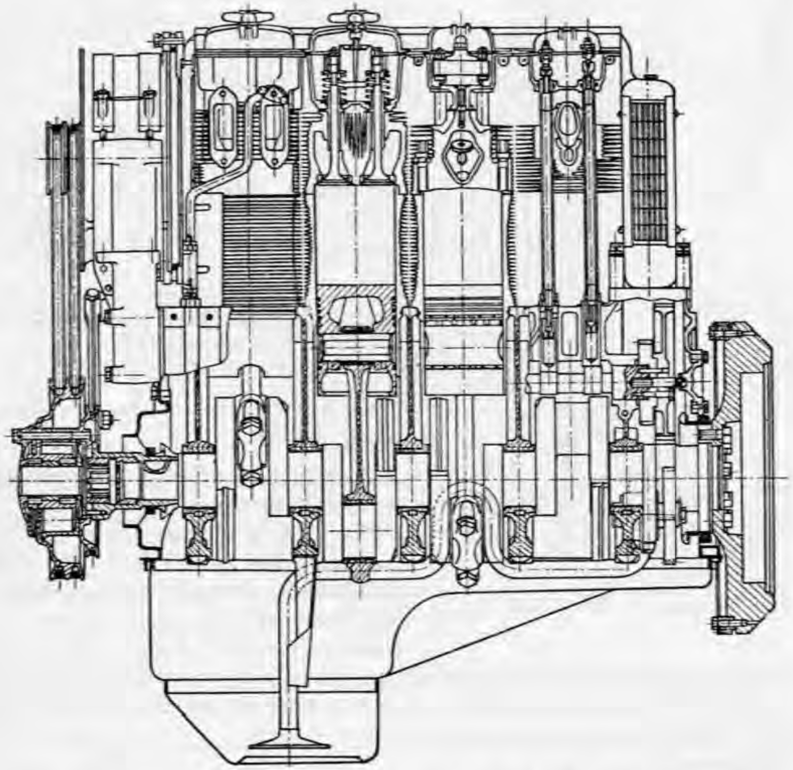


FIG. 216. Longitudinal section of the Deutz F-4-L-514 engine.

of the head. By this, space was saved for finning as no cutaway was needed for box spanners. The bolt holes are drilled in the fins so that no separate machining of the seatings was required. The pressure exerted by the bolts is communicated to the centre of the head by massive fins, thus partly eliminating deformation of the head through mechanical stresses. The bolts must be elastic and reduced in diameter considerably, otherwise thermal expansion of the head would increase the axial forces in the bolts excessively. The injector is in all

these types housed in a boss formed in the thick central fin. The inlet port extends to the upper plane of the head in order to simplify the shape of the cowling. In this instance air flows past the inlet valve to the exhaust. The rocker pivots are housed in a separate rocker box screwed to the cylinder head. An inspection cover is provided for access to the valves for clearance adjustment.

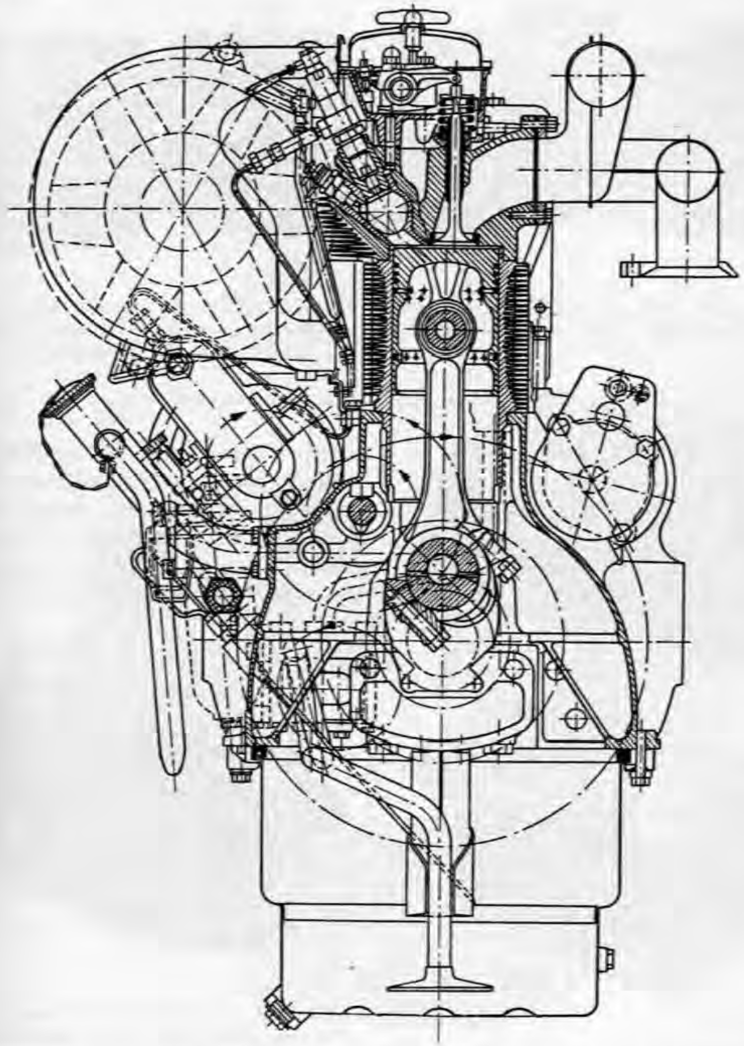


FIG. 217. Transverse section of the Deutz F-4-L-514 engine.

Fig. 225 shows the further development of Tatra compression-ignition engines. The hemispherical combustion chamber does make for high volumetric efficiency but it does not provide sufficient swirl during the induction

FIG. 218. Two types of finning of the cylinder head of the Tatraplan engine.

FIG. 219. Two differently finned cylinder heads of the Tatraplan engine viewed from the spark plug side.

stroke. Besides this, the combustion chamber surface area is large and air compressed during the compression stroke loses its heat rapidly. For this reason the valves were set in an almost vertical position, the ports slightly

FIG. 220. Iron mould casting of the aluminium cylinder head of the Tatra 603 engine.

FIG. 221. Cylinder head of the Tatra 603 engine with inclined finning. Note the poorer finning in the region of the spark plug.

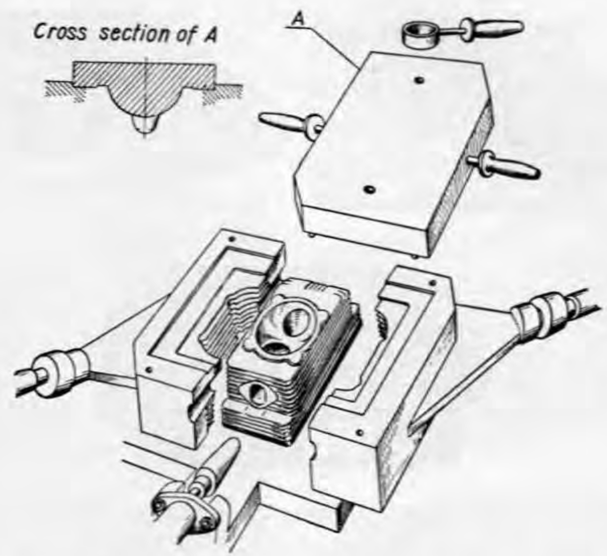


FIG. 222. Illustration of casting cylinder heads into iron moulds.

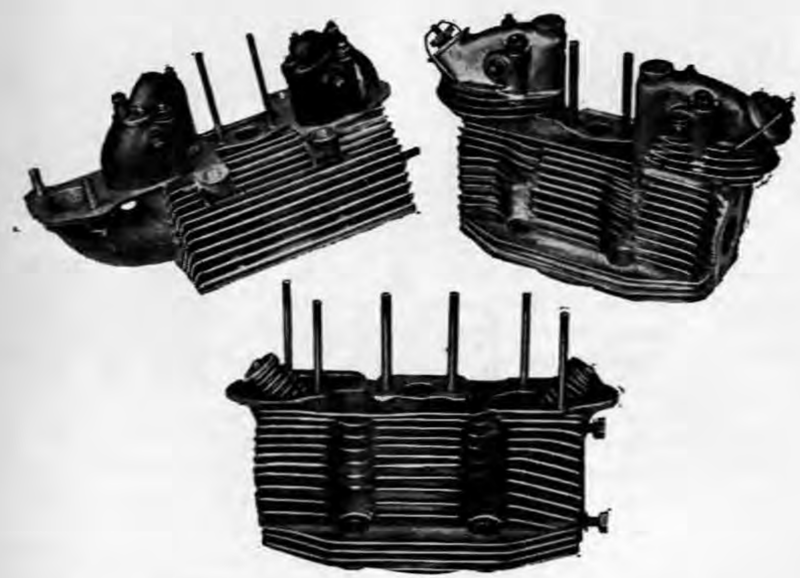


FIG. 223. Development of the Tatra 111 cylinder head (top view).

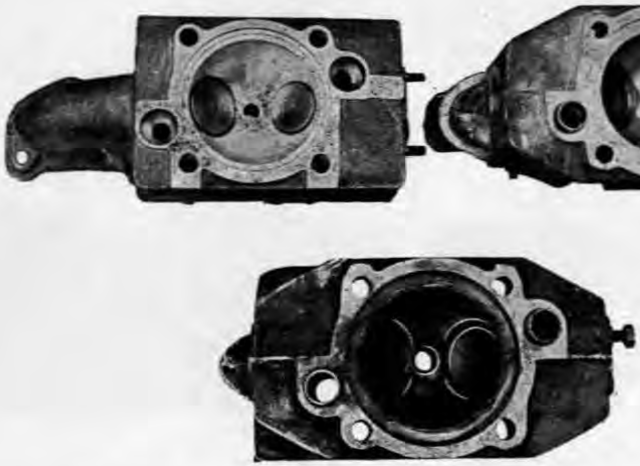


FIG. 224. Development of the Tatra 111 cylinder head (bottom view).

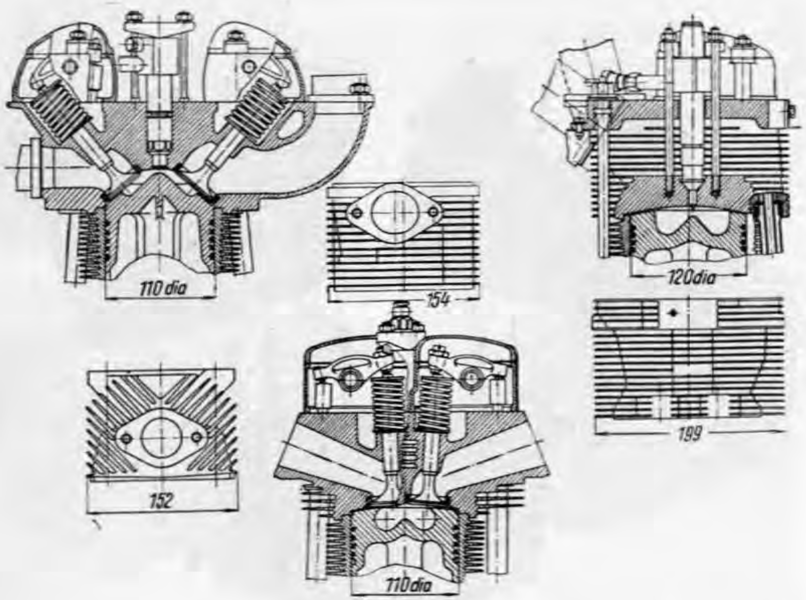


FIG. 225. Development of the cylinder head of the Tatra compression-ignition engine.

inclined away from the cylinder head axis and tangential entry of air was thus ensured. A very compact combustion chamber was formed inside the piston crown and its narrow orifice caused high turbulence during the compression stroke. Traces of varnish provided definite proof of dual turbulence during compression (Fig. 226).

FIG. 226. Double compression turbulence in the toroidal injection chamber in the piston of the Tatra compression-ignition engine.

FIG. 227. Cylinder head of the Tatra compression-ignition engines, series of 110 mm bore, new design.

The flow of cooling air was reversed in this engine, fresh air reaching the exhaust side first; this resulted in a more uniform temperature distribution through the head. In order to make the head rigid and free from vibration, lugs for the fastening bolts were cast on from the upper plate to the combustion chamber. The section between the valves was thus freed for the passage of air and the injector was therefore efficiently cooled. The lugs however presented problems in moulding and therefore a separate plate was formed for each space between the fins and these plates were then superposed (Fig. 227).

For flat engines with 120 mm bore a layout was used in which both manifolds were placed on one side and only one camshaft was employed. The head was fastened by six screws. Because cylinders were spaced far enough apart, the provision of sufficient finning with a large enough flow area presented no difficulty.

A problem in the design of air-cooled engines is the disposition of the exhaust pipe in such a manner as to lead it out of the cooling air stream in the shortest and straightest possible way in order to reduce to a minimum the heat exchange between exhaust gases and the cylinder head. With a short exhaust pipe and the usual type of finning, the pipe emerges into the cooling air duct. In order to prevent warming of the air, particularly where it is directed against the

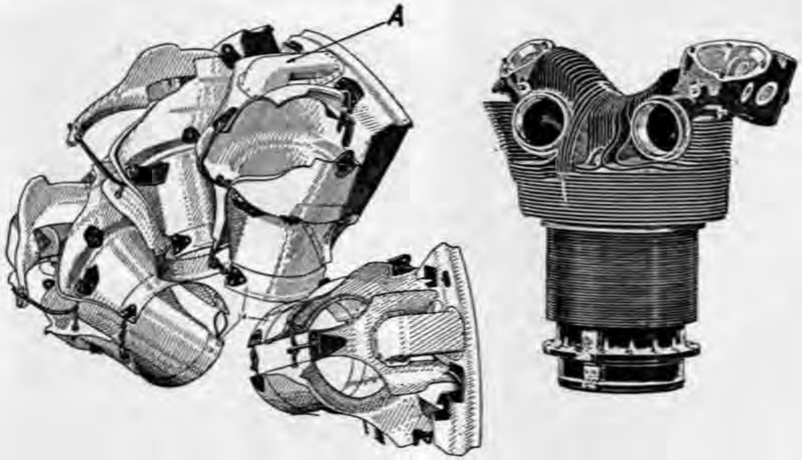


FIG. 228. Cowling of the BMW 801 aircraft engine. Note the special duct A for conveying cool air to the region of the spark plug.

exhaust side of the cylinder head first, the exhaust pipe must leave the duct by the shortest possible route. At the same time the access of cooling air to the cylinder head must not be impaired. This often leads to complex shapes of the cooling air ducts and to difficult assembly.

These difficulties are overcome by the use of a finning layout shown in Fig. 102, in which the exhaust pipe is short and its curvature slight. The upper cooling fin in this design forms a good duct for the air stream and the remaining sheet metal cowling is very simple. Provision is made for extensive finning of the exhaust valve region and the coefficient of heat transmission for this spot is further increased by the curved air stream. The air flowing between the upper cooling fins, which is usually not put to efficient use, comes into direct contact with the exhaust pipe and thus aids in the cooling of the cylinder head. Such a design facilitates the casting of the head in a steel mould.

A separate channel is used to direct a fresh air stream to points subjected to extreme heating in aircraft engine cylinder heads. The illustration of the BMW 801 engine (Fig. 228) shows an example of this. The vertical fins between the valves are interrupted by a fin in the plane of the valves. Air flowing from the front escapes upwards in front of the vertical fin through

sheet metal louvres. A special duct A leads air behind the baffle fin to the sparking plug and between exhaust and inlet valve ports. The illustration also shows the method of attaching the air ducting cowls.

As already stated, a smooth fin surface is desirable. Casting in steel moulds meets this demand by making it possible to produce smooth and thin fins. Fig. 229 shows the influence of the thickness of fins and the quality of their surface upon the amount of air passing between the fins at a given pressure. By reducing fin thickness and polishing their surfaces, the amount of air increased by about 27 %.

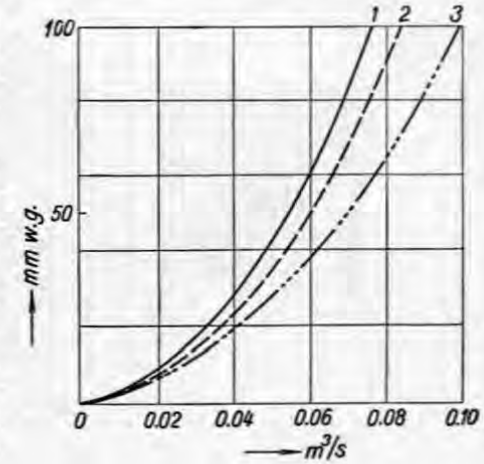


FIG. 229. Influence of surface quality and thickness of fins upon air flow at the static pressure head p_{st}.

1 — Rough heads, thick fins, *2* — rough heads, thin fins, *3* — rough heads, thin smooth fins.

Aluminium alloys are generally used for air-cooled cylinder manufacture. Cast iron is suitable only for engines working under low loads or for motor cycle engines with separately exposed cylinders of small displacement and a large area of fins. Even motor cycle engines are increasingly being fitted with light alloy heads. The type of finning most used today has a spacing of 6 to 5 mm, height of 40 mm and is 2 to 3 mm thick at the base and 1·2 to 1·5 mm at the top. Aircraft engine cylinders are cast with fins up to 80 mm high, 1·6 to 2 mm thick and up to 2·5 mm apart. The casting of such heads is very laborious and a large percentage of waste occurs. But not even such finning is sufficient for highly stressed aircraft engines and recourse has to be made to the use of forged aluminium heads with milled fins. An example of such a head with machined fins is given in Fig. 206.

4. Valve gear arrangement

The simplest form of valve actuation is that used in side-valve designs with the valves disposed beside the cylinder. Such an arrangement is used on the Soviet M-72 motor cycle engine, for instance, a cross section of which is shown in Fig. 230. The valves are slightly inclined to the plane of the cylinder in order to place the camshaft out of the way of the crankshaft and also to obtain a more favourable combustion chamber shape and to facilitate the entry of air. The head of this engine is extremely simple, containing only a plug thread insert. The valves must be set at a sufficient distance from the cylinder for good access of air to the whole periphery of the exhaust valve seat.

The simplest arrangement of overhead valves is in line with the longitudinal axis of the engine and with the camshaft housed in the crankcase.

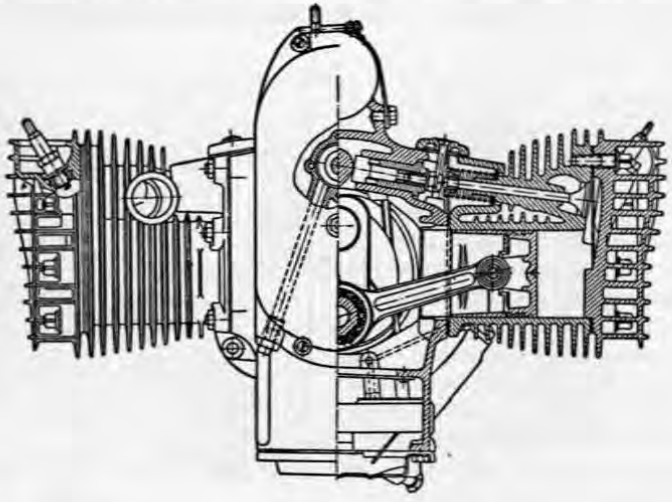


FIG. 230. Transverse section of the Soviet M-72 motor-cycle engine.

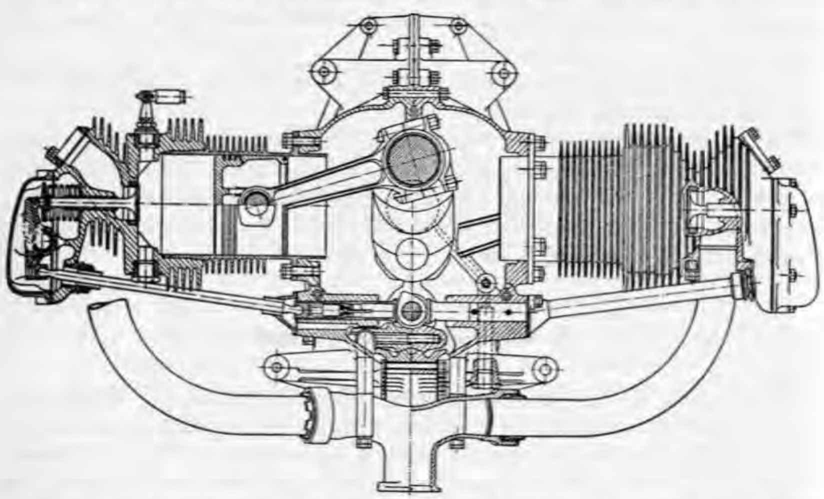


FIG. 231. Transverse section of the four-cylinder Continental aircraft engine, bore 98.4 mm, stroke 95.25 mm. Note the hydraulic tappets for the automatic maintenance of valve clearance.

Such a layout is shown in Fig. 231. Reliable sealing of push rod covers is important in view of the expansion or contraction of the cylinder barrel at varying loads. The seals used must therefore permit a certain amount of movement. The VW design makes use of spring bellows at the ends of the push rod cover tubes. In most designs however, synthetic rubber sealing rings, which permit a certain resiliency of the union and which stand up to high temperatures,

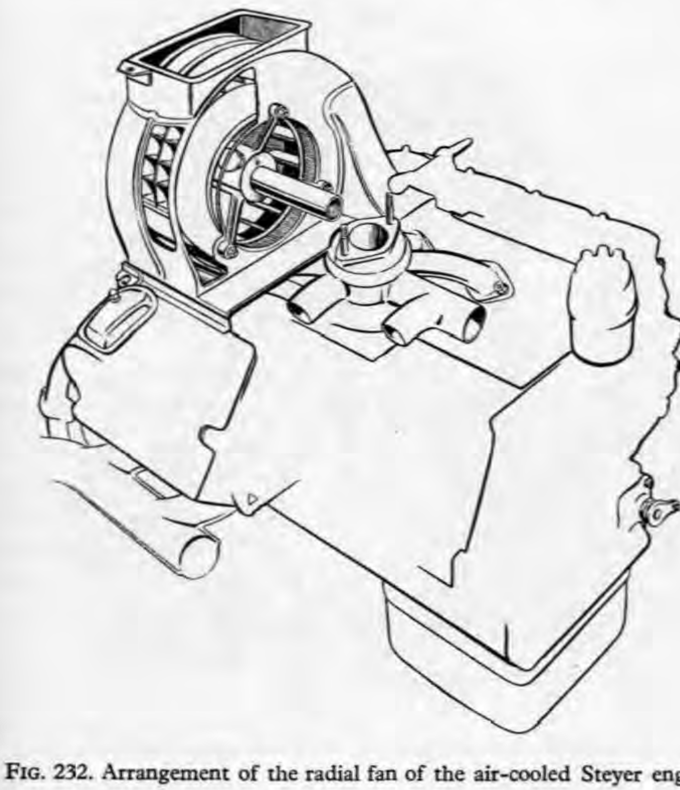


FIG. 232. Arrangement of the radial fan of the air-cooled Steyer engine.

are used. If the temperature of the head is too high for synthetic rubber, the push rod covers may be threaded or pressed into the cylinder head. (Fig. 232.)

Valves positioned in a plane perpendicular to the crankshaft axis require a more complex actuating mechanism. If the plane of the valves is not quite normal to the plane of the crankshaft, the arrangement used in the Tatraplan engine (Fig. 103) or that of the Argus As 410 (Fig. 195) may be found convenient. The plug position of such a layout is inclined and outside the rocker box. With parallel valves, rockers of unequal arm lengths can be used on a common pivot shaft, such as in the head shown in Fig. 102.

With a hemispherical cylinder head with the plane of the valves at right angles to the crankshaft axis, it is convenient to place the camshaft as high as the design permits. The arrangement shown in Fig. 201 may then be used; it is especially suited for V engines and its weight is low.

With closely spaced cylinder barrels, the design of a cylinder head of such a type needs very diligent attention by the designer. Access to the sparking

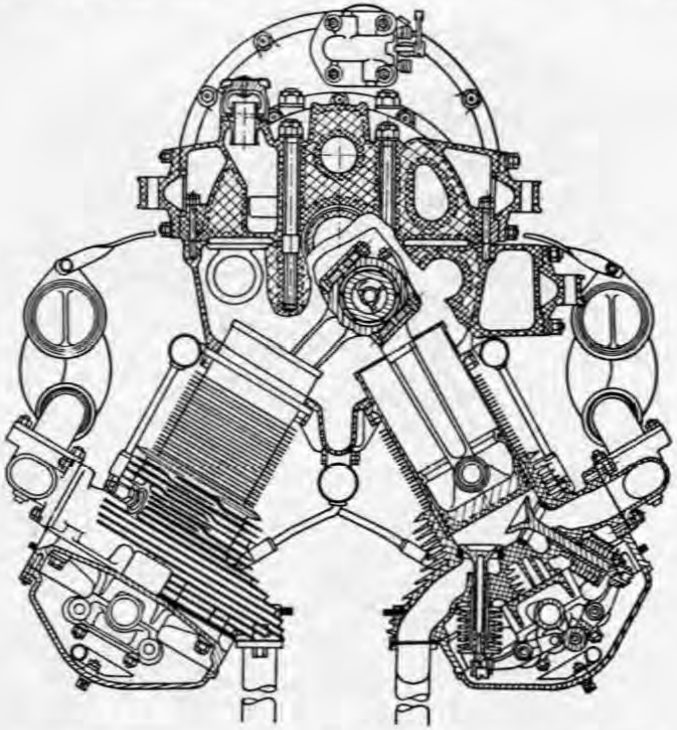


FIG. 233. Transverse section of the inverted Ranger aircraft engine.

plug can be arranged by using a tube threaded into the cylinder head and passing through the cylinder head cover. A separate oil return pipe is needed on the underside of the cylinder.

Overhead camshaft drive eliminates the variation of valve clearance owing to the expansion of heated cylinders. The Tatra 87 arrangement shown in Fig. 384 may be taken as an example of this. The overhead camshaft common for all cylinders of one bank is mounted in its own housing and the drive is transmitted from the crankshaft through a roller chain. The camshaft box mounting on the cylinder heads must be resilient enough to provide for variation in the lengths of individual cylinders which can be caused when one

cylinder misfires (thereby remaining cool) and which would otherwise cause deformation of the camshaft housing or even seizure of the camshaft bearings. The camshaft drive is shown in detail in Fig. 383.

Figs. 233, 234 show the camshaft drive of the Ranger inverted aero engine. The camshaft case is common for a bank of cylinders. The finning of the exhaust valve guide boss is notable. The valve guide inside the boss extends a long way toward the valve head protecting the valve stem from flame. Sparking plugs are disposed laterally. The dismantling of separate cylinder heads in overhead camshaft engines is less convenient.

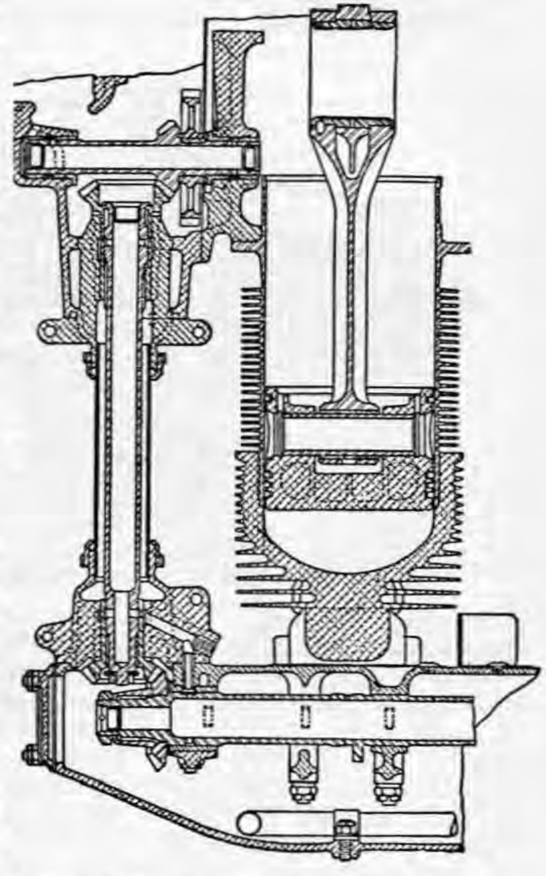


FIG. 234. Camshaft drive of the Ranger aircraft engine.

5. The gas seal between the cylinder barrel and head

All cylinder heads of automobile engines are detachable but the joint between the cylinder barrel and head must be sealed perfectly. When the joint is formed between an aluminium head and a cast iron barrel, no gasket need be used. For the seal between a cast iron head and a cast iron barrel copper sheet or copper-asbestos gasket material is used.

In order to provide a reliable gas seal, the pressure of the surfaces in contact must be sufficient and evenly distributed. Too high a joint pressure leads to permanent deformation and to damage to the seating, the area of which should amount to 20 to 25% of piston crown area in the case of petrol engines and 35 to 40% in the case of compression-ignition engines. Local temperature increases lead to loss of strength in the material and to seating damage.

In order to ensure an even distribution of stress over the whole seating area, the spacing of the retaining bolts on the periphery of the cylinder barrel must be regular. Four bolts are usually sufficient for bores up to 110 to 120 mm. If the cylinders are held in place by bolts from crankcase to the heads, bolts with a reduced spindle diameter must be used to maintain constant stress during thermal expansion. In Tatra oil engines, for example, the

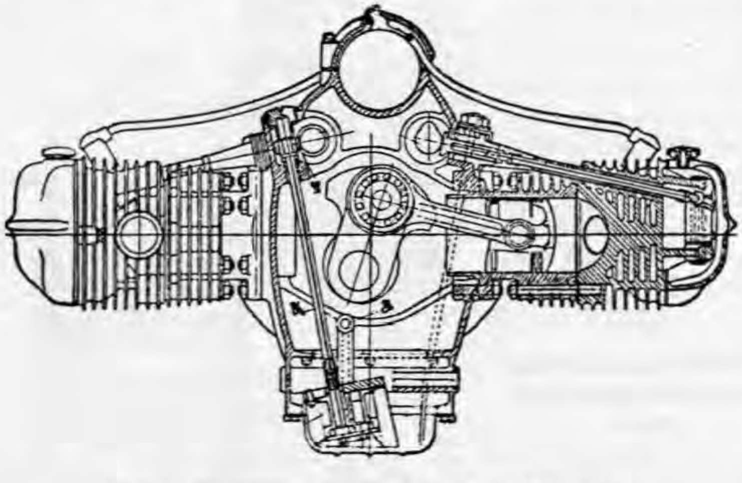


FIG. 235. Transverse section of the BMW R 51 engine.

diameter of bolts with an M 16 (16 mm thread) is reduced to 10 mm. Local overheating of any portion may lead to distortion and blow-by so that the adjacent parts of any two cylinders must be given due attention.

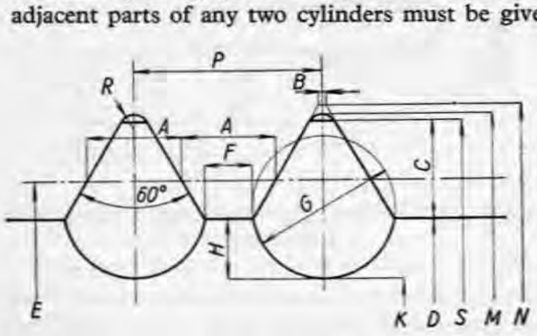


FIG. 236. Special bronze wire for connecting cylinders to heads.
$A = 0.5\,P$, $B = P/24$, $R = 0.072\,P$, $C = 0.525\,P$. $D =$ = basic diameter, $E = D + 0.4\,P$, $G = 0.75\,P$, $H = 0.3\,P$, $S = D + 1.05\,P$, $M = D + 1.122\,P$, etc.

Attachment of the cylinder head to the upper cylinder flange by bolts or studs and the transmission of stress from the cylinder head to the crankcase through the cylinder walls carries some disadvantages. Not only is such an arrangement heavier, but upper cylinder cooling is also impaired. It is rarely possible to thread the studs into the cylinder flange and to pass them through

the cylinder head, for they would obstruct the inlet and exhaust ports and general design and moulding are unfavourably affected. With the nuts located under the upper cylinder flange, fins have to be cut away in the very place where they are most needed. The reduced cooling of the upper cylinder can cause gumming of piston rings. In order to prevent this, long bosses are sometimes formed below the flange and the nuts are then placed in such a location, where local cooling is less important. An example of this is given by the BMW R 51 engine (Fig. 235). The cylinder base flange must be of great thickness in order to prevent deformation.

If the cylinder walls of a barrel clamped between the head and crankcase are thin, uneven bolt loads may even cause piston deformation and piston seizure. The bolts must therefore be tightened evenly or their load must be checked by a torque wrench.

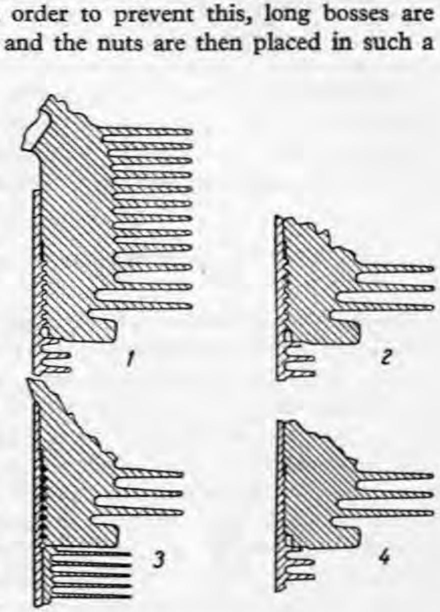


FIG. 237. Several examples of thread connection between cylinder and head.
1 — Thread with 60° angle, *2* — stub thread, *3* — special wire according to Fig. 236, *4* — fine thread.

The head-to-cylinder seal of large aircraft engines proved a source of trouble and for this reason steel cylinders, screwed onto cylinder heads came into general use. The cylinder is screwed into the heated cylinder head and can never be separated from it again. The bolt holes are drilled into the cylinder flange after threading it into the head as otherwise they would not mate. Before the cylinder is screwed on, the head is heated to 300 to 350 °C and the cylinder barrel may also be cooled. The two parts are joined by a special high-strength thread or a special coil of bronze wire is inserted between the steel barrel and aluminium head, shaped as shown in Fig. 236. This wire acts as a packing preventing the threads from working loose.

When the cylinder head is heated during running, its pressure on the upper cylinder is relieved and the upper part of the cylinder expands. This had been provided for in the manufacture of the cylinder, the bore of which is not cylindrical, but bevelled by 0·3 to 0·45 mm in its upper reaches. Such a cylinder cannot be rebored and must be replaced by a new one.

The cylinder is machined so as to ensure its true cylindrical shape at operating temperature. The reduction in diameter in the pressed-in part of the cylinder is still apparent some 50 mm below the base of the cylinder head. After heating the bore is exactly cylindrical. The thread in the head must be formed in such a manner as to bed down perfectly with the head hot. Owing to the difference in the rate of expansion of aluminium and steel, the thread pitch at normal temperature must be smaller in the head than in the cylinder. If the thread pitch were the same on both parts, the thread would engage perfectly when hot only in its few upper turns, the middle turns being relieved and the lower turns bearing against the opposite side. Several illustrations of cylinder head to cylinder joints are shown in Fig. 237.

The thread base should be rounded off in order to prevent a concentration of strain. The relative thickness of the steel cylinder wall and the aluminium alloy head should be at least 1 : 4·5. The strength of both materials at working temperatures must always be taken as a basis.

6. Temperature measurements

Mention must also be made of temperature measuring which is specially important in the case of the air-cooled engine. Methods used differ with the aim in view. Very often, for example, the aim does not lie in determining the absolute temperature but its relative distribution. Such a case arises when determining the suitability of the finning layout and the utilization of single fins, when the information sought relates to the temperature of individual fins, the distribution of temperature over the cylinder head surface and the plotting of isotherms. Surface temperature can be measured readily by the use of coatings which change in colour at a certain temperature. Thus the boundaries of areas of two or more differing temperatures may be seen over the whole surface. This cannot be attained by any other accurate temperature measuring methods. The use of such means has its advantages for gaining information quickly. "Thermocolor" colours are most widely used and they show the variation of one single or several temperatures by a change in colour. The most used of these colours are given in Fig. 238.

The use of Thermocolor does not give precise results, the variation being ± 5 °C. If the colour is exposed to temperature for a long time, it begins to change at lower temperatures. According to the change in colour shown in Fig. 238, the given temperatures are arrived at after 30 minutes heating. If heat is applied for 120 minutes, the change of colour will take place at a temperature some 10 °C lower and with Thermocolor Nos. 6, 7 and 8 even at temperatures 25°, 30° and 50 °C lower.

Fig. 238 shows the distribution of temperatures over the surface of an aircraft engine cylinder as shown by Thermocolor. On the exhaust side of the cylinder head Thermocolor No. 7 was applied, on the inlet side Thermocolor No. 30 and on the cylinder barrel Thermocolor No. 4. While not being very exact this method provides a good survey of temperature distribution on the surface.

It will be realized that Thermocolor cannot be used for precision measuring in the same manner as e.g. thermocouples. Its great advantage lies in the

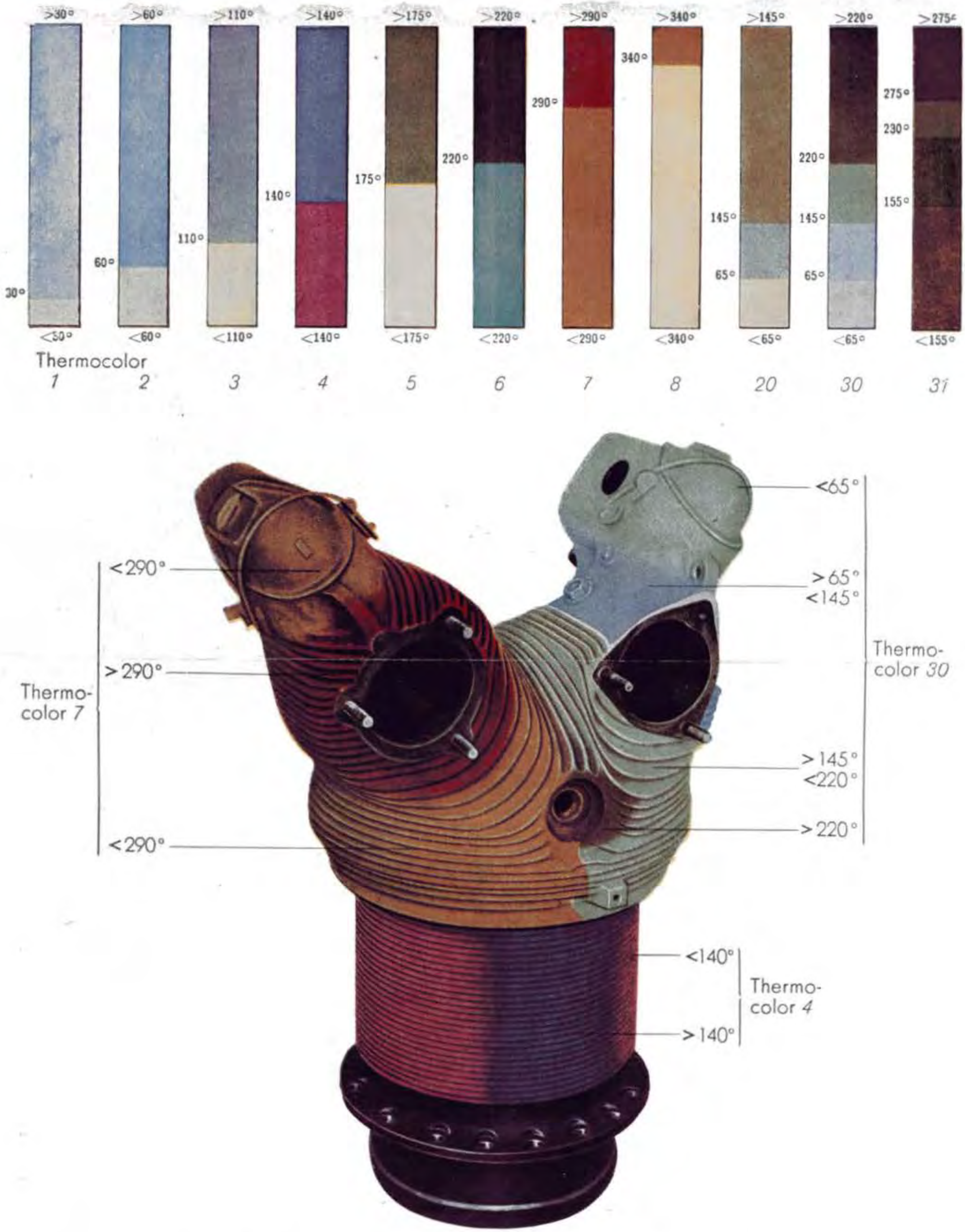


FIG. 238. Example of measuring cylinder temperatures by Thermocolor.

fact that it permits the observation of temperature distribution over a surface and the drawing up of isotherms. To do this a very large number of thermocouples would be required – a condition which cannot be met in practice, as for example, it is very difficult to fix the couples to the thin fins of an air-cooled engine, whereas paint can be spread over the thinnest of them without difficulty. For such reasons this method cannot be overlooked for it offers

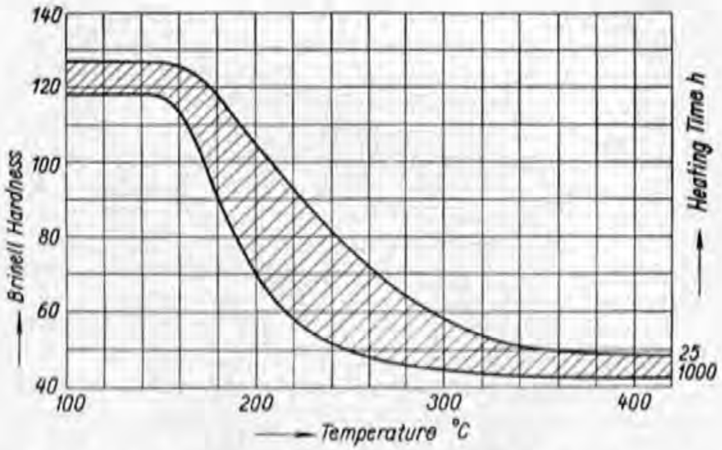


FIG. 239. Reduction of hardness of a typical aluminium alloy for cylinder heads in relation to time and temperature.

great advantages. The designer often needs to determine the degree of utilization of the fins and for this purpose the method is well suited.

Thermocolor paint is supplied in powder form containing also the binder and it is prepared for use by dissolving in alcohol. The paint dries quickly and is impervious to petrol and oil. The thermal insulation effect of a minute layer of the paint is negligible owing to its low coefficient of heat transfer to air. The approximate temperatures prevailing in the area to be painted must be measured before applying Thermocolor in order to select the right grade. Often several grades of paint have to be used. In the example shown in Fig. 238, three grades of paint were used to measure the temperatures of an air-cooled engine cylinder. Thermocolor Nos. 20, 30, 31 are used here to show several temperatures each. A record of the experiments may be made by black-and-white photography, but for greater clarity it is suggested that isotherms be marked on the surface by chalk lines.

In order to gain rough information on the temperatures prevalent inside aluminium cylinder heads and cylinders, use can be made of the property of heat-treated materials which causes a reduction of their hardness after a certain time depending on the temperature and the time during which the part is exposed to the temperature [67]. The reduction in hardness with temperature and time of the materials commonly used for cylinder heads is given in Fig. 239. The values are determined after letting the engine run for 25 hours, for instance, at the required load and then dismantling the engine and cutting the parts under test (cylinder head, piston, crankcase etc.). By measuring the hardness on the faces formed by cutting, the reduction in hardness will be obtained and

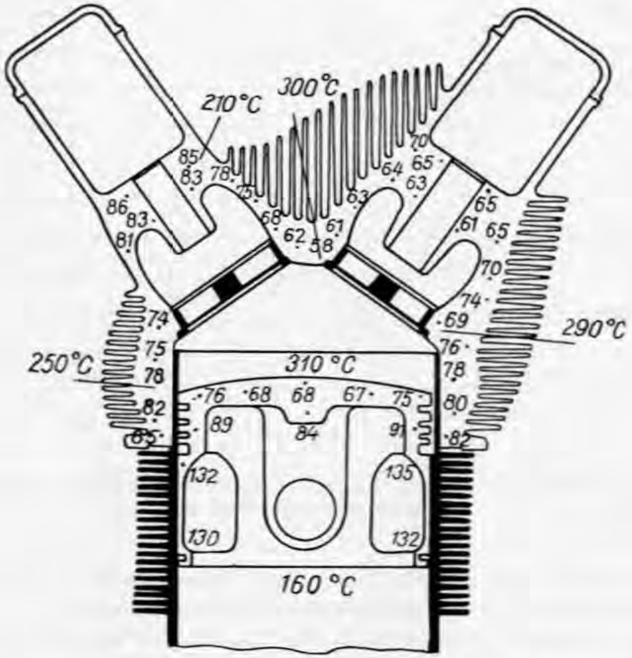


FIG. 240. Points of different Brinell hardness on the cylinder head of an aircraft engine and corresponding temperatures estimated from the drop in hardness.

the operating temperature of the points in question determined with a great degree of precision. Naturally the part must not have been subjected to the effects of heat before the beginning of the test. An example of such measuring conducted on an air-cooled aircraft engine cylinder is shown in Fig. 240.

The common means of measuring temperature of metals is by thermo-couples, which are to be preferred for their small size, precision and for the fact that they provide instantaneous temperature readings. An example of a thermo-couple suitable for measuring the temperatures of air-cooled engine cylinders and cylinder heads is shown in Fig. 241. An exact hole must be drilled at the required spot to the required depth of 1 to 1·5 mm under the surface of the cylinder or combustion chamber and the thermo-couple is

pressed into the cavity in such a way as to ensure good metal-to-metal contact with its surroundings. The exact temperature of the area can be found by measuring the current produced by the two heated metals. With closely spaced fins it has to be borne in mind that the flow of co oling air may be hampered by the leads from the thermo-couple, thus affecting the temperature of the measured area.

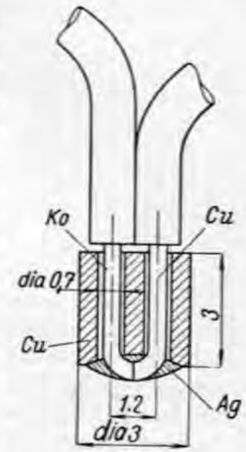


FIG. 241. Dimensions of a thermocouple normally used for measurements of cylinder temperatures.

The measuring of oil temperature is relatively simple and resistance or mercury thermometers are used. Greater problems are set by the need to measure the temperature of air, particularly of the warm outgoing stream from the engine, for the temperature is unevenly distributed over the whole section and air velocity varies. Usually one must be content with measuring the air temperature at several locations and obtaining mean temperature by a rough approximation. In order to measure the mean temperature of the whole section, a resistance wire net can be used, the value being calculated from readings of the resistance of the net. The temperature of air entering the fan and the carburettor is measured with mercury thermometers shielded from heat radiated by the warm engine parts.

Exhaust gas temperature is measured by thermo-couples, resistance thermometers or by direct observation through pyrometers.

CHAPTER XII

THE CYLINDER BARREL

1. Materials and design

The cylinder barrels of air-cooled engines are generally made of cast iron. Cast iron has good frictional properties, high resistance to wear and is easily machined. In low performance engines, fins cast integrally with the barrel are satisfactory (Fig. 242). On motor cycle engines care has to be taken to ensure that cooling air penetrates to the base of the fin and the fins must therefore be widely spaced (10 to 12 mm). The necessary cooling area is gained by a large fin diameter made possible by the fact that the cylinder stands unobstructed as a rule without the height of finning being limited by considerations of space.

On side-valve cylinders, the whole circumference of the exhaust valve seat must be well cooled. Sufficient clearance must be provided for air to pass between the valve port and the cylinder wall and the port must be extensively finned. An illustration of such a cylinder used on a motor cycle engine is given by Fig. 243. Note the shape of the countersunk valve seat and the holes in the lower part of the cylinder of this flat engine for additional lubrication. (See also Fig. 230)

FIG. 242. Cylinder of an air-cooled compression-ignition engine with cast fins.

Automobile engines, working under higher stress, cannot make use of cast fins and therefore the fins are machined by turning from solid. In this instance sets of cutting tools combined in comb fashion turn all fins simultaneously (Fig. 244). In automobile engines the fins generally have to be intersected in order to provide room for retaining bolts and for push rod covers. Such niches are cast in the rough shape of the barrel so that no additional machining is needed after turning the fins. Fig. 245 shows (left) the casting of a Tatra compression-ignition engine cylinder barrel and (right) the same barrel with machined fins. The machining operation takes 20 minutes. Machined fins of cast cylinder barrels are usually spaced 5·5 to 3·5 mm apart with a height of 15 to 25 mm. Thickness is usually chosen as small as practicable, being about 1·5 to 1 mm at the extremity of the fins.

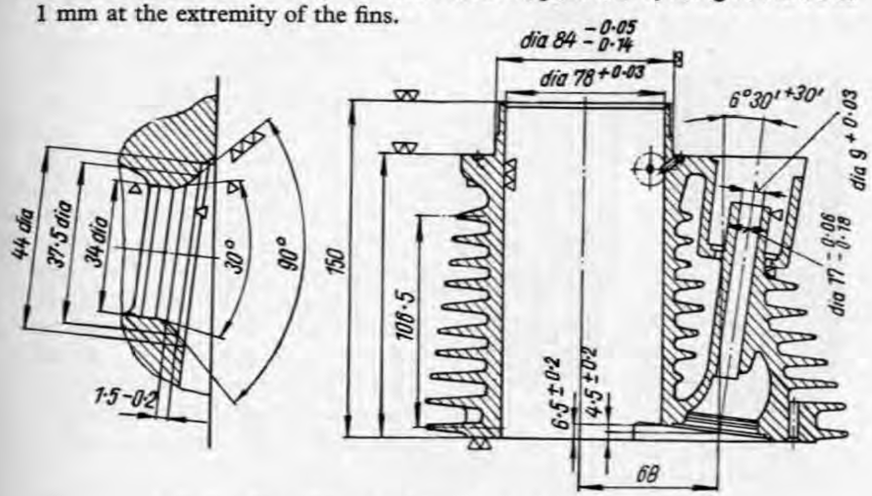


FIG. 243. Cylinder of the Soviet M-72 motor-cycle engine.

In the case of an in-line engine the choice of finning height is dictated by considerations of space. As this must be smaller, the fins are at reduced distances apart in order to achieve the same coefficient of heat transfer. Fig. 85 shows three forms of finning, all having the same base coefficient of heat transfer. If conditions permit the use of a height of fin amounting to 37·4 mm, the fins may be spaced at 12·7 mm apart, but with the height of fins of adjacent cylinders limited; proportionally reduced spacing must be used in conjunction with a reduced fin height. Cast fins are used in automobile engines up to spacings of about 6 mm.

FIG. 244. Machining of fins of an air-cooled engine cylinder.

Densely spaced fins can be cast using the plastic shell moulding process. An accurate mould is produced by this method in which the metal pattern is moulded in a mixture of phenol-formaldehyde resin with silica sand of 0·1 to 0·06 mm grain size and heated up to about 200 °C. A mould of strong but porous material is thus obtained and it is further hardened at about 400 °C. The moulding and hardening of this mould takes about 3 to 4·5 minutes in all. Such a mould is shown in Fig. 246. This method is extremely convenient for casting cylinders of air-cooled engines and the castings are noted for their precision and smooth surface. The saving of material by the use of plastic shell mould casting as compared with machining is considerable.

FIG. 245. Cylinder of the Tatra 111 compression-ignition engine prior to and after machining.

In order to prevent gas leakage between the piston and cylinder, the latter must be perfectly cylindrical at operating temperatures. The maximum tolerated ovality of large cylinders is 0·1 to 0·15 mm; with smaller bores this is reduced. Ovality may be caused either by imperfections of machining or by thermal and mechanical distortion. The distribution of temperatures must be uniform over the cylinder circumference. Uneven heating can cause distortion and the cylinder in question will not provide a reliable gas seal when hot. Uniform temperature over the whole cylinder surface can be obtained by correctly designed cool air ducting. Air entering the cylinder can also reduce the temperature at that side of the cylinder wall, against which it is directed. With temperature differences on the periphery of the cylinder exceeding 40 to 50 °C, there is danger of considerable deformation through heat. The choice of suitable finning has already been discussed.

Mechanical distortion of the cylinder may be caused by uneven spacing of retaining bolts on either

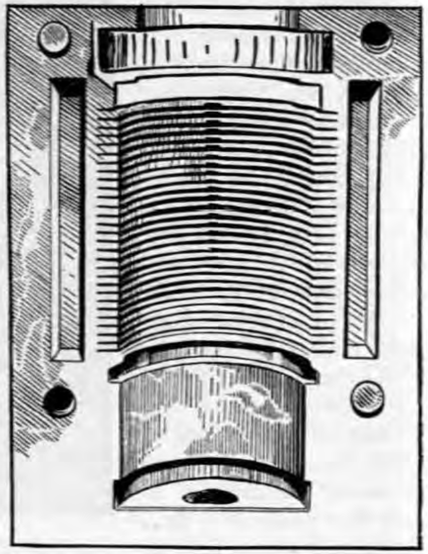


FIG. 246. Shell mould for casting cylinders.

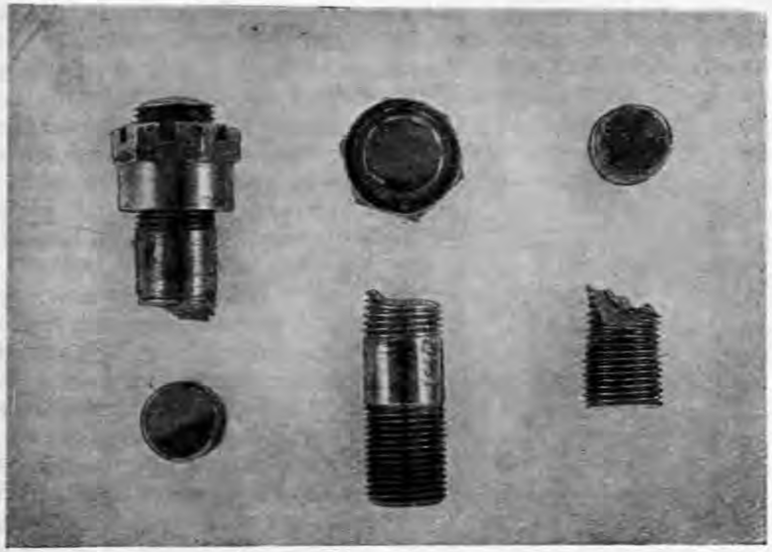


FIG. 247. An M 12/1 bolt fractured in operation. Left: fatigue fracture, right: static fracture.

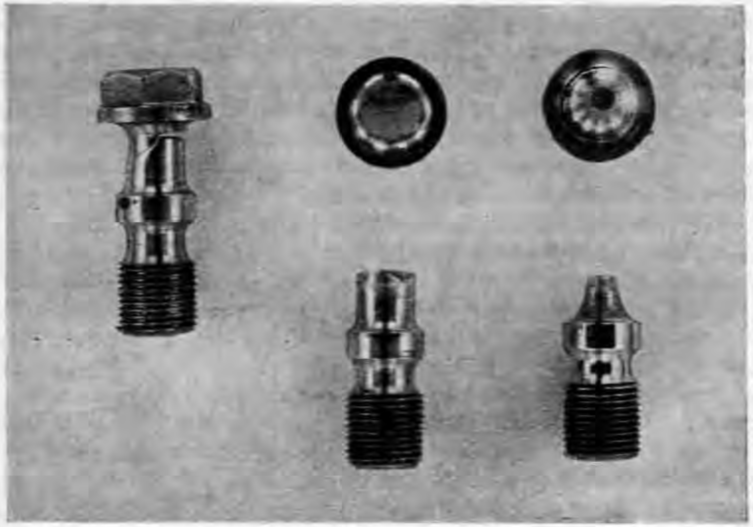


FIG. 248. Modified M 12/1 bolt satisfactory in operation. Left: fatigue fracture, right: static fracture.

the upper or the lower flange. Uniform tightening of the bolts is important and elastic bolts (of reduced diameter) reduce the risk of deformation by mechanical causes. Short and stiff bolts are unsuitable under all conditions. Breakage of bolts fastening either the lower flange to the crankcase or the upper to the cylinder head are generally caused by their rigidity, the rule being that the harder the washer, the more resilient must be the bolt used. Frequent breakages can therefore usually be cured either by fitting thinner bolts or by increasing their length.

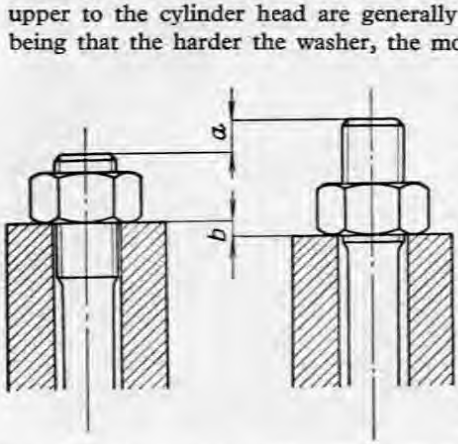


FIG. 249. By screwing up the nut the bolt is expanded by a, and the washer compressed by b.

An instance of such an increase in the tensile strength of studs is given by Figs. 247 and 248. The stud originally used had an M 12 × 1 thread and breakages were frequent. After altering its section as shown in Fig. 248, tensile strength increased by 21% as proved during fatigue tests. The material used was unchanged and weight was reduced by the modification. The studs proved reliable in service. The improvement may be accounted for in the following way.

If the nut (see Fig. 249) is tightened, the bolt increases in length by a and its washer is compressed by b, the increase in bolt length and the reduction of washer thickness being proportional to the initial tensile load produced by tightening the nut. This stage can be plotted in the graph of Fig. 250 with the tension applied plotted on the vertical axis in kg and the increases in bolt length on the horizontal axis (right) and the compression of the washer (left). By subjecting the bolt to an initial tensile load V, the bolt increases in length by a and the washer is compressed by b. By applying a force P to the bolt, the force is not simply added to load V, but it causes a further increases in length by c, but the washer can expand by the same amount and its load will decrease to Z. The resulting force acting on the bolt is then $P + Z$.

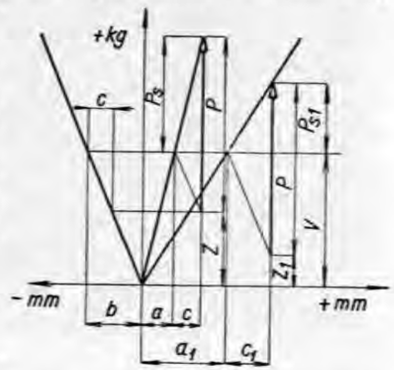


FIG. 250. The function of elastic bolts.

By reducing the bolt diameter, length will increase to a for an equal load V. As there is no change in the washer, its compression will again equal b. By applying a force P to the bolt as before, the increase in length of the bolt will be c_1 and washer tension will be reduced to Z_1. The resultant force within the bolt will be $Z_1 + P$ — i.e. less than in the first instance.

The alternating load of the bolt will fluctuate within the range of P_s in the first case and only of P_{s1} in the second. The stress of the bolt will therefore be reduced at the expense of the washer. The initial load of the washer Z must however always be a positive value, otherwise the flange would be lifted off the crankcase.

This example serves to prove that the use of short and stiff bolts is wrong in principle. The bolts should be long and reduced in diameter (resilient) and the washer should be stiff.

With a flange fit of cylinders to the crankcase (such as is the case with motor cycle engines), the flange must have sufficient rigidity in order to prevent distortion of the cylinder wall owing to the stress applied by the retaining bolts. Clear traces of deformation are shown by mirror polished areas of cylinder walls in the vicinity of the bolts. The base flange of aluminium cylinder barrels must be particularly thick and rigid.

Even in designs in which the cylinder barrel is compressed between the cylinder head and crankcase, its wall thickness near the upper and lower seatings must be increased in order to avoid possible deformation (see Fig. 260/II). A thin wall in the middle section makes for light weight and increases the cooling air flow area between the fins.

The spigot aligning the respective head and cylinder centres must not be too close a fit in order to prevent the transfer of distortion from the cylinder barrel to the head. A number of bolts or studs exceeding 4 is beneficial but it poses a problem in head design if the bolts are to avoid the spaces needed for ports, plug and sealing flanges.

From the point of view of cooling, light alloy cylinder barrels are highly desirable. The heat conductivity of aluminium is nearly three times that of cast iron. Aluminium is also lighter than cast iron and easy to cast and machine. The high coefficient of thermal expansion of aluminium is beneficial, for aluminium pistons may be fitted with a very small clearance which does not increase with the warming up of the engine. This advantage is put to good use in chronium plated bores. The thin layer of chrome on the inner aluminium surface impairs neither cylinder expansion nor heat transfer.

Other conditions being unchanged, the change from cast iron to aluminium as a constructional material for cylinder barrels, reduces cylinder temperature by about 20 °C. Owing to this improvement in cooling, the compression ratio may be raised and thermal efficiency increased [40] [37].

It is interesting to note that in the quantity of heat removed, there is hardly any difference between an aluminium and cast iron cylinder. Heat transfer, it will be remembered, is dependent on outer surface area, which is not affected by a change of material. But owing to improved heat conductivity, the temperature difference in the cylinder wall is reduced and the temperature of the cylinder inner surface is also reduced. The very favourable heat conductivity of aluminium aids tangential heat flow and the distribution of temperature over the cylinder circumference is more uniform. Chromium plated aluminium cylinders have brought about a great improvement in air-cooled engines.

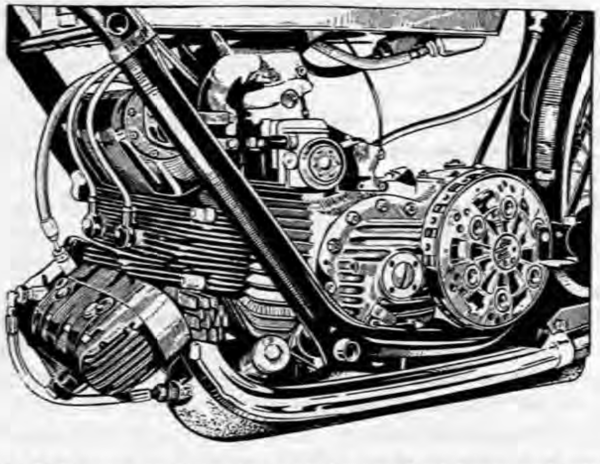


FIG. 251. The AJS motor-cycle engine with longitudinal finning of cylinders. Note the interrupted finning of the cylinder head in the region of the exhaust valve.

The Al-Fin process has the advantage of making use of the desirable properties of both aluminium and cast iron. The bonding of aluminium with a cast-in iron or steel liner is so complete as not to hinder the transfer of heat. The good frictional properties of cast iron are used in conjunction with the good heat conductivity of aluminium fins.

FIG. 252. Intermittent finning of the cylinder of the Triumph motor-cycle engine.

When pressed-in cylinder liners are used for aluminium cylinders, the interference for pressing in must be chosen correctly in order to prevent the liner from working loose in a heated cylinder. As soon as the insert is freed inside the cylinder when heated, the air space thus created forms an obstacle to heat transfer and local overheating of the liner may ensue. This problem is not as acute in small bore cylinders as it is with larger bores. Wall thickness of the liner is usually about 1·5 to 2 mm. Thicker liners must be used for two-stroke engines in order to prevent inward bulging of the liner at the

partition between the ports. A high nickel content alloy is sometimes used for its high coefficient of thermal expansion. The liner does not lose its initial load imparted to it by pressing in when the cylinder is heated and retains close contact. The higher coefficient of expansion of the liner has the further advantage of making possible a smaller piston clearance.

Cylinder finning must be such as to promote unhindered cooling air flow. On aircraft and automobile engines, only fins normal to the cylinder axis are used. The direction of the fins of motor cycle engines is sometimes modified to suit the direction of air flow. On very inclined cylinders longitudinal fins are therefore used, as shown in Fig. 251. The intermittent finning of this cylinder head should be noted in view of the already stated fact that such an arrangement improves the conditions of cooling. For the same reason intermittent finning is also used for cylinders, the two stroke TWN B 125 (Fig. 252) being an example on which the alternating intermittent fin layout is visible.

The striving for thinner fins has led designers to consider cast-in sheet metal fins. The production of such designs for automobile engines meets with difficulty in that the apertures for studs and push rod tunnels must be moulded in the fins before casting. The efficient bonding of sheet steel fins with the cast iron cylinder is of utmost importance if transfer of heat is not to be impaired. For production (moulding) the use of two semicircular fins is convenient, the amount of waste being reduced. Experiments so far conducted offer hope for possible future use of this technique.

2. Aircraft engine cylinders

The good cooling of cylinders is of prime importance in large aircraft engines and for this reason, only machined cylinders with dense finning are used. Fin thickness at the top is generally 0·75 to 0·5 mm. As the cylinder barrel transmits the load from the cylinder head to the crankcase, steel is used exclusively, making possible machining of thinner fins than would be the case if cast iron were used. An example of an aircraft engine cylinder is shown in Fig. 209. The studs are divided into four clusters for better accessibility. On cylinders screwed onto their heads the bolt holes are drilled and flanges relieved after screwing.

The amount of waste is naturally high when cylinders are turned from rough forgings, frequently as much as 90% of the material being removed by machining. For the production of the BMW 801 aircraft engine cylinder a new technique was

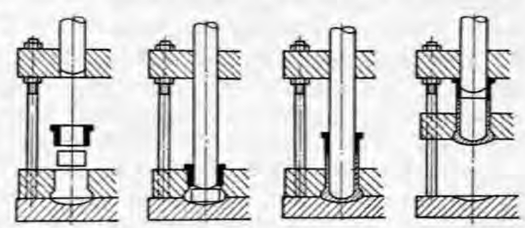


FIG. 253. Extrusion of a cylinder liner from a cylindrical blank.

evolved, resulting in a large saving in material. [10] Open hearth steel 1252 containing 1·4% Cr, 0·8% Mn, 0·45% C was used; satisfactory results were also obtained by the use of low alloy steels.

The liner was extruded from circular blanks of 172 mm dia by 62 mm. After uniform heating by induction the blanks were inserted into a special tool and extruded in a sequence of operations shown diagrammatically in Fig. 253. A disc of 50 mm dia was cut out of the heated pot bottom and the resulting flange was widened to a cylindrical shape. (Fig. 254.)

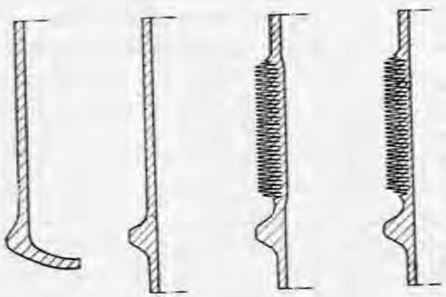


FIG. 254. Production operations of rolled fins on the cylinder of the BMW 801 aircraft engine.

Of great interest is the process whereby the fins were given their final shape by rolling. The sequence of production of the liner is shown in Fig. 254. The heated liner rotated on a core and 33 discs were pressed into it at a pressure of approximately 20 tons. The liner material was thus forced to "flow" to a height of 19 mm, forming fins of 0·5 mm thickness at the top and 0.9 mm at the base and spaced 3·6 mm apart. The lay of metal fibres in the flange and fins thus obtained is ideal. The only remaining machining operations consisted of cylinder boring, flange machining and of machining the threaded section of the cylinder by which it is attached to the cylinder head.

Experiments with sheet metal fins brazed to a steel cylinder barrel did not meet with such success as it proved impossible to maintain even spacing between the fins. Better results were obtained by turning a spiral on the cylinder circumference and by brazing a spiral of copper sheet to this. The brazed areas were cleaned by high-pressure steam. Brazed-on fins were of 0·75 mm gauge sheet, 25 mm high and spaced 2·5 mm apart. Good results were obtained on an experimental single-cylinder engine. The weight of the cylinder was, however, increased by 4·4 lb (2 kg). In order to take full advantage of the excellent heat conductivity of copper, the fins would have to be paper-thin — a requirement which cannot be met in production.

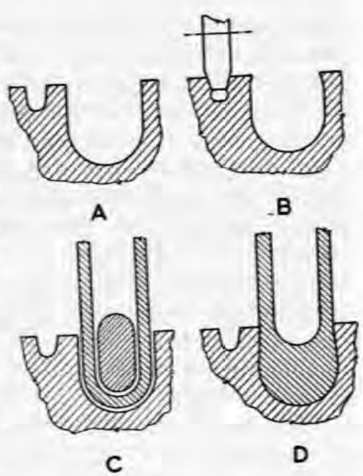


FIG. 255. Connecting U-shaped sheet metal fins to a steel cylinder.

For this reason the use of brazed-on aluminium fins which proved equally efficient for cooling was investigated, but so far no reliable method of brazing the aluminium fins has been developed.

Better results were obtained by mechanical means of attaching the fins to the cylinder barrel. [11]. The turning of grooves for separate thin sheet metal fins was, however, difficult and the tool used wore rapidly; to overcome this a wider groove, housing two fins, was then turned. A helical 3·9 mm wide groove was turned on the cylinder circumference as shown in Fig. 255. In between the grooves a slot was turned and by rolling out its edges as shown in Fig. 255 B, the fin groove walls were sloped. From aluminium 0·6 mm gauge strip a helical U shaped double fin was produced on a special machine, wound into the groove and fixed into position by aluminium wire (Fig. 255 C). The last operation consisted of rolling down this wire and thus ensuring a perfect joint between the aluminium and groove walls. With no weight increase, the cooling area of the fin was thus increased by 80%.

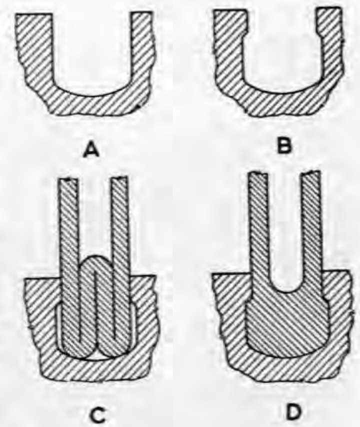


FIG. 256. Connecting W-shaped sheet metal fins to a steel cylinder.

An improved method illustrated in Fig. 256 was adopted after some experience with such fins. This makes use of circular grooves (as distinct from helical), which make for easier and quicker machining. By moving the forming tool to both sides, the base of the channel is widened (Fig. 256 B). Semi-circular fins of W section are then inserted into the channels (Fig. 256 C).

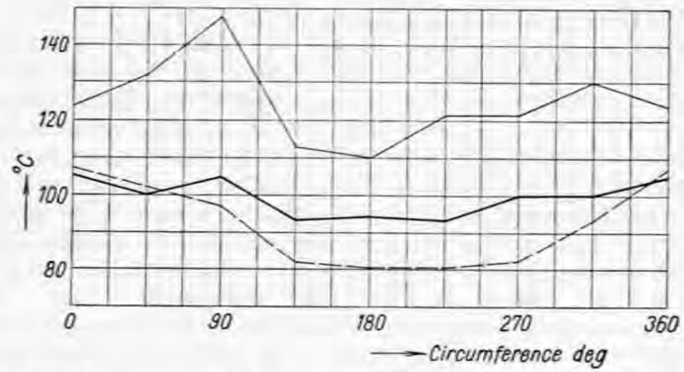


FIG. 257. Temperature of the cylinder surface circumference at the fin base. Upper full line – steel fins, lower full line – aluminium fins, dashed line – copper fins.

The last operation consists of rolling down the aluminium and thoroughly filling the groove, the final stage being shown in Fig. 256 D. This process has the following advantages:

1. there is no need to use aluminium wire;
2. thin sheet fins must be reinforced in order to prevent damage during handling. The dividing slit between two adjacent semi-circular fins is bent at 90° with the ends deflected so as to contact the adjacent fin and act as a reinforcement of the finning;
3. machining of grooves is speedier;
4. individual damaged fins can be replaced.

The results of tests with such fins are given in Fig. 257. Temperature distribution over the cylinder circumference is very even and temperature is lower than with steel fins. The comparison was made at equal power outputs and an equal pressure head. Such finning proved most satisfactory in actual service. Fig. 258 shows a cylinder with forged and machined head and a cylinder with sheet metal fins.

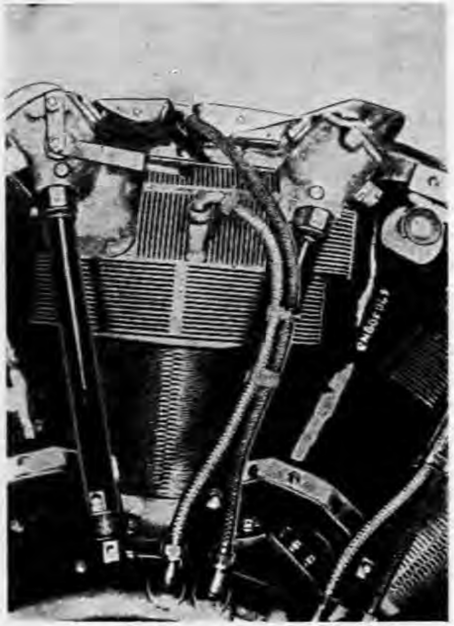


FIG. 258. Cylinder of the Wright aircraft engine with forged cylinder head, machined fins on the head and aluminium sheet fins on the cylinder.

3. Cylinder wear

Perfect air filtration is a prerequisite for small cylinder wear. Dust contains hard crystals of silica, which together with oil on the cylinder wall form a perfect grinding paste. Good air filtering is particularly necessary in tractor engines and on rear mounted vehicle engines. The engine compartment of rear-engined cars must be sealed against the ingress of dust, or air must be ducted from the front of the vehicle or from a source where it is sufficiently clean. In addition to these precautions, good oil-filled air filters must be used in both instances.

Cylinder wear can be reduced by the use of chromium plated piston rings. The smooth surface of a run-in chromium plated ring reduces friction and cylinder surface wear.

The first compression ring has the greatest effect on cylinder wear reduction and for this reason a chromium plated ring is generally fitted only in this position.

Cylinder wear is not only the result of mechanical abrasion but its major cause is the corrosive effect of the products of combustion. With a cylinder temperature below approx. 65 °C exhaust gases condense on the cylinder walls, corroding them and increasing wear. Most rapid wear occurs before the cylinder walls warm up after starting from cold. The influence of cylinder wall temperature on the rate of wear is shown in Fig. 259 [66]. At temperatures below 65 °C, the rate of wear increases rapidly and it is therefore important for the engine to operate below this temperature only for the shortest possible time.

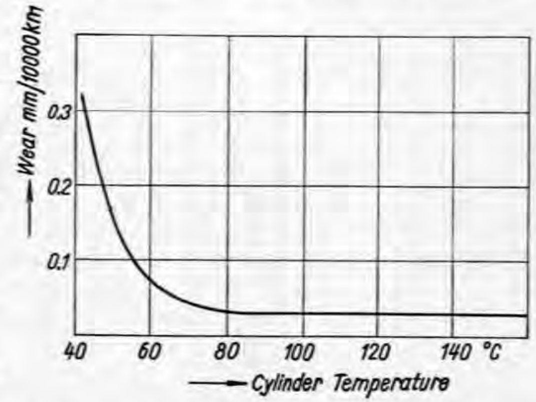


FIG. 259. Influence of cylinder wall temperature upon wear. Note the rapid increase of wear at temperatures below 65 °C.

A smaller mass is concentrated around the cylinder of an air-cooled engine than in a water-cooled engine. The air-cooled cylinder therefore reaches its operating temperature sooner. The warming-up period was discussed in the chapter on cooling control. The temperature of an air-cooled cylinder does not drop below the critical boundary of 65 °C even at idling speed. All these considerations result in a rate of wear which amounts to one third to one half of the wear shown by a water-cooled engine — an observation which is supported by evidence from companies producing both the air and water-cooled variations. [31], [52].

The fact that cylinder wear is chiefly caused by corrosion through condensation is shown in the diagram in Fig. 260 (after Dr. Seifert); according to which the wear in an air-cooled engine amounts to only about a quarter of the wear in water-cooled engines. The use of high duty (detergent) oils results in a reduction of cylinder wear in water-cooled engines by almost one half, the main reason of this reduction being the increased protection against corrosion afforded by detergent oils.

No improvement in the rate of cylinder wear resulted from the use of high duty oils in air-cooled engines.

It is obvious that the cylinder surface must be well lubricated in order to restrict cylinder wear. Plentiful cylinder lubrication is preferred today with surplus oil being removed by the oil-control ring. Good oil-control rings with high specific pressure and a narrow sealing surface in conjunction with large holes for the return of oil below the ring ensure low oil consumption together with efficient lubrication of the cylinder wall.

In horizontal engines a separate jet of oil must be directed against the upper cylinder region of the right-hand bank of cylinders. With oil cooling of the piston crown employed, the cylinder wall is generally well lubricated by oil dripping off the piston.

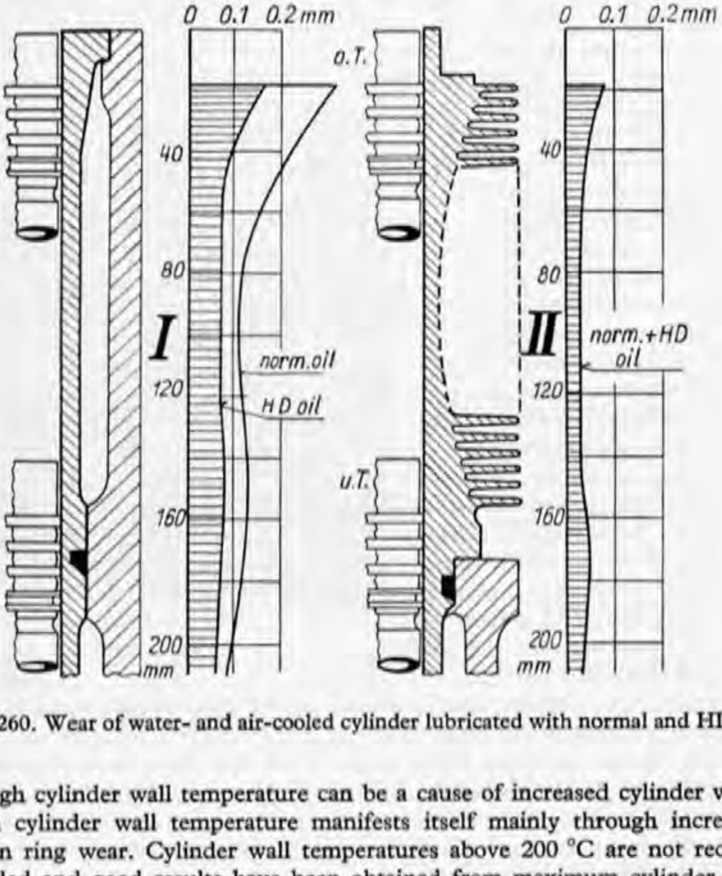


FIG. 260. Wear of water- and air-cooled cylinder lubricated with normal and HD oil.

High cylinder wall temperature can be a cause of increased cylinder wear. High cylinder wall temperature manifests itself mainly through increased piston ring wear. Cylinder wall temperatures above 200 °C are not recommended and good results have been obtained from maximum cylinder wall temperatures of 180 °C. Higher temperatures are, however, permissible with high grade oils and high quality materials used for the manufacture of cylinders.

CHAPTER XIII

THE CRANKCASE

The crankcase forms the foundation of the whole engine. It is the heaviest engine part and deserves special attention. The chief purpose of a crankcase is the provision of a rigid base for the crank mechanism. Loads from the crankshaft bearings are transmitted through the crankcase to the cylinder heads. For this reason a crankcase must be designed to:

1. transmit loads from the cylinder heads to the crankshaft bearings,
2. resist bending stresses,
3. resist torsion loads.

These properties must be attained with a minimum of constructional material and the design must be suitable from the point of view of production. The crankcases of modern water-cooled engines are always cast in one with the cylinder block, resulting in a very rigid unit.

In air-cooled engines built-up construction is used. This results in the separate mounting of each cylinder and a considerable reduction in the depth of the upper part of the crankcase above the crankshaft axis. This is a basic difference from water-cooled construction practice and one which must be respected by the designer.

For reasons of weight reduction, cast iron crankcases of water-cooled engines are split in the plane of the crankshaft axis, the sump being formed by a light sheet metal stamping. Such a layout cannot be used for air-cooled engines of built-

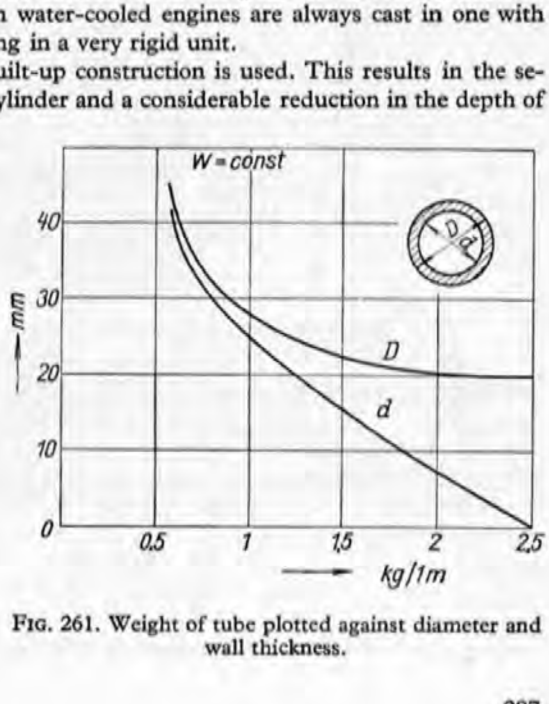
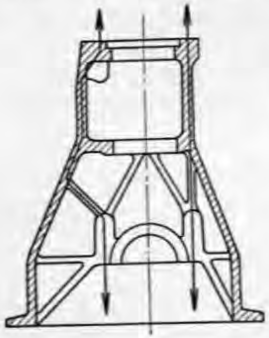


FIG. 261. Weight of tube plotted against diameter and wall thickness.

up construction. The following two forms of crankcase are chiefly used:



1. the crankcase open from below, but with its bottom face deep below the crankshaft axis and,

2. the tunnel-like crankcase into which the crankshaft is assembled endwise from behind or from the front and the sump of which is formed by a light cover.

FIG. 262. Bolt loads distributed according to the left hand side of the illustration involve the use of more material. A more favourable load distribution is shown on the right hand side.

Crankcases of the first group do not differ substantially from these used in water-cooled engines; the most notable differences being in the greater width

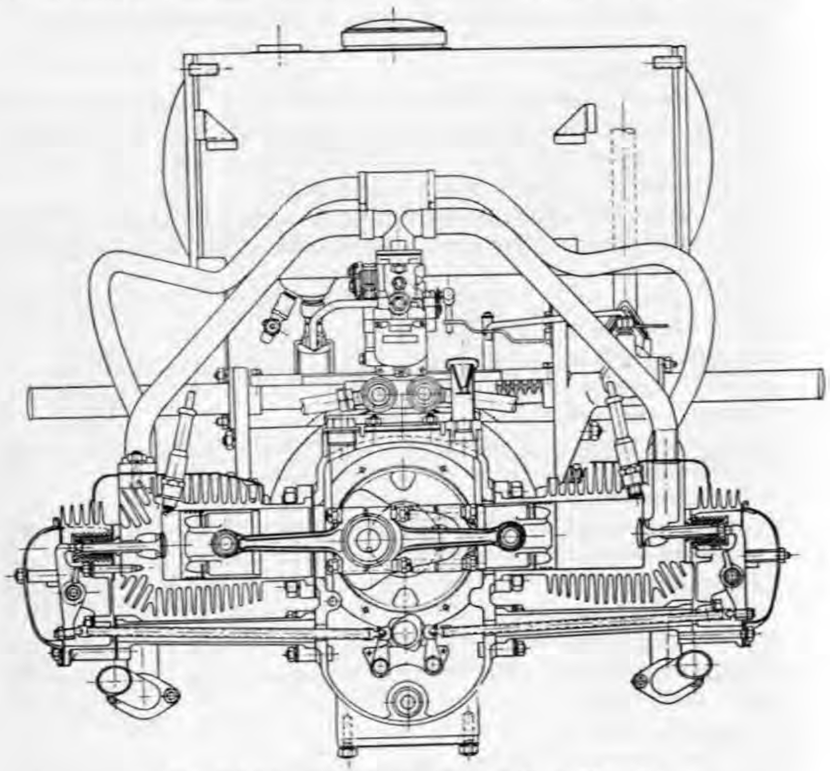


FIG. 263. Transverse section of the Tatra 57 engine.

and the lower location of the bottom face of the air-cooled engine crankcase. Examples of crankcases so designed are shown in Fig. 216 and 217. The great width and depth of the crankcase makes possible a structure which resists bending and torsional stresses with a small quantity of material. To illustrate this, Fig. 261 shows the dependence of the weight of a tube on its diameter with a constant moment of resistance. The graph clearly indicates the rapid

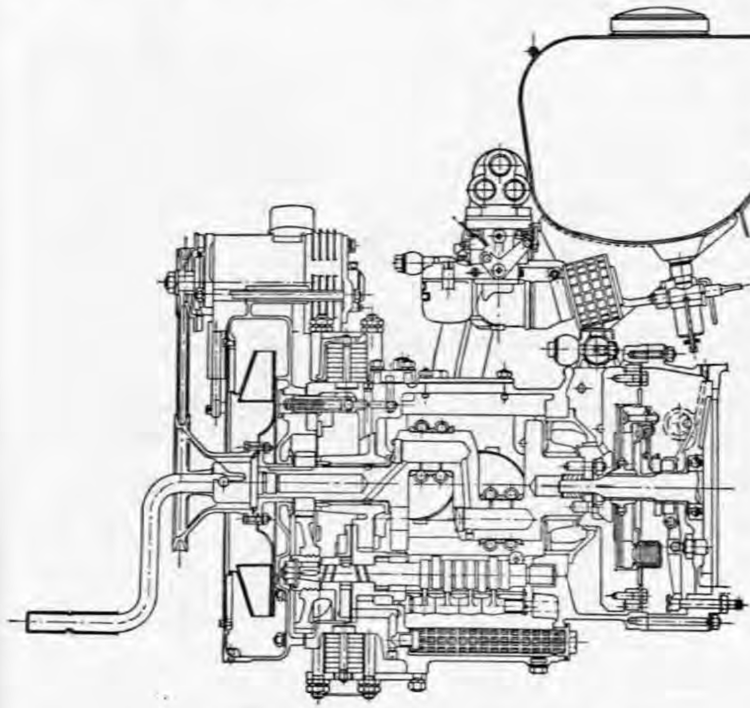


FIG. 264. Longitudinal section of the Tatra 57 engine.

rate at which the necessary wall thickness is reduced with an increase in diameter and the reduction of weight per unit of tube length.

Crankcase wall reinforcing ribs and webs must be used with the greatest caution. The reinforcement of a straight wall by a single rib does increase the rigidity of the wall, but the upper edge of the rib is under great stress whereas the remainder of its mass is not fully used. More desirable would be the use of I section ribs, but these unfortunately are not convenient because of casting complication. Better results are obtained by curvature of the walls and, where necessary, by a gradual increase in wall thickness near the flanges and at the curvatures leading into flanges and partitions or sidewalls.

The problems of crankcase construction were thoroughly investigated by the Continental concern manufacturing aircraft engines and the result of this development is reflected in a crankcase without reinforcing ribs which is stiffer and lighter than its ribbed predecessors. Experience gained with the design of aircraft engines was used in the construction of an open-bottom crankcase for a V 12 air-cooled compression-ignition engine which is remarkable for its small overall dimensions and low weight (Fig. 393).

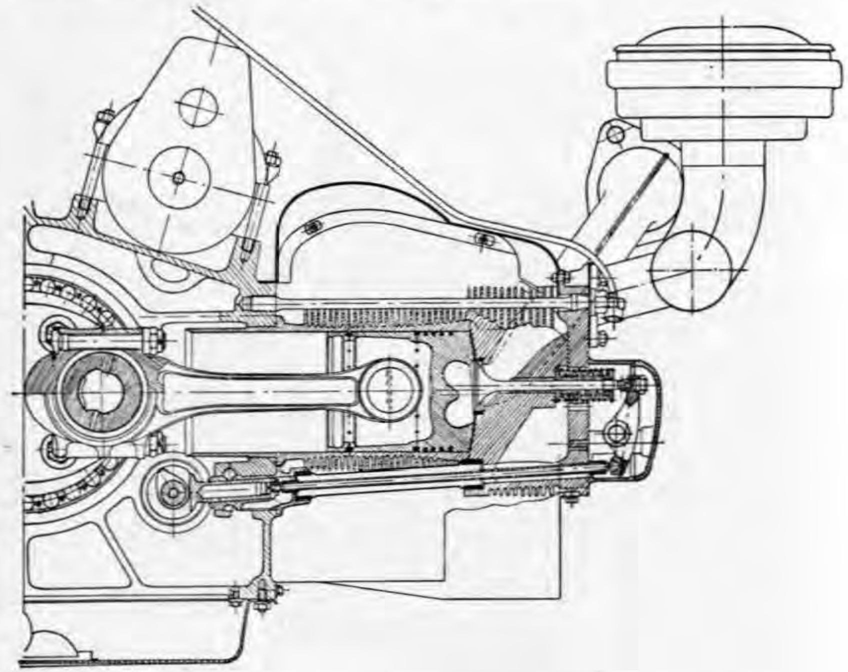


FIG. 265. Transverse section of the flat six-cylinder Tatra 116 engine.

Further attention must be paid to the transmission of loads acting upon the cylinder head to the crankshaft main bearings. The path of such forces should be as short and straight as possible. Fig. 262 shows the correct and incorrect path of a load acting upon the crankcase. The indirect path results in bending stress on the crankcase with the accompanying incomplete utilization of constructional material which makes for increased weight. The most convenient means is a straight, wide rib stressed in tension. If long bolts are used and the cylinder anchored at the main bearing housings, the crankcase will be free of stress.

If the crankcase partitions incorporate breather holes, the periphery of these holes must show an increased thickness, otherwise a similar concentration

of stress takes place as on the upper edge of a simple stiffening rib. Cracks appearing in crankcases often originate from such holes.

The tunnel-type crankcase offers many advantages which commend it for use in air-cooled engines. In principle such a crankcase is similar to a tube with its high torsional and bending resistance for a minimum given weight. An example of such a design is provided by the Tatra 111 engine crankcase (Fig. 407). As the crankshaft must be inserted into the crankcase from one end, the use of roller bearings and a built-up crankshaft is advisable. Plain bearings can be employed only when the crankshaft has two main bearings only as in the flat-four Tatra 57 engine (Figs. 263, 264) or several other small engines. The crankshaft is introduced into the crankcase during assembly through the rear end cover aperture.

With a design incorporating roller bearings, it must be borne in mind that the assembled crankshaft must pass through the inner circumference of the outer bearing ring. In such a case a built-up crankshaft is suitable. The separate crankshaft members are steel castings. The small width of roller bearings enables a close grouping of cylinders even when two connecting rods share a common crankpin, which is an arrangement offering advantages in manufacture.

For flat engines, the monoblock tunnel-type crankcase may also be employed, as shown in Fig. 265. A large sump cover facilitates assembly. The nuts of big end bolts on flat air-cooled engines always face the pistons, facilitating the removal of any individual cylinder and connecting rod assembly.

CHAPTER XIV

THE VALVE GEAR

1. Valve gear arrangements

The purpose of the valve gear is to ensure the best possible charging of the cylinder during the induction stroke, to seal it during the compression and expansion strokes and to make possible a complete evacuation of the products of combustion during the exhaust stroke. By their method of operation, valve mechanisms may be classified into two groups:

1. the poppet valve systems,
2. rotary and sleeve valve systems.

The great majority of engines today is fitted with the former type, which has undergone a long period of development. Poppet valve layouts may be divided into the following:

1. side valves (s.v.),
2. overhead valves — pushrod operated (o.h.v.),
3. overhead valves actuated from an overhead camshaft (o.h.c.),
4. combined (F-head).

Good cylinder charging is possible only when the air entering the cylinder meets with a small resistance. Resistance is caused by the friction of air against inlet manifold walls, by changes in the direction of flow and by variations of inlet pipe flow area. The induction manifold should therefore be short, of constant cross section area and without sharp bends. The resistance to flow is at its highest value at the choke and valve and increases with increasing gas velocity. The gas velocity at the valve should therefore be chosen as small as conditions permit. With 197 ft/s (60 m/s) velocity throttling already occurs at the valve and with gas velocity exceeding 262 ft/s (80 m/s), resistance rises rapidly. The gas velocity in induction pipes of petrol engines should be sufficiently high (equal to about 70 m/s) in order to prevent the condensation of fuel on the inlet manifold walls. The section of inlet air pipes in compression-ignition engines is not limited in so far as pressure pulsations in the induction tract are not employed to improve cylinder charging.

For the calculation of air velocity at the valve only comparative values are generally used, reckoned from the ratios of the piston crown area A cm^2, flow section area through the valve a cm^2 and mean piston speed c_m m/s. The flow velocity at the valve

$$v_1 = \frac{c_m \cdot A}{a} \tag{50}$$

The formula for the calculation of flow area at the valve is dependent on the amount of valve lift as stated in literature referring to this subject. [36]. For a valve seat angle of 45° the following formula gives satisfactory results:

$$a = d \cdot h \cdot 2.22 + h^2 \cdot 1.11 \text{ (cm}^2\text{)} \tag{51}$$

where d denotes the internal diameter of the valve seat (cm)
h „ „ valve lift (cm)

Besides the flow area of the valve, the flow coefficient is also important. If two valves are of equal flow area, it does not follow that an equal amount of air will pass through them even if the pressure head is also equal. A good flow coefficient is provided, for example, by the valve shown in Fig. 183. If this valve is parallel to the cylinder axis and near the cylinder wall, conditions may be much less favourable. In the case of the inlet valve particularly, care should be taken to ensure free entry of air into the cylinder over the entire valve circumference and without sharp turns. Further advantages may be gained by the smallest possible deviation of the inlet pipe axis from the valve stem axis, even at the expense of a larger and heavier valve.

Good results are also obtained from inlet ports parallel with the cylinder axis, such an arrangement being used by the aircraft engine illustrated in Fig. 205.

The coefficient of flow can be ascertained from a continuous flow of air through the valve. Tests have shown that results obtained in this way are also applicable to the pulsating flow in a running engine.

Particular notice should be taken in air-cooled engines of the induction pipe wall temperature. By heating the incoming charge, its quantity by weight is reduced and with it, engine output.

The induction pipe should under no circumstances be closely associated with the exhaust pipe; both manifolds should be separated by a stream of cooling air.

For cylinder charging the inlet valve should be larger than the exhaust valve. The valve port may be reduced in diameter up to 90% behind the valve seat without an unfavourable effect on breathing and tapering in front of and behind the valve seat has a beneficial effect on flow through the valve, preventing the air flow breaking away from the seat.

Valve timing depends on the required torque curve, the required maximum engine speed and the shape of the induction pipe. In slow-running engines the exhaust valve opens later in order to permit gas pressure to act on the piston for a longer time and the inlet valve closes earlier after bottom dead centre in order to prevent blow back of the charge into the induction pipe. Little valve overlap at t.d.c. is used under such conditions.

In engines operating at high crankshaft speeds the exhaust valve must open sooner in order to permit a large enough pressure drop before the piston begins to travel upward in the exhaust stroke. The inlet valve must close at a later stage after bottom dead centre as at that piston position a low pressure exists within the cylinder and the influx of air continues owing to inertia, despite the upward movement of the piston. The valve should close at the precise moment at which the pressure inside the cylinder prevents further flow through the valve. In engines with a simple induction system (e.g. single-

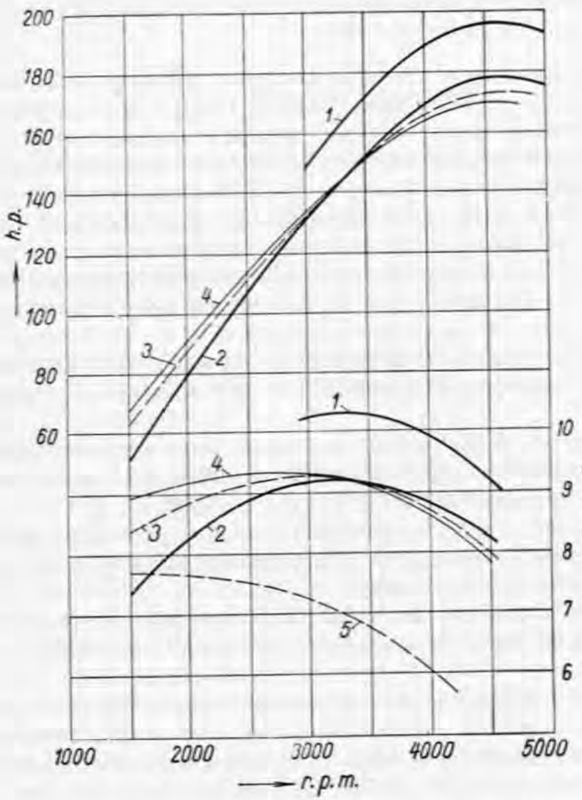


FIG. 266. Influence of valve timing upon the performance of an eight-cylinder engine, bore 84 mm, stroke 98·5 mm with different camshafts according to Table 39. Curve 1 represents engine performance with alcohol fuel.

cylinder motor cycle engines) the pressure pulsations may be used to improve breathing.

Valve timing overlap at top dead centre is fairly large in high-speed engines and use is made of the kinetic energy of gas in the exhaust pipe for scavenging the combustion chamber. In compression-ignition engines, valve timing overlap is limited by the gap available above the piston crown at t.d.c. The

influence of valve timing on engine output is given in Fig. 266 and Table 39 containing data of the timing of camshafts under test [23].

Table 39

Effect of Valve Timing Upon Engine Performance (Fig. 266).

Mark	I. O.	I.C.	E.O.	E.C.	Intake period	Exhaust period	Overlap
1. Super Harmon	30	78	64	26	288	270	56
2. Super Winfield	24	68	61	24	275	265	52
3. Racing	26	64	59	21	270	260	47
4. Semi-racing	21	59	54	16	260	250	37
5. Series	0	44	48	6	224	234	6

All values are given in degrees of crankshaft position.

Breathing efficiency is not only dependent on valve timing, but also on the rate of valve opening and closing, high velocities being desirable. Large valve timing overlap at top dead centre and a long period of valve lift are not favourable to engine starting and idling.

2. Cams

Harmonic cams are frequently used for automobile engines (Fig. 267). The following formulae expressing lift s, velocity v and acceleration a are valid for harmonic cam profiles:

$$s_b = (r_3 - r_1) \cdot (1 - \cos\alpha) \text{ (mm)} \tag{53}$$

$$v_b = \omega(r_3 - r_1) \cdot \sin\alpha \frac{1}{1000} \text{ (m/s)} \tag{54}$$

$$a_b = \omega^2 (r_3 - r_1) \cdot \cos\alpha \frac{1}{1000} \text{ (m/s}^2\text{)} \tag{55}$$

ω denotes the angular velocity of the cam.

With symbols as in Fig. 267 and all dimensions being in millimetres, the following equations relate to the cam nose:

$$s_s = h - s(1 - \cos\beta) \text{ (mm)} \tag{56}$$

$$v_s = s \cdot \omega \cdot \sin\beta \frac{1}{1000} \text{ (m/s)} \tag{57}$$

$$a_s = s \cdot \omega^2 \cdot \cos\beta \frac{1}{1000} \text{ (m/s}^2\text{)} \tag{58}$$

The course of lift, velocity and acceleration with a harmonic cam are given in Fig. 267 (right). The minimum radius of the cam follower must be calculated

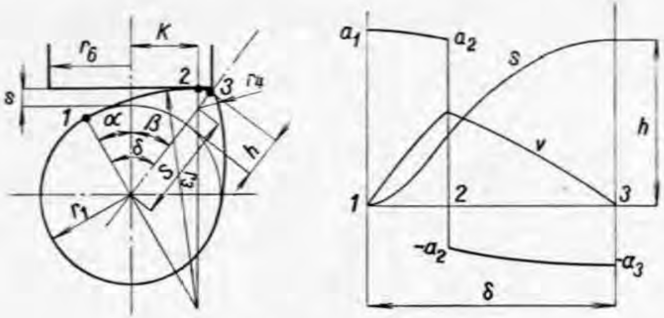


FIG. 267. Harmonic cam.

on the assumption that during the transition from the accelerating to the decelerating flank of the cam, the follower must be in continuous contact with the cam flank.

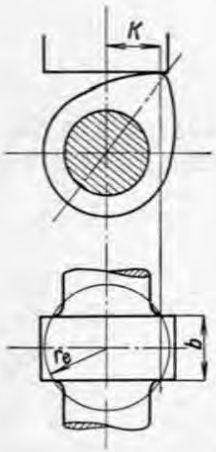


FIG. 268. Minimum diameter of the harmonic cam follower.

This position is indicated in Fig. 268 and the relative minimum radius r_e is

$$r_e = \sqrt{K^2 + \left(\frac{b}{2}\right)^2} \qquad (59)$$

where $K = (r_3 - r_1) \cdot \sin \alpha_2$

b denotes the cam width,

α_2 „ „ apex angle of the accelerating cam flank.

A harmonic cam profile may be obtained provisionally by a very simple geometrical construction shown in Fig. 269. The cam profile is drawn enlarged, in a scale of 10 : 1 for example. For such geometric construction it is sufficient to mark only the centre of the base circle, centre of the nose and of the flat flanks.

The amount of lift of the tappet on the flank and cam nose is found by drawing at right angles to the tappet axis a tangent to the cam profile and by measuring its distance s from the cam base circle.

The velocity and acceleration on the cam flank is determined by drawing a semicircle above the line $\overline{O_1 O_3}$ and by drawing a chord parallel with the

tappet axis through centre O_3. The intersection 1 of this chord with the semicircle indicates the velocity and acceleration of the tappet, since

$$v = \omega \cdot \overline{O_1 1}\ (\mathrm{m/s}); \quad a = \omega^2 \cdot \overline{O_3 1}\ (\mathrm{m/s^2})$$

When the velocity and acceleration at cam nose are to be found (Fig. 269), the semi-circle is drawn about the line joining centres $\overline{O_1 O_4}$. The chord

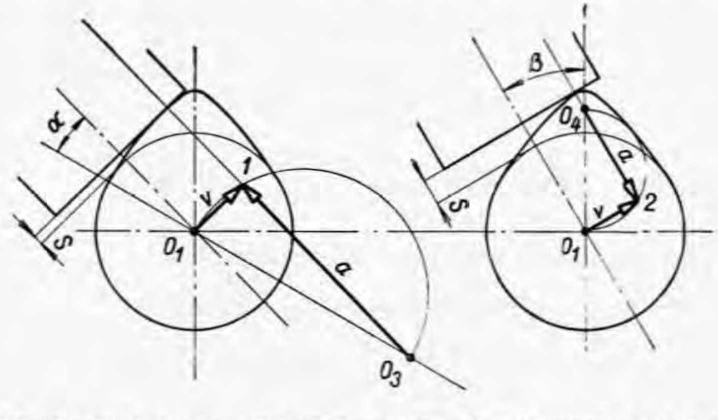


FIG. 269. Graphical determination of lift, velocity and acceleration at the flank and summit of the cam.

parallel with the tappet axis passing through point O_4, intersects the semicircle at point 2 and then

$$v = \omega \cdot \overline{O_1 2}\ (\mathrm{m/s}); \quad a = \omega^2 \cdot \overline{O_4 2}\ (\mathrm{m/s^2})$$

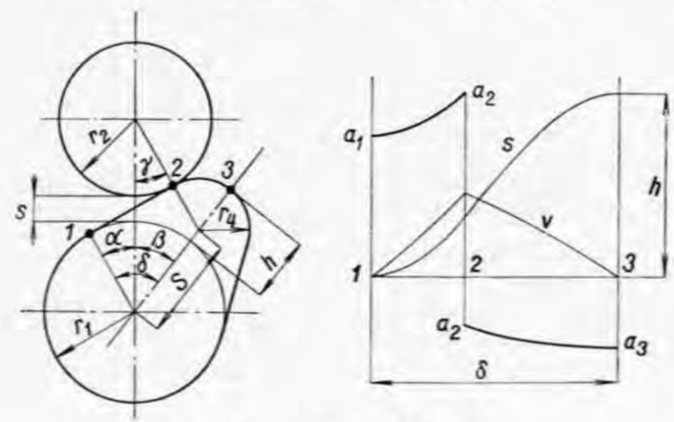


FIG. 270. Tangential cam.

The dimensions thus obtained must be divided by the scale (e.g. by 10) and the real dimensions in metres must be substituted in the equations.

The tangential cam is formed by two straight flanks joining the base and nose circles.

With symbols used as in Fig. 270, the distance, velocity and acceleration on the flank section are the following:

$$s_b = (r_1 + r_2) \cdot \left[\frac{1}{\cos\alpha} - 1\right] \text{ (mm)} \tag{60}$$

$$v_b = \omega\,(r_1 + r_2)\frac{\tan\alpha}{\cos\alpha}\,\frac{1}{1000} \text{ (m/s)} \tag{61}$$

$$a_b = \omega^2\,(r_1 + r_2)\frac{2 - \cos^2\alpha}{\cos^3\alpha}\,\frac{1}{1000} \text{ (m/s}^2\text{)} \tag{62}$$

For the cam nose the following equations are valid:

$$s_s = (r_2 + r_4)\cos\gamma + s\cos\beta - (r_1 + r_2) \text{ (mm)} \tag{63}$$

$$v_s = -\,s\,\omega\,\frac{\sin(\gamma + \beta)}{\cos\gamma}\,\frac{1}{1000} \text{ (m/s)} \tag{64}$$

$$a_s = -\,s\,\omega^2\left[\frac{s}{r_2 + r_4}\,\frac{\cos^2\beta}{\cos^3\gamma} + \frac{\cos(\gamma + \beta)}{\cos\gamma}\right]\frac{1}{1000} \text{ (m/s}^2\text{)} \tag{65}$$

Cams with convex flanks are used in motor cycle engines where roller followers are replaced by tappets with cylindrical bases. The following equations relate to lift, velocity and acceleration (Fig. 271):

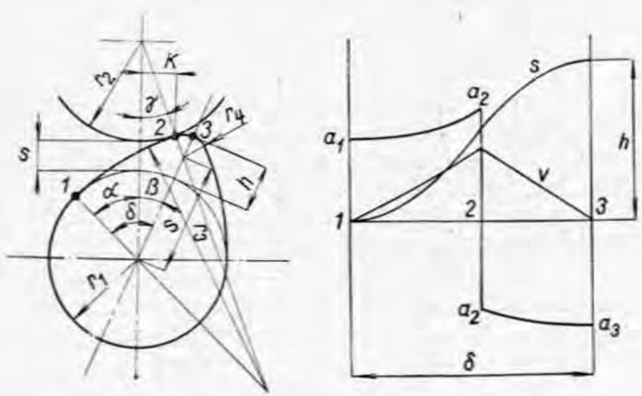


FIG. 271. Cam with convex flank.

$$s_b = (r_3 + r_2)\cos\gamma - (r_3 - r_1)\cos\alpha - (r_1 + r_2) \text{ (mm)} \tag{66}$$

$$v_b = \omega\,(r_3 - r_1)\frac{\sin(\alpha - \gamma)}{\cos\gamma}\,\frac{1}{1000} \text{ (m/s)} \tag{67}$$

$$a_b = \omega^2\,(r_3 - r_1) \cdot \left[\frac{r_3 - r_1}{r_3 + r_2}\,\frac{\cos^2\alpha}{\cos^3\gamma} - \frac{\cos(\alpha - \gamma)}{\cos\gamma}\right] \cdot \frac{1}{1000} \text{ (m/s}^2\text{)} \tag{68}$$

The same equations as given for tangential cams (63), (64), (65) are relative to the nose of this cam. The course of acceleration is shown in Fig. 271 (right). The shortest length of the convex follower is 2 K (non-rotating).

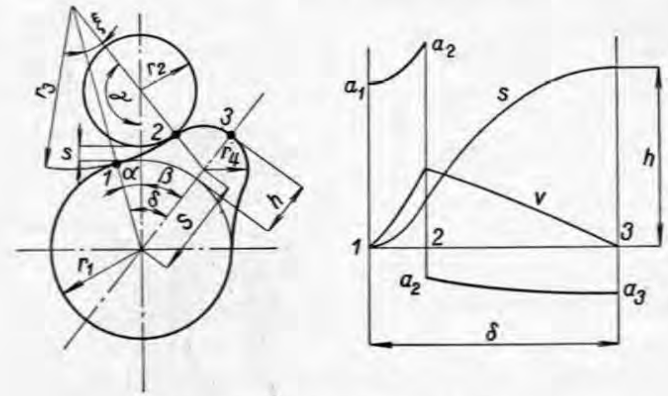


FIG. 272. Cam with concave flank.

Concave flank cams are found on the cam rings of radial engines and the following equations (Fig. 272) are applicable:

$$s_b = (r_3 - r_2)\cos\gamma + (r_1 + r_3)\cos\alpha - (r_1 + r_2) \text{ (mm)} \tag{69}$$

$$v_b = -\,(r_1 + r_3)\,.\,\omega\,\frac{\sin(\alpha + \gamma)}{\cos\gamma} \cdot \frac{1}{1000} \text{ (m/s)} \tag{70}$$

$$a_b = -\,(r_1 + r_3)\,\omega^2\left[\frac{\cos(\alpha + \gamma)}{\cos\gamma} + \frac{r_1 + r_3}{r_3 - r_2}\,\frac{\cos^2\alpha}{\cos^3\gamma}\right] \cdot \frac{1}{1000} \text{ (m/s}^2\text{)} \tag{71}$$

The right side of Fig. 272 shows the curve of acceleration. The respective nose equations (63), (64), (65) are the same as for a tangential cam.

The number of cams and the gear ratio between the crankshaft and cam ring are given in Table 40. The radius of the concave flank must be smaller than that of the grinding disc. An example of the cam ring drive is shown in Fig. 273.

After the amount of necessary cam lift and the valve timing have been decided upon, the cam profile must be established. It depends, in this case

Table 40

Transmission to Cam Rings of Radial Engines

Number of cylinders		3	5	7	9
Rotation reverse to that of the crankshaft	Number of cams	1	2	3	4
	Ratio of gear from crank	1 : 2	1 : 4	1 : 6	1 : 8
Rotation in the same direction as crankshaft	Number of cams	2	3	4	5
	Ratio of gear from crank	1 : 4	1 : 6	1 : 8	1 : 10

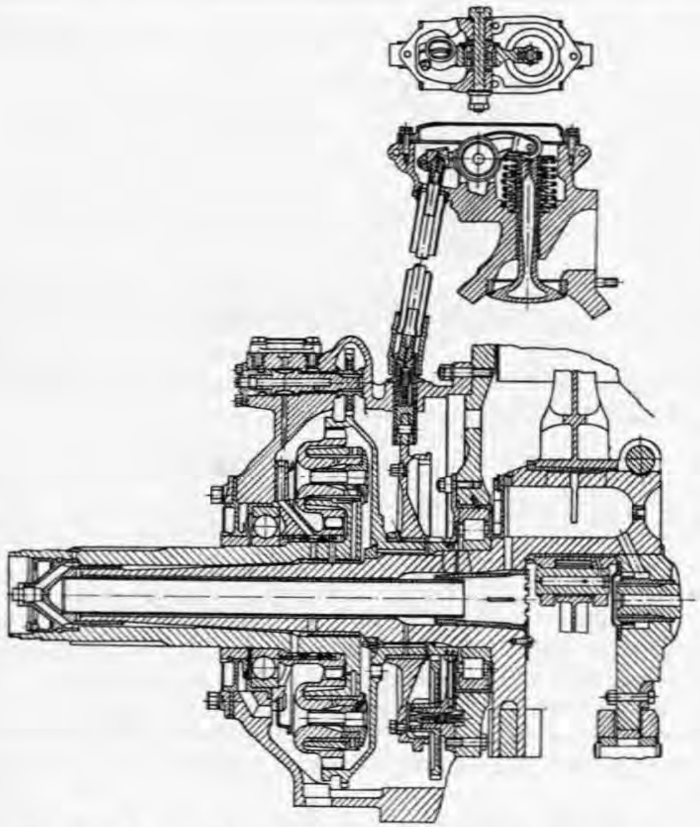


FIG. 273. Camshaft drive at the Wright Cyclone aircraft engine.

also, on the required rate of valve opening. As stated previously, rapid valve opening and closing is to be preferred for better breathing characteristics with a given valve timing. The acceleration on the lifting flank should therefore be chosen to be higher than deceleration on the closing flank. Acceleration about three times as high as deceleration is generally employed, an acceleration of five times the deceleration being the limit.

The method of designing harmonic cams is as follows (Fig. 274). The cam axes are drawn and any desired distance, e.g. $\overline{01} = 20$ mm is entered on the vertical axis. At the angle corresponding to the half-open position of the cam (e.g. 70°) a straight line is drawn and upon it the distance $\overline{02}$ is entered in such a way as to provide a ratio of acceleration on the cam nose a_3 to acceleration on the cam flank $a_1 = \overline{01} : \overline{02}$. From point *2* as centre, a circle is drawn through point *1* the intersection of which with $\overline{20}$, point *4*, determines the dependence of cam lift on the base circle as distance $\overline{13}$ gives the cam lift.

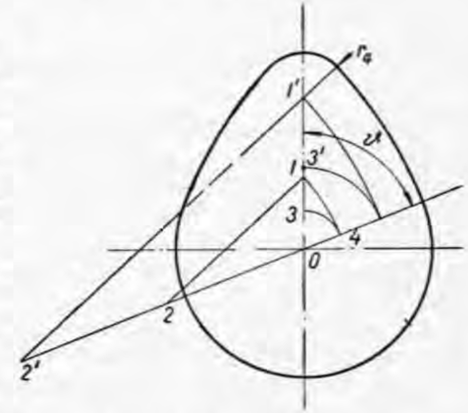


FIG. 274. Method of designing harmonic cams.

This diagram of the required cam is then constructed. The following correlations hold:

$$\overline{13} : \overline{10} : \overline{02} = \overline{1'3'} : \overline{1'0} : \overline{02'},$$

$\overline{1'3'}$ being the real (required) cam lift.

With point *1'* as centre, a circle of the required radius of the cam nose r_4 is described, the rest being completed by circles with centres at *2'* and *0*. Every value equidistant to this curve shows equal valve lift and an equal rate of acceleration. The choice of cam nose radius is governed by the specific pressure between the cam flank and the cam follower. The radius should not be less than 2 mm. These requirements affect the ultimate choice of the base circle radius.

An exact calculation of the cam including flanks will be indicated below, with cam lift for every 1° to the nearest 1/1000 mm.

Cams constructed of circular and straight line sections result in sudden changes from acceleration to deceleration. This sudden change in acceleration has a detrimental effect on resilient valve operating mechanisms and may cause unwanted vibration of the valves and even valve float. With the rate of valve lift almost unaffected, the abruptly phased rate of cam lift may be supplemented by a continuous rate of lift, accompanied by a slight increase of the period during which the valve is off its seat. The calculation of a cam with a continuous increase in the rate of acceleration is highly complex and several ways of producing a suitable rate of acceleration exist [63]. Such continuously accelerating cams are in use today in high speed engines, especially in conjunction with self-adjusting hydraulic tappets.

3. Valve clearance and its determination

Valve clearance is needed to enable the valve to seat properly and to form a gas-tight seal with the valve-seat face. With a lack of clearance, the valve cannot form a proper seal and escaping hot gases rapidly heat and damage the valve and its seat. With a view to silent valve gear operation especially in air-cooled engines, it is advisable to keep this clearance as low as possible and ensure that its variations during running are kept down to the lowest possible amount.

Valve clearance varies during engine warming-up and cooling-down. To demonstrate this, the overhead valve engine, shown in diagrammatic form in Fig. 275 may be studied. With the engine temperature rising, changes take place which are described below by the use of the following symbols:

t_a denotes the mean valve temperature

t_b „ „ mean tappet and camshaft temperature
t_c „ „ mean crankcase temperature
t_d „ „ mean cylinder temperature
t_e „ „ mean push rod temperature
t_f „ „ mean cylinder head temperature
t_n „ „ original temperature
α_a „ „ coefficient of thermal expansion of valve
α_b „ „ coefficient of thermal expansion of tappet and camshaft
α_c „ „ coefficient of thermal expansion of crankcase
α_d „ „ coefficient of thermal expansion of cylinder
α_e „ „ coefficient of thermal expansion of push rod
α_f „ „ coefficient of thermal expansion of cylinder head.

The distance of the valve stem and surface from the cam centre increases by

$$L_1 = C\alpha_c(t_c - t_a) + D\alpha_d(t_d - t_n) + A\alpha_a(t_a - t_n) \quad \text{(mm)}$$

The push rod and tappet increase in length by

$$L_2 = B\alpha_b(t_b - t_n) + E\alpha_e(t_e - t_n) \quad \text{(mm)}$$

The rocker axis to cam centre distance increases by

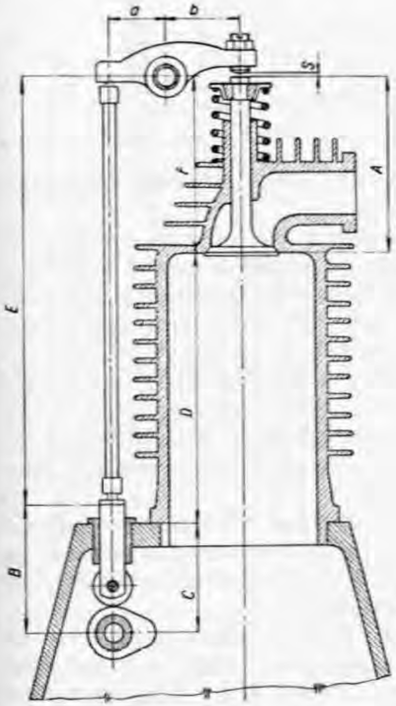
$$L_3 = C\alpha_c(t_c - t_n) + D\alpha_d(t_d - t_n) + F\alpha_f(t_f - t_n) \quad \text{(mm)}$$

Valve clearance with engine hot is

$$s = L_3 + (L_3 - L_2) \cdot \frac{a}{b} - L_1 + 0{\cdot}05 \quad \text{(mm)}$$

assuming clearance with engine cold to be adjusted to 0·05 mm. Dimensions calculated as in Fig. 275.

With these temperatures known, the variation of valve clearance may be calculated. On a side valve layout, clearance is usually reduced on a hot engine,



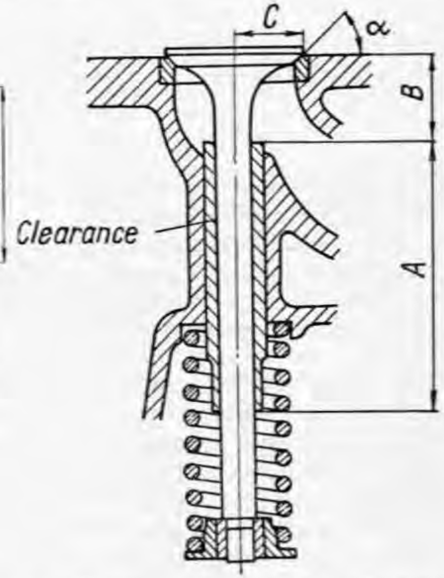


FIG. 276. Jammed position of valve in the guide.

FIG. 275. Schematic representation of an ← OHV valve gear.

while on an overhead valve system it increases. A side-valve engine therefore operates silently with small valve clearance when hot. In order to reduce valve gear noise, either precautions must be taken to prevent clearance fluctuation during the warming up period, or a ramp of a height equal to the increase in clearance must be formed on the cam. Fluctuations of valve clearance can be prevented, or at least reduced, by the choice of suitable constructional materials for the various valve gear parts. For a small clearance on an overhead valve engine, low expansion cylinders and high expansion push rods are desirable. Clearance fluctuations can be reduced by attaching the rocker pivots to the cylinder head by retaining bolts, these bolts being supported by the head only at its base near the joint flange. Expansion of the aluminium alloy head does not then affect rocker pivot position. An oblique position of the push-rods also reduces the influence of cylinder barrel expansion.

Owing to the clearance of the valve stem in its guide, one side of the valve head tends to seat earlier than the other, the valve being usually in a slightly jammed position in its guide, as will be seen in Fig. 276 and as is expressed by the formula

$$s = \frac{(0{\cdot}5\,A + B)\,.\,\tan\alpha + C}{A}\,\mathcal{J} + I,$$

where s denotes the required valve clearance
A, B, C, α — as in Fig. 276
$\mathcal{J}$ „ „ valve guide clearance (mm)
I „ „ deflection of the valve disc when seating

(0·05 mm) increased by an amount due to the leakage of the hydraulic tappet (about 0·025 mm) if employed.

With normal tappets, the required clearance necessary for dealing with expansion of the engine through heat must be calculated instead of allowing 0·025 mm. As far as the closing face of the cam is concerned, the influence of valve gear deformation (chiefly of the push rods and rockers) by accelerating forces and by valve spring pressure must be taken into account, together with factors previously mentioned. Owing to this resilience, the push rod is reduced in length and the valve will seat earlier than expected by theory. On the valve opening side of the cam the effects of this deformation are not as harmful, as they result in slower opening of the valve, but they can put the whole valve gear in a state of resonance.

With large valve clearance, considerable shocks are transmitted through the valve gear and the valve contacts the seat at high velocity thus buffeting the seat. Valve seat buffeting increases with the cube of seating velocity which must therefore be kept within permissible limits.

The seating, or, respectively, opening velocity increases rapidly with increasing valve clearance with a normal cam. As a result, wear and noise increase rapidly. In order to prevent this, the foot of the lifting face of the cam is provided with a ramp ensuring constant velocity within the clearance variation which accompanies the engine warming-up period.

Fig. 277 shows the variation of valve timing and valve opening velocity with the variation of valve clearance. The full line relates to a cam with ramps on both faces which keep valve opening velocity constant throughout the range of valve clearance variation. The stress on the valve gear and the noise emitted by it therefore do not differ with varying valve clearance, but large fluctuations in valve timing are caused.

According to the illustration the timing of both cams with the engine warm is the same, $IO = 10°$ and $EC = 15°$. These figures are valid for a clearance of 0·4 mm. On a cold engine with 0·1 mm valve clearance a rampless cam

is timed $IO = 15°$ $EC = 20°$, whereas a cam with ramps is timed $IO = 25°$ and $EC = 30°$. With the engine hot, however, the valve opening velocity of a rampless cam is greater after the valve is lifted off its seat. The effective opening is therefore not the same for both cams and the quieter running of the cam with ramps is accompanied by smaller valve lift, a condition which has to be provided for in design. Larger fluctuation of valve timing occurs with

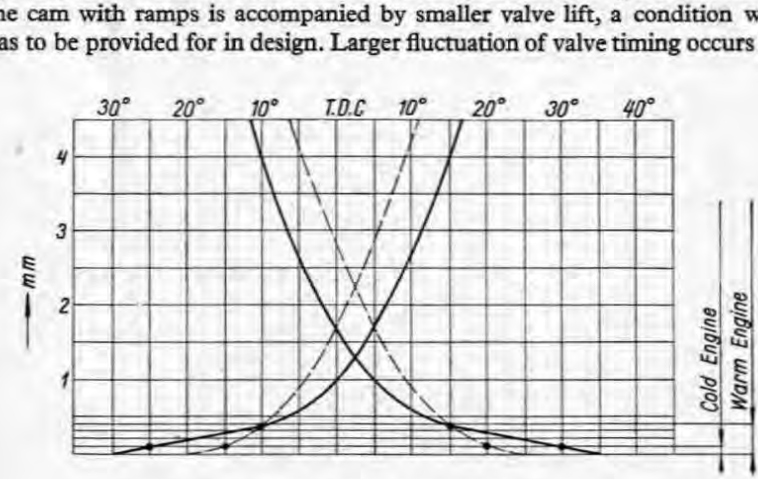


FIG. 277. Variation of valve timing with valve clearance.

the cam with ramps, in this instance the increase being 15°. The large variations of valve timing with such ramps with clearance variations must be borne in mind when checking the valve timing. A clearance variation of 0·1 mm results in considerable variation in timing, expressed in degrees. For this reason it is convenient to give valve timing data relating to an adjustment with greater valve clearance than is normally employed when the engine is cold in order to be able to measure timing on the more inclined opening face of the cam instead of on the ramp.

The simplest method of obtaining the profile of a ramp compensating for valve clearance is shown [63]. The cam profile is drafted without regard to the ramp and valve lift curve and the ramp is then interposed at the point at which valve velocity equals the required ramp velocity. The following velocities are recommended for opening-side ramps:

automobile high-speed engines	1·0 to 3·0 m/min,
commercial vehicle and low-speed engines	2·0 to 4·0 m/min,
compression-ignition engines, small aircraft engines . .	4·0 to 6·0 m/min,
large compression-ignition and large aircraft engines	6·0 to 20·0 m/min.

These velocities are given for an engine speed of 900 rev/min. With higher speeds the noise of valve gear is higher, but still lower than that of other sources of noise in the engine. The ramp should be shaped as an involute curve, i.e. with a uniform increase of lift with the angle of cam rotation.

The construction of such a ramp is illustrated by Fig. 278. The lift curve of a valve without ramps marked in dotted line relates to the actual (effective) valve lift h_e. Valve opening velocity is entered in the same diagram. The ramp speed v_n is now chosen and a line parallel with the base is drawn at this

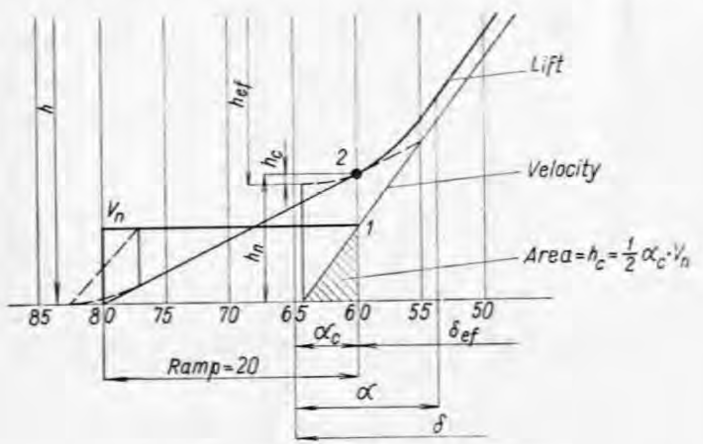


FIG. 278. Design of the cam ramp.

distance. At the intersection of both velocity curves the ramp cam should be adjoined to the valve-lift curve of the cam. Velocity curves will intersect at point *1* and the ramp will therefore be adjoined to the lift curve at *2*. The ramp is represented at point *2* by a tangent, the slope (ram steepness) of which equals the required velocity. If the required height of ramp h_n is entered from *2*, the size of the base circle is obtained. Valve lift on a cold engine will then be

$$h = (h_e - h_c) + h_r \tag{72}$$

where h_c — lift correction $= 1/2\,\alpha_c$, multiplied by the steepness.

The effective valve opening is then

$$\delta_e = \delta - \alpha_c \tag{73}$$

$$\sin \alpha_c = \frac{\frac{360}{2\pi} \cdot v_r}{r_3 - r_1}, \tag{74}$$

$$r_3 - r_1 = \frac{s^2 - (s - h_e)^2}{2\,(s - h_e - s \cdot \cos \delta)} \tag{75}$$

A certain acceleration of the tappet must take place from the beginning of ramp travel. If the slope of the ramp is smaller than 0·01 mm per degree of crankshaft rotation, no alteration of the beginning of the ramp is required, the transient section being produced automatically during grinding.

If valve clearance fluctuation is large and the specified velocity on the ramp section small, a prolonged ramp will ensue, resulting in great timing variations between cold and hot engine conditions. This has an undesirable effect on engine starting and idling. The designer should therefore try to eliminate variations of valve clearance by some automatic means, the most widespread and most convenient being the use of hydraulic tappets with automatic clearance control.

The hydraulic tappet operates in the manner shown in Fig. 279. It incorporates a cylinder with a non-return ball valve at its base and a plunger which is pressed by a spring towards the valve. With the valve closed the cylinder volume under the plunger is filled with fluid and the ball valve is closed. The tappet base is in contact with the cam and the plunger is forced towards the engine valve.

At the beginning of valve lift the non-return valve does not permit the oil in the cylinder to escape and thus the valve begins to lift. The force of the cam acts on the valve through the fluid and plunger. During extreme valve lift the ball valve is still closed and normal valve operation takes place, although a small portion of the oil passes around the plunger owing to leakage. The maximum permissible plunger downward movement during one stroke is 0·02 mm.

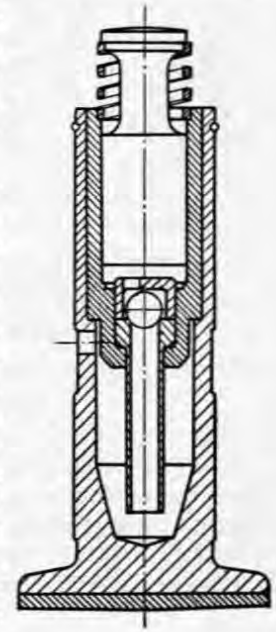


FIG. 279. Cross section of the hydraulic tappet.

When the stroke is at its end, the tappet base is still in contact with the cam and the plunger is displaced by its spring towards the poppet valve. The ball valve opens, permitting the ingress of oil which was lost by leakage during the previous stroke. After this "topping up" of the cylinder, the ball valve closes and the tappet is ready for a new cycle.

As poppet valve temperature rises, the plunger is pressed by the spring in the direction of the poppet valve and with the expansion of the valve gear, oil which takes up the clearance repeatedly enters into the cylinder. Were the plunger absolutely leakproof, the engine valve would remain open after a drop in engine temperature. The amount of leakage past the plunger is however such as to ensure the proper seating of the valve.

The hydraulic tappet controls valve clearance and its oil cushion damps the vibrations of the valve actuating mechanism, preventing resonant vibration.

As a result, valve opening velocity may be increased and ramps omitted and a rise in engine output thus effected.

Maintenance is simplified as there is no valve clearance adjustment to attend to. The use of hydraulic clearance adjustment on aircraft engine rockers is shown in Fig. 280.

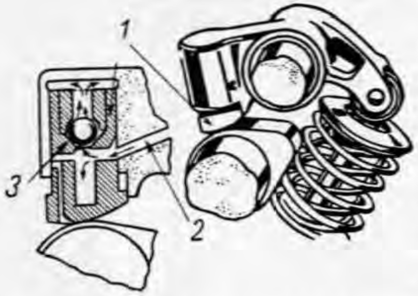


FIG. 280. Hydraulic tappet in the rocker of an aircraft engine.
1 — hydraulic tappet, 2 — oil inlet, 3 — non-return valve

At high revolutions a valve actuating mechanism of imperfect design begins to vibrate and the tappet is bounced off the cam. At every bounce oil enters into the tappet cylinder thus "pumping" it up until the poppet valve ceases to close properly. This results in loss of output, valve scorching and in some cases a possible collision of the valve with the piston. Valve drive incorporating hydraulic tappets must therefore be designed to a high standard of perfection in order to rule out resonant vibrations.

4. Forces acting upon the valve

The velocity of valve opening is limited by the accelerating forces acting upon the valve. The accelerating force is mass multiplied by acceleration. However, first of all the weight and acceleration of all moving parts must be reduced as acting on the valve or cam. The following equations are valid for the reduction of mass, acceleration, velocity and lift:

$$m_1 = m_2 \left(\frac{J_2}{J_1}\right)^2 \tag{76}$$

$$a_1 = a_2 \cdot \frac{J_1}{J_2} \tag{77}$$

$$v_1 = v_2 \cdot \frac{J_1}{J_2} \tag{78}$$

$$c_1 = c_2 \cdot \frac{J_1}{J_2} \tag{79}$$

where mass $= \frac{W}{g}$ (kg s²/m)

W denotes the weight (kg)
g „ „ gravitational acceleration (m/s²)
a „ „ acceleration (m/s²)

v denotes the velocity (m/s)
c „ „ lift (mm)
y_1 „ „ rocker arm length on the valve side (cm) (see Fig. 281)
y_2 „ „ rocker arm length on the cam side (cm)
1 „ „ subscript indicating the valve side
2 „ „ subscript indicating the cam side

For the reduction of the force exerted by the valve spring the following equation is used:

$$P_1 = P_2 \frac{y_2}{y_1} \tag{80}$$

Thus the force of the spring acting at the cam side P_2 may be reduced to the valve. If the constant of the spring on the cam is K_2, then at the valve side it will be

$$K_1 = K_2 \left(\frac{y_2}{y_1}\right)^2 \tag{81}$$

This formula makes clear the advantage of locating the spring at the valve instead of at the tappet.

With valve opening, accelerating force acts against the direction of motion at first, and is manifested as compression between the cam and tappet. The overall accelerating force acting on the cam will be

$$P_2 = \frac{a}{g}\left[W + \frac{1}{2}W_1 + \left(W_2 x^2 + \frac{W_1}{2} + \frac{W_3 k_1^2}{y_2^2}\right)\cos^2\beta\right] \text{ (kg)} \tag{82}$$

and the corresponding force at the valve:

$$P_1 = \frac{a \cdot x \cdot \cos\beta}{g}\left[\frac{W + \frac{1}{2}W_1}{x^2 \cdot \cos^2\beta} + W_2 + \frac{W_1}{2x^2} + \frac{W_3 k_1^2}{y_1^2}\right] \text{ (kg)} \tag{83}$$

If $\beta = 0$, then the equations are simplified to

$$P_2 = \frac{a}{g}\left(W + W_1 + W_2 x^2 + \frac{W_3 k_1^2}{y_2^2}\right) \text{ (kg)} \tag{84}$$

$$P_1 = \frac{a \cdot x}{g}\left(\frac{W + W_1}{x^2} + W_2 + \frac{W_3 k_1^2}{y_1^2}\right) \text{ (kg)} \tag{85}$$

The same formulae are used to calculate the accelerating force acting upon the valve spring during the second half of lift. The symbols used in the formulae are:

W denotes the weight of the complete tappet (kg)
W_1 „ „ weight of push rod (kg)

W_2 denotes the weight of the valve and one half of the spring weight (kg)
W_3 „ „ weight of rocker (kg)
k_1 „ „ radius of gyration of the rocker (cm)
x „ „ rocker arm ratio
a „ „ cam acceleration (m/s^2)
β „ „ angle included by the tappet axis and the push rod at half-lift

The valve springs must exert a pressure exceeding the force of acceleration in view of valve guide friction and the possibility of running the engine above the maximum speed.

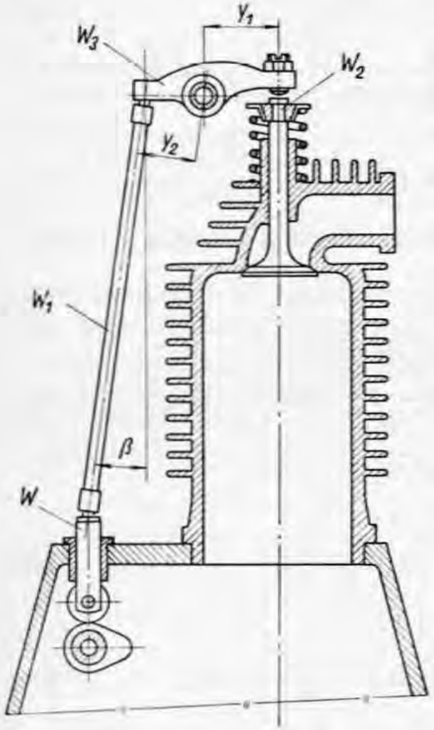


FIG. 281. Illustration of valve gear.

The amount of friction in the valve guide depends on side thrust in the guide. This thrust should be diminished by suitable rocker design. The following values must be added:

for friction in the valve guide, approx. 10%,

for speed — 10% in excess of maximum, approx. 21%,

for speed — 20% in excess of maximum, approx. 44%.

The spring pressure with valves closed must be greater than

$$P_1 = p_1 \,.\, A_e \quad \text{(kg)}$$

where p_1 denotes the maximum under-pressure in cylinder during induction stroke (kg/cm^2)
A_e denotes the exhaust valve area (cm^2)

Beside the force of acceleration, the force of valve springs also acts upon the inlet and exhaust cams

$$P_s = (S + K \,.\, s \,.\, x \,.\, \cos\beta) \,.\, x \,.\, \cos\beta \quad \text{(kg)}$$

where S denotes the force of the spring with valve closed (kg)
K „ „ force of the spring compressed by 1 cm, i.e. the spring constant (kg/cm)
s „ „ cam lift (cm).

The overall cam load

$$P = \frac{a}{g}\left[W + \frac{1}{2}W_1 + \left(W_2x^2 + \frac{W_1}{2} + \frac{W_3k_1^2}{y_2^2}\right)\cos^2\beta\right] + \\ + (S + K \,.\, s \,.\, x \,.\, \cos\beta) \,.\, x \,.\, \cos\beta \quad \text{(kg)} \tag{86}$$

and with $\beta = 0$

$$P = \frac{a}{g}\left(W + W_1 + W_2x^2 + \frac{W_3k_1^2}{y_2^2}\right) + (S + K \,.\, s \,.\, x)\,x \quad \text{(kg)} \tag{87}$$

In addition to this the pressure of exhaust gas acting upon the valve is transmitted to the cam

$$P_e = p_2 \,.\, A_e \,.\, x \,.\, \cos\beta \quad \text{(kg)}$$

where p_2 denotes the exhaust gas pressure during valve opening (kg/cm^2)
A_e „ „ area of exhaust valve (cm^2)

The overall load on the exhaust cam being

$$P_e = \frac{a}{g}\left(W + \frac{1}{2}W_1 + \left(W_2x^2 + \frac{W_1}{2} + \frac{W_3k_1^2}{y_2^2}\right)\cos^2\beta\right] + \\ + [S + K \,.\, s \,.\, x \,.\, \cos\beta]\,x \,.\, \cos\beta + p_2 \,.\, A_e \,.\, x \,.\, \cos\beta \quad \text{(kg)} \tag{88}$$

With $\beta = 0$, then

$$P_e = \frac{a}{g}\left(W + W_1 + W_2x^2 + \frac{W_3k_1^2}{y_2^2}\right) + \\ + (S{+}K \,.\, s \,.\, x)\,x + p_2 \,.\, A_e \,.\, x \quad \text{(kg)} \tag{89}$$

Tappets

In order to calculate the pressure of a plain, curved or roller cam follower of a tappet against the cam, the normal force N (Fig. 282) must first be found:

$$N = \frac{P}{\cos\eta}, \quad B = P \,.\, \tan\eta$$

where P denotes the axial thrust on the tappet
B „ „ side thrust on the tappet guide
η „ „ angle enclosed by the line connecting the curve centres of the curved surfaces in contact, with the tappet axis

In order to keep side thrust on the tappet guide low, $\eta > 30°$ should be avoided. The specific pres-

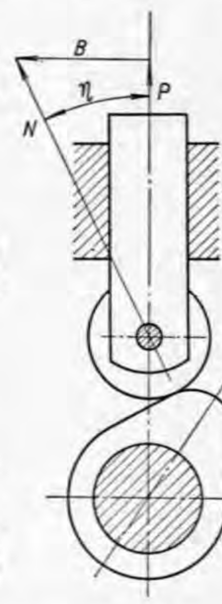


FIG. 282. Side thrust on the tappet guide.

sure at the contact of the cam with follower, given for two convex surfaces as

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{r_3 + r_2}{r_2 \cdot r_3}} \quad (\mathrm{kg/cm^2}) \tag{90}$$

one concave and one convex face

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{r_3 - r_2}{r_2 \cdot r_3}} \quad (\mathrm{kg/cm^2}) \tag{91}$$

one flat and one convex surface

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{1}{r_2}} \quad (\mathrm{kg/cm^2}) \tag{92}$$

where b denotes the cam flank width in cm

r_2 and r_3 denote the curvature radii in cm.

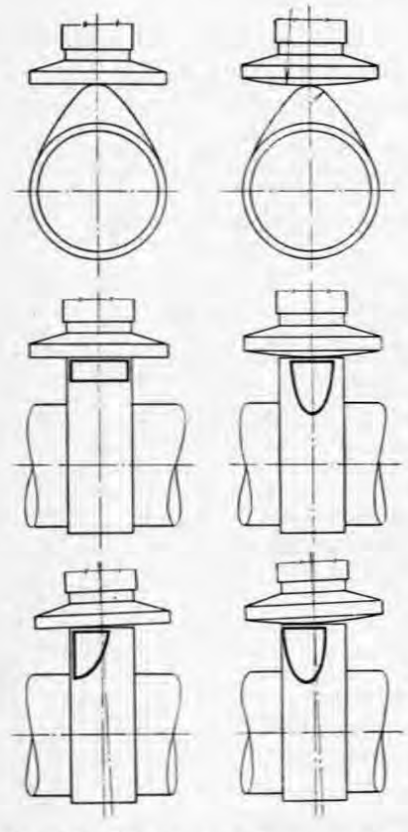


FIG. 283. Distribution of pressures between cam and tappet in the case of a flat and convex follower with the tappet axis not normal to the cam axis.

The use of two different materials for the follower and cam is advisable. With a combination of cast-iron with steel, greater thrust may be applied with safety than with an identical material used for both parts.

Practical experience has shown that a greater thrust can be imposed on part-spherical followers with a large radius of curvature than on straight surfaces. With a radius of about 29·5 in (750 mm), the contact area between cam and follower is reduced and the maximum stress thereby increased, but the possible effects resulting from imperfect follower in relation to the cam or from cam deflection are thus eliminated. Fig. 283 shows the distribution of pressures in the contact areas of a flat and spherical follower with a precisely perpendicular setting and with the tappet axis slightly inclined. If the tappet is not precisely at right angles to the cam, high boundary stresses may result. This is absent on a follower of spherical shape.

With cast iron tappets, the flat fol-

lower with hardened surface should not be subjected to loads exceeding 535 ton/in² (8500 kg/cm²) (according to results obtained by experiment) with a cam width of 12 mm. If the cam is wider, its full width can be taken into account only if an absolutely perpendicular tappet can be ensured together with cam rigidity. With the use of a spherical cam follower surface having a radius of 29·5 or 97 in (750 or 2500 mm), the load should not exceed 857 ton/in² (13 500 kg/cm²). With a non-rotating cam follower, values reduced by about 5% are applicable.

For a follower having a spherical surface the following equations apply:

$$K = \frac{a}{b}, \quad A = \frac{1}{r_2}, \quad B = \left(\frac{1}{r_2} + \frac{1}{r_4}\right), \quad D = (A + B),$$

$$\frac{B}{A} = \left(\frac{r_2}{r_4} + 1\right), \quad G = 2\,\frac{\dfrac{1-\sigma_1^2}{E_1} + \dfrac{1-\sigma_2^2}{E_2}}{D}, \quad C = \frac{G \cdot P}{a^3},$$

$$a = \left(\frac{G\,P}{C}\right)^{\frac{1}{3}}$$

where a denotes the long half axis of the ellipse
b ,, ,, short half axis of the ellipse
E ,, ,, modulus of elasticity
σ ,, ,, Poisson's ratio
P ,, ,, thrust in kg
r_4 ,, ,, cam summit radius
r_2 ,, ,, spherical follower radius

area of contact $= \pi \cdot a \cdot b$.

Specific pressure over area of contact

$$p = \frac{1{\cdot}5\,P}{\pi \cdot K \cdot a^2} = \frac{1{\cdot}5\,P^{\frac{1}{3}}}{\pi \cdot K \cdot \left(\dfrac{G}{C}\right)^{\frac{2}{3}}} \tag{93}$$

Valve springs

The required spring load is

$$P = m \cdot a_3 + 10\% \text{ to } 20\% \text{ reserve}$$

where m denotes the mass of all moving parts of the valve gear reduced to valve,
a_3 ,, ,, acceleration at cam nose reduced to valve.

The valve spring is best utilized when its constant K is proportional to the course of acceleration, when

$$K = \frac{m\,(a_3 - a_2)}{s_3 - s_2} \quad \text{(kg/cm)} \tag{94}$$

where a_3 denotes the maximum deceleration at the valve (m/s^2) at maximum lift s_3 (cm)

a_2 „ „ minimum deceleration at the valve in m/s^2 with lift s_2 (cm)

If the spring constant does not comply with this equation, the spring is not fully utilized. Hence it is necessary to take the characteristic of the spring to be used into account when designing a highly stressed gear.

The following equations are applicable for the calculation of cylindrical springs made of round wire:

$$d = \sqrt[3]{\frac{8 \,.\, P \,.\, D \,.\, \varphi}{\pi \,.\, k_t}} \quad \text{(mm)} \tag{95}$$

$$P = \frac{\pi \,.\, d^3 \,.\, k_t}{8 \,.\, D \,.\, \varphi} \quad \text{(kg)} \tag{96}$$

$$f = \frac{8 \,.\, P \,.\, D^3 \,.\, i}{G \,.\, d^4} \quad \text{(mm)} \tag{97}$$

$$k_t = \frac{8 \,.\, P \,.\, D \,.\, \varphi}{\pi \,.\, d^3} \quad \text{(kg/mm}^2\text{)} \tag{98}$$

Since the diameter of wire d is relatively large compared with the spring diameter D, the Wahl coefficient of correction φ must be used in the equation. Its value can be determined from Fig. 284.

$$\varphi = \frac{\dfrac{D}{d} - 0{\cdot}25}{\dfrac{D}{d} - 1} + \frac{0{\cdot}615}{\dfrac{D}{d}}$$

The following symbols are used in the equations:

D denotes the mean coil diameter (mm),

d „ „ diameter of spring wire (mm),

i „ „ number of coils,

P_1 „ „ load of spring as assembled (kg),

P_2 „ „ load of spring during maximum lift (kg),

f „ „ spring compression (mm),

k_t „ „ maximum permissible torsional stress (kg/mm^2),

G denotes the modulus of rigidity (shear) = 8200 kg/mm^2,

H „ „ spring constant $\dfrac{P_2 - P_1}{l_1 - l_2}$ (kg/mm).

The number of end coils is 1·5 to 2 the total number of coils of the spring thus being

$$i + 1{\cdot}5 \text{ to } 2.$$

The number of spring coils should not be a whole number, so that the end coils of the spring are seated on opposing sides, thus preventing lateral spring bulge, which is especially dangerous with long springs.

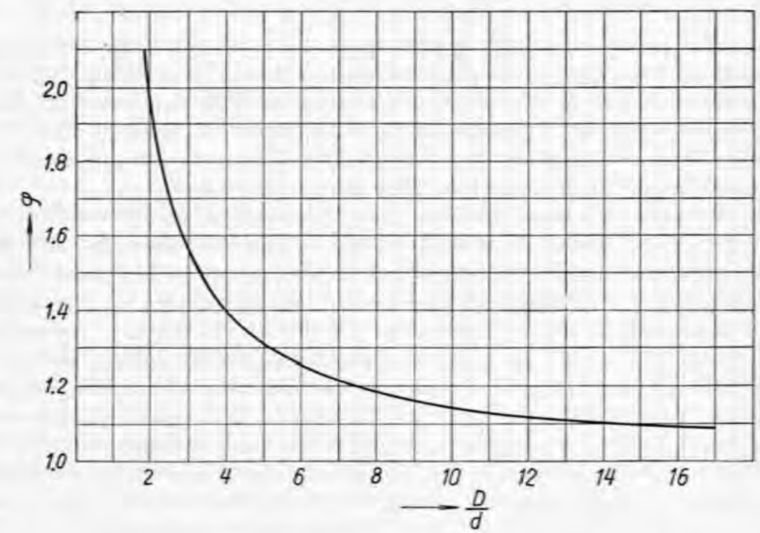


FIG. 284. Correction coefficient for the calculation of cylindrical springs.

The ratio of $P_1 : P_2$ should not be below 1 : 4 in the case of valve springs, otherwise fatigue fractures are frequent.

When designing the springs, spring load and the ratio of loads with the valve open and closed (the stress of a compressed and fitted spring) must not be overlooked. Fig. 285 shows the limits of permissible loading for safe operation obtained by tests of large numbers of springs.

The inherent advantage of hairpin springs is the reduction of the weight of moving parts in the valve actuating mechanism. Their requirements on space in the direction of the valve axis is much smaller than with coil springs, being determined by the valve lift only. It is therefore possible to shorten the valve stem considerably, thus further saving weight. If hairpin springs are not encased, their cooling is better than that of coil springs, but their demands on space are greater and encasing is more difficult.

Hairpin springs are attached by brackets with two slots for the spring wire and the bottom ends of the spring are inserted into holes in a washer.

The natural frequency of springs can be calculated from the formula

$$n = 485\,140 \frac{d}{i \, . \, r^2} \qquad \text{per min} \tag{99}$$

where d denotes the wire diameter (cm)

i „ „ number of coils

r „ „ mean coil radius (cm)

In order to ensure a high frequency of the spring, the maximum permissible torsional stress k should be chosen to be a sufficiently high value of the order of 31·7 to 38 ton/in² (50 to 60 kg/mm²). With fast running sports engines this value may be increased up to 50 ton/in² (80 kg/mm²). The outer spring should have from 3·5 to 5·5 coils. The lifetime of the spring can be increased considerably by shot blasting the spring surface.

Good results have been obtained by the pre-stressing of valve springs. In such a case the spring is stressed beyond the limit of elasticity. Finished springs are compressed solid several times before assembly. During compression, permanent deformation of the surface fibres of the spring takes place and when the spring is releaved, pre-stress remains in the inside of the spring material, but the surface fibres are under negative stress. This ensures a better utilization of spring material. The original length of the spring will thus be reduced before assembly, but no further reduction takes place during service. When designing a pre-stressed spring, the stress for compression to solid should be chosen at 54 to 66 ton/in² (85 to 105 kg/mm²).

It is obvious from Fig. 285 that to a certain extent the pre-stress of a spring depends on the maximum load with the valve open. The larger the maximum spring load, the higher must be the pre-compression with the valve closed. With higher pre-compression a higher number of turns must be used for a given valve lift, this having an unfavourable effect on the frequency of the spring. For this reason the use of maximum stress of spring material is of no great consequence.

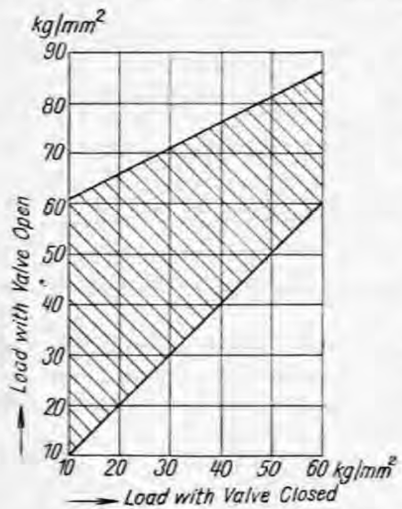


FIG. 285. Recommended stresses in valve springs.

When cams having a continuous course of acceleration in conjunction with a dynamically well designed valve gear are employed, it is advisable to use springs the natural frequency of which is at least above the 9th harmonic at maximum engine speed. With unsuitable valve gear dynamics, the frequency of the spring should be above the 11th harmonic at maximum engine speed. If the frequency lies between the 9th and 11th harmonic, progressive pitch springs are recommended with the pitch smaller at the stationary spring end. If the spring frequency is below the 9th harmonic the use of a friction oscillation damper is advised.

In air-cooled engines, heat insulation of the springs is beneficial. This can be effected by the use of a washer of a material with good thermal insulating properties or by the provision of an air space under a large portion of the spring washer.

The frequency of valve springs commonly used average 12 000 to 20 000/min. The frequency of the spring must be high enough in order to prevent spring resonance during engine running, which is the cause of considerably increased load. The impulse to such vibration might be given by the period of valve lift or the period of lift acceleration. The valve is open for about 1/3 of the revolution of the camshaft. The frequency of valve springs being always safely above three times the camshaft revolutions no resonant vibrations occur.

A more dangerous source of valve spring vibrations is the impulse given by the acceleration period during the first stages of valve lift. If this acceleration lasts over 36° of crankshaft rotation, then resonant vibrations arise if the frequency period of the valve springs happens to coincide with the time taken to turn the camshaft 72°. The frequency of the valve spring must therefore be at least ten times higher than camshaft revolutions in order to avoid the risk of vibrations. In order to prevent resonant vibration, the acceleration period should not be an even multiple of 720°. Periods of 30°, 36°, etc. are therefore unfavourable. Furthermore the period of deceleration should not be an odd multiple of the period of acceleration. For an accelerating period of 30° a decelerating period of 150° is therefore unsuitable. At high engine speed this can lead to vibration in the overhead valve actuating mechanism, as will be explained later.

The effects of valve spring vibration are unfavourable to engine running, resulting in noisy operation, irregular actuation of valves and possible valve fracture. Whenever it occurs a cure can be effected by the use of higher frequency springs or by changing the harmonic cams.

Effective damping of spring vibration may be obtained by coiling the spring so that its pitch progresses with the number of turns. During the compression of such a spring the coils are progressively compressed solid so that the number of coils varies during valve lift and with it varies the natural frequency; under such conditions resonant vibrations are impossible.

Valve gear is usually calculated under the assumption that it is perfectly rigid but during valve opening, the whole mechanism is subjected to great accelerating forces which are greater than the load of the valve spring. Under such load distortion takes place in the valve gear by bending the camshaft between supporting bearings, the compression of the push rod and bending of the rocker and sometimes even of the rocker shaft. These distortions result in conditions greatly differing from those arrived at by calculation and can be detrimental to engine running.

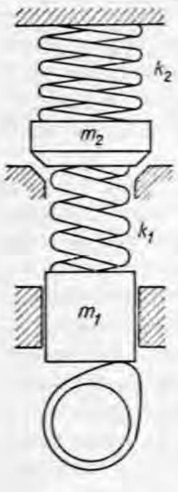


FIG. 286. Schematic representation of a resilient valve gear.

It is therefore necessary to take the resilience of the valve gear into consideration in the case of high speed engines and steps must be taken to reduce its effects. The overhead valve layout is the most flexible, others, i.e. the side valve and overhead camshaft being stiffer.

The whole valve gear may be imagined to be an elastic system depicted in Fig. 286. The mass of the entire gear is divided between the valve and the tappet, masses m_1 and m_2 resulting, k_2 being the spring constant of the valve spring and k_1 the constant of the valve actuating gear. The mass m_2 is interposed between two springs with spring constants k_1 and k_2.

The spring constant k_1 can be measured by applying a static load to the valve gear (Fig. 287) and measuring distortion. Frequency can then be calculated from the formula

$$\omega = \sqrt{\frac{k_1}{m_2}}$$

The precision of the result obtained depends on the correct reduction of masses, a much more reliable way being the direct measurement of vibration of the valve actuating mechanism. This is done on the engine in the following manner. The crankshaft is rotated until the valve under consideration lifts off its seat for at least one half of its total lift. Then a strip of sheet metal about 1·5 mm thick is inserted between the valve stem and the rocker arm.

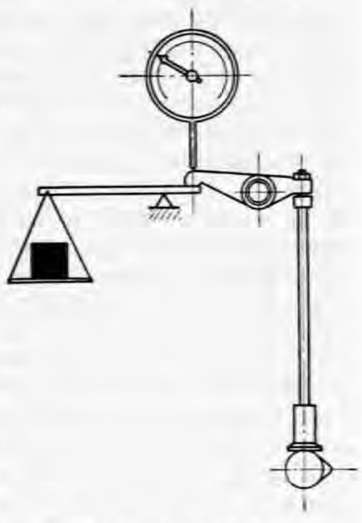


FIG. 287. Measurement of the elastic deformation of the valve gear by static load.

An electro-magnetic pick-up is then brought near to the rocker and connected with an oscillograph. By a brisk withdrawal of the strip insert, the valve stem and rocker pad are brought into collision and the whole system vibrates. The frequency of the system can then be calculated from the oscillograph readings and time recording. The frequency should be about 40 000 to 50 000/min. With valve gear having small rigidity, much lower values down to 20 000/min are obtained. In such a case the danger of resonant vibrations of the system becomes real.

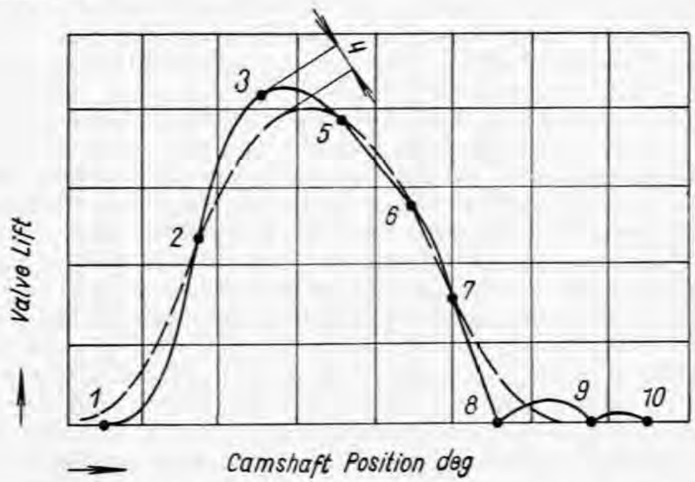


FIG. 288. Theoretical (dash line) and real (full line) course of the valve lift showing the effect of valve gear resilience.

The valve gear will begin to vibrate when a force acts upon it at regular intervals which are a multiple of the natural frequency of the system. Since vibration is caused by elastic deformation of the valve actuating gear, a variation in valve lift must obviously follow. An example of such a variation in valve lift is shown in Fig. 288. The dotted line denotes valve lift according to cam shape, obtained by cranking the engine or during idling speeds. At high engine speed the variation drawn in full line may take place in engines with resilient valve gear. During the valve opening phase considerable delay occurs, as when the lifting force comes into action, distortion takes place and only after it has reached a certain stage will the valve begin to lift off its seat at point *1*. There is, therefore, some delay in comparison with the ideal valve opening curve. The distorted valve gear now acts in much the same way as a compressed spring. After the largest initial acceleration of the gear imparted to it by the cam is over, the valve actuating gear will begin to show a reduction in the amount of distortion and will appear to expand at both ends. As, however, one end is pressing against a rigid cam, the expansion will all be apparent in the direction of the valve. Real valve lift will therefore catch up with ideal valve lift at point *2*. Expansion of the valve gear will, however, continue owing to the effects of inertia, for at point *3* the acceleration of the valve is larger than that corresponding to the cam shape. The valve (tappet) will therefore "float" off the cam by the magnitude of *4* and will contact the cam again at *5*. Similar valve float may occur on the closing face of the cam at section *6—7*.

A further distortion of the valve gear takes place owing to the acceleration imparted to it during the closing of the valve. Owing to the great decelerating force of the valve, the actuating mechanism is compressed once more and the valve will contact its seat at *8*, whereas it should still be open by the amount of distortion as shown by the dotted line. The rapid seating of the valve is the cause not only of noise, but also of rebound of the valve, which seats again at *9* and possibly bounces again to seat finally at *10*.

If the amount of valve float is not large enough to cause collision with the piston, it manifests itself through noisy operation and increased wear. If, however, hydraulic tappets are incorporated, they will be "pumped up" and correct valve operation will be impaired. When the valve floats or bounces, oil enters the tappet and cannot escape as the ball valve closes and the valve will remain open during the remaining phase of the stroke and will not seat properly when it is supposed to. During the next cycle this "pumping" action is repeated, valve lift being increased again and the valve remaining continuously open. Blow-by to the exhaust and inlet ports results, causing irregular running. Special care must therefore be taken in the design of valve gear incorporating hydraulic tappets, in order to exclude the possibility of valve float. In engines of current size, the danger of "pumping-up" commences at speeds of about 4500 to 5000 rev/min.

The critical multiples of engine speeds at which the greatest danger of valve float exists, are different for different cam profiles, depending on the course of acceleration. Fig. 289 shows two typical courses of acceleration: 1) constant acceleration, which is very closely attained by harmonic cams and *2*) continuous course of acceleration.

In both cases the critical multiples of natural frequencies are also stated. In the instance of constant acceleration unfavourable conditions arise when the period of acceleration coincides with a 1/2 cycle of the natural frequency of the valve gear, i.e. $N=0{\cdot}5$, N being the ratio of acceleration period to the period of one oscillation of the valve gear. As the natural frequency of the valve gear is independent of engine revolutions and the acceleration period is dependent on engine speed, the critical unfavourable engine speeds may easily be calculated.

With a cam with gradual acceleration the critical conditions arise when $N = 1$, i.e. when the frequency period equals the period of acceleration. Fig. 289 shows by dotted line the theoretical course of acceleration and in the full line the real course of acceleration of the valve taking into account the resilience of the valve gear and the effects of damping. The illustration shows that

in the first case, owing to resilience in the valve gear, the constant acceleration is transformed into a continuously changing acceleration, the maximum of which is, however, about 75% above that of the original constant acceleration. This is not dangerous and results in an increase of thrust between the cam and follower. The influence of the resilience of the gear, however, is also apparent in the decelerating section and here it is highly inconvenient.

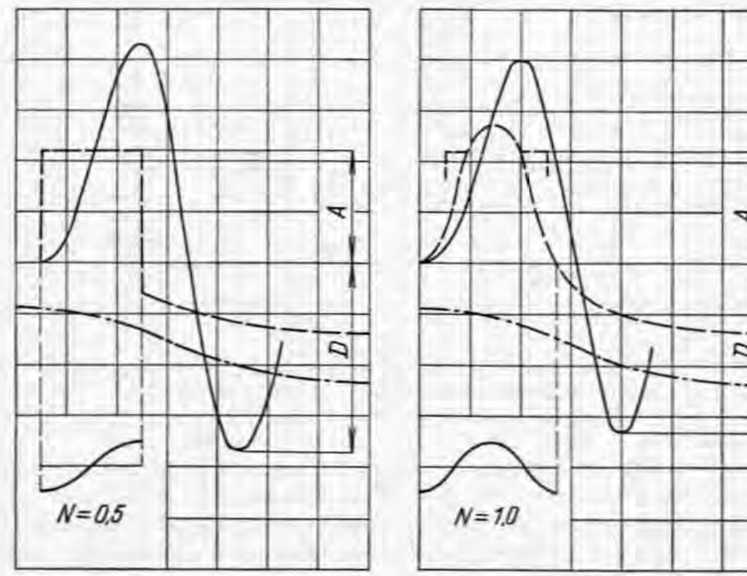


FIG. 289. Theoretical (dash line) and real (full line) course of acceleration of a cam with constant acceleration (left hand) and with a continuous course of acceleration (right hand). The dash and dot line represents the valve spring load.

It will be apparent from the illustration that deceleration D is greater in magnitude than theoretical acceleration A and several times greater than theoretical deceleration D_t.

The dot and dash line marks the deceleration which is in balance with the load of the valve spring at maximum engine speed. This deceleration is considerably greater in magnitude over the whole speed range than theoretical deceleration and valve float should therefore not occur. The deceleration transferred under the influence of valve gear resilience is, however, much greater than the reserve of the spring and float therefore does occur. The real deceleration with valve gear oscillation is independent of the theoretical deceleration but it does depend on the magnitude of theoretical acceleration. Its magnitude D is therefore compared with the magnitude of theoretical acceleration A.

Fig. 290 shows the dependence of the ratio D/A on the value of N with a resilient valve gear with damping by internal friction assumed.

Interesting conclusions may be drawn from this diagram. With a constant acceleration cam, valve gear vibrations may arise even if the valve gear frequency is high, since with $N = 1{\cdot}5$, the real deceleration D equals theoretical acceleration A. With $N = 0{\cdot}5$, the value of D is about $1{\cdot}7\ A$, such a case being possible

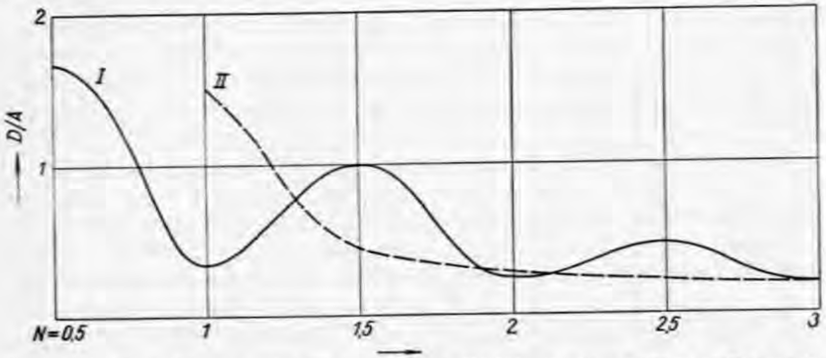


FIG. 290. Ratio of deceleration to acceleration D/A due to the resilience of the valve gear plotted against natural frequencies of the valve gear.
I — cam with constant acceleration, *II* — cam with gradual acceleration.

in very fast running sports engines with a valve gear which is insufficiently stiff.

In the case of the cam with gradual acceleration with $N = 1{\cdot}5$ D is low and there is no risk of valve float. For this reason cams with a gradual acceleration are to be preferred for high speed engines and especially where hydraulic tappets are employed. When the value of N exceeds 1·3, gradual acceleration cams are more suitable than constant rate cams, under this point they are however at a disadvantage. The basic difference between the two types however lies in the fact that with gradual acceleration cams, the value of D/A is continuously decreasing, whereas with cams of constant acceleration, it fluctuates so that while the valve gear may operate satisfactorily at maximum speed, dangerous vibration may arise at lower engine speeds.

During the development of the air-cooled Tatra 603 engine for sports-car use, impressions of a floating inlet valve appeared on the piston crown at speeds from 7 000 to 7 500 rev/min. The piston is at t.d.c. 40° after I O and the valve acceleration phase in the engine in question was about 30°. The rate of acceleration was nearly constant. The frequency of the valve gear was measured and it was found that at maximum speed the value of $N = 0.5$ was being approached. The cam profile was therefore modified so that the acceleration phase was increased by about 5°, the rate of valve lift remaining practically unchanged. Valve timing was somewhat affected (by an increase in the period for which the valve was off its seat), but the magnitude of acceleration was reduced. A two-fold improvement was thus gained.

1. By increasing the period of acceleration the critical speed range, at which the period of acceleration would cover 1/2 of the natural frequency, was raised.

2. The reduction of the magnitude of acceleration resulted in a reduction of the induced deceleration due to the resilience of the valve gear which depends on the magnitude of acceleration.

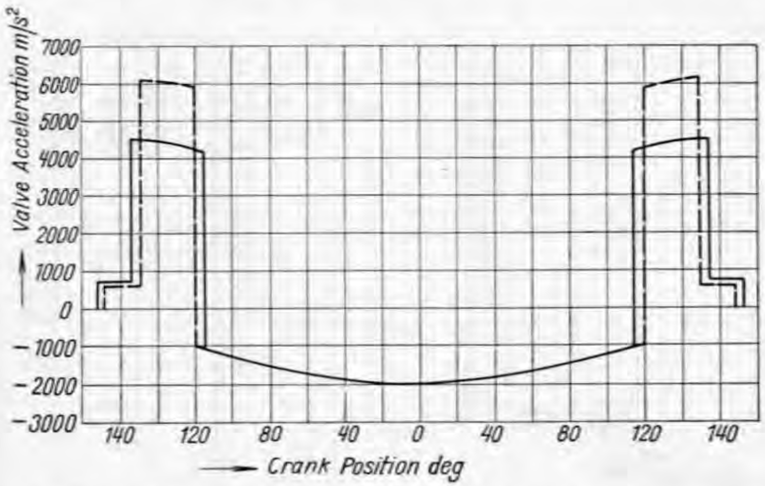


FIG. 291. Theoretical course of acceleration in a T 603 sporting engine at 7,000 rev/min, original (dash line) and modified (full line).

The original rate of acceleration (dotted line) and the modified acceleration (full line) are shown in Fig. 291. This modification eliminated valve float. The use of a cam with a gradual rate of acceleration would have brought no improvement in this case. An increased valve gear rigidity would have been beneficial. With a cam of constant acceleration, the speed at which the acceleration period is 1·5 times that of the natural frequency of the gear must be checked. At such revolutions the spring must be strong enough to prevent float with an induced deceleration of considerable magnitude.

The important point is to avoid the promotion of valve gear vibration through resonance at critical speeds. The deceleration period therefore should not be an odd multiple of the period of a half-wave of the natural frequency of the valve gear and the period corresponding to one camshaft revolution should not be a multiple of the period of one wave of the natural frequency. This means in practice that the deceleration period expressed in terms of degrees of rotation of the camshaft must not be an odd multiple of the acceleration period and that 360° must not be the even multiple of the double or 2/3 period of acceleration.

5. Rotary and sleeve valves

The chief advantage of rotary and sleeve valve systems is the absence of the hot exhaust valve in the combustion chamber and thus the possibility of increasing the compression ratio without an accompanying increase in the risk of detonation. In rotary or sleeve valve engines mixture enters the cylinder

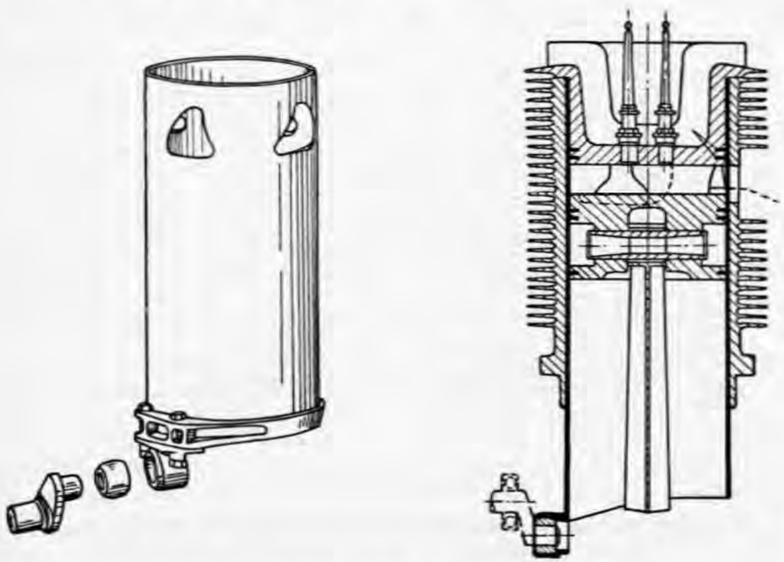


FIG. 292. Actuation of the Burt sleeve valve.

without great changes in the direction of flow through the port in the cylinder wall or in the cylinder head and it is not pre-heated by the exhaust valve. The coefficient of flow of such ports is therefore favourable.

Another advantage of rotary valve engines is the fact that no valve maintenance, such as clearance adjustment, valve grinding etc., is required. A more complex valve actuating mechanism is sometimes necessary. Such valve systems are not used in automobile engines but in aircraft engines they are used with success, particularly by Bristol.

The Burt sleeve valve arrangement employs only a single sleeve which is given both axial and rotational movement. The sleeve is driven through a short crank and a ball and socket joint, as is apparent in Fig. 292. Every point on the sleeve describes an ellipse, the major axis of which is at optimum conditions at a 3 : 2 ratio to the minor axis. A disadvantage of this layout is the fact that heat must be abstracted from the piston through a sleeve and the two oil films formed on its surfaces.

The flow areas of the ports, the port arrangement and sleeve movement are shown in Fig. 293. The sleeve moves in such a manner that during the compression stroke the sleeve port is positioned inside the head and above the sealing rings, proper sealing thus being ensured. A disadvantage of many sleeve valves is absent in this case, namely that of the sleeve port opening the

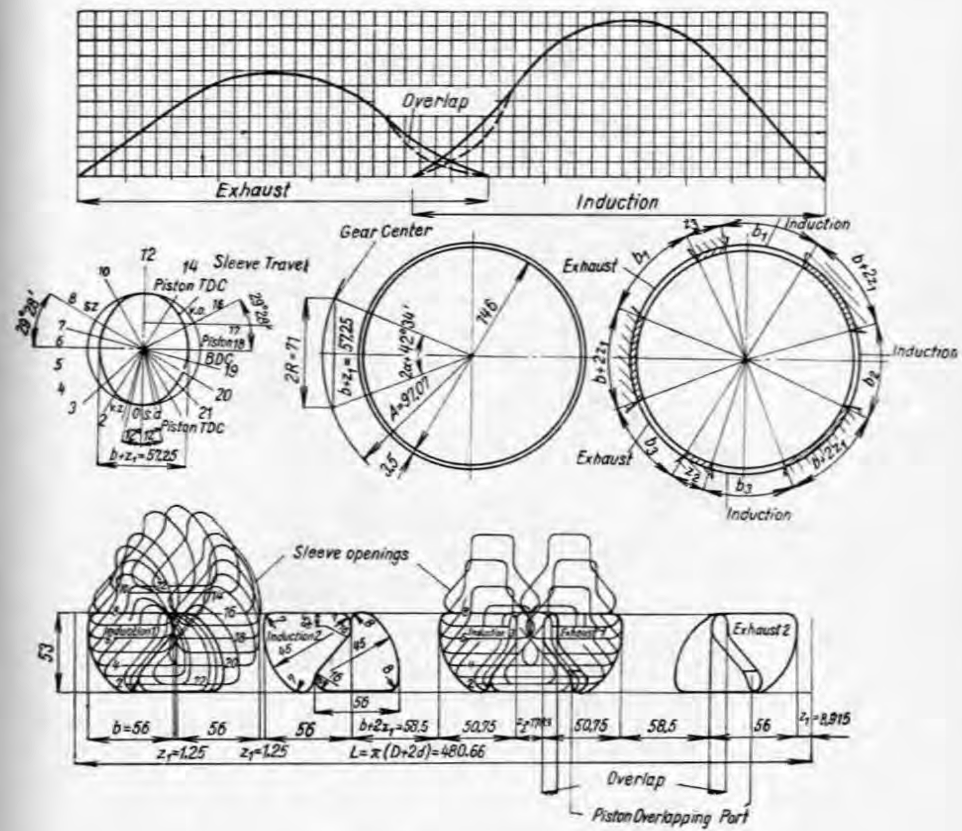


FIG. 293. Bristol sleeve valve gear.

way for burning gas to the cylinder wall or head within which the sleeve is operating. With some other layouts this results in the burning of oil in such exposed places, and in the formation of carbon which causes sleeve guide wear and can even result in seizure.

The sleeve is in contact with the cylinder over a large area. During engine starting from cold at low temperatures the oil offers a large resistance.

An engine equipped with this valve gear has many other disadvantages and troubles, especially with cylinder head cooling etc.

A very good example of the development of an air-cooled cylinder head with a limited base area for finning is given by the Bristol engine. The outer surface of the cylinder head upon which fins can be formed is small and is not exposed to a direct air stream. The cylinder head and fins originally formed an

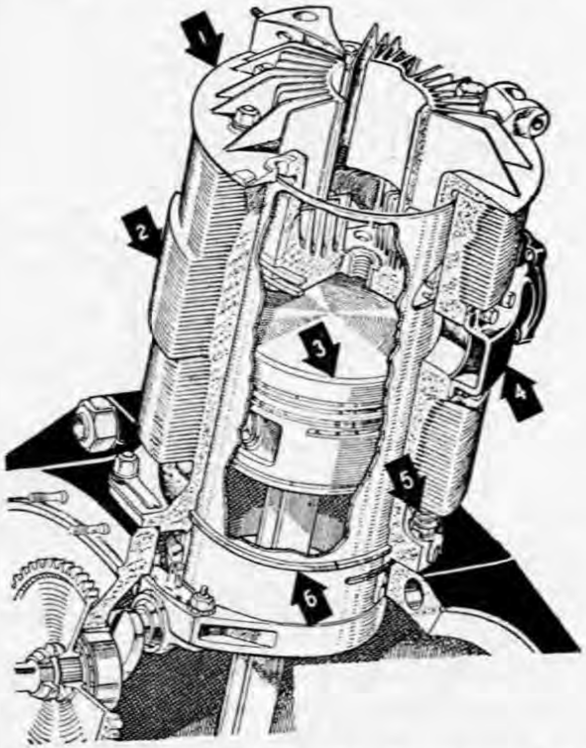


FIG. 294. The original head and cylinder of the Bristol sleeve valve engine.
1 — cylinder head with sealing rings, *2* — aluminium cylinder, *3* — piston with piston rings, *4* — induction pipe, *5* — securing bolt of cylinder, *6* — scraper ring of the sleeve valve.

integral casting with a cooling area of 539 in² (3480 cm²) and a weight of 9 lb (4.3 kg). By improved casting into steel moulds the height of finning was increased and so was the cooling surface to 655 in² (3750 cm²) with a resultant weight of 9·5 lb (4.54 kg) (Fig. 294).

No further increase in the cooling area of a cast head was possible making necessary recourse to a split head [47]. The head consisted of a base-plate which was pressed into the upper, flanged part. The base was of Y-alloy and

on its upper part, parallel fins, 3·8 mm apart, were cast. This brought an increase of 147 in² (950 cm²) in cooling area, total cooling area per head being now 728 in² (4700 cm²). The weight of the head was reduced to 9 lb (4.33 kg). The increase in cooling area resulted in a temperature reduction of 15 °C compared with the standard series type. Further improvements in foundry technique made possible a finer spacing of 3 mm and the resultant increase in cooling area to 775 in² (5000 cm²).

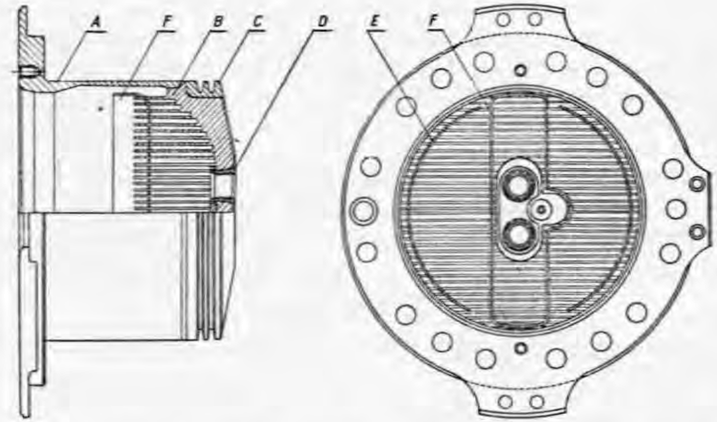


FIG. 295. Final design of the cylinder head of the Bristol sleeve valve engine.
A — steel sleeve with flange, *B* — finned base plate of a copper-chromium alloy, *C* — cast iron holder of sealing rings, *D* — Monel metal spark plug quill, *E* — anti-vibration stiffener of fins, *F* — channel and air duct.

By a further increase of engine performance the temperature of the cylinder head was raised once more and material strength was used to the limit. Experiments with bronze heads resulted in reduced temperature but at the expense of doubling cylinder head weight. Further tests were conducted with the head as an integral casting with cast-in copper cooling spikes. In all 173 spikes were employed, of 3·2 mm diameter and 37 mm length, giving a cooling area of 100 in² (650 cm²) and the overall cooling area being 670 in² (3850 cm²). The spikes were cast-in to a depth of 7 mm into the base plate. Cylinder head weight was 9 lb (4·3 kg) and temperature was reduced by 10 °C in comparison with the preceding type.

Further experiments were conducted with 40 cast-in fins of 0·66 mm gauge copper sheet, spaced 2·5 mm apart. The copper cooling area was then 546 in² (3180 cm²) and the overall cooling area 1005 in² (6150 cm²). The fins were cast-in to a depth of 6 mm, the base-plate thickness being increased by 3 mm. The weight of this cylinder head was 10·8 lb (4.94 kg) and in comparison with the split production head, temperature was reduced by 14 °C.

Because of sparking plug trouble, further modifications became necessary to reduce the temperature of the middle section into which the plugs were screwed. A screwed copper sleeve of 75 mm diameter was used and screwed into the cylinder head as far as the combustion chamber. Copper spikes were then brazed to its outer surface. The rapid corrosion of the copper sleeve in direct contact with burning gases and corrosion of the plug thread made further modifications urgent. The surface of the copper sleeve inside the combustion chamber was nickel plated and the plug holes bushed with Monel metal. Sheet metal air ducts were silver-brazed to the tips of the copper spikes.

The screwed-in sleeves did not prove suitable owing to the impossibility of placing the plug in the most convenient position. Pressed-in sleeves were, therefore, used and this modification reduced the temperature of the middle portion of the head by 15 °C. Meanwhile sparking plug development caught up with the demand, making it unnecessary to introduce the modification to mass production. On the grounds of the good results obtained, however, experiments were continued with a copper plate screwed into the flanged steel barrel. 47 mm high fins were machined on the upper part of the copper (Hidural 6) base plate, which was 12·5 mm thick. In order to prevent the corrosion of the copper by tetra ethyl lead contained in the fuel, the combustion chamber walls were again nickel plated and monel-metal quills were used for the plug boss.

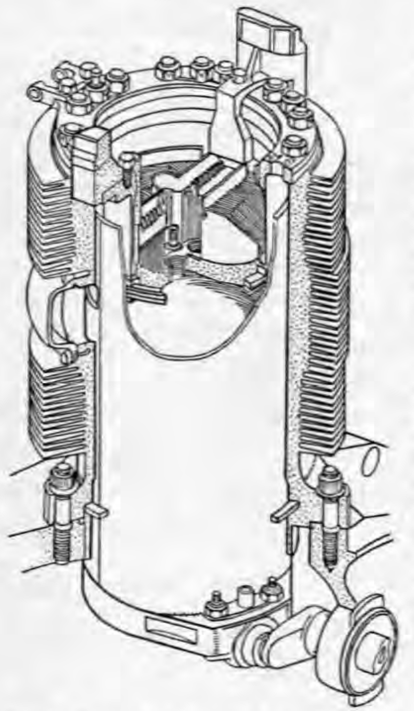


FIG. 296. Final design of the cylinder of the Bristol sleeve valve engine.

In order to reduce the overall weight of the cylinder head, a base plate of only 8·8 mm thickness was used with 40 brazed-on fins of 0·9 mm gauge copper sheet and spaced 2·5 mm apart. With a cylinder head weight of 13·7 lb (6·25 kg) the cooling area amounted to 744 in^2 (4800 cm^2). The copper base plate was protected from corrosive effects by a 0·5 mm layer of nickel and the compression rings were carried by a cast iron (DTD 485) pressed-in ring carrier. The threading of the sleeve did not prove to provide a gas-proof joint and therefore recourse was made to brazing. It was not found possible to combine in actual production both brazing operations, i.e. sleeve to head and fins to sleeve.

The development of the cylinder head reached the stage shown in Fig. 295 before it went into series production. In this the head is built-up of the steel barrel with flange A to which the copper-chrome alloy Y (0·45 to 0·8% Cr) is brazed. The cast iron ring carrier is pressed into the base plate and the sparking plug holes are bushed with monel-metal quills D. The upper base plate surface carries 36 machined fins 1·06 mm thick and 3·5 mm apart. Protection of the thin fins against fracture is provided by a brazed-on steel strip with slots E and a sheet metal channel F, forming an air duct. The bottom wall is protected from corrosion by a layer of 0.25 mm thickness of nickel. Cooling air is directed by ducts and after passing between the fins, it is led away from the head on the rear side. The design of the cylinder head of this sleeve valve engine is shown in Fig. 296.

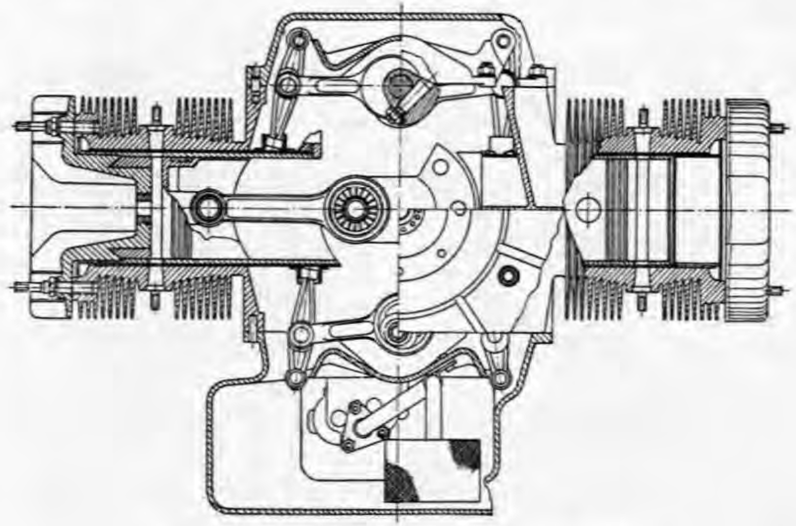


FIG. 297. The Jack and Heinz sleeve valve engine.

An interesting sleeve valve design is produced by the Jack and Heinz concern. Two semi-circular valve elements are used here, performing only axial motion. Between the half-sleeves and the piston, a liner attached firmly to the crankcase, is interposed. This liner reaches above the upper compression ring of the piston. Above the liner, there is a sealing ring with apertures for inlet and exhaust ports. The sealing ring expands during the power stroke and forms a reliable gas seal by its pressure against the inlet and exhaust sliders. A seal is formed above the upper face of this ring by resilient rings under compression from Beleville springs via the stationary liner.

The sliders are driven by timing shafts (Fig. 297) rotating at quarter crankshaft speed, the upper shaft actuating the exhaust sliders and the lower the inlet. For every revolution of the timing shaft, the slider uncovers its port in the cylinder wall twice. An automobile version of this engine with an ingenious cooling control system is shown in Fig. 115. The aluminium cylinders of this engine and the crankcase, split vertically in the crankshaft axis, form one integral unit.

Rotary valves perform only uniform motion. Peak engine speed is therefore governed only by limitations of the crank mechanism. No accelerating forces act on the valve except during speed variations.

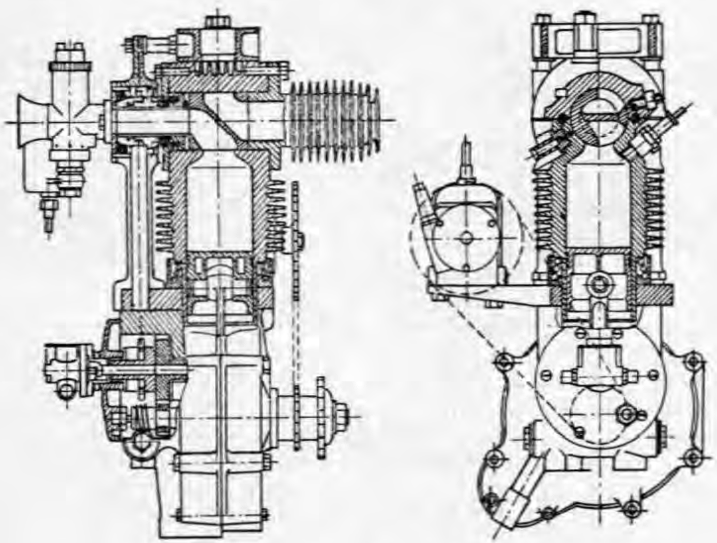


FIG. 298. The Cross sleeve valve engine.

The Cross rotary valve gear is of some interest as the cylinder is not rigidly connected with the crankcase, but sprung away from it and bears on a rotary valve made of nitrided cast iron and located in the cylinder head. Mixture and exhaust gases flow in an axial direction. The cylinder head bears the whole force of the explosion and it is connected with the crankcase by two long bolts and a special bridge-piece, to which it is connected by a pivot free to rotate and unsymmetrically located. This ensures a more uniform distribution of pressure on the upper half of the rotary valve (Fig. 298).

The specific output of such an air-cooled single cylinder four-stroke engine of 248 cm^3 displacement is 71·2 b.h.p./per 1000 c.c. at 6000 rev/min with a compression ratio of 11 : 1, fuel consumption being as low as 0·85 lb/b.h.p.h (189 g/b.h.p. h). The largest flow area of the inlet port is 1·2 in^2 (7.95 cm^2) i.e. 25.3% of piston crown area.

The Aspin rotary valve gear employs a conical rotary valve with an apex

angle of 60°, housed in the cylinder head (Fig. 299), the axis of valve rotation being identical with the cylinder axis. Since the thrust of the explosion would press the valve into its seat with such a force that a great effort would be required to drive it, the load is taken by two bearings with tapered rollers which maintain a minute clearance between the head and rotary valve, taken up by the lubricating and sealing film of oil. This clearance is adjustable

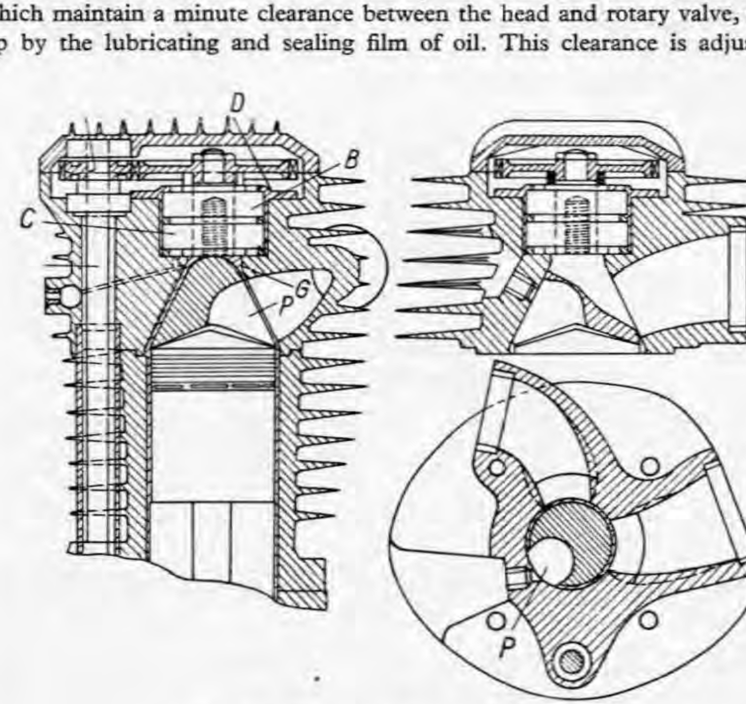


FIG. 299. The Aspin sleeve valve engine.

through the bearings *B* and *C* affixed to the head by a small cover *D*. Proof of the small amount of friction of the rotary valve is said to have been given when the valve driving shaft fractured owing to valve inertia when the engine was suddenly throttled down.

The valve consists of a Cr-Ni steel shell *G* filled with light alloy. A suitable combustion space *P* is formed in the side of the valve and the shell is also cut out for this purpose. The surface area of this channel equals 23·6% of the piston area. At top dead centre the piston reaches as far as the rotary valve and the whole charge is displaced to the space *P*, with accompanying high turbulence. During compression and combustion, the rotary valve is thrust against the inlet and exhaust channels and a good seal results. Due to pressure inside the cylinder, the free edges of the steel shell are pressed against the cylinder wall, thus improving the pressure seal. During the induction stroke the rotary head lifts slightly off its seat, this being caused by the under-pressure in the cylinder and having a beneficial effect on the formation of a sufficient oil film. The sparking plug is connected to the combustion space only for a short period during the last phase of the compression stroke and for most of the duration of the cycle is covered by the valve. This results in a low plug temperature and ordinary touring plugs may be used with exceptionally high compression ratios and high engine speeds.

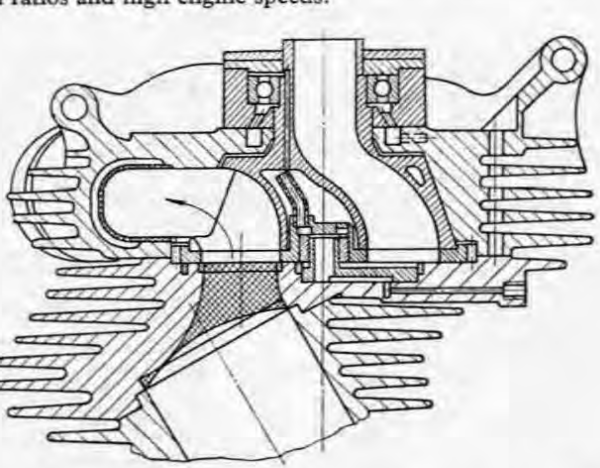


FIG. 300. Section of the cylinder head of the NSU motor-cycle engine with sleeve valve gear.

An experimental single-cylinder engine of 67 mm bore and 70·5 mm stroke with a displacement of 250 cm^3 developed 30 b.h.p. at 10 000 rev/min, i.e. 120 b.h.p./litre. This is a really remarkable feat for a normally aspired engine. The compression ratio used was 14·1 : 1 and the fuel consumption is stated to have been 0·32 lb/b.h.p.h (145 g/b.h.p. h).

An explanation of such an exceptionally low fuel consumption is given by the fact that the combustion space is offset to the axis of valve rotation, thus causing a rich mixture layer on the outer periphery of the valve adjacent to the sparking plug. This difference in mixture distribution is caused by centrifugal force which forces droplets of fuel outwards. The mixture is therefore rich on the outside and leaner nearer the centre. The percentage of fuel in the mixture may be reduced so long as ignition remains reliable, as the mixture surrounding the plug is richer, the overall mixture ratio can be leaner.

Rotary valves are also used experimentally by the NSU company on its motor cycle racing engines. In principle this layout employes the rotary valve in the cylinder head. The compression space port is closed by a disc rotary valve and a seal provided by a sealing ring in the manner of a piston ring. Gas pressure inside the cylinder acts upon the sealing ring from below,

pressing it against the disc. The end gap of this ring is specially shaped for these particular conditions. Gas sealing is effective, but accompanied by excessive wear. In order to improve cooling and to make a more convenient plug location possible, the rotary valve was inclined to the cylinder axis and offset to the side. The inlet port passes through the valve axis and the exhaust port opens directly to the cylinder head. The transfer of heat from exhaust gases to cylinder head is prevented by a sheet metal duct separated from the cylinder head by an air space. The rotary valve is driven through gears, teeth being formed on its circumference. The drilled bearing shaft of the rotary valve is used for the delivery of cooling oil which passes through the disc and at its upper extremity enters the ball bearing through which it passes before returning via a return channel under the bearing. A slight increase in performance was gained over poppet-valve engines but tests are not at an end at the time of writing. A sectioned experimental single cylinder head is shown in Fig. 300. A twin engine with a common valve for both cylinders is also in the experimental stage.

Prospects of sleeve and rotary valve engines

The advantage offered by the Burt sleeve valve layout lies with the possible use of an aluminium cylinder, as the piston does not slide inside the cylinder, but inside the sleeve. The sleeve, however, must be made of a material having a high coefficient of thermal expansion in order to ensure a rate of expansion equal with that of the cylinder. This requires a high grade nickel alloy. Difficulty is encountered in the efficient cooling of the head.

The Cross and Aspin rotary valves are subject to frictional wear by carbon deposit and imperfect gas sealing. This defect will probably be difficult to overcome. The possibility of the development of a rotary valve free of these imperfections is however by no means ruled out.

The size of flow area is also limited by considerations of design in rotary and sleeve valve engines (the dimensions of the valve, timing etc.). In Fig. 301

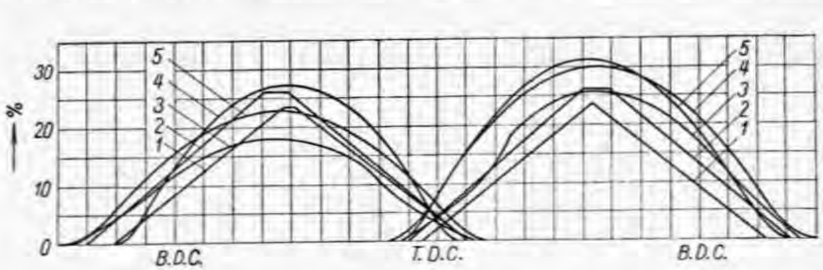


FIG. 301. Comparison of the sleeve valve gear with a poppet valve gear. Rates of port openings are given in percentage of the piston crown area.

1 — Aspin bore/stroke = 67/70 mm, *2* — Cross 62/82 mm, *3* — Napier Sabre 127/120.6 mm, *4* — Mercedes-Benz four-valve 66/70 mm, *5* — two-valve hemispherical cylinder head 55/52.5 mm.

a comparison is made of the rate of exhaust and inlet port opening of such an engine and the orthodox poppet valve type. There is no appreciable difference, the advantage being slightly with the poppet valve. The rate of port opening in rotary valve engines is uniform and therefore lower than in the case of the poppet valve.

The Burt sleeve valve opening rate is superior, but the flow areas involved are not larger than those in the poppet valve engine.

CHAPTER XV

THE COOLING FAN

1. The types of fan employed

Fans can be classified into the two basic types of radial (centrifugal) and axial (propeller) fans.

In radial fans, air flows through the wheel in a plane perpendicular to the fan axis in a radial and centrifugal direction, for which reason these fans are also referred to as centrifugal fans. The blades may be curved forwardly or backwardly. A higher efficiency is attained by blades curved backwardly. With blades curved forwardly the outlet velocity of air is higher and its kinetic energy can only be converted into a static pressure head with low efficiency.

The dimensions of a fan with the blades curved forwardly are smaller than those of a fan with blades curved backwardly for equal speed and pressure head. For this reason automobile cooling fans usually use the former. The outlet velocity pressure can be converted into static pressure with low efficiency, particularly if space limitation does not permit the provision of a suitably shaped spiral casing. The efficiency of radial blowers is therefore generally low. In order to prevent air turbulence the outlet velocity should be specified higher than the inlet velocity which is attained by a gradual reduction of the blade width toward the wheel periphery. Basic types of radial fans are shown in Fig. 302.

The use of a radial fan is indicated by low speed, a small volume of air to be delivered and high pressure head.

One advantage of the radial fan is the possibility of using a direct drive from the crankshaft, the obtained pressure head being sufficient for cooling

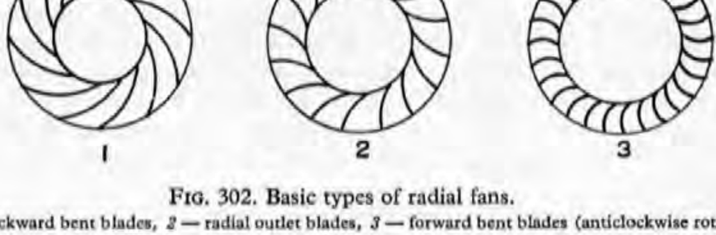


FIG. 302. Basic types of radial fans.
1 — backward bent blades, *2* — radial outlet blades, *3* — forward bent blades (anticlockwise rotation).

purposes. Large bends in the air ducts have to be incorporated immediately behind the fan. With an in-line engine the provision of uniform air supply for all cylinders is difficult. The vanes in the air duct are not equally efficient at varying engine speeds and, therefore, the most suitable shape of air duct must be found experimentally (Fig. 303).

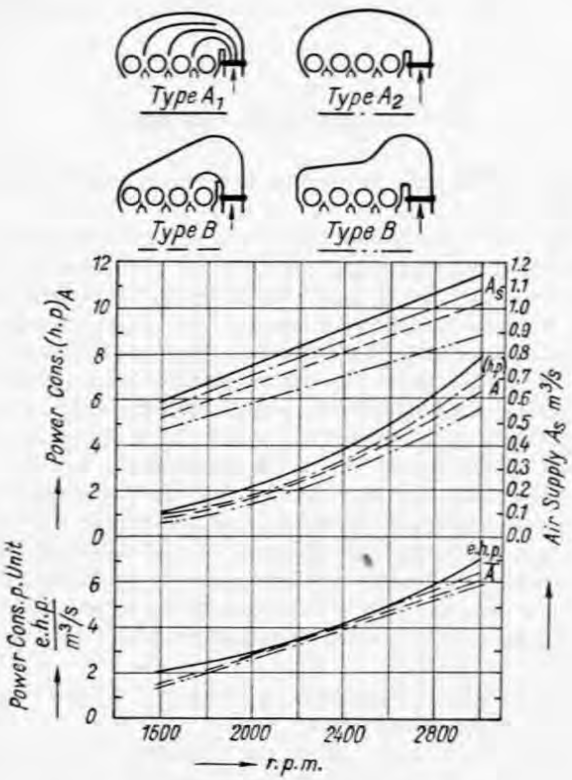


FIG. 303. Suitable shape of air duct.

The pressure of air supplied by the fan is proportionate to engine speed, outside diameter of the wheel, and shape of blades. If a small engine requires a high pressure head (flow area being small and finning dense), the diameter of a fan driven by the crankshaft would be too large. The diameter may partly be reduced by turning the blades forwardly and if this does not suffice, fan speed must be increased. Fig. 304 shows a simple stationary engine with a radial cooling fan.

The flow of air in axial fans is parallel to the fan axis. During its flow through

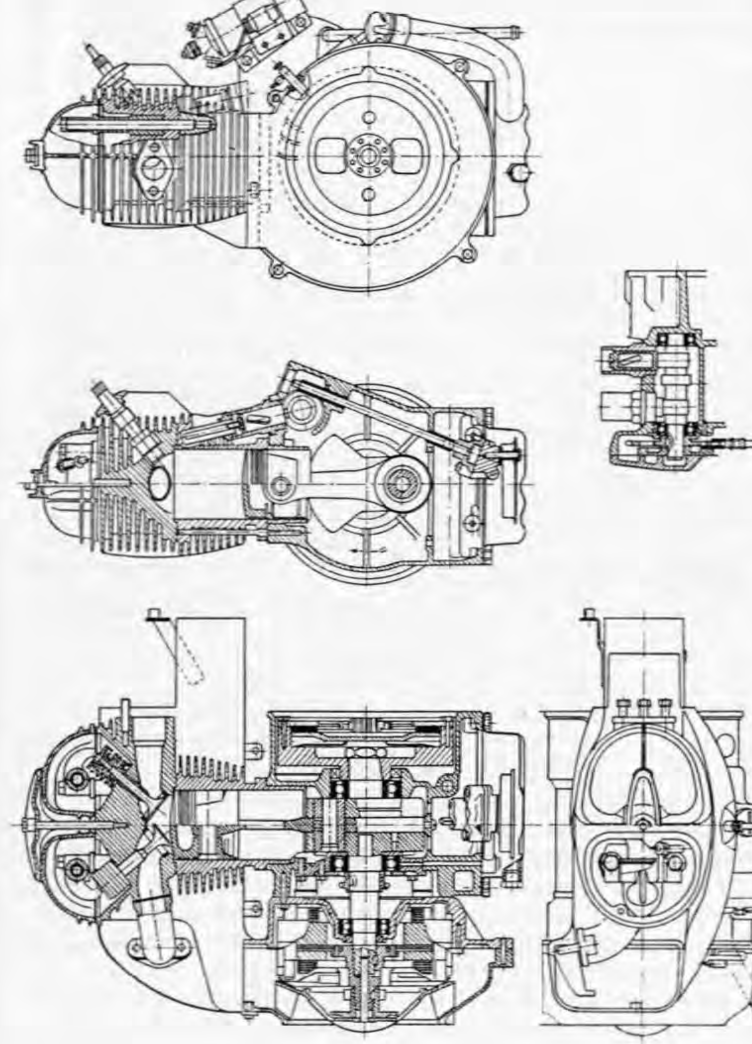


FIG. 304. Sections of an air-cooled BMW engine, bore 72 mm, stroke 73 mm, performance 13 b.h.p. at 5,800 rev/min.

the fan, the air does not change its direction frequently and good efficiency can be attained. With small pressure heads no guide vanes are necessary and the simplest propeller fan is sufficient [29].

For the higher pressures required in multi-cylinder air-cooled engines, a guide wheel must be included and it is placed before the impeller (in the direction of air-flow). The efficiency of the fan is thus considerably increased

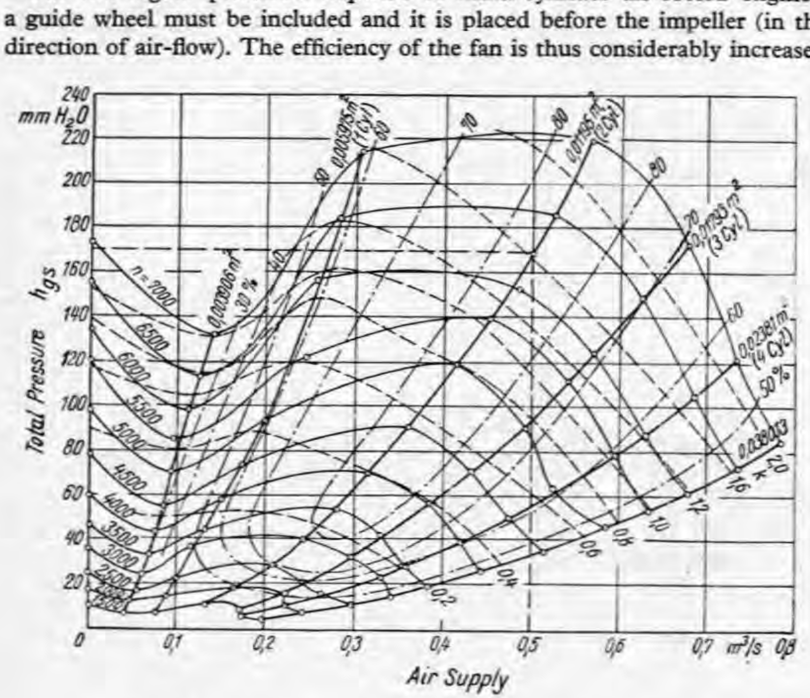


FIG. 305. Example of a universal fan for air-cooled Robur engines.

The characteristic of this axial flow fan includes curves of air supply for the 1, 2, 3 and 4 — cylinder engine. The fan operates most favourably with the two-cylinder engine with a transition area of 0·01195 m² and an efficiency of over 80 %.

and within a certain speed range an axial outlet can be attained. Air pressure is proportionate to circumferential velocity and small fans must therefore be run at high speeds. The relative loss caused by the clearance between the wheel and fan housing is larger with small fans. For the supply of an equal amount of air at an equal pressure, the dimensions of an axial fan are smaller than those of a centrifugal. (Fig. 305.)

In order to attain high efficiency in axial ventilators, the ratio of dynamic pressure to total pressure should not be too high. This ratio is expressed by the dimensionless pressure number [16] which should not exceed 0·6. With higher pressure numbers, e.g. 0·8, the fan diameter is reduced or lower speed may be employed, but the fan is sensitive to resistance variations at the outlet side.

If the outlet velocity of a fan is too high, efficiency is generally reduced, for the dynamic pressure cannot be utilized with any degree of economy. A high dynamic pressure is also usually the cause of uneven air distribution to individual cylinders.

High fan speeds make for a reduction in diameter, but they are often the cause of noise. V belts are usually used for driving axial fans but with a large transmission ratio and a given minimum fan pulley diameter, the crankshaft pulley diameter is inconveniently large. Losses in the belt drive are considerable, especially where high transmission ratios are employed, and have to be taken into account when calculating the necessary power input for cooling.

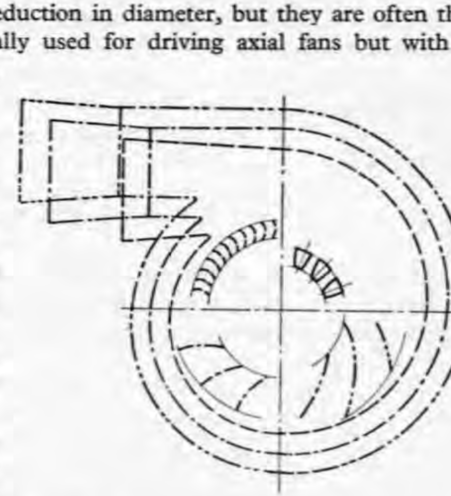


FIG. 306. Comparison of dimensions of different types of fans.

All four types are designed for an equal delivery of 0,7 cu. m/h and delivery head 380 mm w.c. The smallest is an axial flow fan of a diameter of 200 mm operating at 8,900 rev/min. The second larger one is a fan with forward bent blades of a diameter of 275 mm and 3,800 rev/min. The third one is a fan with radial outlet blades 330 mm diameter and 3,800 rev/min. The largest is a fan with backward bent blades, diameter 390 mm and 3,800 rev/min.

If high pressures (above 200 mm of water column) are needed for cooling, the circumferential velocity of the wheel becomes too high and the fan becomes noisy in operation. In such cases, twostage blowers are sometimes used (e.g. in the SLM flat-eight engine).

The advantage of the small overall dimensions of the axial fan are most apparent on V engines with the fan at right angles to the crankshaft axis. Such an example is given in Fig. 393 which shows an air-cooled V 12 Continental engine used as a prime mover in tanks by the United States Army. The fan takes up a minimum of space and the passage of air through the armoured compartment is the shortest possible and without sharp bends. Warm air is drawn from the engine and is forced out of the engine compartment by the shortest route. The two axial fans of this 750 b.h.p. engine are 660 mm in diameter and their axial depth is very small. A pressure head of 150 mm water column is needed to cool this compression-ignition engine, the temperature of the cylinder heads being kept at 175 °C. Considerable resistance to air flow is produced by the entry and exit openings for the cooling air which are protected by louvres, so that total fan input is twice as high as would be normally necessary for cooling the engine. Cooling air passes not only over the cylinders, but also through coolers of engine oil and hydraulic transmission gear oil. Satisfactory cooling is guaranteed up to external temperatures of 52 °C. The power consumed by the fan is 110 b.h.p. i.e. 14·7% of the total b.h.p. performance. For water-cooled engines the fan input under similar conditions would be even greater.

Fig. 306 compares the dimensions of four types of fans of an equal output [18]. The largest is a radial fan with blades curved backwards and the smallest is the axial fan. All three types of radial fan have the same speed of 3800 rev/min whereas the axial fan rotates at 8900 rev/min. Axial fans are generally preferred for automobile air-cooling, radial fans being only used on engines with a small number of cylinders, or when the fan impeller is mounted on the crankshaft. For the calculation of fans reference should be made to specialised literature [16].

2. Fan and engine balance

By expressing the fan output in terms of its speed it will be found that the volume of air supplied is directly proportionate to the fan speed, air pressure increases as the square of fan speed and power consumed as the cube of speed.

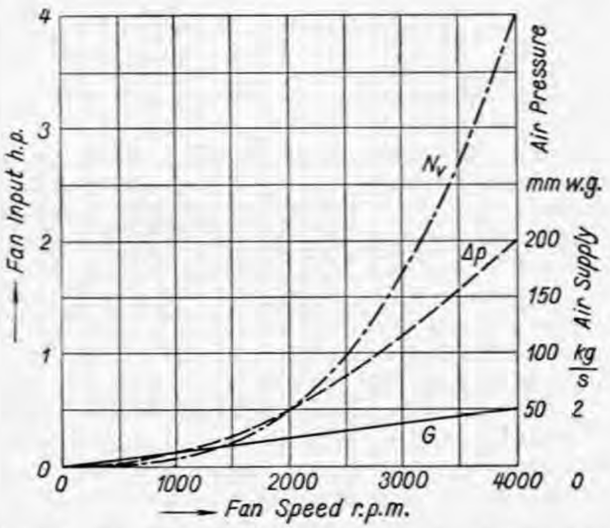


FIG. 307. Air flow, air pressure and fan input plotted against speed.

These correlations are given in Fig. 307 and they apply to an arrangement in which the air is forced through an opening with a constant flow area, this being the case with an air-cooled engine. If twice the amount of air is to be forced through the aperture, air pressure must rise 2^2 times, i.e. four times. These conditions are met by the fan.

The increase in power consumption with the 3rd power of speed is very rapid and must be given due consideration in high speed engines.

The characteristic of the fan which expresses the correlation between the pressure and quantity of air supplied is of utmost importance (Fig. 308). The conditions shown apply to constant fan (engine) speeds. It is apparent from the diagram that if the amount of air supplied is reduced at constant fan speed (e.g. by throttling) air pressure increases according to the curve *a*. If, however, the amount of air supplied is reduced below a certain limit, a sharp drop in pressure ensues as a result of blade overloading. Air velocity through the impeller is then too low and since there is too great a static pressure head in the impeller, air cannot follow the under-pressure side of the blade and "breakaway" occurs, manifesting itself by turbulence and consequent noise.

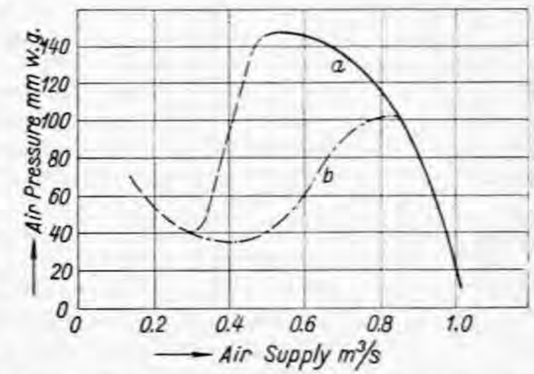


FIG. 308. Characteristic of an axial flow fan and its operational harmony with the engine.

During the design of a fan the conditions under which it will operate in the engine must be considered. If, for example, the engine requires a pressure head of 120 mm water column and a flow of 14·1 ft³/s (0·4 m³/s) at maximum speed and a fan with the characteristic given in Fig. 308 is used, unsatisfactory cooling will result. This fan can supply under a pressure of 120 mm water column a volume flow of 27·5 ft³/s (0·78 m³/s) of air at an efficiency of about 88%. It may therefore be considered to be a very good fan. When used in conjunction with an engine through the cooling ducts of which only 14·1 ft³/s (0·4 m³/s) of air can be forced at 120 mm water column, this fan is completely unsatisfactory. With the amount of air flow reduced below 17,6 ft³/s (0·5 m³/s), breakaway conditions arise and pressure drops to about 40 mm of water column. At this pressure, however, 14·1 ft³/s (0·4 m³/s) will not be forced through, but an amount proportional to pressure, which is 8 ft³/s (0·23 m³/s), which is far below the flow required for satisfactory cooling. The fan under consideration would therefore be satisfactory when used with an engine of double the number of cylinders and hence a double flow area.

This example proves that fan design must be adapted for the requirements of the engine. The fan described is however not even suitable for an engine twice as large with a flow requirement of 28·2 ft³/s (0·8 m³/s). If cooling is regulated by throttling, the rate of flow could under low temperature conditions be below 17·6 ft³/s (0·5 m³/s) with resultant breakaway. This does not seem as dangerous, for with an increase of temperature, the thermostat would open up the regulating louvres and airflow would increase and balance the demand.

But if the amount of air flowing under breakaway conditions is increased, the rise in pressure will not follow the same curve as with throttling, but curve b. In the range between these two curves the fan is unstable, i.e. the same rate of flow can be attained at various pressures. Were the characteristic of the fan as unfavourable as shown in Fig. 308, the fan would not be clear of this unstable range even at full duct opening and breakaway conditions would persist. Only by increasing the flow rate above 29·2 ft³/s (0·83 m³/s) and a reduction in pressure can the fan be made to operate clear of this range of instability. This can be effected in practice by prising up some of the air ducting shrouds thus increasing the flow area.

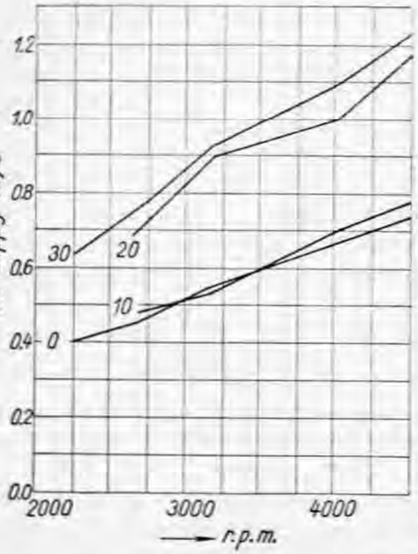


FIG. 309. Influence of an obstacle behind the fan upon the output of the fan. Figures at the curves indicate distances in mm of the obstacle from the fan wheel.

Requirements on a well designed fan are: the absence of breakaway and the narrowest possible range of instability. The fan must be effective down to very small flow rates without breakaway occurring and the two curves should be very close together in the region of instability both during throttling and acceleration.

The breakaway of air from the fan blades manifests itself by increased noise and by pulsating induction. The increase in noise is sometimes difficult to note with the engine running, but if a hand or a strip of paper is placed near the guide wheel the reverberations of inducted air can be clearly felt. This condition may be caused by a reduction of the flow area such as for example occurs when the fins are cast with smaller spacings than required or when some of these spaces are blocked; an immediate temporary cure can be effected by prising off parts of the air ducting to the engine. Some of the air intended for engine cooling does escape unused, but owing to the rise in volume supplied a pressure rise ensues and a greater flow past the engine results.

Shaped (cast) blades can operate under higher load and are not so sensitive to breakaway as sheet metal blades, which also have a reputation of being noisier in operation. On welded sheet metal blades, the welds must be neat. Rough and projecting welds enhance breakaway conditions and the same may

be said of projecting seams or ripples on cast blades or of sharp edges produced by machining, etc. Conversely, a smooth blade and housing surface reduces resistance and promotes laminar air flow and an increased stability of blades. Any turbulence in the inducted air (caused, for example by the interference of an object obstructing the inlet) reduces efficiency and stability. Objects hindering the free outlet of air also reduce fan performance. Fig. 309 shows the reduction in fan output through the approach of an obstacle to free air outlet.

In order to reduce the overall width of in-line engines, fins are sometimes left to protrude into the outlet duct of the fan. This does not reduce fan output if the cylinder is not too near to the air outlet and if the fins are without baffle.

The dynamic air pressure is generally utilized for cooling aircraft engines. On some modern aircraft engines with very dense finning a high pressure is necessary. During take-off, when engine output is high and the speed of the aircraft low, such engines tend to overheat.

If cooling arrangements are governed by take-off requirements then during level flight at high speeds the air resistance to the aircraft is unduly increased by the large air intakes; in such cases cooling fans are used for supplying cooling air. Most unfavourable conditions are found on helicopters, the air speed of which is relatively low. Cooling is sometimes assisted by the use of exhaust gas ejectors, the exit of hot air with exhaust gases adding to the propulsion of the aircraft.

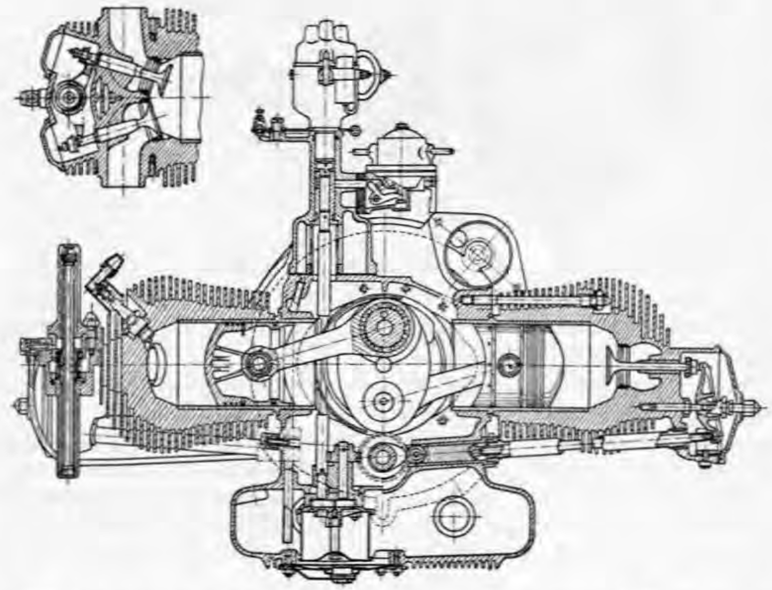


FIG. 310. Sectional view of a two-cylinder Panhard-Dyna car engine. Note aluminium cylinder with liner.

3. Design details of fans

For small pressure heads axial, propeller type fans, without a guide wheel and casing, are employed. These are in general use in water-cooled engines and are rare in air-cooled engines. Fig. 310 shows a propeller fan. In order to reduce noise, the blades are sometimes distributed at irregular intervals around the circumference, care being taken to balance the fan as a whole.

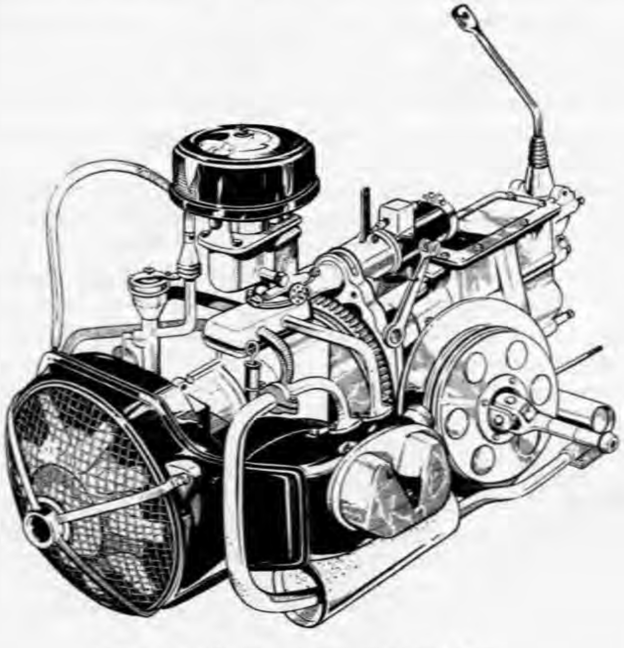


FIG. 311. Citroen engine with axial fan.

For larger pressure heads axial fans, mounted in casings but without guide wheels are used. (Fig. 311.)

For large pressure heads above 80 mm water column, axial ventilators are fitted with guide wheels. The guide wheel vanes are usually of sheet metal and their number and that of the fan blades should be chosen so as not to have a common divisor in the interest of noise reduction. A guide wheel and impeller of an axial fan made of sheet metal pressings are shown in Fig. 312 (Tatra 111). The sheet metal fan is simple to manufacture and light in weight. The welds at the outer ends of the guide vanes must be neat in order not to hinder air flow.

If the fan is fixed by means of a steel band, it must be gripped about the circumference of the guide wheel. Were the unit attached by the circumference

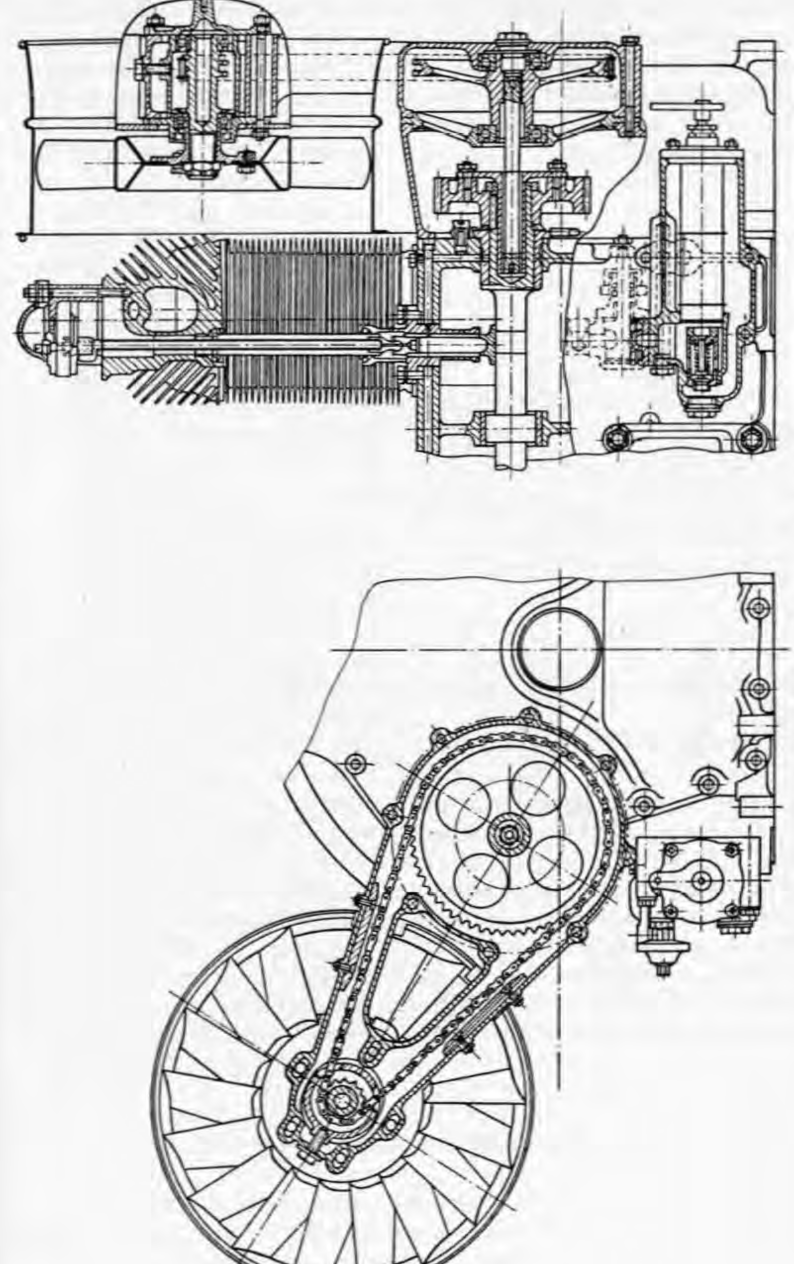


FIG. 312. Propeller fan of the Tatra 111 engine.

of the impeller casing the casing would be subjected to compression and the blades could come into contact with it. The casing must be sufficiently rigid in order to resist deformation and in order to maintain a small and uniform clearance between the casing and blades; the clearance should be as small as possible in order to minimize its considerable and unfavourable effect on efficiency.

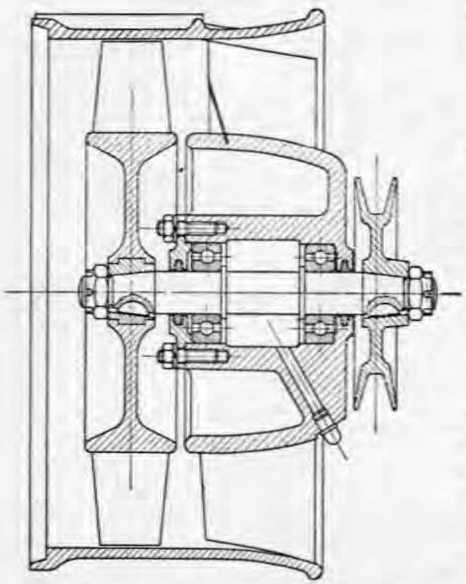


FIG. 313. Fan with cast iron casing.

A fan, produced by casting is shown in Fig. 313. The casing is cast aluminium with cast-in steel sheet blades. A dynamo can be mounted inside the cast hub of the guide wheel. The impeller is cast of aluminium together with the shaped blades. It can also be produced of plastics. The fan is fixed by a band or lug and a belt tightening screw is provided.

Fans are usually driven by V belts which are flexible and permit belt slip during sudden changes of speed. They are silent in operation, but efficiency is generally low, up to a third of the fan input often being absorbed by the belts, which results in a short service life.

A roller chain drive gives longer life and its efficiency is higher. It is, however, less silent in operation and requires a suitable chain cover and efficient lubrication. As the chain is not elastic, a resilient member must be included in the drive, in the form of a torsion bar, (Fig. 312) or a slip clutch.

Gear and shaft drive is an advantage for its very low rate of wear, servicing and efficieney. An illustration of such a drive on the Tatra 924 engine is shown in Fig. 117. where an hydraulic coupling is incorporated at the fan end protecting the gears from overload during changes in speed. An example of a bevel gear driven fan is shown in Fig. 386, such an arrangement resulting in a convenient flow of air around the engine without sudden changes of direction.

4. Air ducting and pressure losses

The cooling air stream must be directed from the fan by suitable air ducts or cowls to the engine cylinders and from the cylinders to atmosphere. The fan must not only overcome the pressure head required for the air flow around

the cylinders, but also the total resistance of the air ducts. The distribution of air pressure about the outer surface of the vehicle should also be investigated as the pressure head thus created can assist or impair the flow of air past the cylinders. The cooling air outlet should be placed in an underpressure region of the body and the inlet duct in the region of the highest dynamic pressure. The dynamic pressure of air in conjunction with a suitably

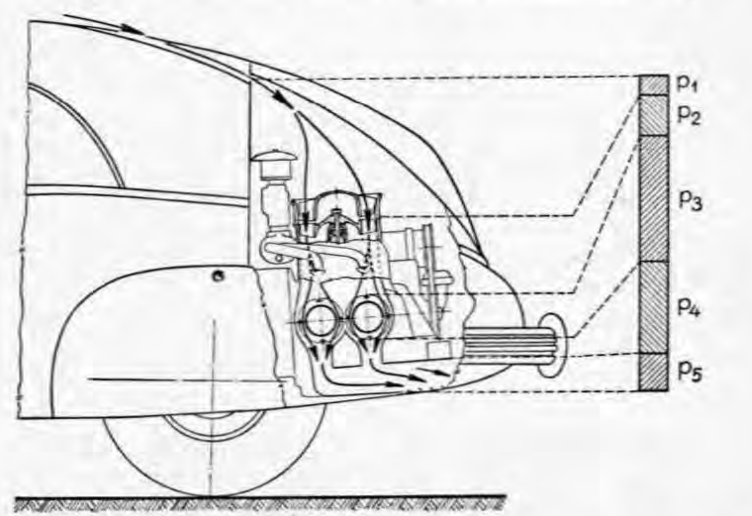


FIG. 314. Pressures developed during the passage of cooling air through the vehicle

chosen air outlet location can often suffice for the good cooling of racing engines without the use of a fan. On aircraft engines dynamic pressure is generally sufficient.

Dynamic pressure increases as the square of speed. Air-cooled engines usually reguire a pressure of 100 mm of water column for cooling purposes and this corresponds to a speed of 92 m.p.h. (145 km/h). For this reason dynamic pressure alone cannot be relied upon for engine cooling, as for instance at a speed of 18·6 m.p.h. (30 km/h) when driving up-hill it amounts only to 4·34 mm. The dependence of dynamic pressure on speed (km/h – m/s) is given in Table 41. Dynamic pressure should always be fully utilized.

The distribution of pressure loss in the air flow during its passage through the vehicle is shown in Fig. 314. The first loss p_1 occurs on entry into the inlet duct, this opening sometimes being too confined. It is either obstructed by a decorative grill or by control shutters, etc. This resistance to air flow should be kept as low as possible under all circumstances. In special purpose vehicles, resistance in the inlet ducts is considerable its values for various duct shapes being given in Fig. 315.

A further pressure loss occurs in the air duct between the fan and cylinder finning, being caused by the conversion of kinetic energy of the air into static pressure and by the changes in flow velocity and direction. These losses are marked p_2 in the diagram.

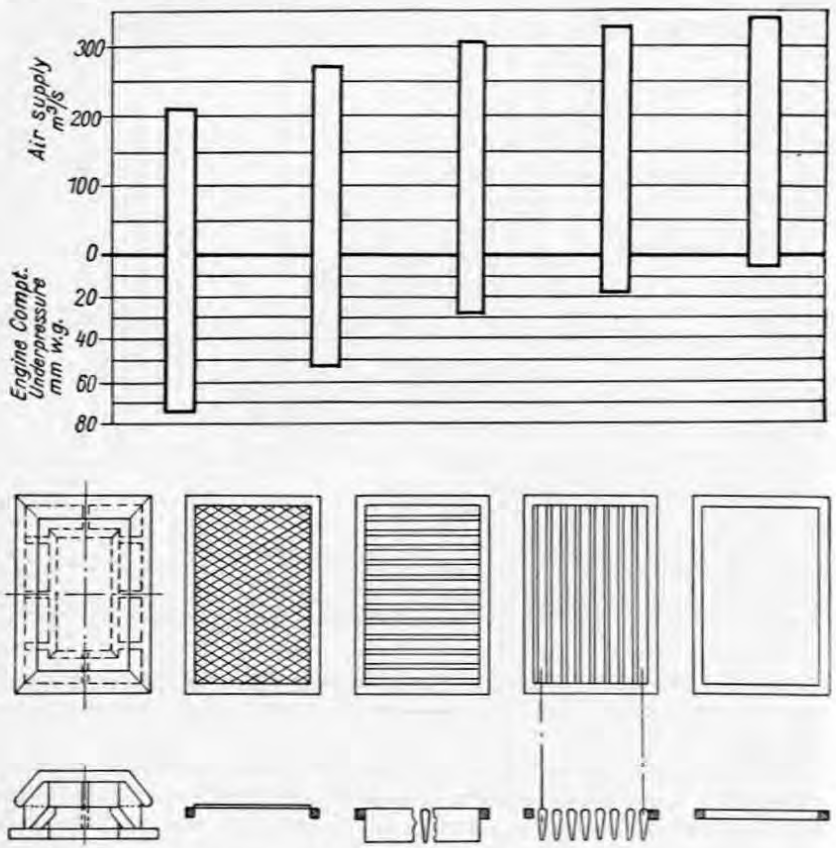


FIG. 315. Air resistance of protective gratings in armoured vehicles.
The streamlined design of grating reduced underpressure to one third with a simultaneous increase of air flow by 50 %.

The greatest pressure loss p_3 occurs in the cylinder finning interspaces. It includes losses through friction, variation in velocity, variations in flow direction, losses through turbulence and at the confluence of two air streams behind the cylinder.

A large pressure loss p_4 occurs at the outlet of air from the engine ducting; this can be reduced only to a small extent by the use of a suitable diffuser.

Table 41

Dynamic Head of Air

Velocity		Dynamic head
km/h	m/s	mm w.g.
10	2.78	0.48
20	5.56	1.93
30	8.33	4.34
40	11.11	7.71
50	13.89	12.05
60	16.67	17.36
70	19.44	23.63
80	22.22	30.86
90	25.00	39.09
100	27.78	48.22
110	30.56	58.34
120	33.33	69.43
130	36.11	81.48
140	38.89	94.50
150	41.67	108.5
160	44.44	123.4
170	47.22	139.3
180	50.00	156.2
190	52.78	174.1
200	55.56	192.9
210	58.33	212.6
220	61.11	233.4
230	63.89	255.1
240	66.67	277.7
250	69.44	301.3
260	72.22	325.9
270	75.00	351.5
280	77.78	378.0
290	80.56	405.5

The last pressure loss p_5 occurs in the remaining duct from the engine to the exterior of the vehicle.

In automobile engines the pressure loss occuring in the flow of air around the cylinders is further increased by

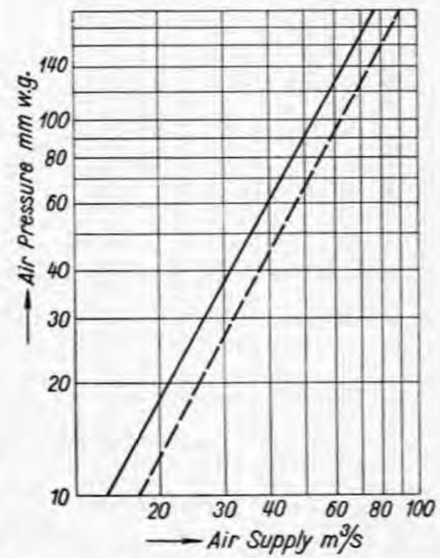


FIG. 316. Effect of cylinder bolts upon the pressure head across the cylinder.
Full line — cylinder with bolts, dash line — cylinder without bolts.

the bolts which obstruct the flow between the fins. This must be taken into account when determining the flow resistance of cylinders. The difference of resistance between cylinders with and without bolts is shown in Fig. 316.

All this provides sufficient proof of the fact that the pressure loss at the cylinder is only part of the total pressure loss which must be taken into account when designing the fan. Furthermore, losses through leakage, which amount to about 20 to 40 % of the quantity of air needed for cooling must be allowed for. If high pressure heads are used for cooling, loss through leakage result in a large increase in fan input. All shrouds therefore should be a leakproof fit to reduce the pressure loss to a minimum.

Air under pressure can also be used for cylinder charging, but as the pressure head is negligible from the point of view of cylinder charging, its in-

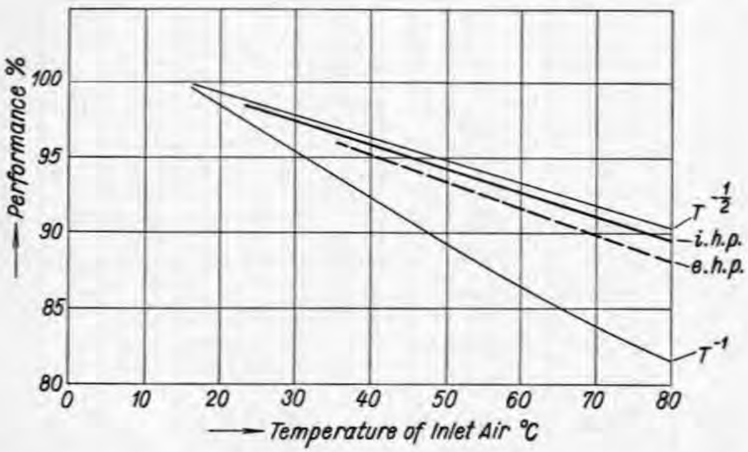


FIG. 317. Influence of temperature of air inducted into the engine upon engine output.

fluence on engine performance is practically nil. There is also the danger of the air in front of the cylinders being warmed by heat radiation and the advantage of increased charging pressure is thus lost. The air entering the cylinders must pass through an air cleaner and the whole layout is more difficult to install. With petrol engines the float chamber must under such conditions be pressure sealed and its breather hole connected to the induction manifold.

The temperature of the inducted air exerts a considerable influence on engine output. The specific gravity of warm air being lower, volumetric efficiency is reduced with the entry of warm air and with it engine performance. The influence of temperature of inducted air on engine performance is shown in Fig. 317. This influence is not equal for petrol and compression-ignition engines. In petrol engines it approaches b.h.p. $\propto T^{-1}$ in oil engines it is closer to b.h.p. $\propto T^{-\frac{1}{2}}$. The diagram shows the results of measurements made on a petrol engine. It follows that it is beneficial to induct air of the lowest possible temperature which does not, in the case of petrol engines, impair the formation of a readily inflammable mixture.

The air ducting must be arranged in such a way as to divert an equal flow

to all cylinders and to ensure uniform cooling over the whole circumference of each cylinder.

Difficulties in uniform air distribution to each cylinder increase with the number of cylinders in one bank but good results can be achieved by the use of suitable baffles. "Suction" cooling is very convenient from the point of view of a regular air flow distribution among cylinders.

A regular distribution of temperature can be attained by the use of deflector baffles on the exit side of the cylinder (see Fig. 98). Difficult conditions prevail with multi-cylinder aircraft engines with a very large number of cylinders.

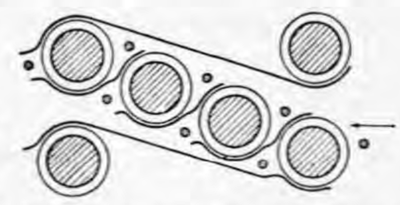


FIG. 318. Cowling of multi-line radial engines. Arrow indicates direction of air flow.

A diagrammatic drawing of the air ducting of a multi-row radial aircraft engine is shown in Fig. 318. The cylinders are arranged on the crankcase in spiral form and every row has its own air intake and exit.

CHAPTER XVI

THE CRANK MECHANISM AND LUBRICATION

1. The crank mechanism

The requirements on the crank mechanism of an air-cooled engine are identical with those in a water-cooled engine. A rigid crankshaft, free from torsional vibration, is required together with high mechanical efficiency and low weight of the crank-mechanism.

The crankshaft must be checked for torsional vibration by calculation; this is outside the scope of this book and reference may be made to abundant literature on this subject. In engines with a short crankshaft for four-cylinder in-line engines or V eight cylinder engines, a crankshaft mounted in properly dimensioned bearings is sufficiently strong and rigid.

Table 42

Average Dimensions of Crankshafts

	Main Shaft Bearing		Crank Pin Bearing		Web		Cylinder spacing
	diameter	length	diameter	length	width	thickness	
In-line petrol	0·7	0·6	0·6	0·6	0·9	0·5	1·12
engines	0·8	0·4	0·7	0·45	1·2	0·2	1·35
Petrol engines V	0·75	0·45	0·6	0·55	0·9	0·45	1·2
	0·85	0·60	0·7	0·4	1·2	0·2	1·6
In-line oil engines	0·7	0·65	0·65	0·6	1·0	0·4	1·25
	0·8	0·55	0·7	0·45	1·3	0·25	1·4
Oil engines V	0·7	0·5	0·7	0·6	1·0	0·35	1·45
	0·8	0·6	0·75	0·4	1·35	0·2	1·65

All dimensions are given as multiples of bore D

Table 42 shows the distances between cylinder centres, bearings dimensions and crankshaft web dimensions of petrol and compression-ignition engines. The diameters of main bearing journals and crank pins are restricted by

circumferential velocity and considerations of design, a current requirement affecting crank pins (to quote an example) being the possibility of removing the connecting rod through the cylinder bore. A certain increase in crank pin diameter may be attained through splitting the big-end of the connecting rod obliquely.

An advantage inherent to air-cooled engines is the fact that after detaching the cylinder, the crankcase mouth opening is increased by double the spigot thickness, this leaving a larger diameter for dismantling the connecting rod. For this reason a horizontally split big-end can be used in air-cooled engines with crank pins of large diameter. A horizontally split big end is simpler to manufacture and stiffer for the same weight than one split obliquely.

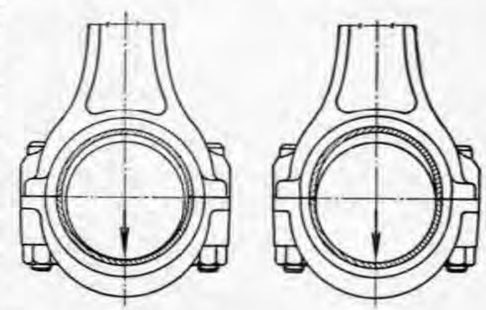


FIG. 319. Distortion of the connecting rod due to the accelerating forces of the piston. The left hand picture illustrates the shape of the connecting rod at low speed, the right hand one shows the deformation of the big end bearing due to large accelerating forces at high speeds.

The dimensions of compression-ignition engine crank pins do not vary much from those used in petrol engines. The increased bearing load is dealt with by the use of bearing materials which stand up to higher pressures, such as lead-bronze. The data in the table refers to plain bearings. With roller bearings certain improvements may be effected.

In the construction of air-cooled engines the V and the flat layout have gained favour as the crankshaft of such types is comparatively rigid and the counterbalance weight relieves the bearing (especially the centre bearing) although it also increases crankshaft weight while reducing the natural frequency of torsional vibration. The balancing of revolving masses need not be perfect in such a case and it depends on the dimensions and properties of the bearing carrying the greatest load.

The radius at the junction of the webs and pins should be large in order to prevent local concentration of stress, minimum radii of 0·025 to 0·04 D being advisable.

In order to secure the best utilization of the crankshaft constructional material highly stressed sections should be avoided. By the rolling in of material at the transition between pin and web the fatigue resistance of the crankshaft can be increased by up to 60%. The concentration of material around the mouths of oil-ways in crankshaft journals also increases fatigue resistance considerably.

Bronze bearings require a hard surface for the main journals and the journals of compression-ignition engine crankshafts are today hardened mainly by the application of direct induction heating. Up to 500 to 550 Brinell hardness can be obtained.

The connecting rod is regarded as a partly reciprocating and partly revolving mass. Its weight therefore must be specified with utmost care. The big end must be sufficiently stiff to prevent distortion through accelerating forces. In highly stressed engines a certain amount of big end distortion is always

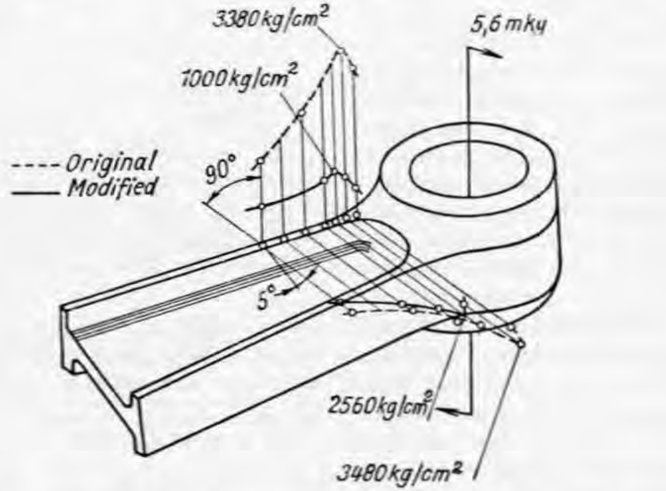


FIG. 320. Stresses in the connecting rod of an aircraft engine.

reckoned with and a slight bearing clearance at right angles to the connecting rod axis is provided in order to prevent the bearing "gripping" the pin or seizure occuring near the split of the bush. The deformation of a bearing by accelerating forces is shown in Fig. 319. Big end bearings are generally bushed with thin walled sleeves. The same observations made about cylinder bolts apply to connecting rod bolts: their axes should be located at the geometric centre of the seating surface of the big end and cap. If this is not the case, the bolts are exposed to bending stress when tightened and the connecting rod is distorted. This is particularly important with light forked connecting rods.

Side thrust at the joint can be taken up by a step in the joint face. In such a case the joint faces are formed with mating straight or curved-wall splines which transmit lateral forces, and make the use of inclined split big ends possible even with large diameter crank pins.

The small end eye mass is reciprocating and must therefore be as light as possible. The correct shaping of the joint between the small end eye and connecting rod shank is important. The stress at this point is greatest and fractures are known to originate from it. Fig. 320 shows the stress at this point of the

connecting rod as determined by actual measurments. The reduction in stress attained by the modification shown in Fig. 321, (right) is marked by full line. The use of a single radius between the connecting rod shank and small end eye (left) which is often used by designers for its greater convenience, is poor design.

The pistons of an air-cooled engine have considerable influence on engine noise. It is important to keep the cold piston clearance at a minimum

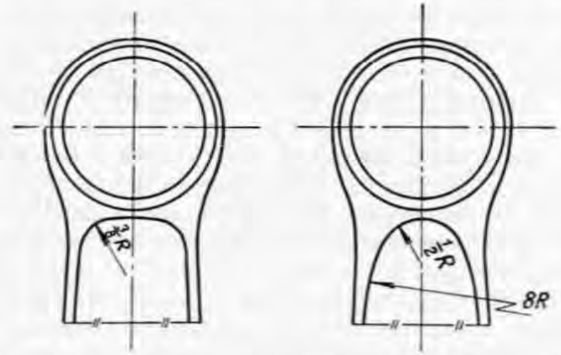


FIG. 321. Modification of the connecting rod providing an increase of strength.

and to reduce its variation as the engine warms up. This may be done by:

1. compensating for piston expansion;
2. flexible pistons with split skirts;
3. use of aluminium cylinders.

Expansion of the piston is usually compensated for by cast-in struts of a material with a small coefficient of expansion; as an example of this type, the Bohnalite-Nelson piston with invar struts may be quoted. Today cast-in rings in the region of the scraper ring are often used and these do not permit the expansion of this section of the piston.

Split-skirt pistons may be fitted into cylinders with a smaller cold clearance. With the expansion of the piston, the split skirt is relatively compressed and seizure is thus prevented. An example of a piston of this type is given in Fig. 322.

FIG. 322. A modern piston o a high-speed engine.

The piston clearance must be specified in the case of the air-cooled engine with special care in order to prevent piston slap and seizure. The use of large piston ovality (up to 0·2 mm) is advisable as this provides clearance for the piston during expansion, especially if steel struts are cast into the piston, acting as bi-metallic elements by changing its elliptic section into a circular one during expansion.

Care should also be taken to provide the correct tolerance between the gudgeon pin and its bosses. A heavy interference fit restricts the expansion of the piston in the direction of the gudgeon pin. Excessive allowance on the other hand, is a cause of knock.

The surface treatment of the piston (graphiting, tinning, anodic oxidation, etc.) is beneficial in the interests of satisfactory running-in with small piston clearances. Partial seizure of the piston is apparent by the knocking noise emitted. If seizure is only slight, knocking disappears after a period.

By slotting the piston below the scraper ring, the transfer of heat from the piston crown to skirt is prevented and the influence of piston crown temperature on skirt distortion is thereby limited, the return flow of oil from the scraper ring also being improved. If the gudgeon pin boss projects directly from the piston skirt without the provision of a recess between them, a thin gudgeon pin can cause a distortion of the piston skirt by its own bending thereby causing local seizure.

A chromium plated aluminium cylinder expands at the same rate as an aluminium piston, which may therefore be fitted with a minimum allowance.

A gudgeon pin offset by about 1·5 mm against the direction of rotation increases the period during which the load on the piston is transferred from one side to the other, thus reducing piston slap at top dead centre.

Piston rings are very important to piston cooling, as heat is transferred from the piston to the cylinder wall mainly through the piston rings. The numerical values of thermal load on the piston rings are given in Table 43. Piston rings are available with normal or extra tangential pre-stress. Rings with high tangential stress begin to vibrate at higher engine speeds. When the ring begins to vibrate owing to the periodical pressure of combustion, the ring ceases to provide a gas seal and fatigue fracture may even occur. Ring vibration can be detected by the increase in blow-by of gases to the crankcase at a certain critical speed. This blow-by should not exceed the value of 0·2 to 1% of the induced volume.

Table 43

Thermal Stresses in Piston Rings and Pistons

Heat transfer between		kcal/m² h °C
piston and side wall of piston	upper side	600
ring	lower side	14,000
piston ring and cylinder		30,000
piston skirt and cylinder		300

Oil control rings prevent piston over-lubrication and remove surplus oil from the cylinder wall. The specific pressure between the piston ring and cylinder, depending on ring stress and the area of the contact surface, is important. The common type of scraper rings has a sealing face 0·75 mm wide and a specific pressure of 49·7 lb/in² (3·5 kg/cm²). Scraper rings produced today have widths of down to 0·1 mm and specific pressures as high as 284 lb/in² (20 kg/cm²), such rings, if provided with sufficient lubrication, give long service life even under arduous conditions. Plentiful lubrication of the cylinder with efficient oil control rings is preferred today.

The oil retained by the ring must be provided with a free passage of flow by arranging a number of holes of sufficient cross section area from the piston ring groove and the leading edge of the ring and also by sufficient notches in U shaped rings.

If the first groove is fitted with a chromium plated ring, wear on the ring itself and cylinder wear is considerably reduced, this being the reason for the recent widespread use of chromium plated rings. Sealing rings should be narrow in order to reduce the amount of running-in required and in order to reduce groove wear. In petrol engines of up to 90 mm bore, a ring width of 2 mm is sufficient.

2. Bearings

Plain bearings are usually employed for the crankshaft, as they are simple, small and silent in operation. Petrol engine bearings are most often bushed with white metal (composition, a Babitt) shells. Babbitt alloys are at their best with a content of 86 to 89% of tin. A valuable property of such alloys is the embedding of foreign hard matter brought by oil with resulting protection of the journal surface. Should oil supply to the bearing fail, it "runs", i.e. melts and the shaft is not damaged.

White metal is cast into stiff thin-walled shells. The overall thickness of such a thin wall shell is about 1·5 to 2 mm, the thickness of the bearing metal layer being about 0·5 mm. The bearing metal is cast onto steel strip and scraped down to the prescribed thickness and only then is the strip cut up into sections of the required length, which are then bent into required shape. As a precaution against turning, the shells are fitted with anchor strips bent outward, as shown in Fig. 323. The bearing housing in the crankshaft or connecting rod must be circular and accurately machined. The diameter of such a flexible thin wall bearing cannot be checked, the only possible measurement being that of the shell thickness and length with the half-shell pressed into

FIG. 323. Shell of a thin-walled bearing.

the semi-circular groove. The rate of wear of such bearings is rapidly reduced by the use of thin Babbitt lining (Fig. 324). Shells with a microscopic layer of 0·05 to 0·12 mm of Babbitt are now being produced and have a reputation of greatly increased life. Three-layer bearings with an intermediate layer of lead-bronze to prevent shaft damage with a run bearing are also produced.

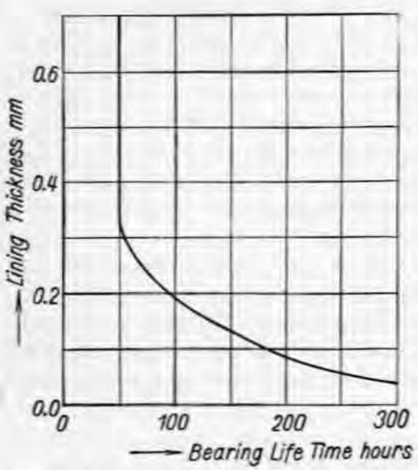


FIG. 324. Service life of a bearing in relation to the thickness of bearing metal.

The normal type of Babbitt bearing will withstand loads of 1280 to 2130 lb/in^2 (90 to 150 kg/cm^2), shells with a skin of about 0·1 mm thickness will resist higher loads. Table 44 gives maximum load figures during laboratory material fatigue tests for an operating life of 100 hours under conditions of ideal lubrication, oil cleanliness, optimum temperature and bearing clearance.

Lead-bronze which will stand up to higher loads than white metal (up to 3550 lb/in^2 — 250 kg/cm^2) is used extensively in compression-ignition engines. The advantages of lead-bronze are more apparent at high temperature as its hardness at 120 °C is 12·6 to 15·8 ton/in^2 (20 to 25 kg/mm^2) (Brinell), whereas with white metal composed mainly of tin, it is only 5 to 7·6 ton/in^2 (8 to 12 kg/mm^2) (Fig. 325) [45].

Table 44

Maximum Bearing Load for a 100 hours Fatigue Test

Type of bearing	Material and thickness of lining mm	Load kg/cm^2
F – 1	Tin-base Babbitt – 0·5	120
F – 23	High-lead Babbitt – 0·5	155
F – 1	Micro – tin Babbitt – 0·1	180
F – 23	Micro – lead Babbitt – 0·1	180
F – 17	Trimetal: lead bronze 0·63, Babbitt 0·15	180
F – 12	Lead bronze – 0·5	245
F – 81	Aluminium, compact (withour bush)	385
F – 2 1	Trimetal: silver 0·33 + lead-indium 0·025	520
F – 77	Trimetal: lead bronze 0·3 — 0·6 + + lead-tin-copper 0·025	520

With the use of lead-bronze for bearing material, the journal must be hardened in order to prevent a rapid rate of wear. The journal must moreover be rigid to prevent whip with the accompanying overloading of the bearing at its edges. Foreign matter is not easily embedded in lead-bronze and efficient means of oil filtration must be provided. Recommended clearances for lead-bronze bearings are given in Table 45.

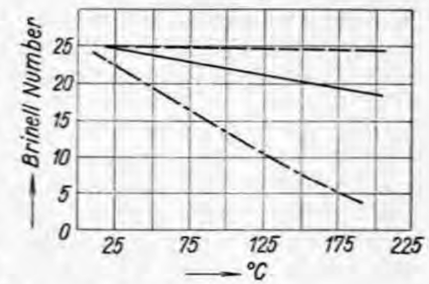


FIG. 325. Influence of temperature on the strength of bearing metals.
Dash line — silver, full line — lead bronze, dash and dot line — lead bearing metal.

Aluminium bearings have suitable frictional properties and are used either as full bushes or cast onto a steel shell. The high heat conductivity of aluminium assists the removal of heat from the bearing. The requisities for aluminium bearings are a hard journal of at least 450 Brinell and clean oil, maximum advisable loads being 3550 to 4980 lb/in^2 (250 to 350 kg/cm^2). Abrasive foreign matter is absorbed by aluminium better than by bronze. The clearance must be sufficient to prevent restraint of the journal with subsequent seizure.

Silver bearings have excellent characteristics and withstand high loads. They are preferred for big end bearings in radial engines. They are excellent in respect of fatigue resistance and freedom from corrosion and their high heat conductivity is valuable. Their properties are further enhanced by the use of a top layer of lead and indium, 0·025 mm thick.

Oil must always be supplied to plain bearings on the relieved side. The loaded bearing side must not incorporate oil distribution grooves, which considerably reduce bearing capacity. Oil grooves should be avoided where possible. Oil is fed to big end bearings to the frontal side of the crankpin

Table 45

Recommended Bearing Clearances

Bearing	Babbitt mm	Lead bronze mm
Main bearings: basic clearance up to 50 mm dia	0·050	0·060
for each further 25 mm	0·020	0·025
Crank pin bearings:		
basic clearance up to 50 mm dia	0·040	0·050
for each further 25 mm	0·015	0·020

viewed from the direction of rotation. The variation of bearing pressure in the big end bearing with varying crankshaft speeds is shown in Fig. 326. At 2000 rev/min the maximum loads imposed on the bearing by gas pressure acting upon the piston F and the load by accelerating forces I is small. At 5000 rev/min conditions are the opposite, the load transmitted to the bearing during the power stroke is small, (being cancelled by centrifugal force) while the highest load is imposed by the accelerating force I during the induction stroke.

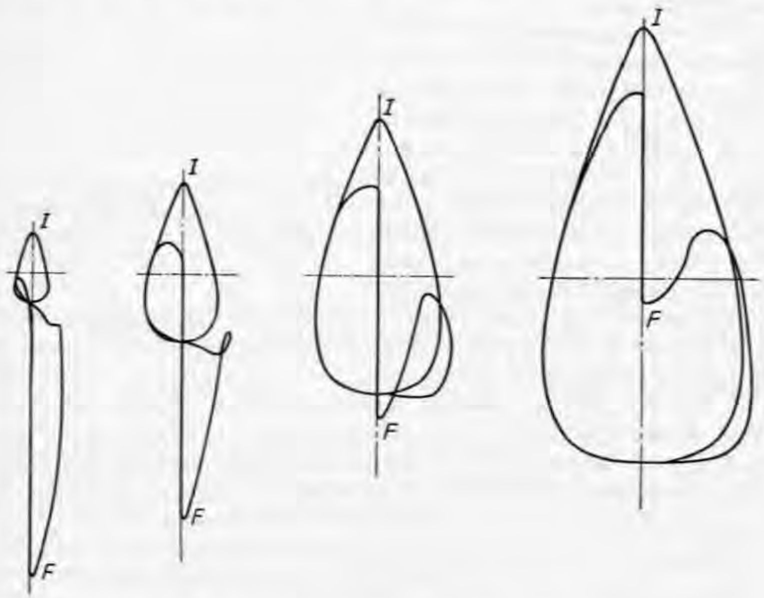


FIG. 326. Variation of load of connecting rod bearing with engine speed.
I — accelerating force, F — force developed by gas pressure upon piston. The diagrams are drawn for speeds of 2, 3, 4 and 5 thousand rev/min respectively.

Rolling bearings reduce friction and increase mechanical efficiency. They do not require the same degree of lubrication as plain bearings, but they are noisier in operation. Owing to smaller frictional loss, oil temperature is kept lower, so that the necessity for an oil cooler is sometimes obviated. A built-up crankshaft is however necessitated by the use of rolling bearings and this is more costly.

The built-up crankshaft scores on account of the relatively simple production of its small parts and because in the event of damage to a single journal, the whole crankshaft need not be scrapped, since the damaged part can be replaced. A built-up crankshaft is also noted for its strong internal damping of torsional vibration.

During assembly all bearings must pass through the first outer bearing ring in the crankcase, this having the disadvantage of limiting the choice of roller size which cannot be made according to the requirements set by the size of the inner and outer bearing rings in order to keep the bearing allowance low. Allowance must be specified in accordance with production tolerances of the bearing surface of the outer bearing ring, in order to guarantee the passage of the caged races of rollers fitted on the shaft. The increase in bearing allowance results in increased running noise. The rollers must be well aligned in their cage as misaligned rollers are the cause of blocking and possible cage fracture.

The oil for connecting rod lubrication must pass through a plain bearing on its way into the crankshaft, this bearing also being utilized to act as an axial crankshaft bearing. The centre of a plain bearing is shifted during engine running in order to make possible the accommodation of an oil film. The centre of a rolling bearing does not change its position and for this reason the combination of both types of bearing on a single shaft is inadvisable and for the same reason the bearing through which oil is supplied is provided with a large radial clearance in order not to transmit high loads and to avoid possible seizure.

The route of the oil passing from the flywheel end to the remote cylinder is long, resulting in an oil pressure loss and oil warming. In order to prevent failures of this last bearing, its correct lubrication must be ensured. If a pressure relief valve is incorporated in the crankshaft at the flywheel end (Tatra 111 etc., see Fig. 408), large quantities of oil flow through the shaft, cool oil is supplied to the remote bearing under pressure unreduced by the valve.

When roller bearings are used on a shaft which is not quite rigid, distortion of the journals resulting in outer edge overload on rollers must be considered. It is obvious that if clearance between rollers and bearing rings diminishes, by the inclination of the journal, to nil or less, the rollers or rings will be damaged. Long rollers are therefore not suitable and roller edges must be rounded off.

In motor cycle engines which use built-up crankshafts and one-piece connecting rods, big end rolling bearings are convenient. Split rolling bearings do exist, but they require a high standard of manufacture. For high bearings loads and high speed operation, caged roller bearings must be used. Roller bearings have a very low rate of wear and make short engine dimensions possible.

3. Lubrication

Cylinder temperature is always higher in air-cooled engines and a greater amount of heat is, therefore, transferred to oil. Should oil temperature be above 100 °C, an oil cooler must be incorporated. In high-performance water-cooled engines the use of an oil cooler is also necessary, for if an oil-to-water heat exchanger is employed, nothing is gained as heat lost to water must ultimately be transferred to the surrounding air so that the water radiator must

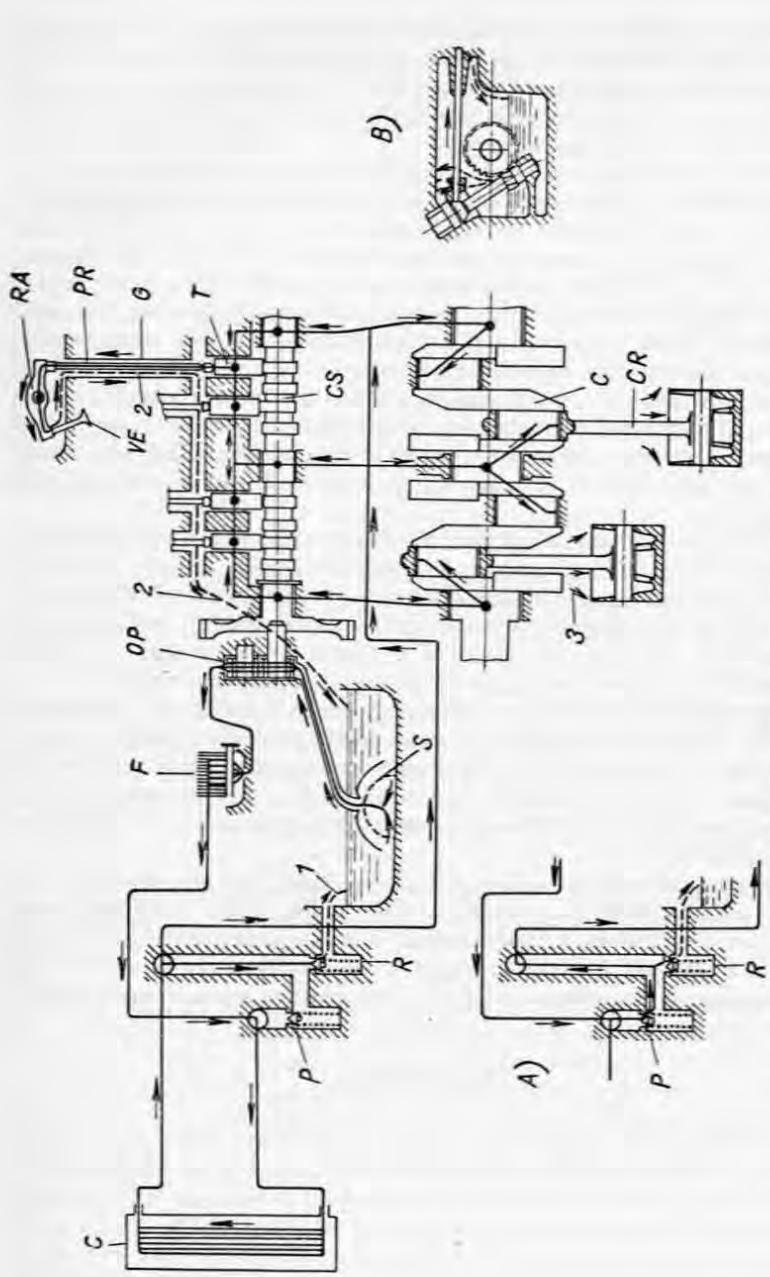


FIG. 327. Lubrication of a Tatraplan engine with an oil sump in the crankcase.
OP — oil pump, F — oil filter, P — relief valve, C — oil cooler, R — reducing valve.

be increased in size proportionally. Thus not only is a heat exchanger added, but also an increased radiator surface and some water.

Oil temperature may reach 110 °C and for short spells up to 120 °C. Temperatures under 80 °C are undesirable, for friction loss increases and there is a risk of oil dilution by fuel.

Lubricating systems can broadly be divided into those with a "wet sump" and those with a "dry sump". With wet sump lubrication (Fig. 327), the bulk

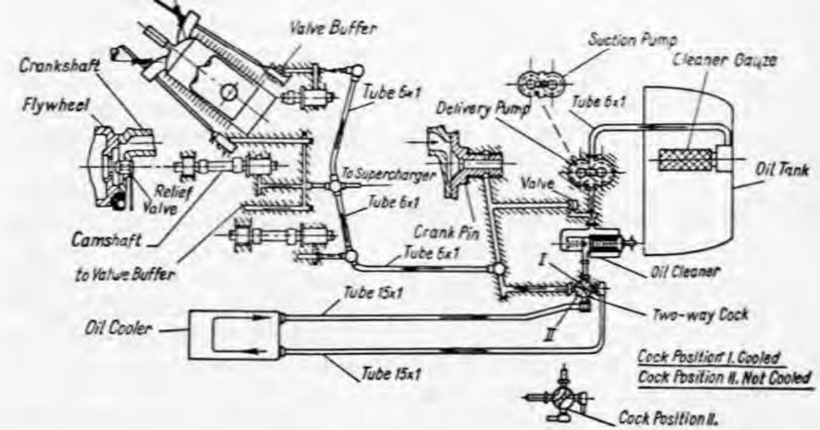


FIG. 328. Lubrication with dry crankcase (Tatra 111). The suction pump delivers the oil from the tank at the flywheel to the oil container.

of oil is in the sump and a single pump circulates the oil. If an oil cooler is employed, it must be interposed between this pump and the bearings. The cooler is then under pressure and must be designed for this pressure. In order to prevent damage to the cooler in winter, when the oil is thick, in a cold engine, a safety by-pass valve is provided in parallel with the cooler and oil passes through this until the oil is sufficiently warmed.

If the engine is intended for off-the-road vehicles in which regular engine running is required even on extreme gradients (up to 60%), the use of a dry-sump system is called for. In such a system the oil supply is in a separate tank remote from the crankcase (Fig. 328). Oil is drawn from the sump and delivered to the tank from the front and rear ends of the crankcase and oil for pressure lubrication of the engine is taken from the tank by the pressure pump. If the tank can be mounted below and in front of or behind the engine, one scavenging pump can be saved and the layout is simplified (Fig. 328). Each one of the scavenging pumps must be of greater capacity than the delivery pump.

If a three-pinion gear pump is used for scavenging, foaming also must be taken into account as oil is delivered to the common return pipe by one pump and air by the other. The oil tank must then be high enough to prevent the loss of foam through the filler orifice. A foam overflow tube to the crankcase may be incorporated. In a lubrication system with only one scavenge pump less oil foam is formed.

An oil flow of 20 to 25 l/b.h.p. h is generally used for petrol engine lubrication and 25 to 35 l/b.h.p. h is specified for compression-ignition engines. In V engines or where roller bearings are employed, the oil flow may be lower. It is an error to assume that with the use of a greater volume of oil, oil temperature will be reduced. If the amount of oil is increased, the amount of heat transferred is also increased but temperature is not affected.

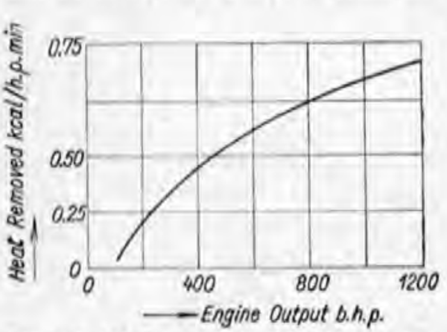


FIG. 329. Heat removed by oil in aircraft engines.

Lubricating oil is sometimes also employed for piston cooling. The oil is either sprayed onto the piston by a jet from the connecting rod small end or from a special jet lodged in the crankcase. The quantity of heat removed by oil in aircraft engines is shown in Fig. 329.

A finned sump does not aid cooling to any great extent as the oil does not flow down the cooled sump walls. Cooling is more intense on the crankcase walls, over which oil flows in a thin film. If cooling by a finned sump is to be efficient, air must be brought to circulate about its cooling area.

Best oil cooling is obtained by the use of a tube-type radiator of sufficient strength and efficiency. Oil should circulate through the cooler several times in order to keep flow velocity in the tubes sufficiently high. Provision must be made for draining the oil out of the cooler. The oil cooler should have approximately the same air resistance as a cylinder in order to utilize air pressure to the utmost.

CHAPTER XVII

ENGINE STARTING

Air-cooled engines are started in the same manner as water-cooled types, but they reach operating temperature sooner, which makes for lower cylinder wear. With ambient temperatures below freezing point, an air-cooled engine is at an advantage, there being no water to attend to.

At temperatures from —20 to —25 °C frictional loss with the oil stiff is equal to the m.e.p. and the engine is therefore unable to overcome its own resistance.

At extremely low temperatures (such as —30 to —40 °C) the engine must be pre-heated directly by a petrol blow-lamp. Heating of cylinders and inlet manifold takes from 3 to 10 minutes according to outside temperature. The overall time for starting an air-cooled engine is about one third of the time needed for a water-cooled engine. A water- cooled engine must be pre-heated with steam at such low temperatures for even hot water poured into the cold engine freezes immediately at temperatures around —40 °C. Condensed steam may also freeze in the engine and preparations are thus delayed. After pre-heating with steam, the cooling system must be filled up with hot water, this all adding to the time needed for engine starting. The induction of warm air during starting is important. For this reason the inlet manifold is heated, or in the case of oil engines, auxiliary combustion is provided in the inlet manifold, by an ignited spray of fuel. The burning mixture of fuel and air enters the cylinder, increasing temperature at the end of the compression stroke. The fuel spray is ignited by a sparking plug (JAZ 204) or by a glowing coil.

During the starting of a compression-ignition engine at very low temperatures, almost double the usual quantity of fuel must be injected into the cylinders and injection must be retarded until almost t.d.c. in order to inject the fuel into heated air. Precautions must be taken to prevent fuel oil from solidifying by heating the injection pump and by leading the exhaust in the proximity of the fuel supply line and the fuel tank.

In tractor engine design, provision must be made by the designer for the generally poor standard of maintenance and an accumulator therefore cannot be relied on to supply current for the usual type of electric starter. Inertia starters are therefore employed. Kinetic energy is accumulated in such a starter by the cranking of a flywheel by hand and the flywheel is then con-

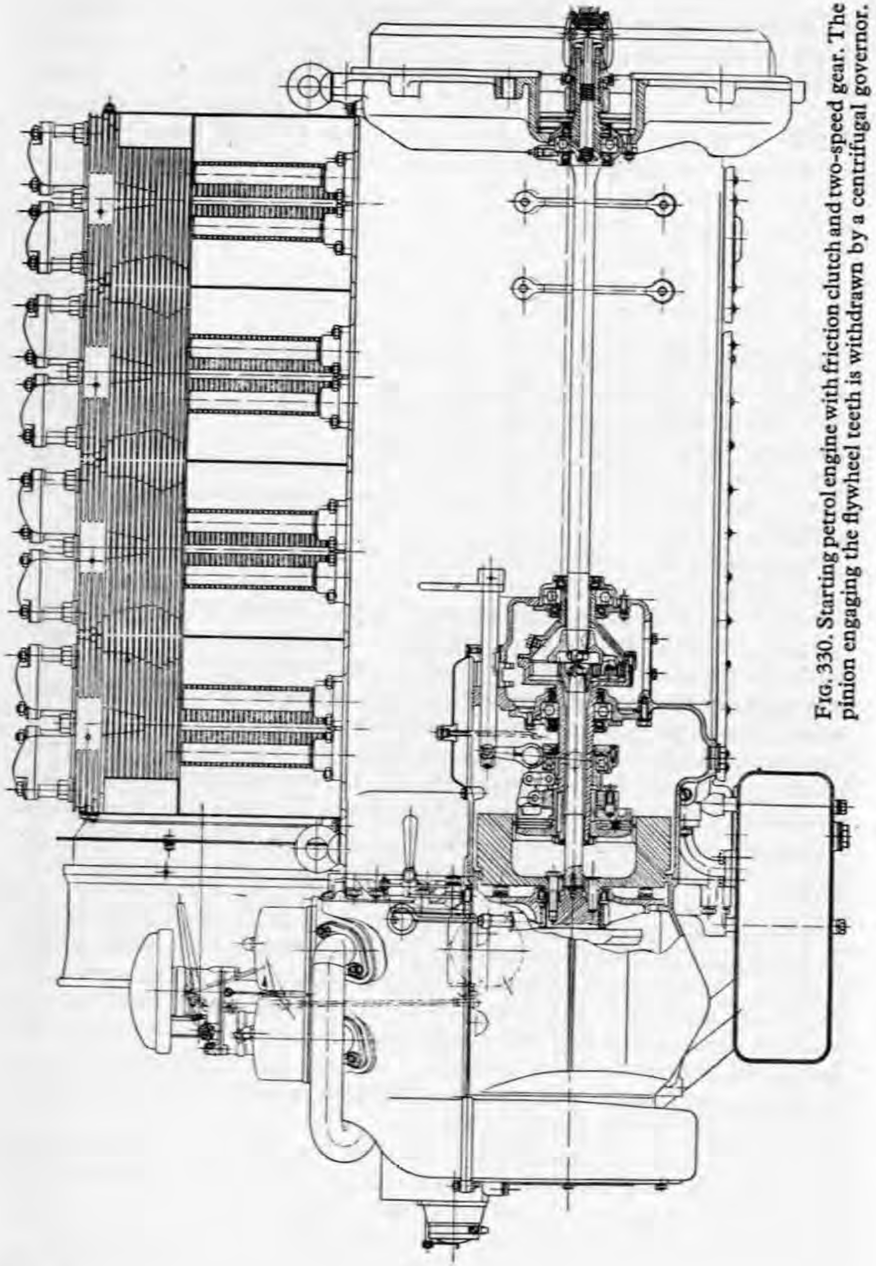


Fig. 330. Starting petrol engine with friction clutch and two-speed gear. The pinion engaging the flywheel teeth is withdrawn by a centrifugal governor.

nected to the crankshaft by a dog clutch. The energy accumulated will turn the engine crank several times; such a method of starting is reliable, but a great amount of physical effort is required when temperature is low [9].

Very satisfactory results have been obtained by the use of an auxiliary starting engine run on petrol. Compression-ignition engines are fitted with a small petrol engine, usually a twin-cylinder type, which is easy to start by hand even at very low temperatures. This auxiliary engine also assists starting by heating the engine by the outgoing cooling air and by heating the induction manifold or oil by its exhaust gases.

This method of engine starting is employed on the Tatra 904 engine shown in Fig. 330. The output of the small engine is transmitted through a friction clutch and two-speed gear to the geared flywheel. The higher gear is for turning the engine, the lower for actual starting. The pinion engaging with the flywheel teeth is withdrawn at a given flywheel speed by a centrifugal governor which releases a spring to withdraw the pinion. Such a starting mechanism is very convenient and has proved its value also on the ČKD caterpillar tractor.

In order to facilitate engine heating with a blow-lamp the cooling air duct should be provided with a detachable cover and a correctly set lamp holder. After heating the engine the duct must naturally be closed again in order to prevent large cooling air losses.

If the vehicle is equipped with a heating system independent of the engine (e.g. South Wind, Eberspächer etc.) the hot air produced may also be used to heat the engine prior to starting.

Starting difficulties in summer may arise when petrol evaporates from the warm carburettor after the engine has stopped. When the usual type of fuel pump driven from the engine is employed, the engine must be run on the starter for a long time before the carburettor is filled up again. The remedy lies in providing good heat insulation for the carburettor, the prevention of vapour loss from the float chamber by a special valve or in the use of an electric fuel pump switched on together with the ignition.

The fuel line to the fuel pump must lead through a cool region and avoid the exhaust pipe area in order to minimise the formation of vapour locks.

CHAPTER XVIII

THE TWO-STROKE ENGINE

1. General considerations

The chief advantage of the two-stroke engine is the fact that it provides twice the number of working strokes given by the four-stroke engine at equal speed. The formula for output per litre of an engine is

$$\text{b.h.p.} = \frac{V \cdot n \cdot p_e}{450}$$

With p_e equal, a two-stroke engine should at a given engine speed, theoretically, produce twice the power of a four-stroke engine, as one expansion stroke takes place during every crankshaft revolution, there being no "pumping strokes" (i.e. induction and exhaust) and air being drawn into the cylinder by other means.

The whole working cycle must be performed in the two-stroke engine during one crankshaft revolution, i.e. the expansion stroke, the removal of burnt gas from the cylinder, the entry of the fresh charge and the compression stroke. The expansion and compression strokes remain as in the four-stroke cycle, but the cylinder must be scavenged and recharged in a brief period during which the piston moves past bottom dead centre.

In a four-stroke engine, the piston approaching t.d.c. displaces the exhaust gases and when moving to b.d.c., it draws in a fresh charge from atmosphere. For this to happen in the short period available about b. d. c., in the two-stroke engine, air under pressure is needed to scavenge and charge the cylinder. Compressed air can be available either from a sealed crankcase or from an independent piston or rotary blower.

The crankcase functions as an uneconomical and inefficient charging and scavenging pump. The amount of air which can be used for cylinder scavenging is less than the swept volume of the cylinder, owing to the low volumetric efficiency of the crankcase which contains a large "dead" space. It is therefore impossible to remove all burnt gas from the cylinder. Beside this, air escapes through the exhaust ports during scavenging, so that the charge at the beginning of the compression stroke always consists of a mixture of exhaust gases with a fresh charge. Only an amount of fuel corresponding to the volume of available air (oxygen) for combustion can be delivered to the cylinder and for this reason the output of the two-stroke engine is not twice that of the

four-stroke; the mean effective pressure is about 42 to 56 lb/in^2 (3 to 4 kg/cm^2).

If crankcase compression is so used for the scavenging of multi-cylinder engines, each cylinder must have its separate crankcase compartment. In horizontally opposed engines with pistons travelling in opposing directions, a common crankcase can be used for both cylinders; both cylinders then must fire simultaneously and the firing strokes of such an engine are at the same intervals as in a single cylinder type. Such an arrangement however, is well balanced (e.g. the IFA BK 350 motor cycle engine).

If the crankcase is used as a pumping chamber, pressure lubrication of the usual type cannot be used and roller instead of plain bearings must be used for crankshaft journals and big ends.

Two-stroke engines employ two systems of crankcase scavenging:

1. the three-port system
2. the two-port system.

A diagrammatic representation of both types is given in Fig. 331. The three port system incorporates three ports in the cylinder, i. e. the exhaust, transfer and inlet ports. With the piston at bottom dead centre, the exhaust and transfer ports are uncovered and cylinder scavenging takes place, while the inlet port is covered by the piston. When the piston reaches top dead centre, the bottom edge of its skirt uncovers the inlet port and air flowing through the carburettor enters into the crankcase.

Suction into the crankcase can begin only after the piston travels for some distance and under-pressure develops in the crankcase. The inlet port must open relatively late for it must remain open for an equal time after the piston has passed t.d.c. If the port remained open for a long time after t.d.c. a large volume of air would escape back into the induction port. The restricted period of induction at t.d.c. lowers the volumetric efficiency of the crankcase.

In the two-port type, there are only two types of ports in the cylinder wall, the exhaust and transfer ports. Induction timing is controlled by a rotary valve, diaphragm valve etc. Induction begins an instant after b.d.c. and ends an instant after t.d.c.

The volumetric efficiency of the crankcase is improved in comparison with the previous arrangement.

The volumetric efficiency of the crankcase is sometimes improved by the use of a separate auxiliary piston, which increases the volume of the crankcase when the working piston is at t.d.c. and reduces it at b.d.c.

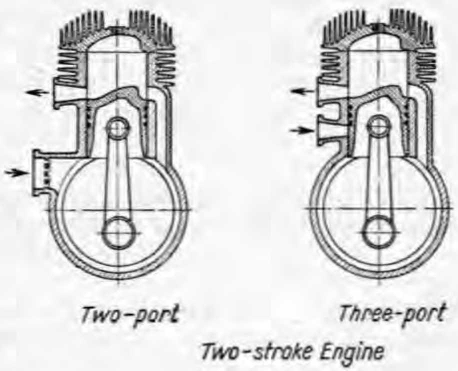


FIG. 331. A two-port and three-port two-stroke engine.

The reciprocating blower driven by the engine crankshaft meets the requirements of a charging pump and on the supply of pressure during scavenging. Its large dimensions and the necessity of balancing reciprocating masses tend to cancel the advantages mentioned to some extent. Reciprocating blowers are used in motor cycle and automobile engines (DKW) and in large compression-ignition marine engines.

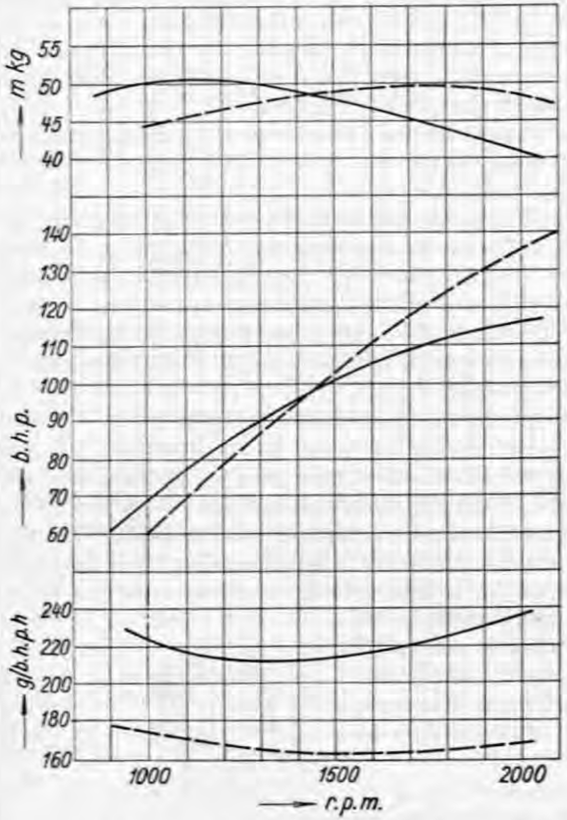


FIG. 332. Comparative characteristics of a JAZ 204 engine with a Roots blower as supercharger (full line) and a Krauss-Maffei engine equipped with a centrifugal compressor (dash line).

The Roots type rotary blower is most commonly used, its advantages being simplicity, low weight and inherent balance. The circumferential velocity of the rotor is restricted to 98 to 164 ft/s (30 to 50 m/s) in order to minimize noise. In large engines a single blower would be too large and the use of several smaller blowers is usually more convenient. The variation of the

amount of air supplied with the variation of speed is favourable, especially at low speeds. With high charging pressures (over 0·5 atm) efficiency of the Roots type blower is low and this is overcome by arranging two blowers in series. The torque curve of such an engine is good and starting easy.

A mechanically driven centrifugal supercharger is preferable if judged by its overall size and weight. Pressure rises (and falls) as the square of speed and some difficulty therefore arises during starting and at low running speeds. This type of supercharger is at its best in engines in which running speed is more or less steady and which work for most of the time at rated speeds.

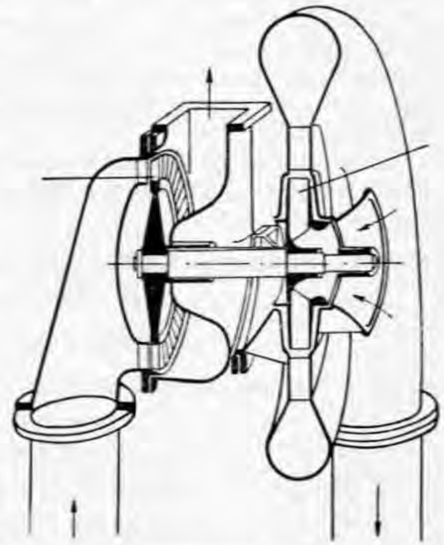


FIG. 333. Exhaust gas turbo-supercharger.

With the increase of engine speed the period for which ports are open is reduced and higher pressure is required if the same amount of air is to enter the cylinder; the centrifugal supercharger meets this demand. This however results in a very flat torque curve with its peak at relatively high speeds, whereas for vehicle propulsion purposes a torque curve which rises with decreasing engine speed is desirable. This can be achieved by the use of a Roots-type blower. The torque curves of an engine equipped with a Roots-type blower (JAZ 204) [43] and with a centrifugal supercharger (Kraus-Maffei) are compared in Fig. 332. The torque curve of an engine with a centrifugal supercharger can be modified by a reduction in the amount of fuel supplied at high speed. Under such high speed conditions the engine thus modified works with a large amount of excess air.

Turbo-superchargers driven by exhaust gases may also be used for two-stroke engines. (Fig. 333.) The scavenging pressure of a two-stroke engine is about 1·10 to 1·4 atm. The power consumed by the scavenging supercharger on high speed two-stroke engines is considerable, sometimes amounting to 15% of the b.h.p. The temperature of exhaust gases of a two-stroke engine is much lower than that in a four-stroke engine owing to dilution by the incoming charge. The output of an exhaust gas turbine is, therefore, also small. In order to reduce supercharger input, it would be beneficial to use a lower pressure for scavenging than for charging, but this would make the installation of a two-stage blower necessary together with a complex lay-out. Furthermore, a two-stroke engine requires the supply of air under pressure before the first power stroke takes place. All these conditions contribute to the difficulties met in the design of such a supercharging system and owing to them, the separate turbo supercharger for the two-stroke engine is so far not beyond the experimental stage.

Recent practice in large two-stroke marine engines has shown the practicability of using a combination of an exhaust driven turbo-supercharger with mechanically driven blowers or superchargers.

The two-stroke engine can also be scavenged by utilizing the kinetic energy of the exhaust gases (e.g. the Kadenacy system). This takes advantage of the pulsation waves in the exhaust pipe for cylinder scavenging and charging. The pulsations in the inlet and exhaust pipes may also be utilized for better charging in four-stroke engines.

Pulsation waves set up in the exhaust pipe are governed by the same laws as sound waves. In a smooth tube blanked off at both ends the frequency of pressure waves is $2\,L_e/a$,

where L_e denotes the length of tube (equivalent)

a ,, ,, sound velocity in a given medium

$$a = C\,(k\,.\,p\,.\,v) = C\,(k\,.\,R\,.\,T)$$

where k denotes the ratio of specific heat values of air (gas)

p ,, ,, mean pressure

v ,, ,, specific volume

T ,, ,, temperature (absolute)

C ,, ,, a constant which decreases with tube diameter.

The velocity of sound is usually between the limits of 1246 to 1476 ft/s (380 to 450 m/sec).

In calculations the vibrating system is substituted by an equivalent length of tube L_e of uniform section:

$$L_e = 2\,(L + C_r)$$

where L denotes the true length of tube (open)

C_r ,, ,, Raleigh correction.

Since rebound will not occur exactly at the end of the tube, about 0·4 of the internal diameter of the tube must be added to its true length. Except in very short tubes, the Raleigh correction may be neglected, whence

$$L_e = 2\,L$$

The natural frequency of an air column in the pipe

$$t = \frac{2\,L_e}{a} \qquad (100)$$

If the exhaust is connected to surrounding atmosphere by a pipe of uniform section then the natural frequency of the pipe

$$t = \frac{4\,L}{a}$$

If this frequency is resonant with the frequency of exhaust impulses, conditions are least favourable as

$$t_e = \frac{60}{n}$$

If the frequency is equal to the period of exhaust port opening, conditions are the most favourable as

$$t_b = \frac{\delta_e}{360}\,.\,\frac{60}{n} = \frac{\delta_e}{6\,n},$$

where δ_e denotes the period of exhaust port opening expressed in degrees of crankshaft rotation.

The optimum exhaust pipe length

$$L = \frac{\delta_e\,.\,a}{24\,n}$$

If the exhaust pipe leads to a chamber of sufficient volume, it can be assumed to lead to the ambient atmosphere and the frequency of any further pipe may be disregarded.

In a two-stroke engine, port timing may be classified as:

1. symmetric
2. asymmetric

Symmetric timing is used more often and it is the result of port opening being governed by the motion of the piston. First the exhaust port opens, and a reduction of pressure in the cylinder results; then the transfer port opens. The ports close in the opposite order which is not favourable as some of the fresh charge can escape through the exhaust port after the transfer port is closed.

Asymmetric timing is possible only with engines with opposed action pistons by an advanced coupling of one crankshaft; on engines with valve timing; and on engines with a modified crank mechanism. A diagram of symmetric and asymmetric port timing is shown in Fig. 334. The timely closure of the exhaust port prevents the escape of the fresh charge from the cylinder with asymmetric timing. Asymmetric timing can also be achieved by the use of rotary, sleeve valves or poppet valves to control induction or exhaust.

Port timing requires precise port shapes in manufacture, and machined ports are therefore favoured. In some motor cycle engines detachable covers are provided for better access for the machining of transfer ports.

Ports of rectangular and rhomboidal section have the largest flow area. But in view of the considerable danger of piston ring breakages, the maximum width of a port opening is 15° and its edges should be rounded off.

Oval ports have rounded corners near the upper and lower surfaces and flow area is partly reduced, but such a cross section tends to reduce piston ring trouble.

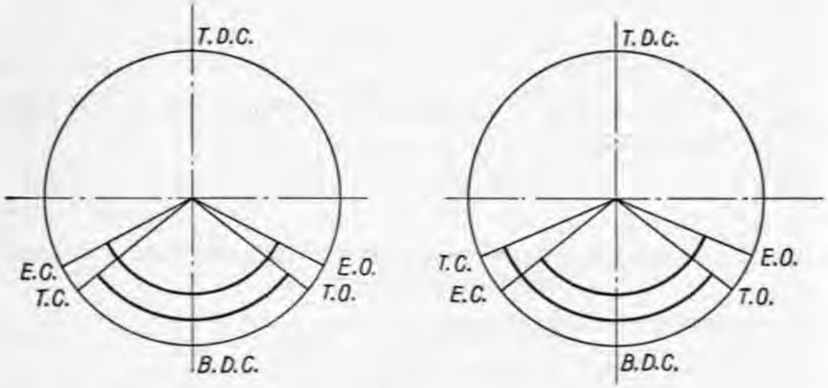


FIG. 334. Symmetric (left hand) and asymmetric (right hand) valve gear of a two-stroke engine.

Circular ports are favourable from the point of view of ring safety, but their flow area is relatively low and they are not suited for exhaust ports owing to the increased rate of carbon deposit.

The tangential arrangement of ports makes for good scavenging. Several rows of ports are sometimes arranged one above the other with each row of different tangential slope. The width perpendicular to the direction of flow must be taken as a basis in calculations of tangential ports.

The openings of the ports should be rounded off in order to promote flow and prevent contraction. The exhaust port edges should be rounded off from inside the cylinder and induction ports from the outside of the cylinder. By such chamfering up to and above 20% improvements of flow can be effected, the gain being the same as when the ports are increased in width by the same amount.

Split ports reduce the flow area and the partitions should therefore be kept to a minimum width which is governed by cooling requirements. The partitions between exhaust ports in some high performance engines are water-cooled. In larger engines the cavities are cast, but in automobile engines they must be drilled or milled. Port overheating can result in piston seizure or increased piston wear and subsequent ring breakages.

The valve gear of a two-stroke engine operates under higher stress than that of a four-stroke engine. The period for which the valve is open equals only about a third of the period in a four-stroke engine, for which reason accelerating forces must be observed closely. The rate of gumming-up of piston rings is greatly reduced by the use of poppet exhaust valves. The use of two smaller

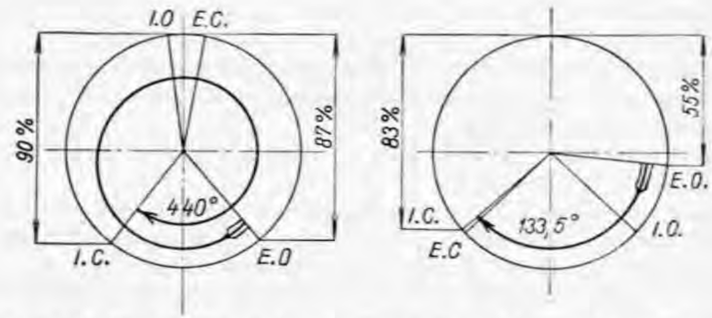


FIG. 335. Difference in timing of a two-stroke and four-stroke engine.

valves is preferable in the interest of good cooling. Both valves open simultaneously and if operated by a common rocker, provision must be made for the separate clearance adjustment of each.

The difference in timing of a two-stroke and four-stroke engine is shown in Fig. 335. In the normal type of four-stroke compression-ignition engine, a crankshaft rotation angle of 440° is available for the removal of burnt gas and the induction of a fresh charge of air. The same must be performed in a rotation angle of 133·5° in the two-stroke engine (quoted figures relating to the JAZ 204 engine). About a third of the interval of a four-stroke engine is available, necessitating the use of large flow area transfer ports and increased pressure.

The effective piston travel during expansion is limited to 55% compared with 83% in the four-stroke cycle; during compression to 83%, compared with 90% which influences unfavourably the overall utilization of the cylinder. The reduction of 1% of piston travel during compression results in a 1% loss of output according to Schweitzer [53] and a reduction of expansion stroke by 1% results in a 0·5% reduction in output. The two-stroke engine actually operates under conditions of slight supercharging.

The main problems of the two-stroke engine lie in the perfect scavenging of the cylinder and its complete charging. The drawbacks of the crankcase as a scavenging pump with its many imperfections will not be discussed here, but other phenomena absent or negligible in the four-stroke engine must be taken into account.

The escape of fresh air through the exhaust ports before scavenging is completed.

Incomplete exhaust, which results if the pressure inside the cylinder and the exhaust port does not drop sufficiently before transfer ports are opened, upon which a part of the burnt charge enters the transfer ports. The quantity of air entering the cylinder is reduced by the volume of this portion of the exhaust gases and moreover exhaust gases heat the incoming charge and the piston crown.

The late closure of transfer ports results in a portion of the air returning from the cylinder under compression, thus reducing the charge.

Too large a flow area of the exhaust ports offers too small a resistance to the passage of exhaust gas, thus impairing the build-up of pressure inside the cylinder during charging.

In order to observe the degree of scavenging effected, the two following ideal cases will be considered.

Perfect scavenging. The air supplied into the cylinder does not mix with exhaust gases and removes an equal volume of exhaust gas from the cylinder. With an air excess some of the fresh air remains in the exhaust gases, for the amount of fuel injected into the cylinder is less than that corresponding to the amount of air in the cylinder. This proportional quantity of air in the exhaust must be added to the fresh cylinder charge. Ideal conditions with scavenging of this type are given in Fig. 336, with $\lambda = 1$. The relative charge R is shown on the vertical axis, the charging ratio P on the horizontal, the two symbols being

$$R = \frac{\textit{quantity of fresh air in cylinder}}{\textit{displacement}}$$

$$P = \frac{\textit{quantity of air supplied by supercharger (blower)}}{\textit{displacement}}$$

With a mixing ratio of $\lambda = 1$, i.e. without excess air, $P = R$. With a mixing ratio $\lambda > 1$ the cylinder charge is increased by the amount of air contained in the exhaust gas not scavenged from the cylinder.

Perfect mixing. In this instance it is assumed that the air supplied into the cylinder mixes completely with the burnt charge and that no air escapes straight through to exhaust. A portion of air, proportional to the mixing ratio of fresh air with exhaust, escapes with exhaust gas from the cylinder. The conditions of such scavenging with $\lambda = 1$ are shown in Fig. 336.

Under real operating conditions neither one nor the other of the described types of scavenging occur. The extent to which various means of scavenging approach these ideal conditions is shown in Fig. 336.

In a two-stroke engine the following types of scavenging are used.

1. Transverse scavenging, with exhaust and scavenging ports on opposing sides of the cylinder wall, scavenging air flowing in a transverse direction and being deflected towards the cylinder head by a suitably shaped deflector formed on the piston crown.

2. Reverse-flow scavenging with scavenging ports on both sides of the exhaust port in one half of the circumference of the cylinder wall, with scavenging air reversing in flow direction after being deflected by the cylinder wall. Flat-top pistons without deflectors are used.

3. Parallel-flow scavenging with scavenging ports at one end of the cylinder and exhaust ports at the other. The scavenging air and exhaust gases move

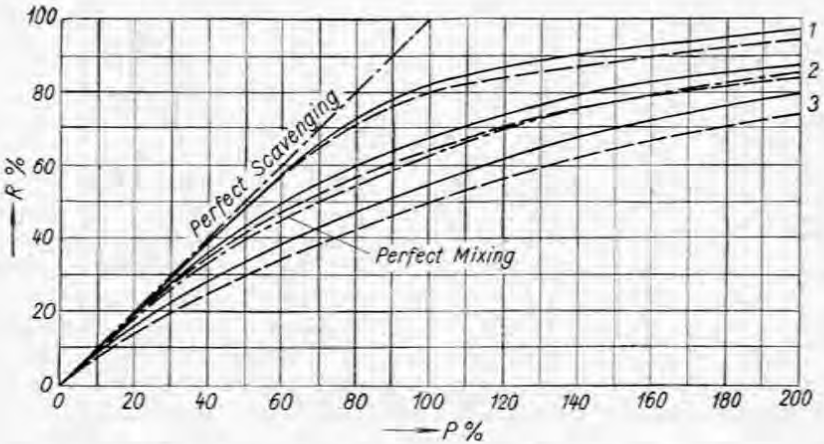


FIG. 336. Different methods of scavenging two-stroke engines.

1 — parallel flow scavenging, *2* — transverse scavenging, *3* — reverse-flow scavenging. Dashed curves hold for a scavenging pressure of 0·7 kg/cm², full curves for 0·1 kg/cm².

in the same direction, parallel with the cylinder axis. The scavenging ports are usually opened by the piston, exhaust ports by poppet valves or also by piston movement. Parallel flow scavenging is also employed in twin-piston U engines with a single crank (split-singles), and with double-piston designs with opposed motion.

Fig. 337 tabulates the most common means of two-stroke cylinder scavenging. The largest flow cross sections of scavenging ports are to be found on parallel-flow types with poppet or rotary exhaust valves. In such a case the whole circumference of the cylinder wall is available and a port flow cross section of about 35% piston area can be achieved without difficulty. With two exhaust valves, the exhaust flow cross section is about 18%, i.e. almost one half.

A certain back pressure in the exhaust system is beneficial with a view to cylinder supercharging.

In the transverse arrangement, scavenging and exhaust ports must be located in the cylinder wall. The size of the scavenging ports is therefore about 25% of piston crown area and that of the exhaust ports about 21% of piston area.

With reverse-flow scavenging (of the Schnürle type, for instance), ports of both kinds must be spaced over only about 3/4 of the cylinder circumference. The size of the scavenging ports therefore amounts to only 18% and that of the exhaust ports only about 14% of the piston crown area.

For better comparison a current four stroke timing layout employing two poppet valves is included in the illustration. The inlet valve flow area of this

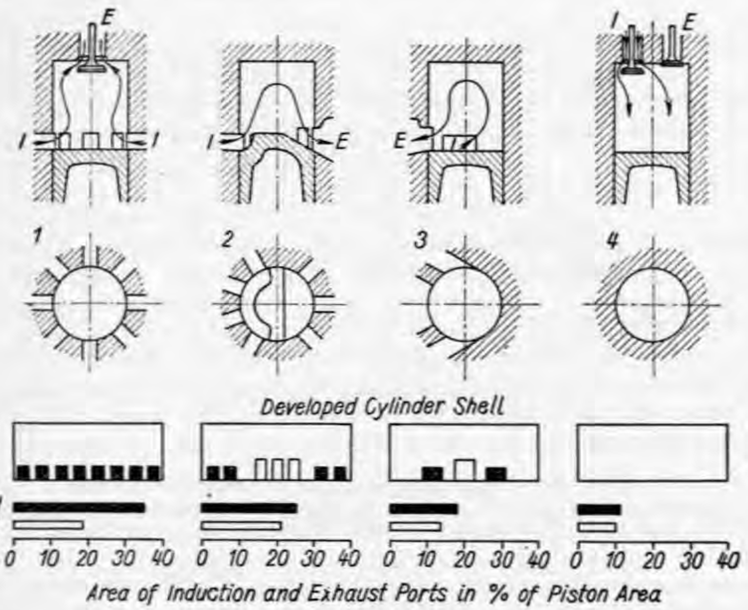


FIG. 337. Comparison of flow cross-section areas for different methods of scavenging.

type amounts to about 11% and that of the exhaust valve to about 10% of piston area. Compared with the same figures for a two-stroke engine the flow section of a four-stroke engine is much smaller; it must however be borne in mind that a period three times longer is available for expelling burnt gas and inspiring a fresh charge.

The examples given prove that the largest scavenging port flow cross sections are possible with the parallel-flow layout with a poppet, rotary or sleeve valve. Conditions with opposed-motion pistons are different (e.g. in the Junkers-type engine). For each kind of port the complete circumference of the cylinder wall is available, but the height of the ports depends on the position of one piston while displacement depends on the position of both pistons. The mean flow cross section in such an engine relative to swept volume is only about one half of the corresponding values achieved in the parallel-flow engine with valve timing.

The construction of a two-stroke engine with parallel-flow scavenging with valve timing or with opposed-motion pistons is no less complex than that of the four-stroke engine. Only in reverse-flow designs, such as the Schnürle type, design is simplified by the absence of valve mechanism or by the provision of only a single crank mechanism. Several types of water-cooled engines of this type have appeared recently, such as the Krauss-Maffei, Gräf & Stift, GM and others, data of which are included in Table 4.

2. Calculation of two-stroke engine timing

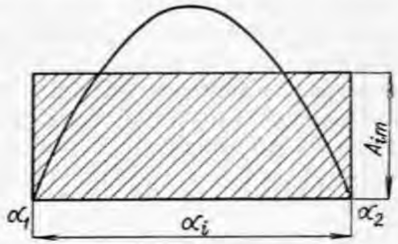


FIG. 338. Port openings in a two-stroke engine.

The inlet (transfer) ports of a two-stroke engine are generally opened by the motion of the piston or by various forms of rotary or sleeve valves. The rate of port opening has an important bearing on cylinder scavenging. The angular travel of the crankshaft is chosen for a basis. If the size of the uncovered area A_i cm² is entered above this base, the rate of opening, given in Fig. 338 is obtained. This graph shows the "time area", which is expressed by the formula

$$\int_{\alpha_1}^{\alpha_2} A_i \, d\alpha = A_{im} \cdot \alpha_i$$

where α_i denotes the duration of port opening expressed in degrees of crankshaft rotation

A_{im} „ „ mean port opening area (cm²)

As all air supplied to the cylinder V_{supp}. must pass through this area mean air velocity v_i is given by these equations:

$$V_{supp} = V_1 \cdot \eta_v = A_{im} \cdot t_i \cdot v_i$$

$$t_i = \frac{60 \cdot \alpha_i}{360\, n} = \frac{\alpha_i}{6\, n}$$

$$V_{supp} = \frac{A_{im} \cdot \alpha_i \cdot v_i}{6\, n}$$

where t_i denotes the duration of opening of inlet port

n „ „ engine speed

v_1 „ „ displacement (cm³)

η_v „ „ volumetric efficiency $= \dfrac{V_{supp}}{v_1}$

The value of $A_{im} \cdot \alpha_i$ is obtained by planimetry from the area of the diagram in Fig. 338:

$$A_{im} \cdot \alpha_i = v_1 \cdot 6\,n \cdot \frac{\eta_v}{v_i}$$

From this equation the "time area" required for the charging of the cylinder at an assumed mean velocity of v_i can be calculated. This area is in direct proportion to the displacement, breathing efficiency and engine speed and inversely proportional to the mean inlet flow velocity. In order to meet given requirements, a large area and short duration can be specified, or a long duration and small area. The first mentioned case is more favourable as it makes for a longer effective stroke. If, however, the ports are opened by piston motion, their height and duration of opening are limited. The position of the upper edge of the transfer ports is specified by timing requirements, i.e. the angle.

For rectangular ports

$$A_{im} = b_i \cdot h_{im} \cdot L,$$

where b_i denotes the total width of scavenging ports,
- h_{im} „ „ mean exposed height of scavenging ports expressed as a percentage of piston travel,
- b_e „ „ total width of exhaust ports,
- h_m „ „ mean uncovered height of exhaust ports in $\% L$,
- h_i „ „ height of scavenging ports in $\% L$,
- h_e „ „ height of exhaust ports in $\% L$,
- L „ „ piston stroke

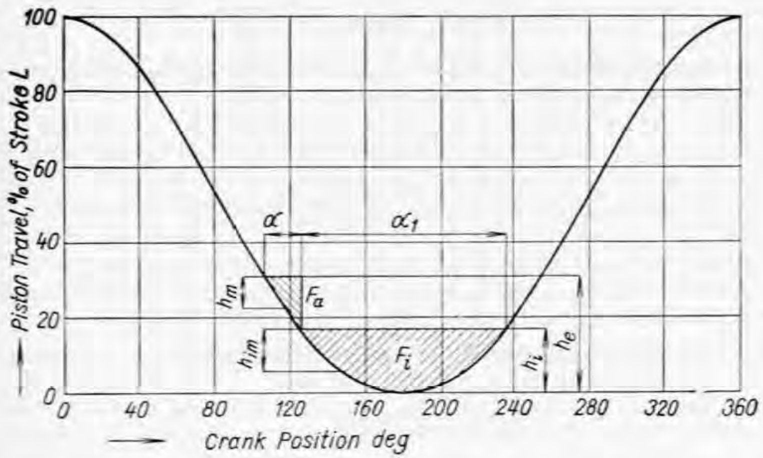


FIG. 339. Restriction of flow area by piston travel at bottom dead centre.

When describing two-stroke engines it is usual to express port height (h_i, h_e, h_{im} and h_m in Fig. 339) as a percentage of stroke. The area A_i in Fig. 339 is limited by the piston travel curve and the height of the transfer port h_i its dimension being $\%°$ (percentage of degrees).

The following formulas apply

$$h_{im} \cdot \alpha_i = A_i; \quad A_{im} \cdot \alpha_i = b_i \cdot L \cdot A_i;$$

$$A_i = \frac{\eta_v}{v_i} \cdot \frac{6\,v_1}{b_i\,L} \cdot n.$$

For speedy calculation the nomogram of Fig. 340 is useful, as the area A_i may be determined, or if A_i is known, then α_i and h_i can be established.

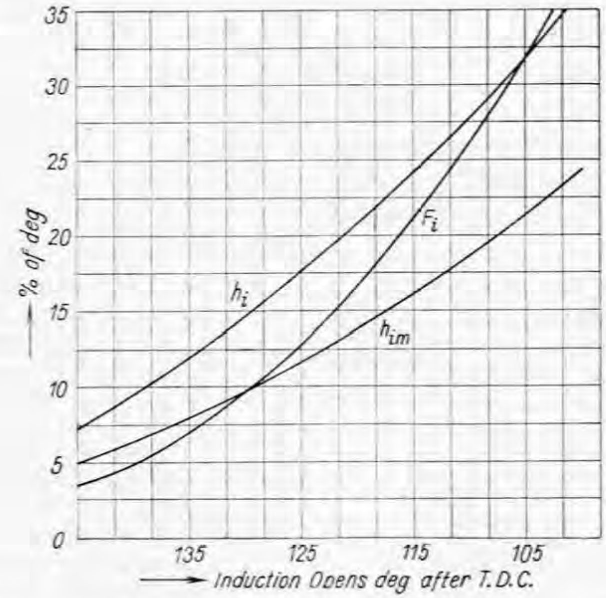


FIG. 340. Nomogram for the calculation of the F_i area valid for a connecting rod length of 4.5 R. The values of h_i and h_{im} are expressed in % of the stroke.

Scavenging pressure can be approximated from the diagram in Fig. 341, where
- D denotes the cylinder bore (m)
- n „ „ engine speed (rev/min)
- β „ „ ratio of the width of entry ports (perpendicular to the direction of flow) to cylinder circumference $\beta = \dfrac{b_i}{D \cdot \pi}$

The pressure required for scavenging varies between 2 to 10 lb/in² (0·15 to 0·7 kg/cm²). With parallel flow scavenging with ports over the entire cylinder circumference β equals up to 75%, but usually does not exceed 65%, with transverse scavenging $\beta = 30$ to 35% and with reverse flow scavenging only 20 to 25%.

Table 46

Values of Flow Coefficient S

	Type of scavenging	Rounded intake edges and smooth passage surface	Rough passage surface	Average value of S
Scavenged engines with symmetric timing	transverse	0·9	0·7	0·8
	reverse-flow	0·65	0·45	0·5
Scavenging with asymmetric timing and ample supercharging	transverse	0·5	0·35	0·4
	reverse-flow	0·35	0·2	0·25
	parallel-flow	0·5	0·35	0·4
Asymmetric timing with slight supercharging	transverse	0·65	0·55	0·6
	reverse-flow	0·4	0·3	0·35
	parallel-flow	0·65	0·55	0·6

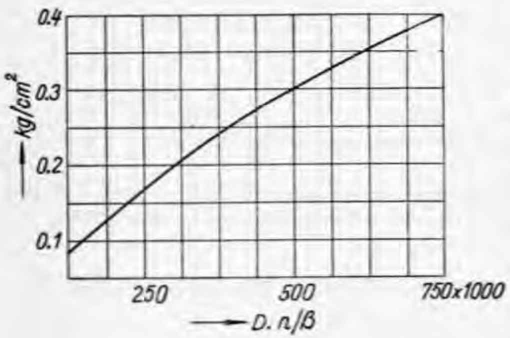


FIG. 341. Chart for the calculation of scavenging pressure.

Horizontal scale = values of $D \cdot n/\beta$ multiplied by 1,000. D = bore in mm, n = speed in rev/min, β = ratio of width of inlet ports b_i measured in a direction normal to flow to bore, $\beta = b_i/D$. Vertical scale = scavenging pressure in kg/cm².

The air flow passing through passages in the cylinder wall is subject to contraction so that the whole surface of the port is not utilized. The magnitude of this contraction is given by the flow coefficient S, which is dependent on port shape, the rounding of edges at the point of entry and on the quality of the surface etc., its magnitude being given in Table 46.

With the scavenging pressure known (from the diagram in Fig. 341) the mean entry velocity v_i can be determined, this being dependent on the ratio of scavenging pressure to cylinder pressure, on absolute temperature of the scavenging air T_1 and on the coefficient of flow S

$$v_i = S \frac{v_0 p_0}{\sqrt{R\,T_1}} \sqrt{2g \frac{k}{k-1} \left(\frac{p_1 + p_0}{p_0}\right)^{\frac{k-1}{k}}} \cdot \sqrt{1 - \left(\frac{p_0}{p_1 + p_0}\right)^{\frac{k-1}{k}}}$$

where R denotes the gas constant for air
p_1 ,, ,, scavenging pressure
p_0 ,, ,, pressure inside cylinder (atmospheric)
v_0 ,, ,, specific volume of air under normal conditions

The dependence of $\frac{v_i}{S}$ upon the pressure difference $\frac{p_1 + p_0}{p_0}$ is shown in the diagram of Fig. 342. The size of the ports depends on the specified scavenging pressure. The duration of port opening must increase with a decreasing specified scavenging pressure, for the port area depends on the duration of opening. A long duration of scavenging has an adverse effect on the effective piston travel during the expansion stroke and thereby also upon the engine output per litre. A higher scavenging pressure is, therefore, to be preferred. In order to supply this, a large amount of power must be consumed by the supercharger or blower. A certain scavenging pressure represents optimum value and if it is further increased, the gain in performance does not equal the rise in supercharger input.

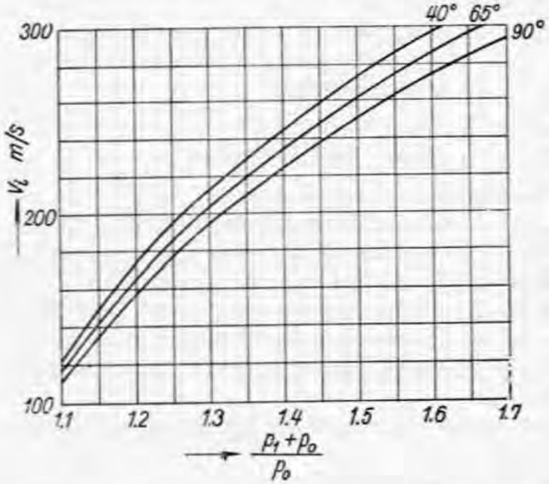


FIG. 342. Chart for the determination of the v_i/S ratio from the pressure ratio $(p_1 + p_0)/p_0$ for different temperatures of the scavenging air.

For scavenging pressures 0·20—0·35 kg/cm² a temperature of 65 °C may be chosen. For lower pressures lower temperatures are taken and vice versa.

If the precise data of the supercharger are unknown, an approximative formula may be used

$$p_k = p_1 \frac{\eta_p}{\eta_k}$$

where η_k denotes the charging efficiency = approx. 1·4 where η_p denotes the overall efficiency of the supercharger approx. 0·7 therefore

$$p_k = 2\, p_1$$

where p_1 denotes the charging pressure

p_k „ „ mean effective pressure of supercharger input

The dimensions of the exhaust ports are determined by the requirement of a reduction in pressure of the exhaust gases in the cylinder up to the instant of port opening to the pressure of scavenging. From this requirement the degree of exhaust port opening advance may be determined if the required size of exhaust ports at the instant of scavenge port opening is known. (Fig. 339.)

For an approximative advance calculation of the magnitude of this area, we may use the formula

$$A_m \,.\, \alpha = 0{\cdot}00013 \,.\, v_1 \,.\, n$$

where $A_m = h_m \,.\, L \,.\, b$ denotes the mean exposed area of ports during angle of advance α for the balance of pressures (cm² per degree)

v_1 „ „ displacement (cm³)

n „ „ engine speed in rev/min

This equation is valid if the following conditions are met:

1. At the instant of exhaust port opening, cylinder volume is 0·8 v_1. If the real volume differs from 0·8 v_1, the percentage by which it is below or in excess of 0·8 v_1 must be added to or subtracted from the result.

2. The coefficient of flow through exhaust ports is 0·422, which represents the mean value found between ports with sharp and rounded inlet edges.

3. The temperature at the beginning of exhaust is 675 °C (which applies to compression-ignition engines).

4. The ratio of exhaust gas pressure to scavenging pressure at the instant of exhaust port opening is 3 : 1.

5. Pressure in the exhaust pipe is atmospheric.

6. The polytropic coefficient of the expansion curve is 1·3.

With the size of the exhaust port area known (Fig. 339) then

$$A_e = h_m \,.\, \alpha = \frac{A_m \,.\, \alpha}{b_i \,.\, L}$$

The area A_e is given in the diagram of Fig. 339 by the curve of piston travel, the beginning of scavenging and the height of exhaust ports h_e. With piston travel, the beginning of scavenging and the magnitude of A_e given, the height of exhaust ports h_e and the angular advance of exhaust α can be found. If this area is sufficient for the reduction of exhaust gas pressure to scavenging pressure, then the remaining area of the exhaust ports is more than sufficient (with a symmetric timing).

The precise area of exhaust ports required for the drop of pressure before the transfer ports open cannot be calculated with any degree of precision and its correct choice must be verified by dynamometer tests. In order not to reduce the length of the expansion and compression strokes however, the exhaust ports should be as low as possible, this leading to the requirement of the fastest possible uncovering of the largest possible area by the piston.

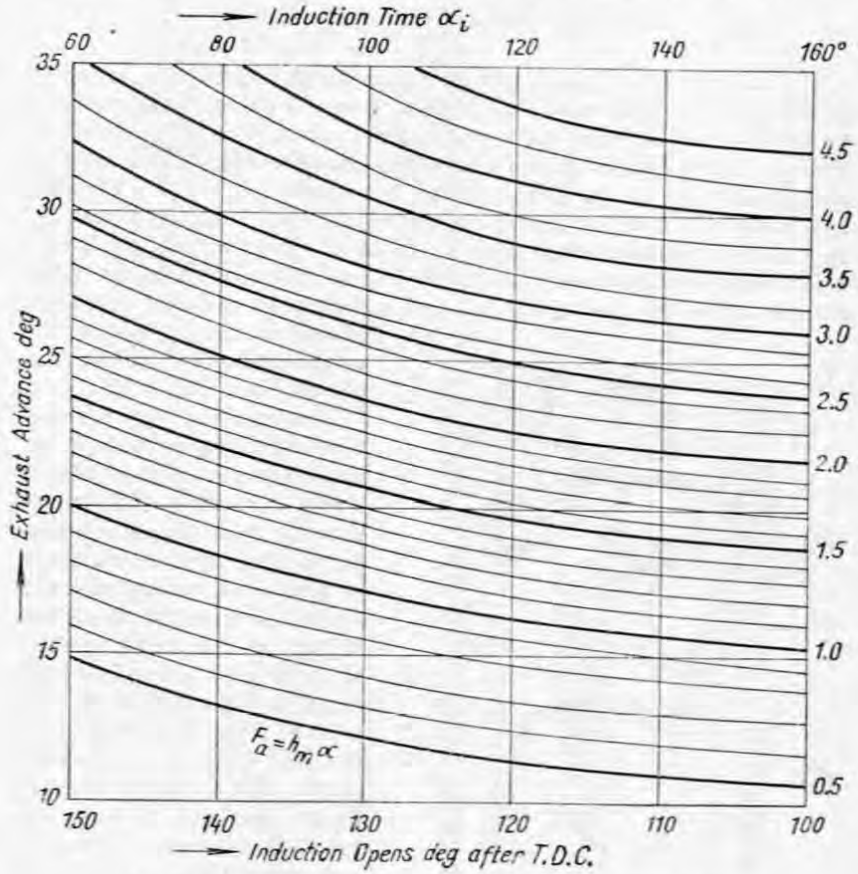


FIG. 343. Chart for the determination of the exhaust advance.

High velocity in the exhaust ports must be specified in order to minimize the angle of advance. The exhaust duct cross section must be of the best aerodynamic shape in order to reduce contraction. This promotes the rapid reduction of gas pressure within the cylinder and also assists scavenging by using the kinetic energy of the exhaust gas.

The magnitude of advance angle α for symmetric timing or for layouts in which the exhaust port is uncovered by the piston can be found in Fig. 343. For a symmetric timing geometric methods of calculation are more convenient. The rate of opening is set by poppet valve lift, by the motion of a sleeve or rotary valve or by the advance of the second piston.

3. Design details of two-stroke engines

Pistons of two-stroke engines generally pose difficult problems for the designer owing to the great heat stress load on them and owing to the necessity to prevent overheating in the vicinity of the piston rings. If the piston is used to uncover exhaust ports, its edges are heated to a high temperature by the hot exhaust gases and the piston rings are then prone to sticking. The Junkers works used a special continuous steel ring referred to as the "Feuerring" to protect compression rings from heat and this measure was effective in combating repeatedly stuck piston rings. In such a case, however, the piston must be of built-up construction, e.g. with a steel crown and light alloy skirt. Means must be provided to take up the clearance at various rates of expansion. (Fig. 344).

The function of such a protective ring during operation is not without interest. A check of such a ring at intervals of 50 operational hours is required by Junkers [64]; for which it is necessary to remove the exhaust pipe and make certain that the ring is free in its groove and check for gas seal along the whole circumference. If any fault is detected, a new one must be fitted. The useful life of the ring is up to 150 running hours. This is obviously closer maintenance and at shorter intervals than is required by a poppet valve of a four-stroke engine. Poppet valves in modern aircraft engines will operate up to 5000 hours reliably. This example may serve to indicate that simple design does not always imply greater simplicity in maintenance or greater reliability in service.

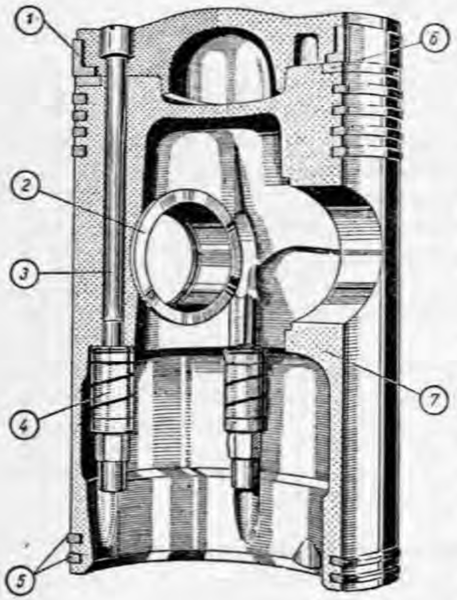


FIG. 344. Piston of the two-stroke Junkers aircraft oil engine.

1 — unsplit freely turnable ring, *2* — pressed-in gudgeon pin sleeve, *3* — bolts, *4* — springs, *5* — scraper rings, *6* — "Niresist" ring preventing wear of aluminium piston skirt, *7* — light metal piston skirt.

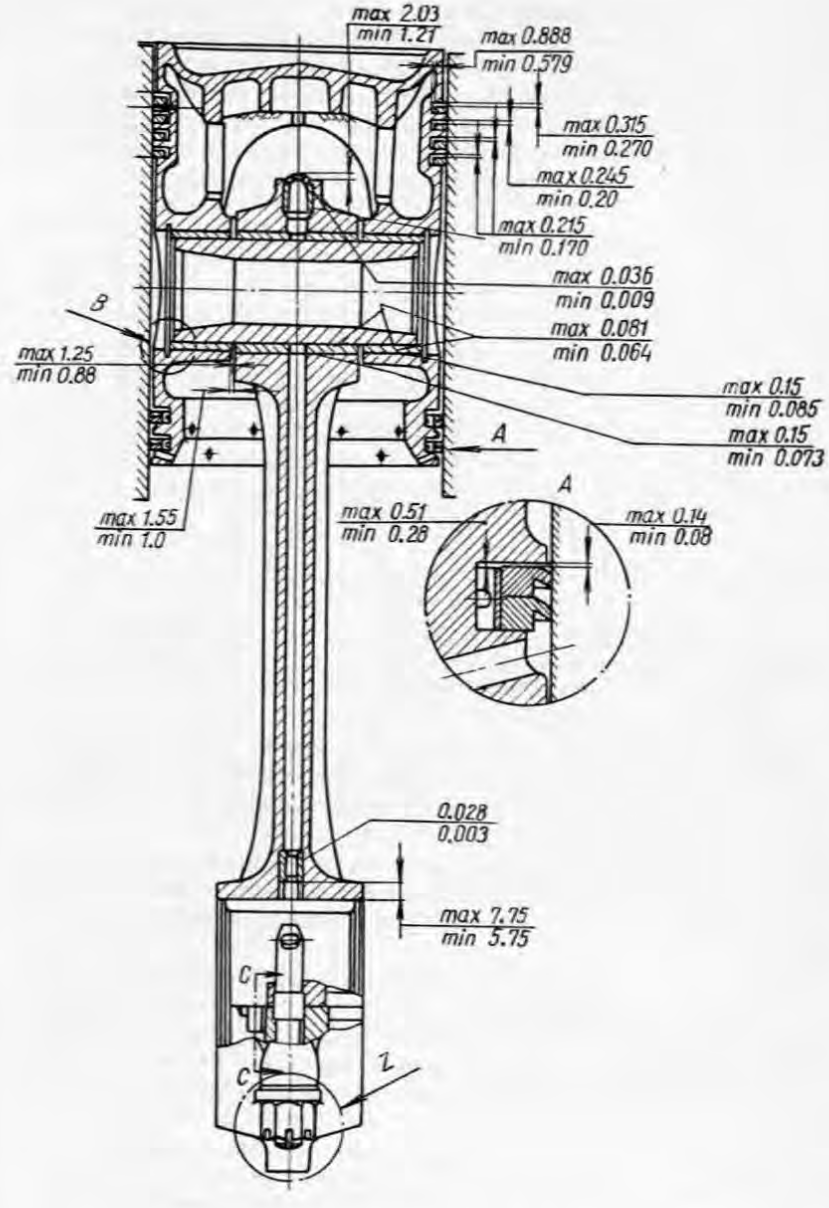


FIG. 345. Piston of the Soviet two-stroke JAZ 204 engine.

Another method of relieving the piston rings of their heat relies on the choice of a material of lower heat conductivity for the piston and removal of heat from the underside of the piston by oil. Such piston cooling is used, for example, on JAZ and GM engines. Oil is sprayed onto the piston from a jet in the small end. The piston is slotted between the crown and ring section in order to minimize the transfer of heat to piston rings which would lead to ring overheating. An example of such a piston is shown in Fig. 345.

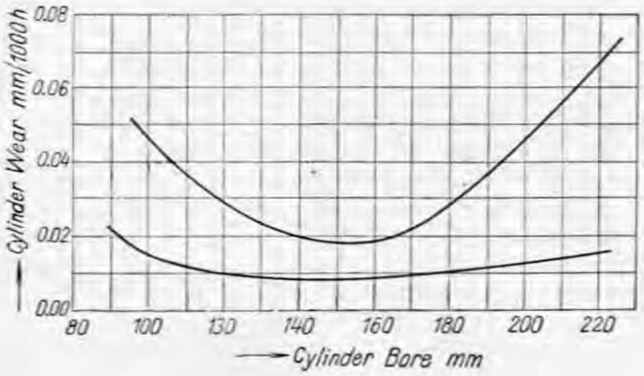


FIG. 346. Cylinder wear of two-stroke and four-stroke engines.
Upper curve — measurements on nine two-stroke oil engines, lower curve — measurements on nine four-stroke oil engines.

The heat removed by the oil must be transferred to air by some suitable means, such as an oil-to-water heat exchanger and a water-air radiator. In larger GM engines other arrangements of piston cooling are used, such as double-crown pistons, fixed jets, etc.

Good results have been attained by the use of a small end bearing which allows for the rotation of the piston. As the piston rotates about its axis, heat is distributed about its circumference more uniformly and wear is considerably reduced.

The height of a two-stroke engine piston must exceed its stroke, otherwise ports would open to the crankcase. Short-stroke design must therefore be employed if piston weight is to be kept low. The weight of a piston in a two-stroke engine is not as critical a value as in the four-stroke engine, as its inertia force is always working against gas pressure at t.d.c. thus relieving the bearing load.

Piston rings, especially scraper rings for use in two-stroke engines, must be manufactured with utmost precision. The bottom scraper rings must prevent the ingress of oil into transfer ports, but simultaneously provision has to be made for efficient lubrication of compression rings in order to prevent a high rate of wear. This is no small task. JAZ rings are shown in Fig. 345. Compression rings are grooved on their periphery and the porous carbon deposit in the groove forms an emergency reserve of oil.

Poor lubrication results in a greater rate of wear of two-stroke engine cylinders. Very thorough experiments with two-stroke and four-stroke engines of varying sizes have been conducted the results of which are shown in Fig. 346 [26]. The rate of wear of two-stroke engine cylinders, as will be noted, is much greater than that common in four-stroke engines.

Poor lubrication of the upper reaches of the cylinder may be improved by the use of a special reciprocating pressure pump which supplies precisely metered quantities of oil directly to the cylinder wall above the ports. The quantity of oil varies according to engine load. The Krauss-Maffei works have solved this problem by governing the rate of oil supply by the position of the injection pump regulator rod; this however, results in a more complicated lubricating system than that of the four-stroke engine.

The high rate of cylinder wear is evident in small engines, particularly in the vicinity of the port partitions. This is due to the combined adverse effects of high temperature and poor lubrication (the wiping away of the oil film) and of the high specific pressure of the ring on the partition or bridge. After wear occurs at these partitions, the ring begins to "breathe" and after a short time it breaks.

Cylinder wall apertures are not desirable if thin cylinder liners are used as the partition between the ports readily over-heats and distorts. Such openings are also inconvenient for the chromium plating of cylinder bores.

The gudgeon pin of a two-stroke engine is prone to seizure, being always loaded in one direction, for which reason the formation of an adequete oil film between the pin and its bush is impossible. Needle roller bearings or special grooved bushes are therefore desirable for the gudgeon pin.

The balance of a two-stroke engine, especially in the in-line arrangement, is worse than that of its four-stroke counterpart.

In view of the order of firing, the crankshaft is asymmetrical and primary and secondary couples (according to the number of cylinders) remain unbalanced. A special arrangement shown in Fig. 347 is employed to balance primary couples.

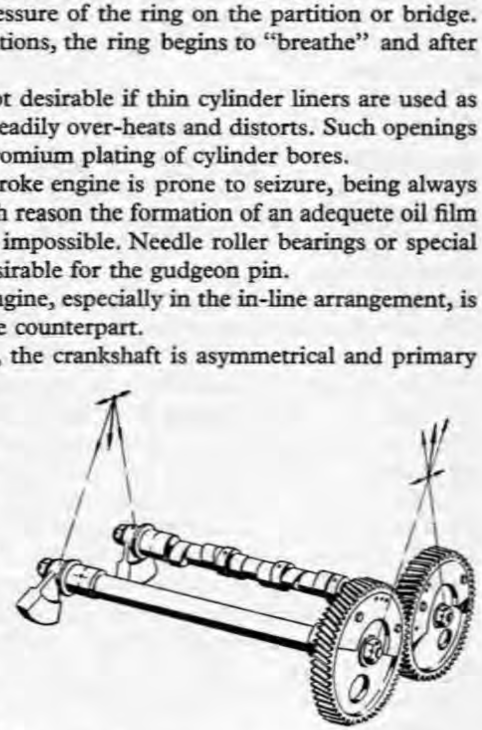


FIG. 347. Arrangement for balancing primary couples in a two-stroke engine.

The 90 °V arrangement of the engine, as well as the opposed piston motion engine, are advantageous for engine balancing.

The injection equipment of two-stroke engines works at higher loads owing to the double number of injections required. Normal type injection equipment is therefore unsuitable and special designs are employed. GM engines are fitted with separate injection pumps in all injector holders, thus eliminating the need for high pressure pipes from injection pump to nozzle. The requirements on the injection pump in the two-stroke engine are extreme and the design of the pump itself is difficult.

4. Air-cooled two-stroke engines

Several difficult problems have to be solved in the design of the air cooled two-stroke engine. If the exhaust ports are to be located in the cylinder wall, the cooling of the partitions will certainly require much attention by the designer. The partitions are narrow and cooling fins of sufficient area which will not hinder air flow are difficult to arrange. Exhaust valves are more convenient, as their surroundings can be extensively finned and experience is at hand from four-stroke practice. The transfer ports in the cylinder wall are not subjected to high temperature and internal cooling through the passing charge is sufficient. With the use of exhaust valves, difficulties with piston crown overheating, experienced when the piston opens exhaust ports, are avoided.

Fig. 348 shows a Stihl two-stroke compression-ignition engine with valves, which is used as a prime mover for tractors. This engine works on the parallel-flow principle. Air enters the crankcase by a tongue-like non-return valve and is passed into the cylinder through ports arranged about the circumference of the bore, which are opened by piston movement. The burnt gases leave via an overhead poppet exhaust valve actuated from a crankshaft mounted cam. The injection pump cam is also on the crankshaft. The injector is laterally disposed with the glow plug arranged opposite to facilitate starting. Cooling air is provided by an axial fan driven from the crankshaft by a V belt. Recent types of this engine employ a radial fan formed in the flywheel.

With a bore of 85 mm and a stroke of 94 mm, the engine produces 12 b.h.p. at 2200 rev/min. The cylinder is of aluminium with a pressed-in cast iron liner. The cylinder head and crankcase are also of light alloy. Starting the engine by cranking is facilitated by throttling the air flow into the crankcase and by a heater plug in the compression space. Oil fed to the big end bearing is metered and passes through a collector ring into the hollow crankshaft journal, whence it is forced into the roller bearing by centrifugal force. The cylinder wall is lubricated by oil delivered through the metering device.

Fig. 349 shows the Orenstein and Koppel two-stroke engine. This two-cylinder engine has a 110 mm bore and 135 mm stroke representing a displacement of 2·56 litres. Engine performance is 36 b.h.p. at 1700 rev/min. Highest speed for car service amounts to 2000 rev/min. A Roots blower is used for scavenging.

Cylinder spacing equals 1·66 D. The combustion chamber is in the cylinder head so that the piston is thermally relieved.

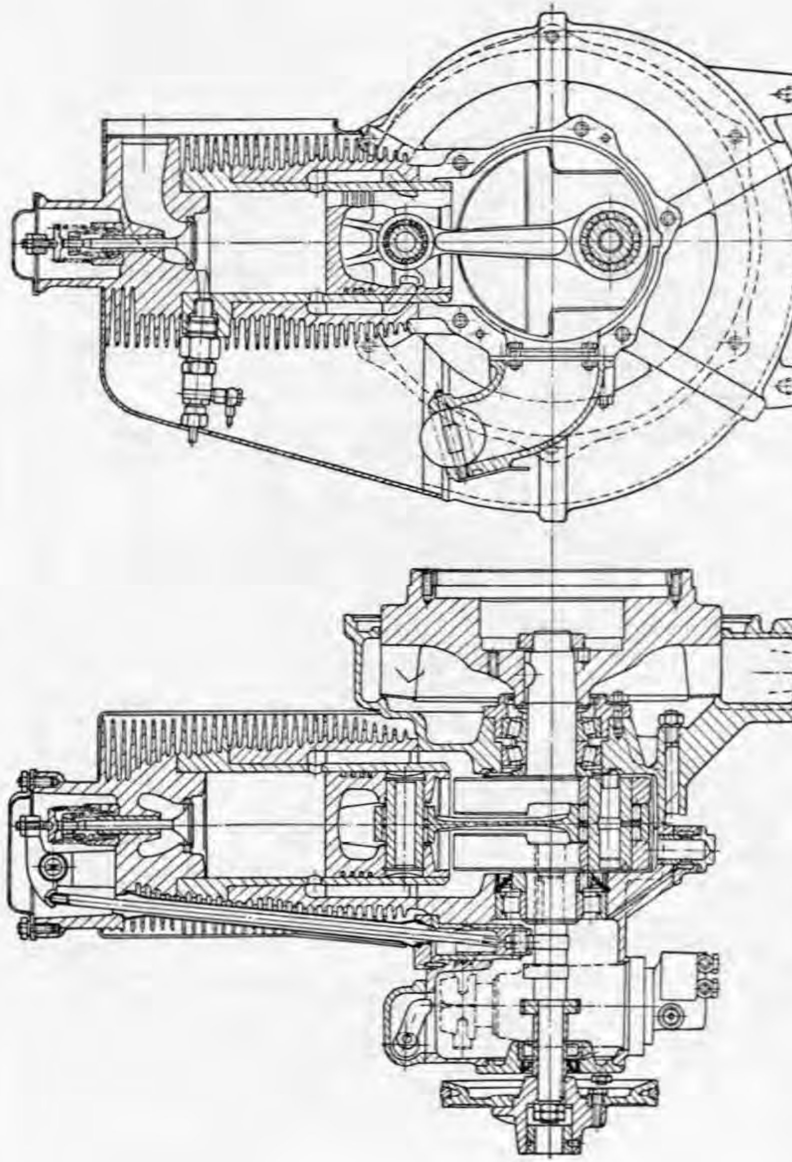


FIG. 348. Transverse and longitudinal section of the two-stroke Stihl engine.

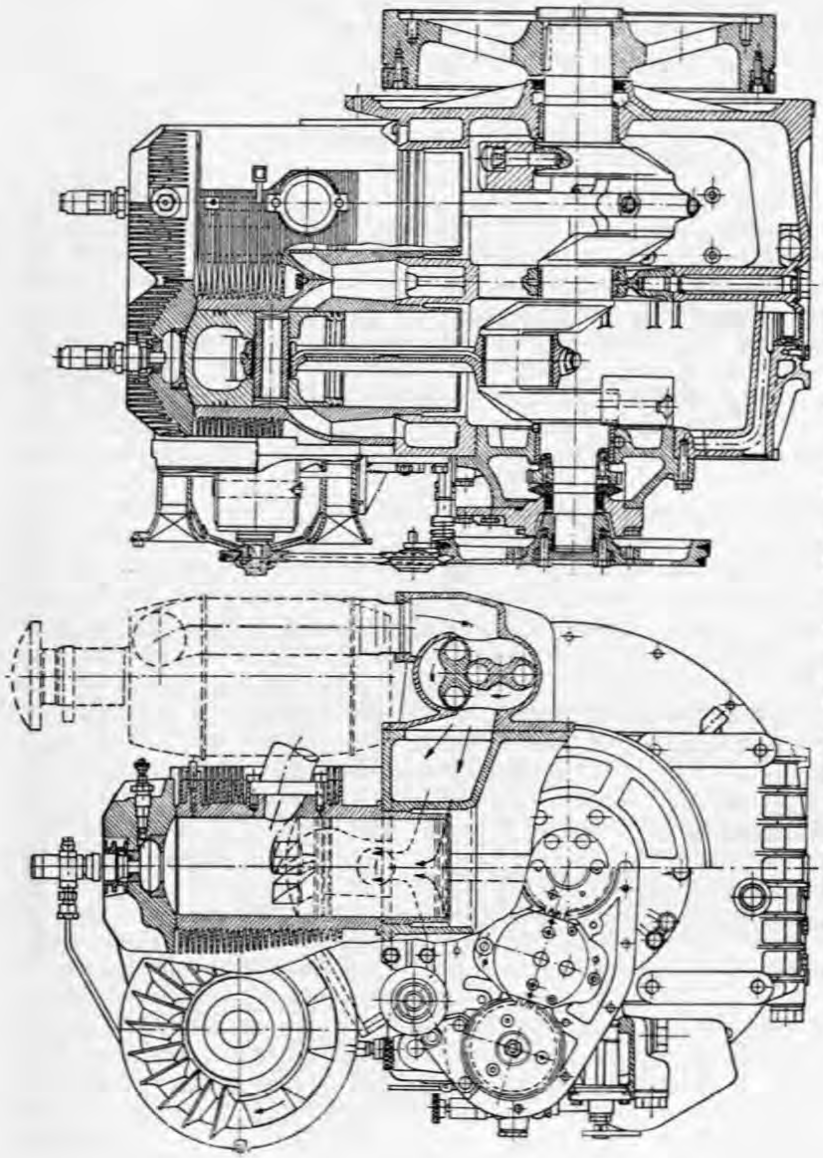


FIG. 349. The Orenstein and Koppel two-stroke engine.

An example of an air-cooled two-stroke aircraft engine is shown by the ZOD type of the Zbrojovka Brno. (Figs. 350 and 351.) Good results were attained with this engine, but its development was not completed. The typical features of a two-stroke engine are shown by the longitudinal section. The massively finned head is screwed to the steel cylinder by a trapezoid thread and inlet ports are arranged in the cylinder wall. The engine is of short-stroke

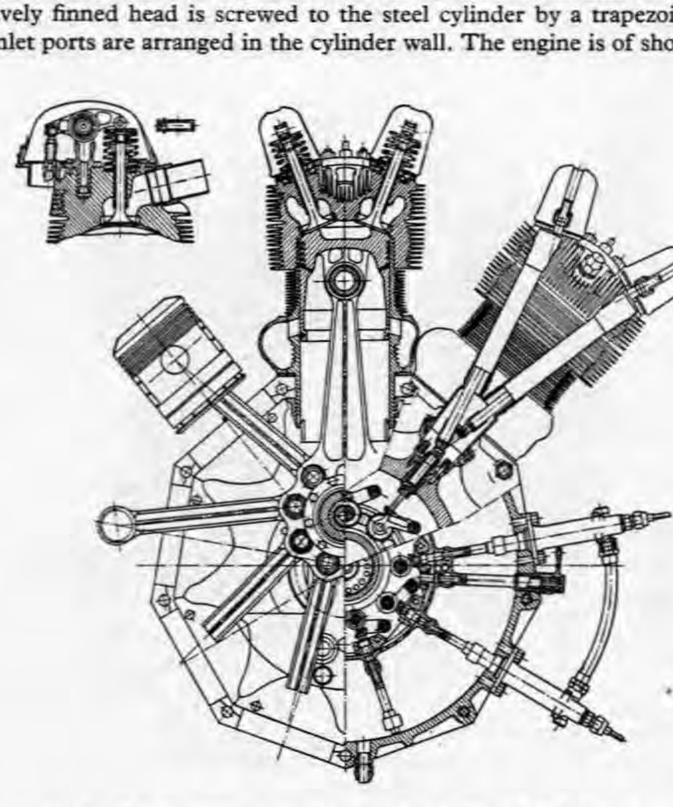


FIG. 350. Transverse section of the two-stroke ZOD engine.

design with a bore of 120 mm and stroke of 130 mm. The pistons are therefore not unduly long, although they cover transfer ports when at t.d.c. The bottom edge of the piston is fitted with a scraper ring preventing the entry of oil into the transfer ports. Compression rings are prevented from turning in their grooves by pegs locating their gaps, in order to prevent the end being trapped in ports with resultant fracture.

The gudgeon pin bosses – sensitive as they are in a two-stroke engine – are pressure lubricated. The bottom part of the cylinder is shrouded by scavenging air brought under pressure from a centrifugal supercharger. The cylinders are attached to the crankcase by large ring nuts.

Each cylinder is provided with its own injection pump, keeping the pressure line to the injectors very short. The amount injected is controlled in the usual way by rotation of the pump plunger. The nozzle has three holes spreading fuel in fan shape into one half of the combustion chamber in the direction of the swirl.

Air swirl is produced by the tangential arrangement of air entry ducts.

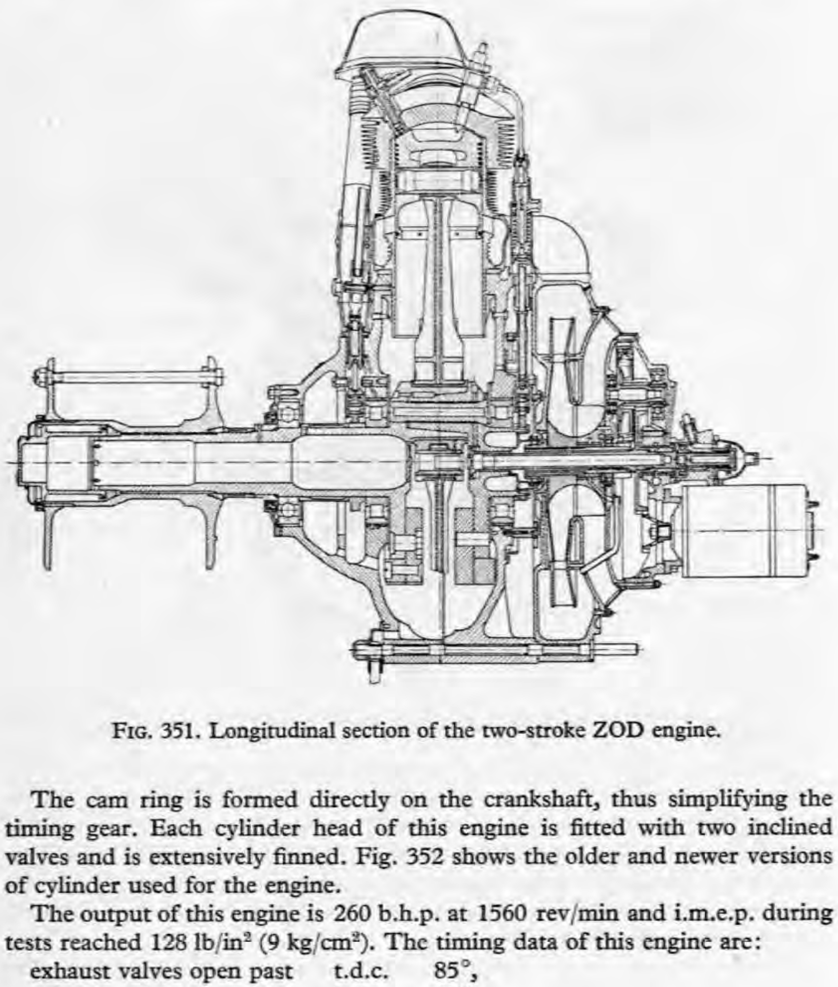


FIG. 351. Longitudinal section of the two-stroke ZOD engine.

The cam ring is formed directly on the crankshaft, thus simplifying the timing gear. Each cylinder head of this engine is fitted with two inclined valves and is extensively finned. Fig. 352 shows the older and newer versions of cylinder used for the engine.

The output of this engine is 260 b.h.p. at 1560 rev/min and i.m.e.p. during tests reached 128 lb/in^2 (9 kg/cm^2). The timing data of this engine are:

exhaust valves open past	t.d.c.	85°,
exhaust valves close past	b.d.c.	60°,
inlet ports open before	b.d.c.	60°,
inlet ports open past	b.d.c.	60°.

Engine starting was effected by air pressure admitted through non-return valves which are apparent in the cylinder head in the longitudinal section of the engine.

A newly designed vehicle engine of similar design is shown in Fig. 353. The bore of this engine is 140 mm, stroke 130 mm. The piston is therefore short and light and has a double crown cooled by oil fed through the connecting

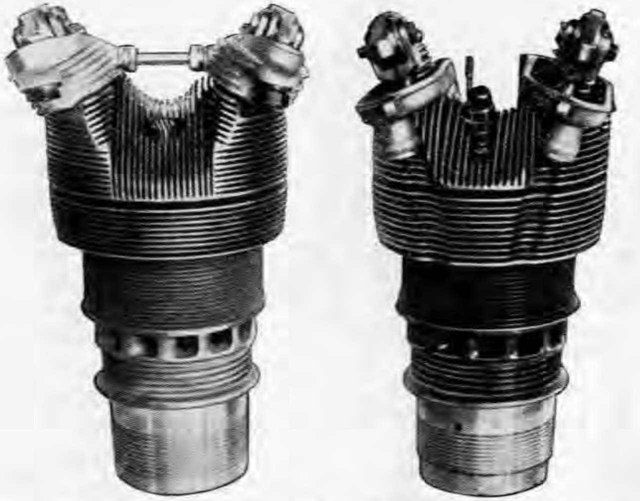


FIG. 352. Old and new design of the cylinder of the ZOD engine.

rod and returned by an overflow tube. The camshaft is located high in order to keep the valve actuating gear light. Scavenging ports are formed in the cylinder wall. The head is attached to the crankcase by six long bolts and the cylinder is clamped between the head and crankcase.

A two-stroke air-cooled engine designed for the Volkswagen (Fig. 354) is also interesting. This is a flat twin with transverse scavenging and with a flat topped piston. Scavenging air is supplied by a Roots type blower mounted in the upper part of the crankcase. The cylinder head, which comprises a squish chamber, is detachable. With bore 85 mm, stroke 100 mm, output is expected to be in the region of 24·7 b.h.p. at 2400 rev/min and weight is estimated at 220 lb (100 kg). Each cylinder is fitted with a separate injection pump.

Another interesting two-stroke petrol engine with opposing motion pistons, intended for light aircraft, is shown in Fig. 355. It is a six-piston three cylinder layout with dual ignition. The gear linking both crankshafts also serves as a reduction gear. The simplicity of design and the widespread use of die castings in this engine will be noted from the transverse section shown in Fig. 356. Scavenging air is supplied by a centrifugal supercharger driven by mechanical means from the rear of the crankshaft. This Culloch engine is still under development. Difficulties may be expected with piston cooling, although it should be possible to overcome them in view of the small dimensions of the cylinders. The engine output is planned to be about 120 b.h.p.

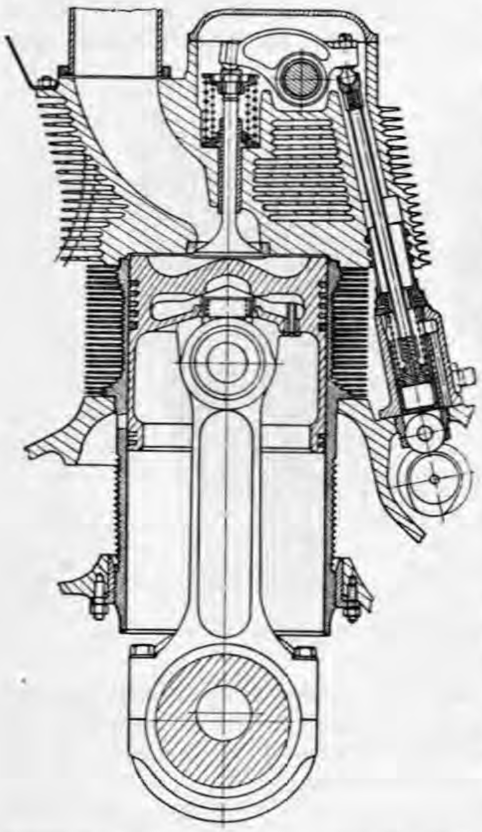
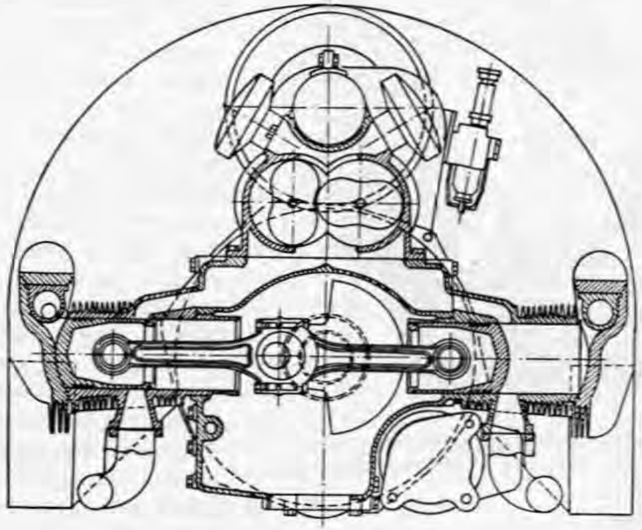


FIG. 353. Design of an air-cooled two-stroke engine of 140 mm bore.

Another interesting illustration is afforded by an air-cooled two-stroke engine intended for auxiliary drive in aircraft. The interesting point about this oil engine is the fact that it may be also run on aviation spirit. When this single cylinder is coupled with a three-stage reciprocating supercharger, air for cylinder charging is taken from the first stage. The axial fan is driven by an exhaust driven turbine. During development work on the engine, it became apparent that an exhaust gas ejector would be more suitable [12].

The scavenging of this engine is worth noting. Round the whole circumference of the cylinder at b.d.c. there are scavenging ports and above them exhaust ports. Such an arrangement has the great advantage of effective piston cooling by the incoming air as soon as the exhaust ports are opened.
Overheating troubles are thus obviated. First tests were conducted in transparent models with coloured water, keeping the Reynolds number at its original value. Ducts arranged in a plane at right angles to the cylinder proved quite unsuitable and the result of a very large number of experiments (with 270 various arrangements) is the arrangement with scavenging and exhaust ports inclined upward and with a conical piston crown. The section of such a cylinder is shown in Fig. 357. According to data gained from literature [57] a mean effective pressure of 104 lb/in^2 (7·35 kg/cm^2) at 2400 rev/min was achieved in the development stage.

One of the most interesting instances of air-cooled two-stroke engine design is apparent in the case of the "Orion" engine. [71] The development of this engine was occasioned by the requirement for an air-cooled compression-



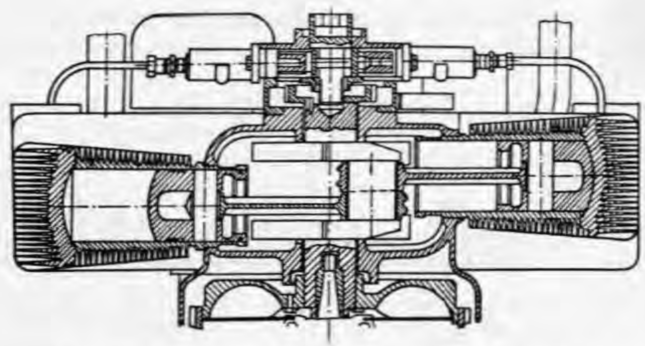


FIG. 354. Design of a two-stroke engine for the VW car.

ignition engine of small overall dimensions and low weight with a power output of 600 b.h.p., which was to be used as the prime mover of a tank for the United States Army. Work on the project began by a thorough study of various combinations of gas turbines with gas generators and piston engines. A volume of 1·7 m^3 was available for the engine, which of necessity had to be very compact with a specific volume of 3,500 c.c./b.h.p. Under such conditions it appeared advantageous to combine the merits of air cooling with the two-stroke cycle.

FIG. 355. Two-stroke three-cylinder engine with opposing motion pistons and reduction gear.

It was decided to employ the two-stroke cycle with continuous flow scavenging with horizontally opposed counteracting pistons. In order to comply with the requirement of a low fuel consumption not exceeding 224 g/b.h.p. per hour, great importance was attached to thermal economy. To this end special attention was given to heat losses through exhaust gases and cooling. The heat content of exhaust gases is nowadays normally utilized for turbocharging. The difficulties involved in the use of turbochargers for two-stroke engines have already been mentioned. In the case of this engine, a layout was employed using a piston engine driven centrifugal supercharger from which compressed gas was supplied to the gas turbine which provided the whole motive power without any direct connection

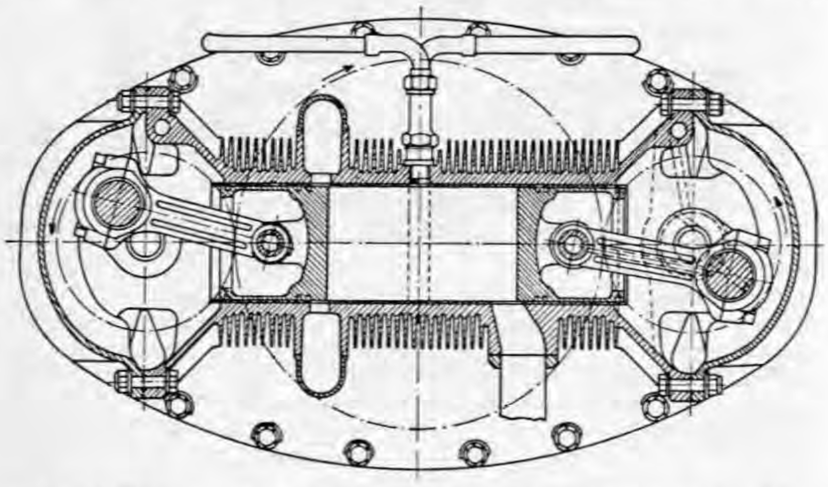


FIG. 356. Transverse section of a two-stroke three-cylinder engine (Culloch).

with the piston engine. Owing to this arrangement exceptional flexibility of operation was obtained, for the pneumatic transmission can be compared to an orthodox hydraulic drive. With regard to the function of the layout, its resemblance to that of a combination of a free piston generator and a gas turbine will be apparent, while it will be equally obvious that the difficulties of starting a two-stroke engine and of supercharging it under partial load have been overcome.

An interesting and original feature of this engine was the effort made to utilize heat loss through cooling to increase the thermal efficiency of the engine. Heat abducted through coolants has, in the case of piston engines in the past, always been regarded as totally lost. Previously it had been found impossible to return at least some of the heat thus lost into the operational cycle – excepting perhaps the slight gain resulting from shaping aircraft radiators as Venturi tubes, in the mid-section of which heat is transferred to air. With such an arrangement the energy regained is just sufficient to balance losses through aerodynamic friction in the core of the radiator.

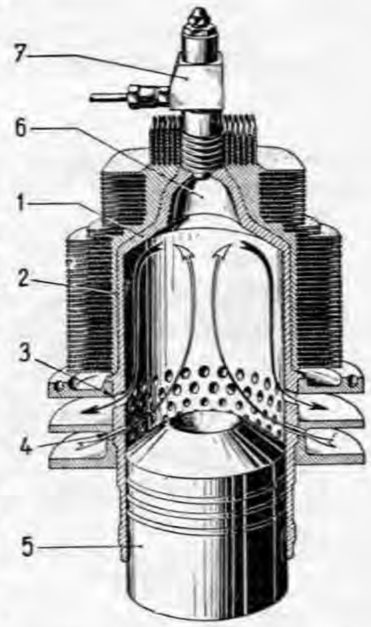


FIG. 357. Cylinder of the air-cooled Barnes and Reinecke engine with reverse flow scavenging.

1 – cylinder, *2* – aluminium fins, *3* – exhaust ports, *4* – scavenging ports, *5* – piston, *6* – combustion chamber, *7* – injection nozzle.

In the case of the "Orion" engine it was proposed to utilize the heat energy of cooling air by passing it through the turbine (Fig. 358). Air coming from the supercharger was split into two streams, one of which passed through the cylinders, and the other over the cooling fins. Owing to lack of space and the requirement of intense cooling, fine pitch finning had to be employed and with a reduction of resistance of the air flowing through the cylinders, this condition could be met. It was found that the engine could be cooled by air with a temperature of 115 °C supplied at a pressure of 2 kg/cm^2.

Development work on the "Orion" engine proved very difficult and interesting and let us at least quote some of the more interesting results obtained. Before the construction of a six-cylinder prototype engine was embarked upon, three experimental single-cylinder units in all had been built. Two-directional cooling had been proposed originally with the fresh stream of cooling air first of all reaching the middle portion of the cylinder barrel, from which it flowed between the fins towards both ends of the cylinder. Later on it became apparent that uni-directional flow could be employed and thus the cooling air stream reached the cylinder barrel in the region of the transfer ports and left the finning near the exhaust ports. The temperature of the exhaust port partitions reached as much as 260 °C. Various means of cooling these partitions were tried, but after combustion had been improved and the cylinder bore diameter increased, they were no longer neccessary.

The transfer ports were drilled in the original cylinder in such a manner as to cause double turbulence. This, as tests proved, had an unfavourable effect

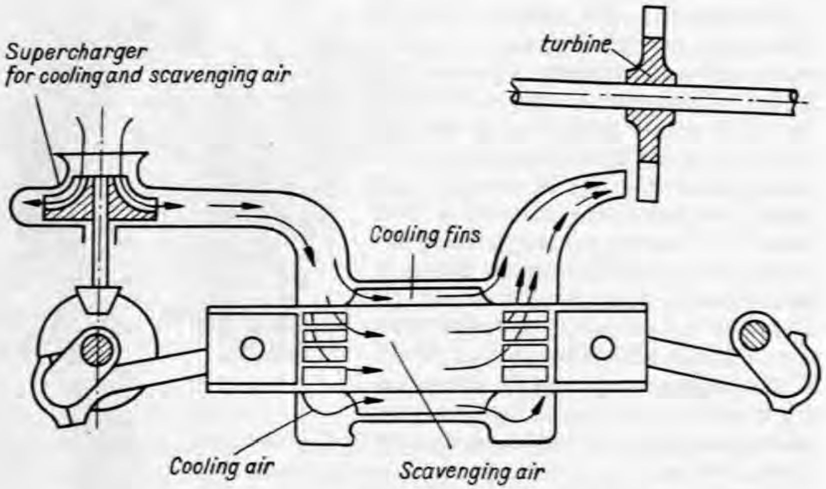


FIG. 358. Arrangement of air supply in a two-stroke aircraft engine.

upon combustion in addition to causing excessive resistance in the transfer passages. The layout was therefore abandoned in favour of tangentially arranged ports, all of which faced in the same direction. Originally a three-piece cylinder liner had been employed. The scavenge and exhaust sides of the liner were of steel with a cast-on aluminium outer sleeve, in which there were machined fins of 3·7 mm pitch and 1·2 mm thickness. Both cylinders were pressed into a central piece of berylium-bronze of high heat resistance and high thermal conductivity. The fins of the mid-sections were of 2·5 mm pitch and 1 mm thickness, their height in both instances being about 25 mm. After assembly under pressure, all three parts were additionally bolted together (Fig. 359). The cylinder bores were provided with a layer of hard chromium plating in order to minimize wear.

In order to reduce the pressure drop through the cylinder, the passages were increased in length and the bore enlarged from 108 mm to 120 mm. This was accompanied by the introduction of uni-directional cooling and the employment of a single-piece liner. The mid-portion of the liner was provided

with a cast-on outer sleeve of aluminium, in which fins were machined in 40 batches of 4 grooves of 1·5 mm width. Fins of 1 mm thickness were of 2·5 mm pitch and the height of the fins was 38·5 mm (Fig. 360). The air flow resistance over the finning amounted to about 700 mm w.h. The air flow

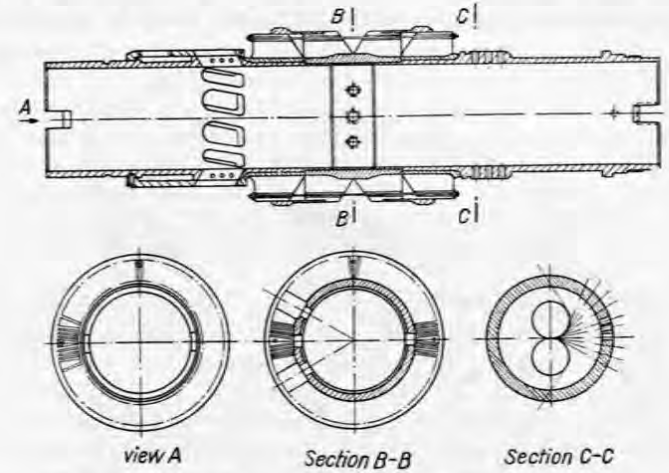


FIG. 359. Original three-piece design of the Orion cylinder.

resistance in the cylinder during scavenging at a rate of 0·24 kg of air per second amounted to 914 mm w.g. in the case of a pear-shaped piston, and about 560 mm w.g. in the case of a flat-top piston

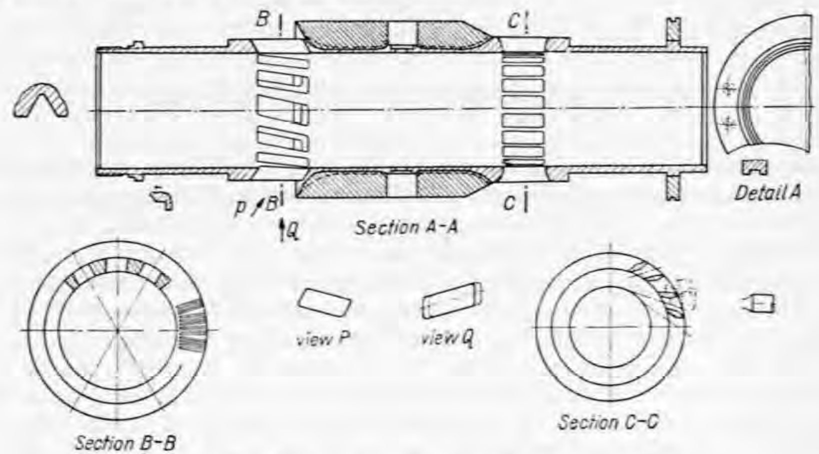


FIG. 360. The final one-piece design of the Orion cylinder.

Much development was also devoted to the pistons of the "Orion" engine. The original piston was provided with oil cooling for the piston crown underside. Oil was sprayed to the oil spray chamber on the underside of the piston crown where it formed a rapid stream which abducted much heat. The amount of heat removed in this manner from the pistons amounted to more than 4 per cent of the total thermal content of the fuel. It will be realized that the

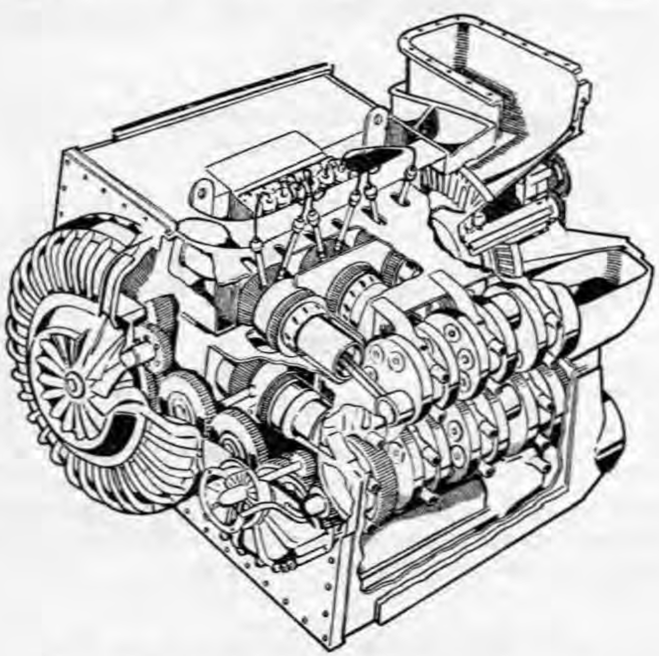


FIG. 361. Sectional view of the Orion two-stroke engine.

heat abducted by air cooling from the surface of the cylinder barrels was not lost in this engine layout, but was utilized in the turbine. Thus the major heat loss was through oil cooling of the pistons, which was undesirable. This led to experimental work on pistons not cooled by oil. The evolution of pistons of this type was also of great interest, but cannot be dealt with in detail in this publication. There were difficulties with the method of attachment of the crown of the piston which was no longer cooled, piston rings stuck in their grooves, etc. The piston, attractive as it had been as a proposition, was hard to produce and under full load it lasted for 5 hours running at the most.

As would be expected, there were also difficulties with oil control (rings) and with the lubrication of the gudgeon pin. Finally it was possible to reduce oil consumption to an acceptable level by modifications of the scraper rings.

Gudgeon pin trouble was obviated by pressure lubrication. Difficulty was also encountered in the design of injection equipment and in the end an accumulator-type system proved satisfactory.

The original idea of using the same supercharger for scavenging and cooling air had to be abandoned, for it was impossible to obtain an equal flow resistance in the ports and cylinder as in the finning, and also owing to the necessity of thorough filtering of induction air. The amount of air required for cooling was 3·5 times that needed for engine breathing and any efficient air cleaner able to cope with such a rate of flow would be far too bulky. The pressure ratio for the induction supercharger was 2·25, that of the cooling blower 2·06. The general arrangement and layout of the "Orion" engine is apparent in Fig. 361. Banks of three cylinders each are superimposed and four geared crankshafts are employed. The supercharging and cooling blowers are arranged in the forward section of the unit, the propulsive turbine is in the rear of the unit.

The development of this interesting engine was not completed, for shortly after the early tests of a prototype engine all development work on the design was discontinued. The reason for this has not been published, but the decision probably was based on the fact that by that time the development of the air-cooled Continental compression-ignition engine of 750 b.h.p. output and approximately the same dimensions had been concluded (see Fig. 393). Despite the fact that the development of the "Orion" engine failed to reach the final stage, it admittedly pioneered a new approach to air-cooled two-stroke engine design and proved the feasibility of the proposed layout on an actual engine.

No mention is made in this section of small air-cooled two-stroke petrol engines of up to 350 cm^3 displacement, which are used with great success in motor cycles. Up to this engine displacement efficient cooling is possible. In recent years reverse flow scavenging in conjunction with flat topped pistons is becoming almost universal in motor cycle engines. The flat piston receives the least amount of heat from the hot gases and is the lightest. Many flat piston scavenging systems giving good results are in use today.

The two-stroke cycle is favoured for small cylinder displacements as there are no cooling problems, the engine is simple (with crankcase actuated scavenging) and the torque curve favourable. Large air-cooled compression-ignition engines however must still undergo extensive development.

CHAPTER XIX

SOME EXAMPLES OF ENGINES PRODUCED

The study of fully developed engine types is a valuable aid to the designer. The development of a newly designed piston engine takes from three to four years' time and this period cannot be reduced unless the engine be derived from a fully developed and proven type. The development of a new type is very costly and before work on such a project is undertaken, suitable conditions must be created to warrant successful attainment of the object in mind.

To illustrate this, the development of the 28-cylinder radial Pratt and Whitney engine may be mentioned. Work on this engine took some 6 years, during which 22 000 test inghours were completed on 23 experimental engines. The progress of the work undertaken over such a long period is shown in Fig. 362. Beside the tests mentioned, experimental work was also carried out on a unit-cylinder test engine. These tests accounted for a further 40 000 hours. Further tests were undertaken over 2 000 hours with a single-row radial experimental engine. The development of the engine in question cost a sum total of 25 million U.S. dollars and this sum does not include the expenditure for developing the 18-cylinder engine, the well proven and highly perfected cylinders of which were used in the larger engine. The aim of the development work mentioned was "only" to group existing cylinders into a larger unit.

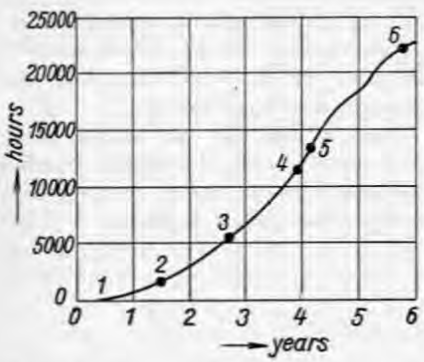


FIG. 362. Time schedule for the development of the Pratt and Whitney aircraft engine in hours of prototype tests.

1 — first run of the engine, *2* — first flight, *3* — production of the first prototype engine, *4* — performance test 150 hours, *5* — first series engine, *6* — performance test 150 hours with 17 % overload.

Such sums invested in development work cannot be recovered in small scale production of engines, as the cost per unit would be excessive. Large aircraft engines were therefore developed almost entirely during the war period, when considerations of cost were of secondary importance. The development costs

of engines for everyday use must however be calculated in the price of every unit. Designers are to be warned most seriously not to introduce new features of design just for the sake of novelty. A large number of able designers have in the past devoted their attention to the development of the reciprocating engine and in the majority of designs, features which have been highly perfected in production are used. Before deciding to adopt a new design, the manufacturer must investigate in detail and consider objectively the pros and cons of the new design. In doing so, the type of production machinery with which the factory is equipped must be borne in mind together with the experience of the design and engineering staff and the standard of specialization of skilled labour.

Much valuable experience for development may be gained from the study of perfected designs. Several interesting and well proven designs are therefore described. They may serve as useful reference for designers.

A. PETROL ENGINES

1. The A.B.C. Engine

This highly interesting design is a flat four-cylinder type with two-throw crankshaft. Such an arrangement results in unbalanced primary couples. To balance them, use is made of a special arrangement entailing the use of two

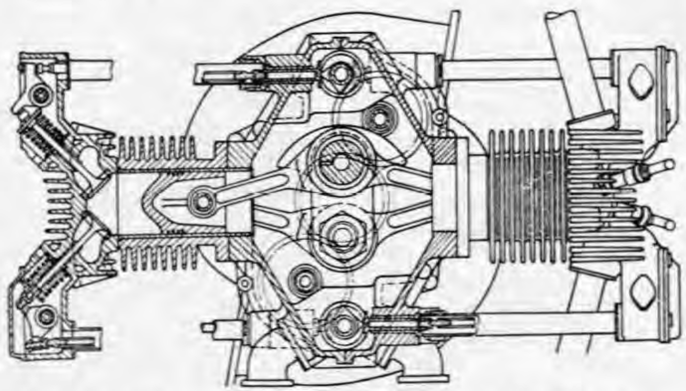


FIG. 363. Transverse section of the four-cylinder A.B.C. engine.

rotating masses (see section on engine balancing). These masses are on two additional shafts which are also used to transmit the drive to the camshafts. They are driven from the front end of the crankshaft through a gear drive and at their other extremities a 2 : 1 gear drives the camshafts. By balancing primary couples of reciprocating masses, perfect engine balance is obtained.

The bore of this engine is 63·5 mm, stroke 76·2 mm, the displacement of the four-cylinder engine is therefore 960 cm³. Maximum performance is 30·5 b.h.p. at 4 000 rev/min. Both crankcase halves, the cylinder barrels and heads are of electron metal castings. Cast iron cylinder liners are used and the connecting rods are of Duralumin. In order to make it possible to locate the cylinders in directly opposing pairs, connecting rods are forked and their unsplit big-end eyes bear directly on floating bushes on the crankpins of the built-up crankshaft. The gudgeon pin is located directly in the little end eye, without bushing, of the Duralumin connecting rod. The valve gear incorporating two camshafts is also of interest. (Fig. 363.)

The weight of the engine without dynamo is 182 lb (70 kg,) which means a power weight: ratio of 2·3 b.h.p. per kg. A radial fan at the front end of the crankshaft supplies the cooling air. The engine is used as an auxiliary unit for use in aircraft. With a performance of 25 b.h.p. at 4 000 rev/min, the temperature of the cylinder head in the vicinity of the sparking plug is 180 C° and that of the tips of the cooling fins is 85 °C.

2. The BMW Engine

The BMW concern has entered the market with a rear-engined light car with an engine of 585 cm³ displacement. This is a flat twin derived from the well known motor cycle engine, the performance of which has been reduced in the interest of longer engine life and a more favourable torque curve from

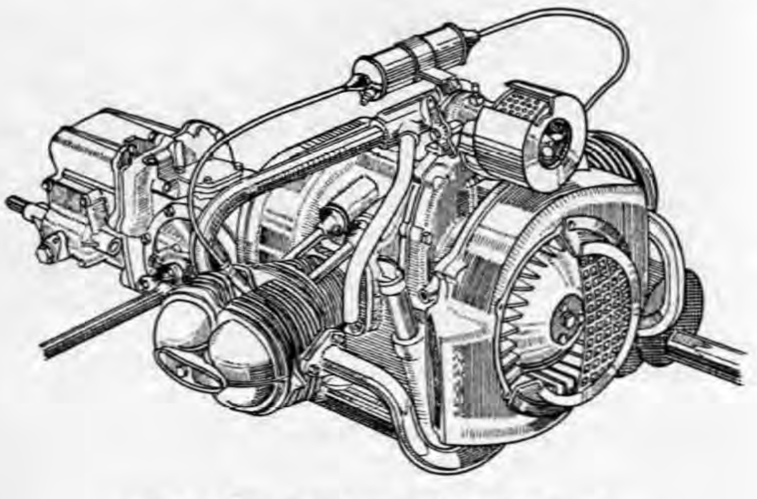


FIG. 354. Two-cylinder air-cooled BMW engine.

28 b.h.p. to 19.5 b.h.p. A radial fan fitted directly to the crankshaft is used for cooling air delivery. The cooling air is directed to the cylinders in the direction of the crankshaft axis, cool air being supplied to the exhaust valves. Cooling is regulated by two air-flaps controlled by a single thermostat. The single horizontal-draught Solex 28 FZP carburettor incorporates an acceleration pump and a starter jet.

With a bore of 74 mm and stroke of 68 mm and a standard compression ratio of 6.5 : 1, the maximum torque is 29 ft-lb (4 mkg) at 2 500 rev/min. The camshaft is located above the crankshaft and the overhead valve layout of the hemispherical head incorporates pushrods and rockers. An electric starter is used and this also serves as a dynamo of 130 W output. The engine is housed in the rear extremity of the vehicle and its general arrangement is apparent in Fig. 364.

3. The BSA Engine

An example of a parallel twin engine for motor cycle application is shown in Fig. 365. The bore of this engine is 70 mm, stroke 84 mm, displacement 646 cm³. The valves are actuated from a camshaft housed in the crankcase. The details of the valve drive are obvious from the sectional drawings. The direction of flow of air between the cylinders should be noted. The flywheel

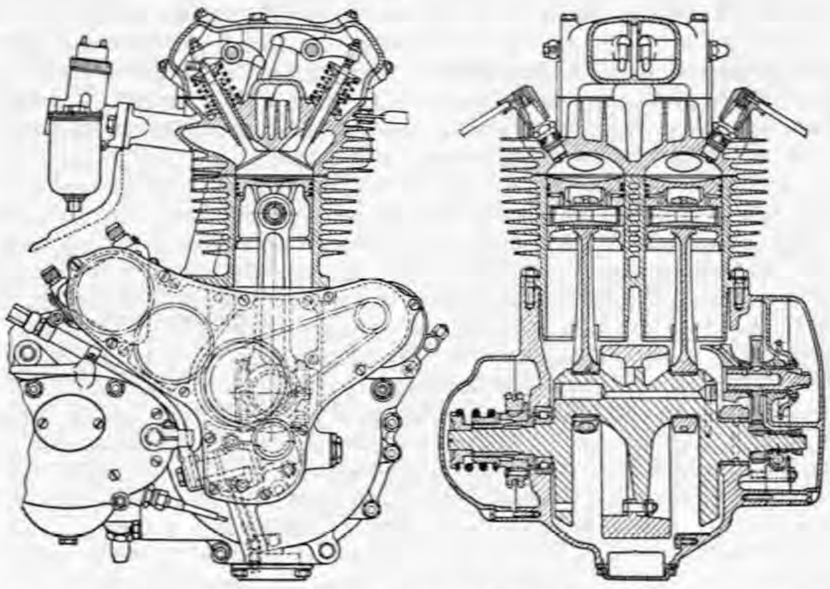


FIG. 365. Transverse and longitudinal sections of the BSA 646 cu. cm motor-cycle engine.

is flange fitted to the centre of the one-piece crankshaft, which revolves in two main bearings, one of which is a plain bearing. Pressure oil is brought to the crankshaft via this bearing.

Both the cylinders form an integral iron casting. As the cylinder axes are far apart for better cooling, it was found possible to place the flywheel between them.

4. The Chevrolet Corvair Engine

This is a six-cylinder air-cooled engine with 85·7 mm bore and 66 mm stroke. In order to reduce the overall height of this flat engine two separate Rochester carburettors are used, one for each side of the engine. The air cleaner mounted on the top of the engine and using shavings of plastic material as a filtering medium supplies air to both carburettors. Easier starting of the engine is made possible by a throttle placed before the filter. Aluminium was used abundantly as a material of construction. The silumin crankcase is split vertically and tightened by 8 bolts located near the main bearings. The main and crank pin bearings are made of nickel bronze with a thin coat of babbitt. The extremely stiff six-throw crankshaft is forged of steel, has no balance weight and its main journals are of a diameter of 53·4 mm. The length of the central and end bearings is 21 mm and of the other ones 19·5 mm. The crank pin bearing has a diameter of 45·7 and a width of 16·5 mm. The length of the connecting rod between eye centers amounts to 120 mm, its shank having an H shaped cross section with a large radius transition to the big end of the connecting rod. The 20·3 mm diameter gudgeon pin is pressed into the connecting rod so that the piston has no Seger rings locking the pin. The flat bottom aluminium piston is slightly tinned and has two compression rings and one U shaped chromium plated scraping ring.

The cast iron cylinders cast into plastic shells have deep fins and are centrally countersunk into both crankcase and cylinder head. The possible use of aluminium cylinders in future is under consideration. The aluminium cylinder head common to all three cylinders on one side of the engine represents an unusual feature. The cylinders are tightened between the common head and crankcase by thin bolts with sprung washers. The wedge-shaped combustion chamber corresponds to a compression ratio of 8 : 1. The engine has relatively small valves with diameters of 34·1 and 31·5 mm for the intake and exhaust valves respectively. The valves are located near the cylinder wall with sufficient room for a partition cooled directly by the vertical fins. Other fins on the wall and exhaust manifold are horizontal. The induction pipe is cast integrally with the cylinder head. The valve seats are pressed into the head with the seat of the intake valve made of nickel alloy steel and that of the exhaust valve of chromium alloy steel. The valve rocker arms, pressed from sheet metal with a spherical bearer directly on the cylinder head bolts, are of a design used also in other types of General Motors engines.

The valves are actuated by hollow push rods from the camshaft supported

by plain bearings without sleeves and housed directly in the crankcase. The valve tappets with plain followers have a hydraulic adjustment of valve clearance in harmony with the thermal expansion of the engine. The tube protecting the push rods is sealed by rubber rings and serves for the return of oil from the cylinder head supplied thereto through the hollow rods. The cast iron camshaft has 6 induction and 3 exhaust cams. Each exhaust cam actuates two tappets of opposing cylinders and is wider than the induction cam. Exhaust ducts in the cylinder are very short in order to prevent the transfer of heat from exhaust gases to the cylinder heads. Steel tubes are pressed into these ducts connecting them with the cast iron exhaust manifold. The geared oil pump is mounted on the extended shaft of the distributor driven by a helical gear from the crankshaft. If the oil temperature exceeds 70 °C, the oil is diverted to the cooler by means of a valve controlled by a thermostat. The total oil content of the engine is 3.8 litres.

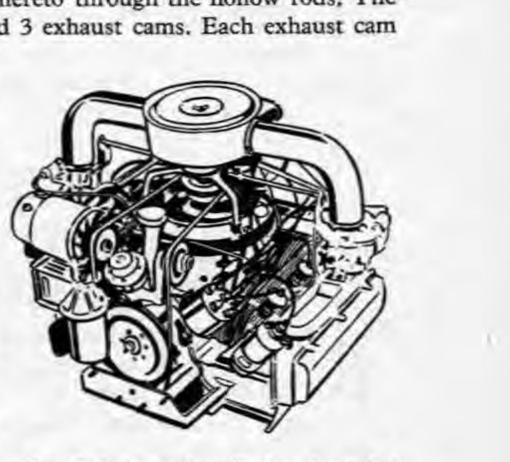


FIG. 366. The six-cylinder air-cooled Chevrolet Corvair engine.

The cooling system is another interesting feature of the engine. A radial fan with a vertical axis is placed directly above the crankcase. The impeller has 24 blades and an outer diameter of 279 mm. It is driven at a speed 1·58 times that of the crankshaft and delivers 51,000 litres of air per minute at 4,000 rev/min. A thermostat located in the stream of outgoing cooling air is connected

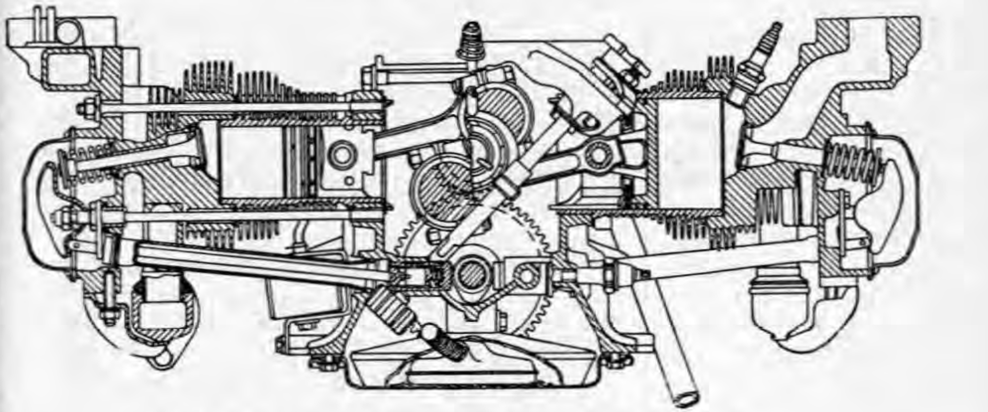


FIG. 367. Transverse section of the Chevrolet Corvair engine.

Air-Cooled Continental Engines

Type	Number of cylinders	Bore mm	Stroke mm	Displacement cm³	Performance b.h.p.	Speed rev/min	Weight kg	Unit performance b.h.p./litre	M.e.p. kg/cm²	Mean piston velocity m/s	Unit weight kg/b.h.p.	Octane number of fuel
AVDS-1790	12V	146·1	146·1	29 400	750	2400	1700	25·5	9·55	11·7	2·27	—
AVS-1790-5	12V	146·1	146·1	29 400	1400	2800	—	47·7	15·3	13·6	—	100
AVS-1790-4	12V	146·1	146·1	29 400	1040	2800	—	35·4	11·4	13·6	—	80
AV-1790-3	12V	146·1	146·1	29 400	810	2800	1080	27·5	8·9	13·6	1·34	80
AVS-1195-2	8V	146·1	146·1	19 600	685	2800	660	35·0	11·3	13·6	0·96	80
AV-1195-1	8V	146·1	146·1	19 600	530	2800	635	27·0	8·7	13·6	1·2	80
AOS-895-1	6F	146·1	146·1	14 700	500	2800	—	34·0	11·0	13·6	—	80
AO-895-2	6F	146·1	146·1	14 700	375	2800	500	25·5	8·2	13·6	1·33	80
AO-536-1	8F	117·5	101·6	8 110	250	3000	403	28·4	8·5	10·2	1·61	80
AO-536-2	8F	117·5	101·6	8 810	200	3000	403	22·7	6·8	10·2	2·00	80
AO-402-2	6F	117·5	101·6	6 600	187	3000	345	28·4	8·5	10·2	1·85	80
AO-402-1	6F	117·5	101·6	6 600	150	3000	345	22·7	6·8	10·2	2·3	80
AO-268-2	4F	117·5	101·6	4 400	125	3000	283	28·4	8·5	10·2	2·28	80
AO-268-1	4F	117·5	101·6	4 400	100	3000	283	22·7	6·8	10·2	2·83	80
4A-084	4F	76·2	76·2	1 400	35	3600	79	25·0	6·3	9·2	2·25	72
2A-042	2F	76·2	76·2	700	17·5	3600	41	25·0	6·3	9·2	2·35	72
4A-032	4F	57·15	50·8	520	10·8	3600	32	20·8	5·2	6·1	2·95	72
2A-016	2	57·15	50·8	260	5·4	3600	18	20·8	5·2	6·1	3·3	72
1A-08	1	57·15	50·8	130	2·7	3600	11·5	20·8	5·2	6·1	4·25	72
1A-03	1	44·45	31·75	50	0·87	3600	6·8	17·4	4·35	3·8	7·85	72

The continuous performance of the smallest engine range is reduced to 57 % of the maximum figure given in the table.

with an orifice which is used for throttling the air inlet. The fan is driven by a V belt running at a 90° bend on two rollers located between the pulley of the crankshaft and that of the fan axle. The left hand roller is used for driving the dynamo at a speed 2·3 times of that of the crankshaft, whilst the right hand one is a guide roller.

The engine delivers 81 h.p. SAE at 4,400 rev/min corresponding to a specific performance of 35·4 b.h.p. per litre of displacement. The total swept volume of the engine amounts to 2,286 c.c. Fig. 366 shows a view of the engine and Fig. 367 the transverse section.

5. Continental Engines

The Continental aircraft engine works have developed three basic ranges of air-cooled engines, mainly for military use. These comprise engines with performances ranging from 0·5 to 1 400 b.h.p. as listed in Table 47.

The range which comprises the smallest engines, does not make use of all parts of standard dimensions, as for instance cylinders of three sizes of bore are used. Nevertheless it is claimed that 6 types of the range can replace the 78 types of engine used by the US Army during the Second World War. Some 800 component parts are needed for the engines whereas previously 23 000 different spare parts were needed for the same range of performance. The new engines are guaranteed to run for 1 500 hours between inspection; maintenance is said to be extremely simple with no tools used. The four-stroke cycle has been preferred to the light two-stroke engine on account of its reliable starting, silent running and low fuel consumption. The two-stroke engine was found to be at a disadvantage despite its weight, owing to excess carbon formation when running on standard army fuels and oils. The engines are used for various electric generating units and the mean effective pressures and piston speeds employed are therefore very low.

A compression ratio of 6 : 1 makes a fixed ignition setting possible. 72 octane fuel is burned without knock and the engines may be started and run on JP-4 jet aircraft engine fuel at temperatures of 0° to 125 °F, i.e. —18° to +52 °C. The crankcase sump on these engines is also cooled. Carburettor freezing is prevented by effective heating through an aluminium duct. There is, however, no formation of fuel line gas locks even at 52 °C. Fig. 368 shows a sectional drawing of the 1A 08 single-cylinder engine with a performance of 1·5 b. h.p. The simple method of splash lubrication by an extended connecting rod cap bolt should be noted.

The big end bearings are of aluminium (Alcoa XC-165) and are tin coated. Push rods are also of aluminium in order to take up the valve clearance as the engine heats up. Very little valve timing overlap is employed in order to facilitate starting and reliable idling.

There is no cooling control and the engine operates without it at temperatures ranging from —54 °C to +52 °C.

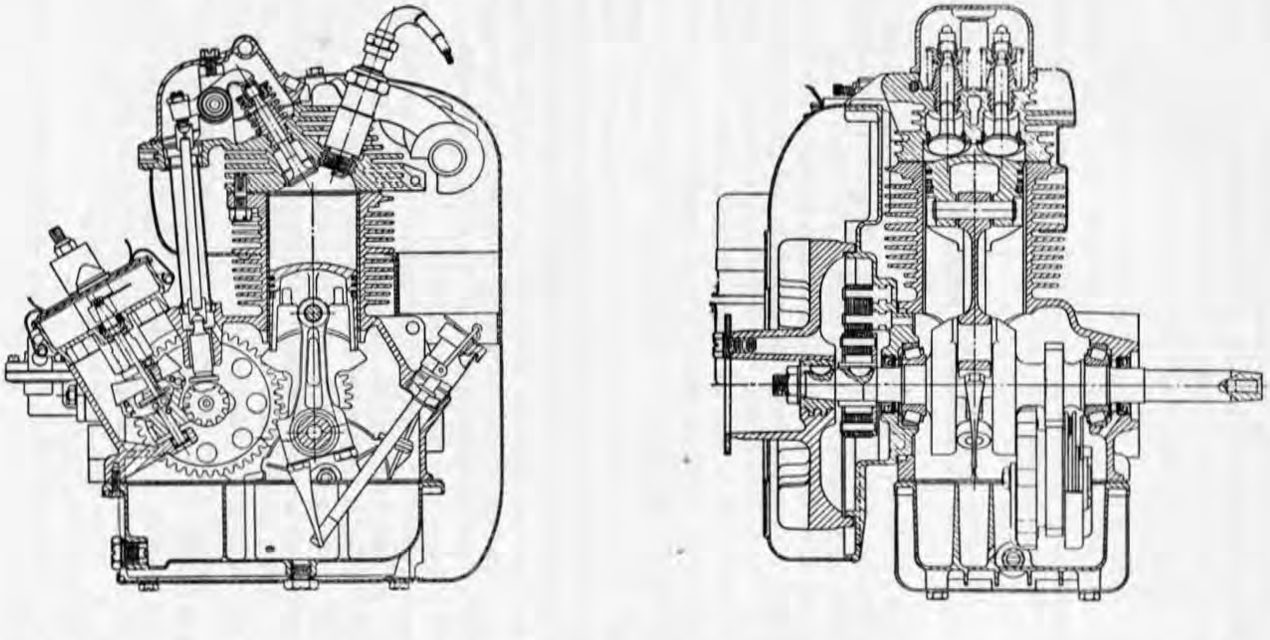


FIG. 368. Transverse and longitudinal sections of the single-cylinder Continental engine having a performance of 1·5 b.h.p.

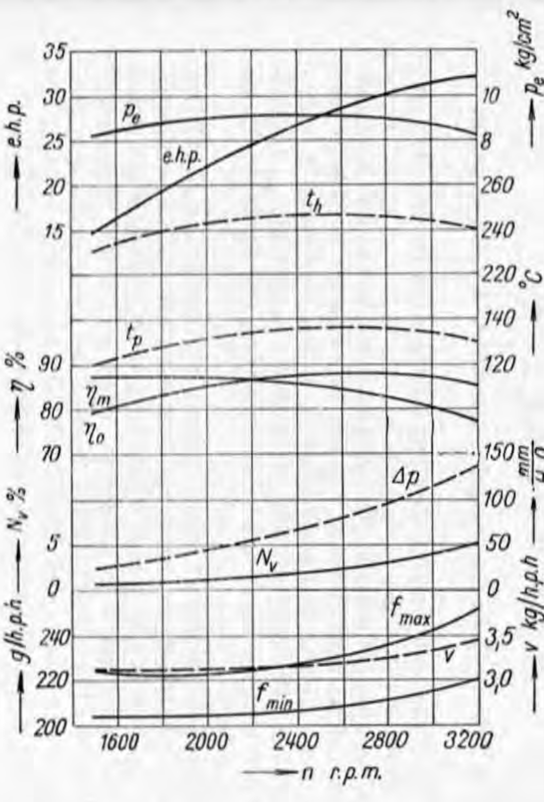


FIG. 369. Characteristics of a cylinder of the medium series of Continental engines, bore 117·5 mm.

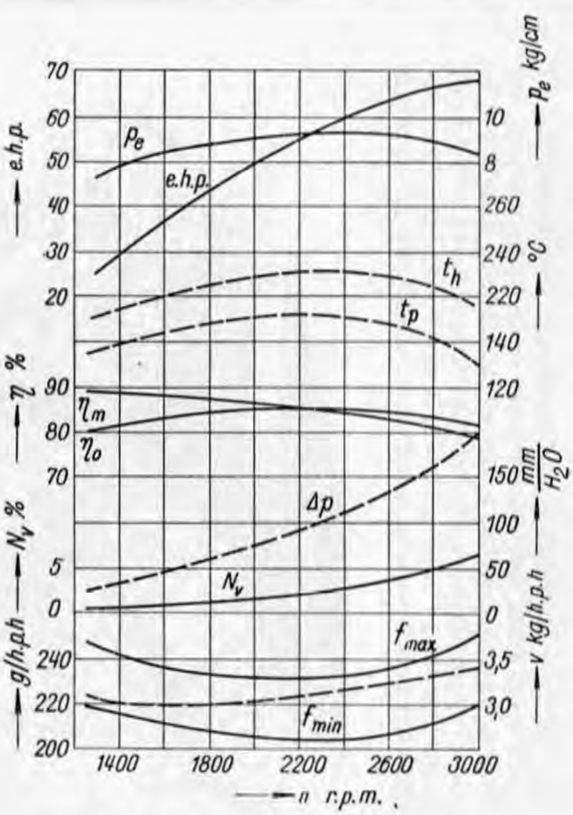


FIG. 370. Characteristics of a cylinder of the large series of Continental engines, bore 146·1 mm.

The engines of the medium range of Continental manufacture employ cylinders with a 117·5 mm bore and 101·6 mm stroke. These o.h.v. engines are produced as a flat four type, or as six-and eight-cylinder units and there is a choice of compression ratios and performances as found in Table 47. The eight-cylinder unit of this range which is used for the GM T 51 land-going vehicle is notable for its arrangement in the vehicle with the crankshaft

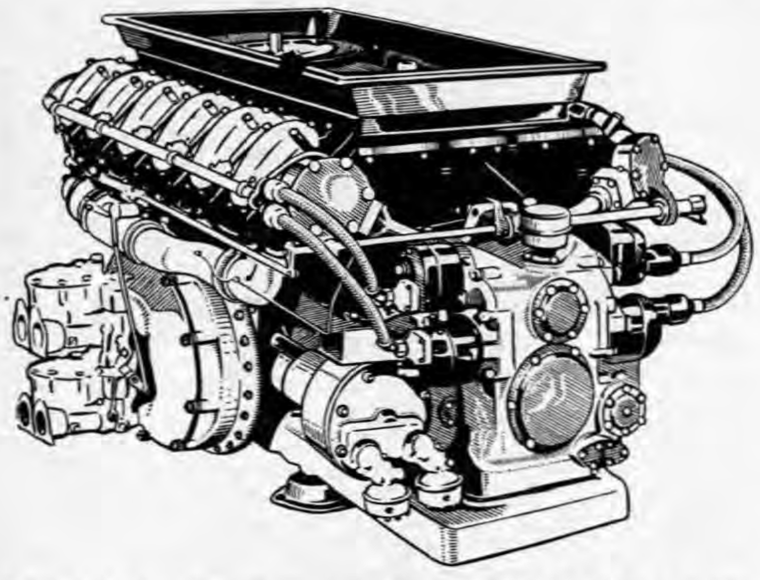


FIG. 371. Twelve-cylinder supercharged Continental AVS – 1790 – 5 engine.

axis vertical. The engine thus occupies very little space in the vehicle in plan view (as shown in Fig. 157).

The characteristics of a single cylinder as employed in this range of engines are shown in diagrammatic form in Fig. 369. This includes the cylinder head temperature t_h, cylinder base temperature t_p, cooling air pressure p, performance consumed by the blower N_v, specific fuel consumption f_{max} at maximum performance, minimum fuel consumption f_{min} and finally specific air consumption v.

The largest range of engines with a common bore and stroke of 146 mm comprises a flat six-cylinder engine and eight-and twelve-cylinder V engines with or without supercharger. The camshafts of these engines are seated eccentrically in the cylinder heads and the valves are actuated by rockers of unequal length. The supercharged version of the twelve-cylinder engine, largest of the range, has a performance of 1 400 b.h.p. at 2800 rev/min. The characteristics of a single cylinder of this range are given in Fig. 370.

The twelve-cylinder engine is interposed between the cooling air inlet and two vertically arranged axial fans driven by bevel gears (Fig. 371). An even flow distribution of cooling air over all cylinders is thus obtained. The fans are light in weight, take up little space and changes in air flow direction are kept down to a minimum. These engines are noted for their light weight. A compression-ignition engine recently developed has twelve cylinders and a performance of 750 b.h.p.

6. The Citroen Engine

This horizontally opposed twin-cylinder engine is used as prime mover for the Citroen light car. With a bore and stroke of 62 mm, the displacement of the engine is 375 cm³. The performance at 4 000 rev/min is 9 b.h.p., i.e.

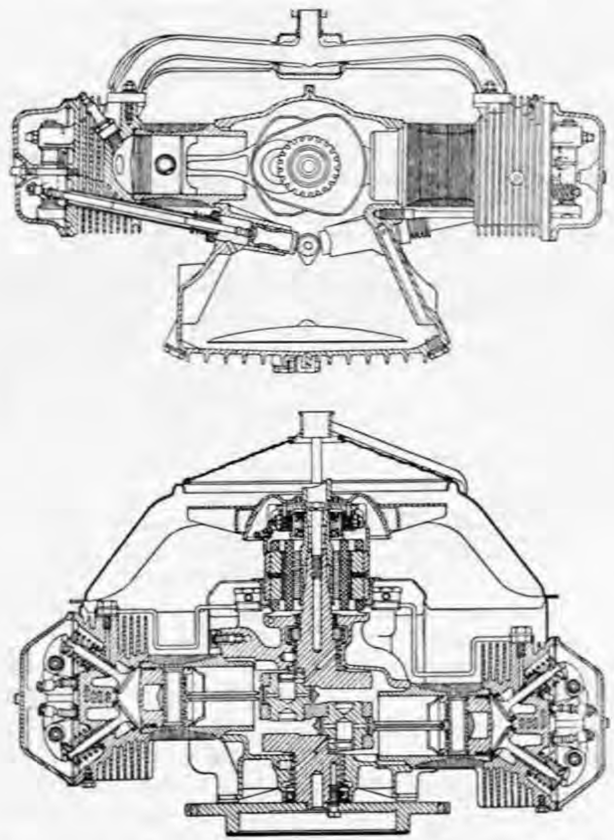


FIG. 372. Transverse and longitudinal sections of the two-cylinder Citroen engine.

24 b.h.p. per litre and an m.e.p. of 86·6 lb/in² (6.15 kg/cm²). The maximum torque of 16·56 ft-lb (2.38 mkg) is developed at 1 800 rev/min. The compression ratio of this engine is 6.2 : 1 (Fig. 372).

The overhead valves in the hemispherical head are inclined at an angle of 70° and actuated by push rods from the camshaft mounted below the crankshaft. The valve lifters are of 24 mm diameter and the push rods are

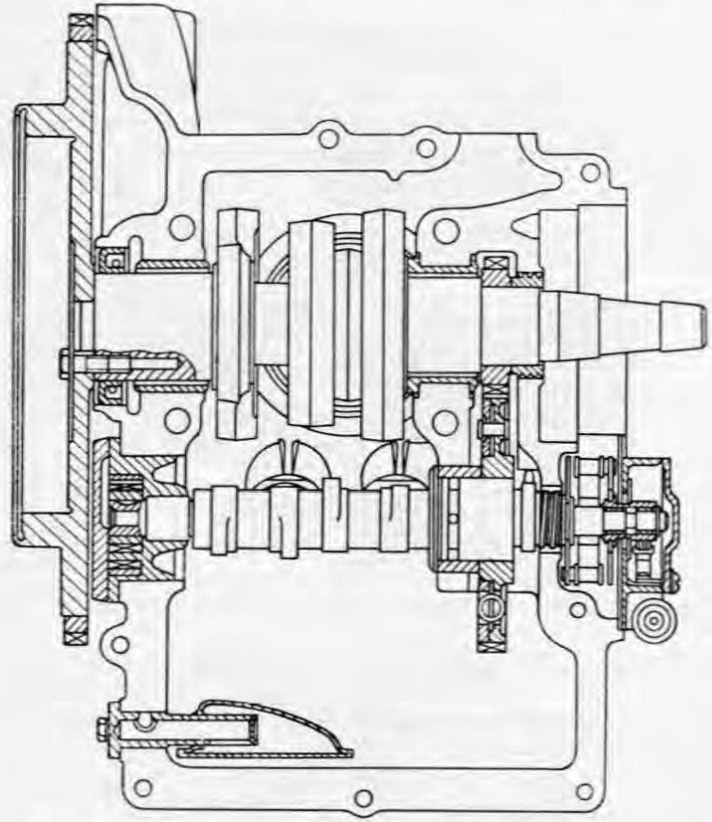


FIG. 373. Longitudinal section of the two-cylinder Citroen engine.

220 mm in length. The cylinder heads are of aluminium and both exhaust and induction pipes are upswept. The cylinder head is connected to the crankcase by three retaining bolts reduced in diameter in their middle portions to 8 mm. There is a sheet copper gasket of 1 mm thickness between the cylinder head and the cast iron cylinder barrel. The valve rockers are fitted with return springs and thus clearance is always maintained between the valve caps and

rockers only. The most recent version of the engine is fitted with pistons having offset domed crowns with resultant compression squish and a lower octane requirement of the engine.

Lubricating oil is brought to the cylinder head into an annular cavity surrounding the exhaust valve guide and hence by a drilled passage to the rocker box. The oil returns to the crankcase through the push rod cover tubes which are pressed into the cylinder heads and sealed by a spring loaded washer at their crankcase ends. The inlet valve seat angle is 60°, the exhaust valve seat angle 45°.

The crankcase is divided in the vertical plane of the crankshaft which is built up of five parts which are press fitted. One-piece connecting rods bear on lead bronze plain bearings; the crankpins are hardened. Such a layout makes for a small distance between cylinder centres resulting in reduced primary unbalanced couples. The dynamo is fitted directly on the tapered front end of the crankshaft, which arrangement obviates the use of a further bearing. Also on the crankshaft is the eight-blade axial fan, in front of which there is a wire mesh intake guard. Simple sheet metal ducting directs the air to the cylinder finning.

The arrangement of the camshaft drive is apparent in the longitudinal section shown in Fig. 373. The pinion on the camshaft has the teeth formed in two separate rows which are forced apart radially by three interposed helical springs. Thus any possible play in the camshaft drive is eliminated. Axial forces are taken up by a ring on the camshaft and by the pinion hub. The front end of the camshaft bears a centrifugal ignition mechanism controller acting on the distributor. The oil pump is driven by the rear end of the camshaft. The pump comprises two gear wheels with outer and inner teeth.

A non-return valve is provided for crankcase breathing ensuring constant under-pressure in the crankcase which prevents any oil leakage. The oil cooler is mounted on the crankcase immediately behind the fan. Oil passing through this is then fed to the main bearings of the crankshaft. These bearings are of the plain type with composition bearing surfaces. The flywheel is attached to the crankshaft by means of five bolts (M 8) and a dowel pin of 8 mm diameter. The starter ring is press fitted on the flywheel and the gear ratio between flywheel and starter pinion is 107 : 9.

7. The Fiat Engine

A parallel twin engine is used in the 500 Nuova FIAT light car. This is of 479 cm³ displacement with a bore of 66 mm and stroke 70 mm. Maximum performance of the engines fitted to the first series of cars was 13 b.h.p. The parallel overhead valves of this engine are arranged in a plane at right angles to the cooling air flow. The downdraught Weber 24 IBM carburettor is fitted with an easy starting device. Plugs 14 mm of 225 rating (Bosch) are used. A single roller chain transmits the drive to the camshaft. A centrifugal oil cleaner is incorporated in the hub of the belt pulley. 12 V coil ignition is used. A radial fan is mounted on the shaft of the dynamo which provides an output of 180 W. A thermostat controlled shutter regulates engine cooling by recirculating heated air through the fan until operating temperature is reached. The general arrangement of the engine and cooling system is shown in Fig. 125. See also Figs. 374 and 375.

8. The Jawa Engine

The best known Jawa motor cycle engine is the 250 cm^3 single cylinder type with two-stroke engine. Single cylinder two-stroke engines can be cooled with safety up to this capacity. The engine in question has a bore of 65 mm and stroke of 75 mm and its performance in 8 b.h.p. at 4 000 rev/min (Fig. 376). The location of the carburettor of this three port engine under a special cover is interesting. The piston is of the flat top kind and reverse flow scavenging is employed.

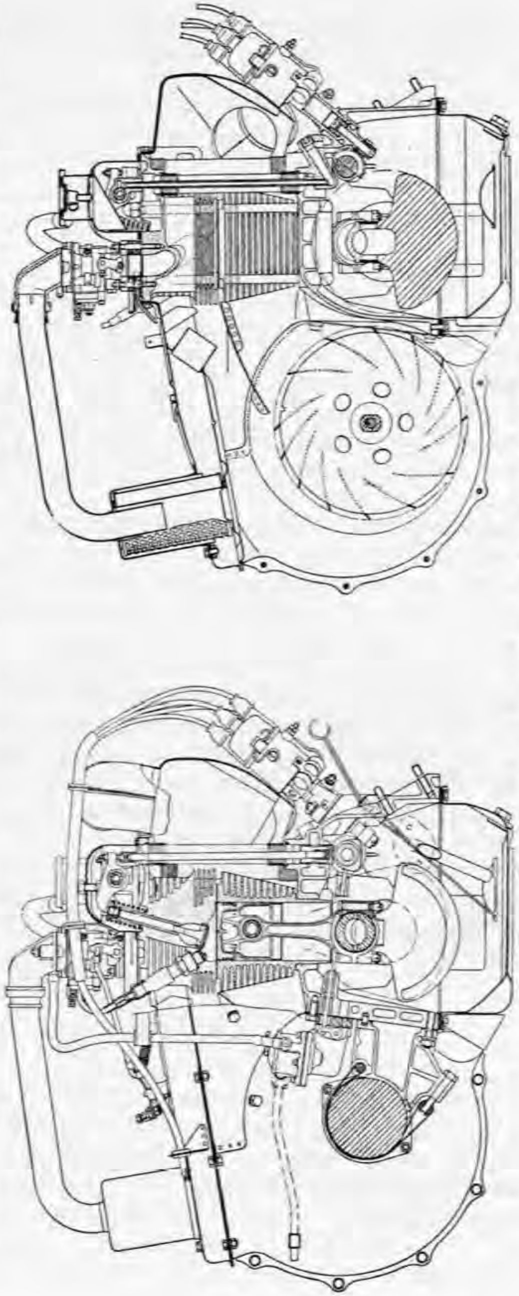


FIG. 374. Two transverse sections of the Fiat 500 engine.

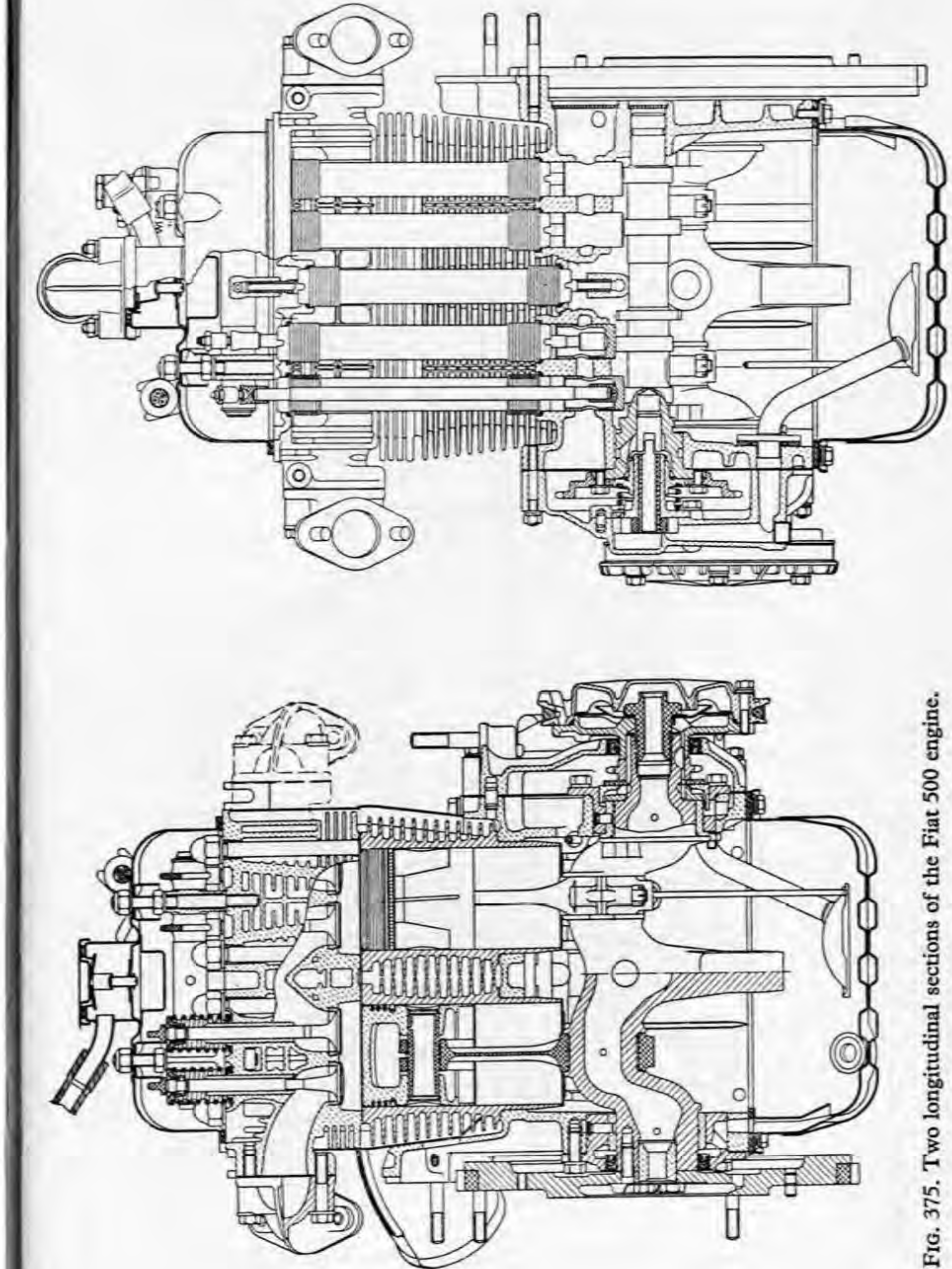


FIG. 375. Two longitudinal sections of the Fiat 500 engine.

The exhaust ports face forward and are efficiently cooled when the machine is in motion.

The cast iron cylinder barrel is mounted on a spigot which reaches deep into the crankcase. The transfer port is cast partly in the crankcase and partly in the cylinder barrel casting. The lower part of the cylinder forms a cast iron

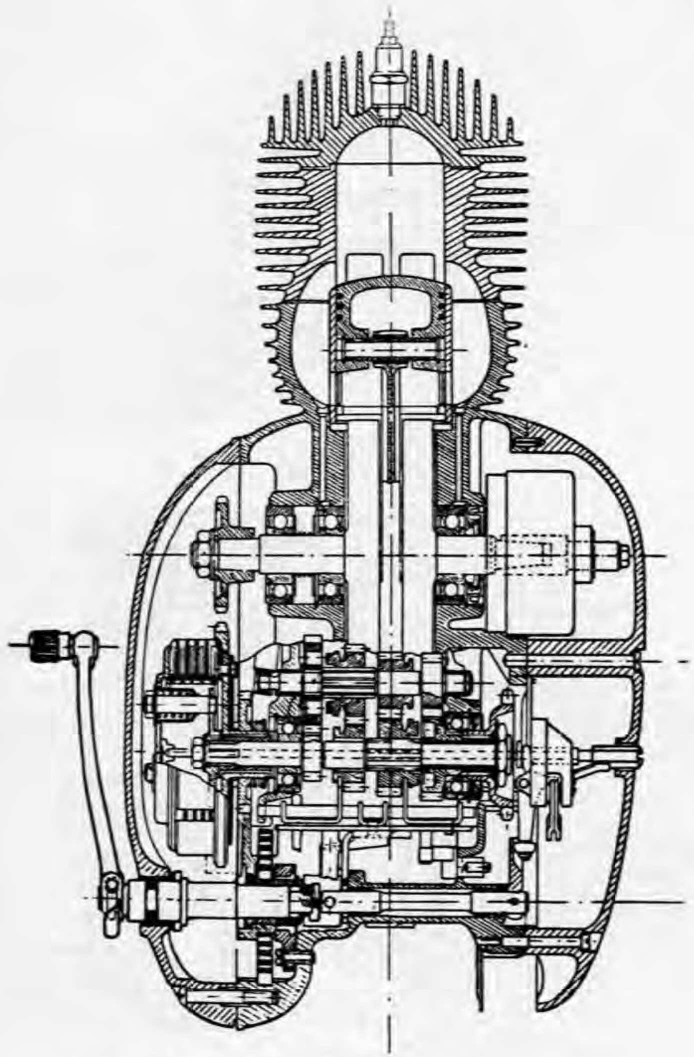


FIG. 376. Sectional view of the Jawa 250 cu. cm engine.

sleeve in the finned crankcase and the inlet port opening cut in the sleeve mates with the inlet port passage in the crankcase casting. The carburettor is fitted to a flange on the crankcase and need not be detached when removing the cylinder barrel.

The 350 c.c. engine has two cylinders (Fig. 377). Each cylinder and head is completely independent and is bolted to the crankcase by 3 long retaining

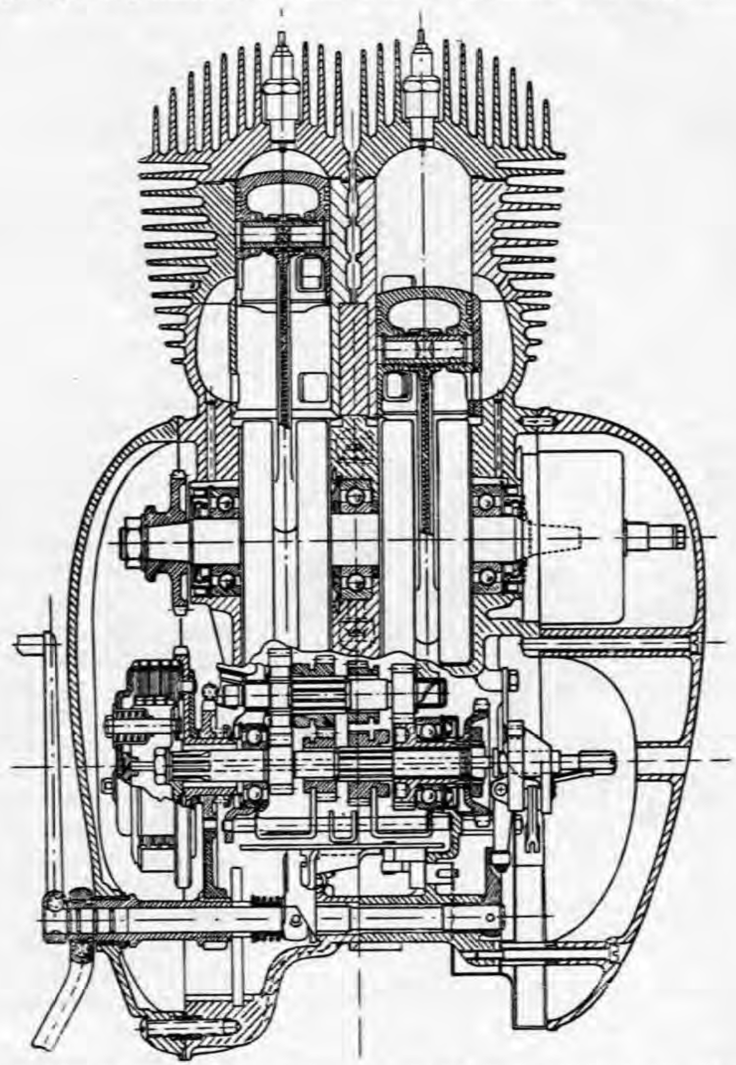


FIG. 377. Two-cylinder two-stroke Jawa 350 cu. cm motor-cycle engine.

bolts. The general design follows the 250 c.c. engine practice. In order to keep the firing intervals regular, the crankpins are 180° crankshaft rotation apart.

Efficient cooling is ensured by an adequate flow area of the cooling air stream between the two cylinders. Both cylinders and the separate crankcase breathing chambers share a common carburettor. The chambers are separated by an inserted partition wall with a ball bearing and gas seal.

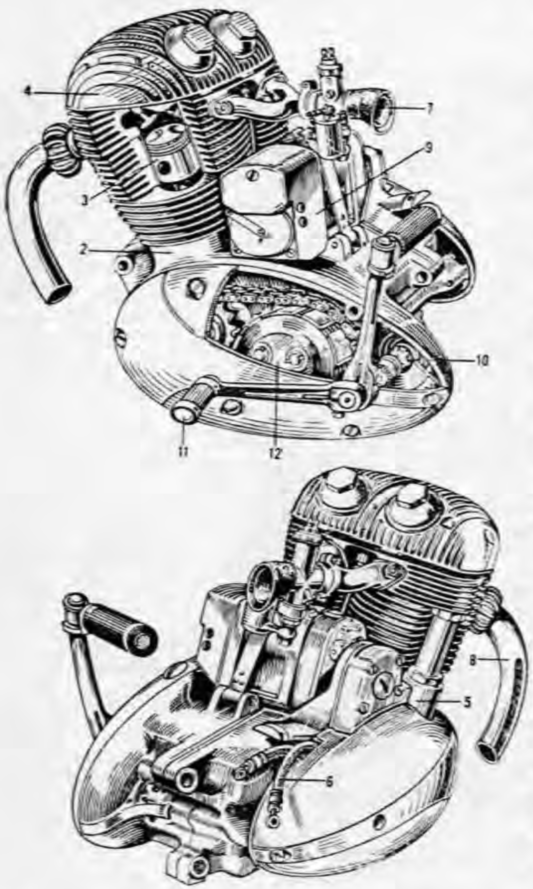


FIG. 378. Sectional view of the Jawa OHC 500 cu. cm motor-cycle engine.

The 500 c.c. Jawa parallel twin engine has a single throw crankshaft with parallel piston movement. The bore and stroke are 65 and 73·5 mm respectively, performance peak is 26 b.h.p. at 5500 rev/min. The light alloy crankcase carries two aluminium cylinders with pressed-in cast iron liners (Fig. 378). Both cylinders form an integral casting and the air flow passage is incorporated in their finning. The light alloy pistons *3* have in the piston crown cavities for the valves. The camshaft is above the combustion chamber in the cylinder head *4* and it is driven through a vertical shaft *5* with a bevel gear at the lower and a worm gear at the upper end. (In recent engines bevel gears are used at both ends.) A hemispherical combustion chamber is employed. Dry sump lubrication is used, oil being circulated from a separate oil tank through oil pipe *6* to the double gear pump.

The engine is very efficient and well proved. In its modification for racing purposes the Jawa 500 motor cycle achieved outstanding results.

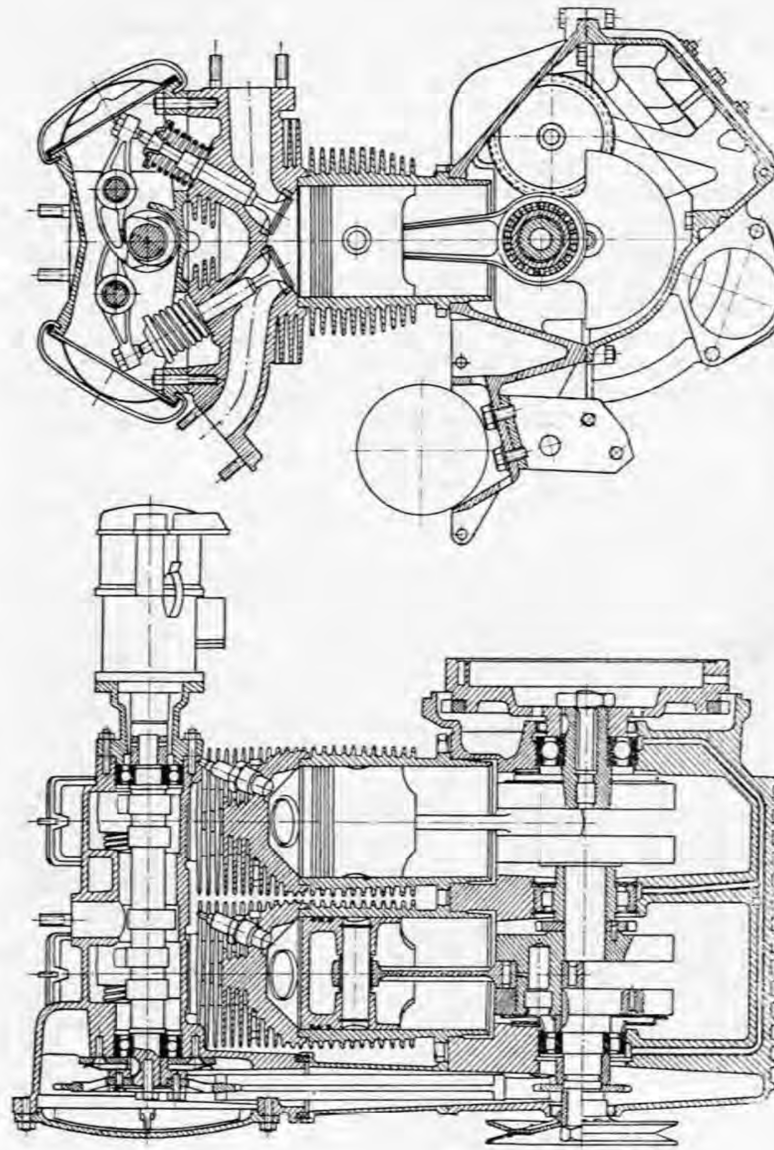


FIG. 379. Longitudinal and transverse sections of the Lloyd engine.

9. The Lloyd Engine

This engine has two parallel cylinders of 77 mm bore and 64 mm stroke giving a total displacement of 596 cm³. The performance of this engine is 19 b.h.p. at 4500 rev /min i.e. 31.6 b.h.p. per litre. A sectional drawing if this engine is given in Fig. 379.

Each cylinder consists of a separately cast iron cylinder barrel and aluminium head with a hemispherical combustion chamber. The compression ratio is 6.6 : 1. The valves are inclined at a considerable angle and are actuated from the overhead camshaft by means of rockers. The light alloy camshaft box is common for both cylinders. A single roller chain with spring tightener transmits the drive from the crankshaft to the camshaft. The valves are set in a plane at right angles to the crankshaft axis. This results in a large air flow area for cooling air. The cylinder centre-to-centre distance is therefore only 1·38 D.

The aluminium crankcase is divided in the plane of the crankshaft. This results in a rigid crankcase and easy fitting of the crankshaft which is seated in roller bearings. The built up crankshaft is assembled by pressing. The

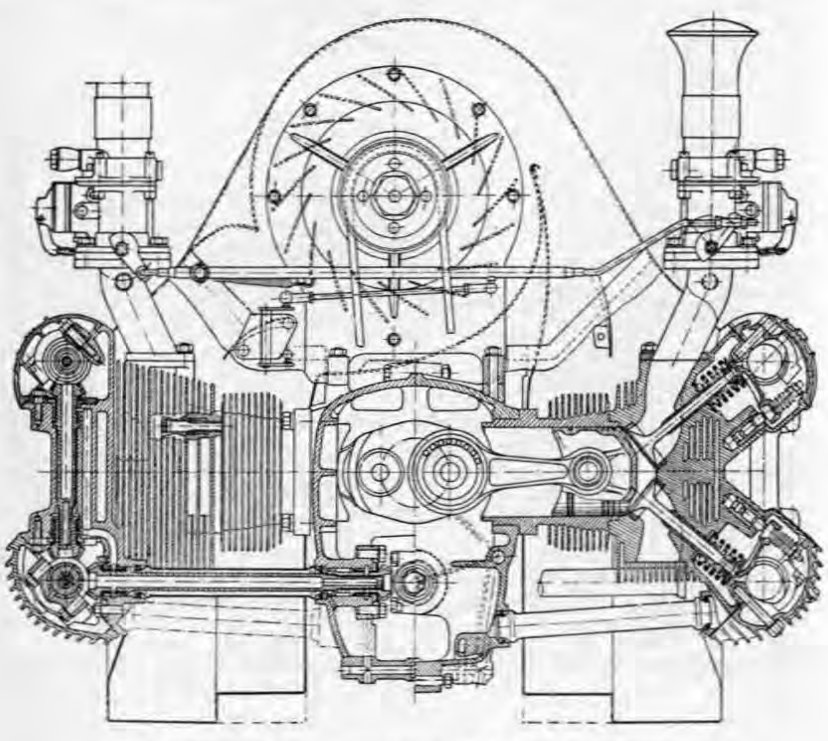


FIG. 380. Transverse section of the Porsche engine.

oil pump is driven from the middle of the crankshaft by spur gears. Crankcase ventilation is effected by a rotary breather valve communicating with the camshaft box. The valve is mounted on the camshaft immediately behind the chain sprocket. Two ports in this rotary valve open the passage to the camshaft box when the pistons are at b.d.c.

10. The Panhard Dyna Engine

This horizontally opposed twin engine of 850 cm³ displacement has a bore and stroke of 85 mm and 75 mm respectively. Engine performance is 40 b.h.p. at 5000 rev/min, i.e. 47 b.h.p. per litre. Aluminium cylinder barrels cast integrally with their heads are employed (Fig. 310). Cast iron liners are pressed into the barrels. As both cylinders are mounted individually on the engine, there is nothing to obstruct high finning; low cooling air pressure is therefore sufficient for effective cooling and this is produced by a simple two-blade fan without any ducting. Torsional valve springs are used. It may be regarded as a highly satisfactory solution of the problem of a light car engine.

11. The Porsche Engine

The Porsche 1500 engine is outstanding in its class by its high performance and low weight. It is a flat four-cylinder type, bore 85 mm, stroke 66 mm and a displacement of 1498 cm³. The performance of this engine is 100 b.h.p. at 6200 rev/min, i.e. 67 b.h.p. per litre. Maximum torque is 87 ft-lb (12·1 m/kg). The combustion chamber is hemispherical and each valve is actuated by an individual camshaft. The light rockers interposed between the cams and valve caps are not pivoted on a bush, but are simply an-

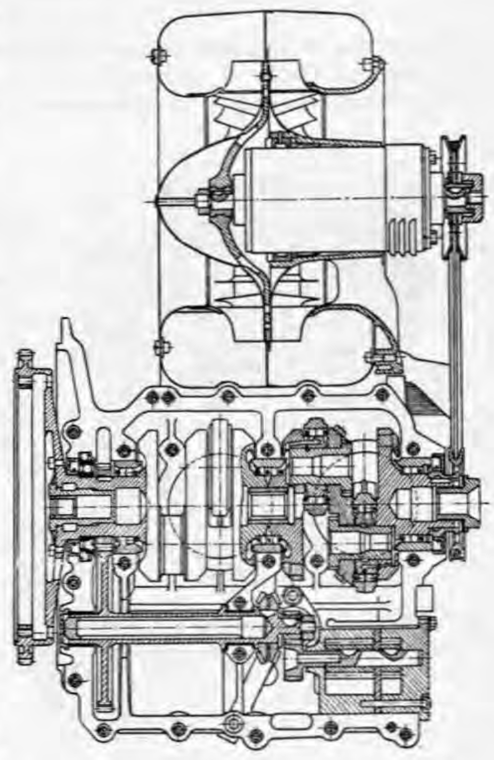


FIG. 381. Longitudinal section of the Porsche engine.

chored by a stud to the cylinder head to which they are pressed by a spring. The valves are at an angle of 78°.

As already stated, the engine incorporates four camshafts, each driven from the crankshaft through bevel gears and inclined shafts. The compression ratio of the standard engine is 9 : 1, demanding fuel of 86 octane rating. Cylinders are cast in aluminium and their bores are hard chromium plated – an arrangement whereby high cooling efficiency is attained. The crankshaft and connecting rods operate in rolling contact bearings. The twin radial fan is driven by the crankshaft through a belt at a ratio of 1 : 1. At 6200 rev/min the fan supplies 4290 ft^3 (66 m^3) of air per minute. Dual ignition is employed with Bosch W 240 T 21 sparking plugs. Two twin-choke Solex 40 PJJ craburettors provide the mixture. The fuel is delivered by two electric pumps. Sectional views of the engine are shown in Figs. 380 and 381.

12. The Sunbeam Engine

This parallel in-line twin cylinder engine for motor cycle use has a bore of 69·85 mm and stroke of 63·5 mm, its displacement being 487 cm^3. With a compression ratio of 7 : 1 the performance is 23 b.h.p. at 5750 rev/min.

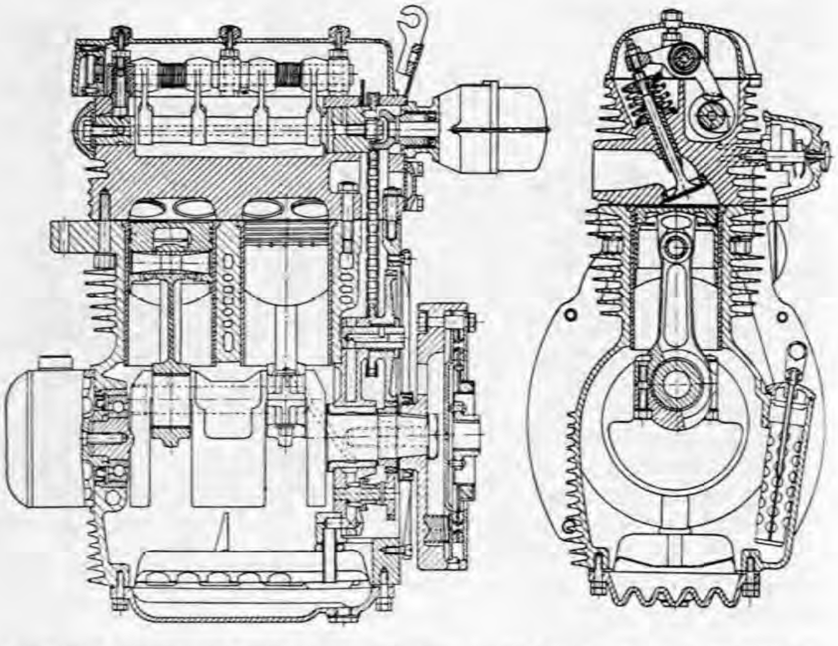


FIG. 382. Longitudinal and transverse section ofthe Sunbeam motor-cycle engine.

The engine design follows current automobile practice. The cylinders form an integral light alloy casting with the crankcase; austenitic iron liners are pressed into them. Air passages are provided between the cylinders. The crankshaft is cast and is fitted with three counterweights balancing primary reciprocating masses to 50%. The large rear crankcase cover facilitates the fitting of the crankcase assembly into the tunnel-type crankcase. The oil pump

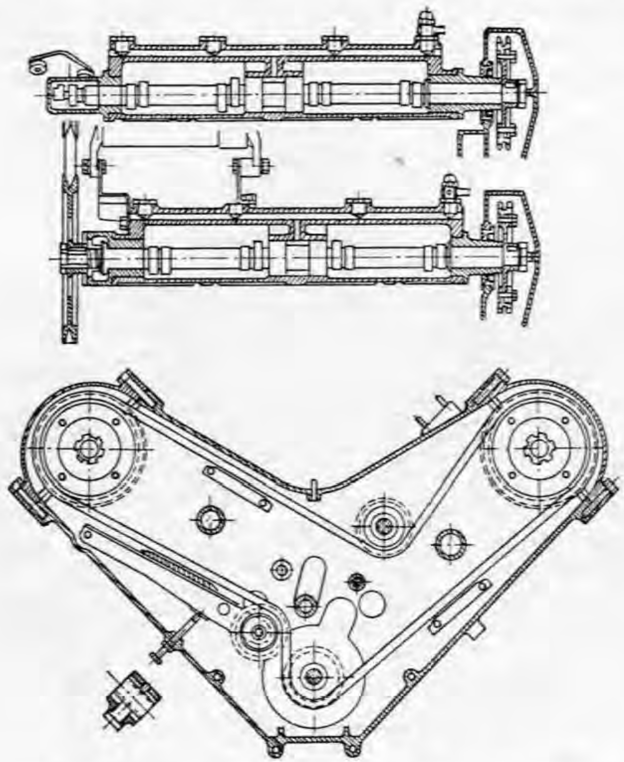


FIG. 383. Drive of the valve gear of the Tatra 87 engine.

is incorporated in this cover together with the pinion driving the overhead camshaft by a single roller chain. The distributor is driven by one end of the camshaft. Valves are actuated by rockers. The front end of the camshaft drives the dynamo.

The valves are disposed in a single row and inclined by 22·5° from the vertical axis of the engine. A suitably shaped combustion chamber with a large squish area results. The valve seats are pressed into the aluminium head, the inlet valve port diameter being 33·3 mm, that of the exhaust port 30·1 mm.

The camshaft is housed direct in the aluminium head. Light alloy connecting rods forged of RR56 of 114·3 mm centre-to-centre length are used. Fig. 382 shows the transverse and longitudinal sections of the engine.

13. The Tatra 87 Engine

This V-8 engine designed as a prime mover for passenger cars has a bore of 75 mm and stroke of 84 mm and a displacement of 2968 cm^3. With

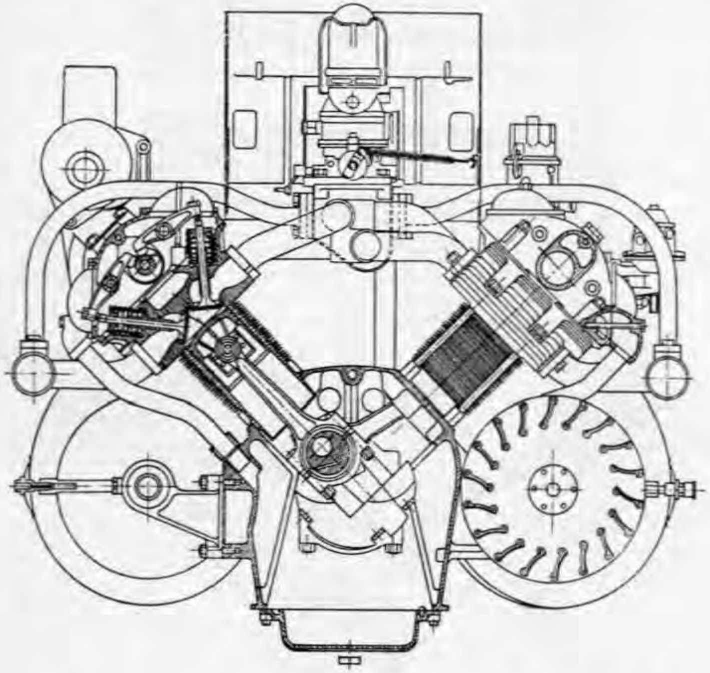


FIG. 384. Transverse section of the eight-cylinder Tatra 87 engine.

a 5·6 : 1 compression ratio, the performance is 70 b.h.p. at 3600 rev/min. The engine weighs 505 lb (194 kg).

This engine has a hemispherical combustion chamber with valves actuated by rockers bearing on an overhead camshaft. Each cylinder with its head forms an individual unit and the camshaft for four cylinders of each bank is mounted in its separate housing (Fig. 383). The attachment of this camshaft housing must provide for small differences in heat expansion which may arise e.g. when one cylinder is out of action.

The camshafts are roller chain driven from the front end of the crankshaft. This form of drive can take up the small variations of distance between the shaft centres with varying engine temperature. Chain guides and a special tightener prevent chain whip. The front cover mounting on the camshaft housing must be resilient. The rear ends of the camshaft are utilized for

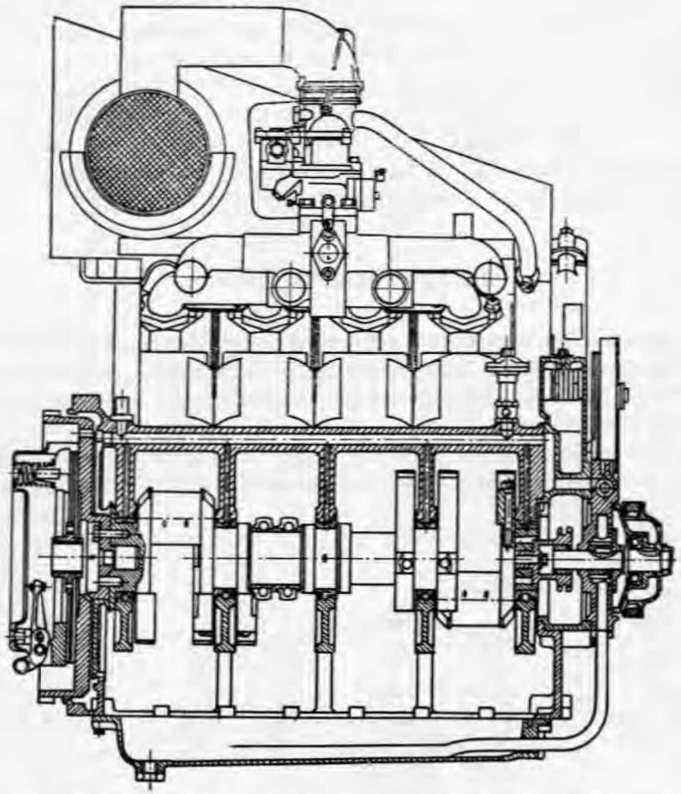


FIG. 385. Longitudinal section of the eight-cylinder Tatra 87 engine.

dynamo and distributor drive. The use of an overhead camshaft prevents valve clearance variation owing to expansion of the engine by heat.

The two banks of cylinders are mounted at an angle of 90°. The crankcase, made of electron metal, has a detachable sump to facilitate crankshaft fitting. The crankshaft is of four-throw type with throws 90° apart and balancing of primary reciprocating masses is achieved by means of bobweights attached to the crankshaft webs. The crankshaft is seated in plain main bearings with white metal surfaces.

Two radial fans arranged on both sides of the engine supply the cooling air. They are driven by V belts which are tightened by pivoting the fan pulleys away from the crankshaft pulley. This is arranged by turning the fan shaft about a swivel bearing located between the impellers. Air leaving the fan impeller is directed by a spiral cowling mainly to the cylinder heads and directional changes of flow are kept to a minimum.

The sheet metal induction manifold is heated by warm air from the engine. Cold air is brought to the twin choke carburettor from the air filter; provision is made for the supply of warm air in winter conditions.

A single pressure pump supplies lubricating oil from the sheet metal crankcase sump. The pump is driven from the front end of the crankshaft. It supplies oil to the oil cooler mounted in the forward end of the car and thence the oil flows through a multi-edge oil filter and further to the lubricated parts of the engine. Other details are apparent in Figs. 384 and 385.

14. The Tatra 600 Engine

This type is a flat, horizontally opposed four-cylinder engine of 85 mm bore and 86 mm stroke with a displacement of 1950 cm^3 and a performance of 52 to 55 b.h.p. at 4000 rev/min. Its compression ratio is 6 : 1 and the dry engine weighs 406 lb (156 kg.)

The combustion chambers of this engine are hemispherical. Valves are actuated by crossed rockers which are operated by push rods from a single

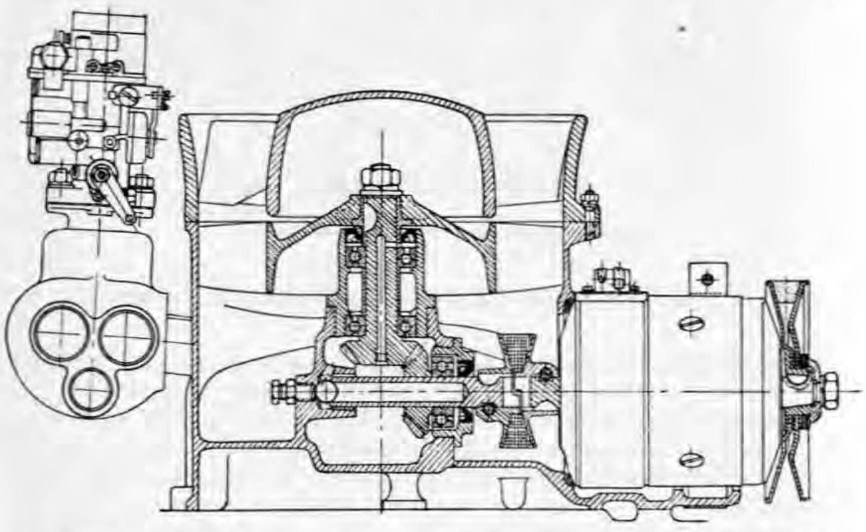


FIG. 386. Longitudinal section of the fan drive of the Tatra 600 engine with vertical fan.

camshaft housed in the crankcase below the crankshaft (Fig. 103). Oil is supplied to the valve mechanism through valve lifters, aluminium push rods and drilled valve rockers; it returns through the push rod cover tubes. Rocker box covers are sheet metal pressings shaped with regard to easy access to the sparking plugs.

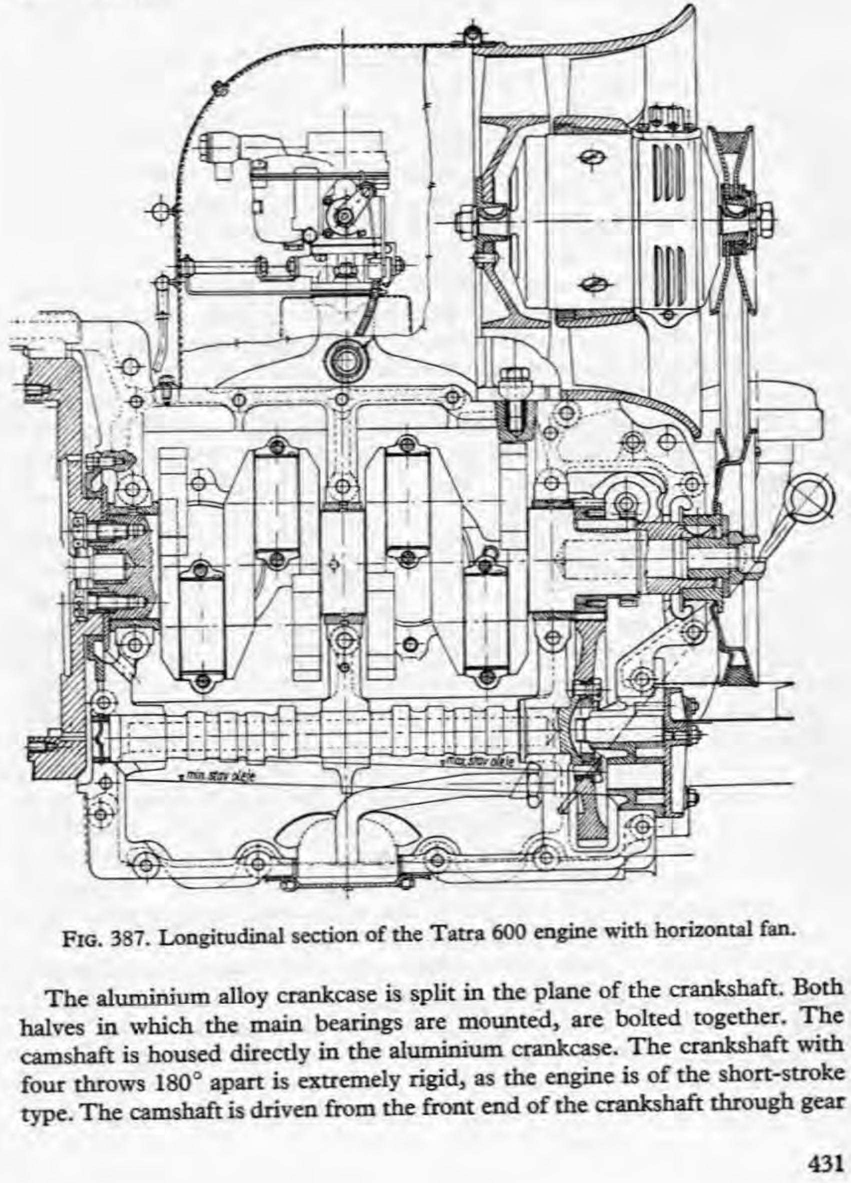


FIG. 387. Longitudinal section of the Tatra 600 engine with horizontal fan.

The aluminium alloy crankcase is split in the plane of the crankshaft. Both halves in which the main bearings are mounted, are bolted together. The camshaft is housed directly in the aluminium crankcase. The crankshaft with four throws 180° apart is extremely rigid, as the engine is of the short-stroke type. The camshaft is driven from the front end of the crankshaft through gear pinions and the ignition distributor through worm gears. The distributor shaft incorporates a fuel pump drive cam.

The lubricating oil pump is driven by the front end of the camshaft, the supply of oil being stored in the finned crankcase. Oil is forced from the pump through the oil cooler mounted in the front part of the car and, through a multi-edge cleaner to the lubricated parts. By-pass pressure valves are provided at the cooler and cleaner.

Cooling air is supplied by an axial fan with a vertical axis, the rotor being driven from the dynamo by bevel gears (Fig. 386). This arrangement was later simplified, the axis of the fan being turned by 90° to horizontal and the impeller fitted direct on the dynamo shaft (Fig. 387). The use of long induction piping was obviated by the use of two carburettors. A sheet metal cowling above the engine directs the air from the fan to the cylinders. The method of cylinder head cooling in this engine has been described in detail in preceding chapters.

The sheet metal exhaust pipe is shrouded by a duct through which heating air is supplied to the car interior. Shutters in the outgoing cooling air duct permit re-circulation of warm air to the engine compartment in winter in order to improve carburation characteristic.

The performance of this engine was increased for sporting car use, by raising the compression ratio to 9:1 and using alcohol fuel, to 85 b.h.p. at 4500 rev/min. A separate carburettor was fitted to each cylinder. No cooling trouble was encountered at this performance.

15. The Tatra 603 Engine

The eight-cylinder Tatra 603 engine has a bore of 75 mm and a stroke of 72 mm, the engine capacity being 2545 cm^3. The engine is mounted in the rear of the car and is designed for this position with a view to accessibility, weight and performance. Small cylinder dimensions and efficient cooling make possible a compression ratio of 6·5 : 1. The dry weight of the engine is 416 lb (160 kg) including fans and oil coolers, which with a net performance of 100 b.h.p. at 4800 rev/min results in a highly favourable weight: power ratio of 4·1 lb/b.h.p. (1·6 kg/b.h.p.) The same engine is also used for the Tatra 805 light truck with performance reduced to 75 b.h.p. at 4200 rev/min. In the latter instance the engine is cooled by air from two axial fans driven by V belts forcing air to the cylinders, cool air reaching exhaust valves first. Warm air leaves the engine past the induction manifold.

In the Tatra 603 F engine for passenger car use the cooling arrangement is modified in that it comprises two axial fans placed at the sides of the engine, which draw the air over the engine. Thus fresh cooling air is brought to the engine past the carburettors and induction manifolds. With such an arrangement the induction side of the cylinder head runs at a lower temperature, which makes for higher volumetric efficiency of the engine. Two twin-choke

Motorpal 30 SSOP carburettors also aid high volumetric efficiency at high engine speeds.

The induction-type cooling layout employed has the advantage of an even distribution of cooling air over all cylinders. An additional air stream entering

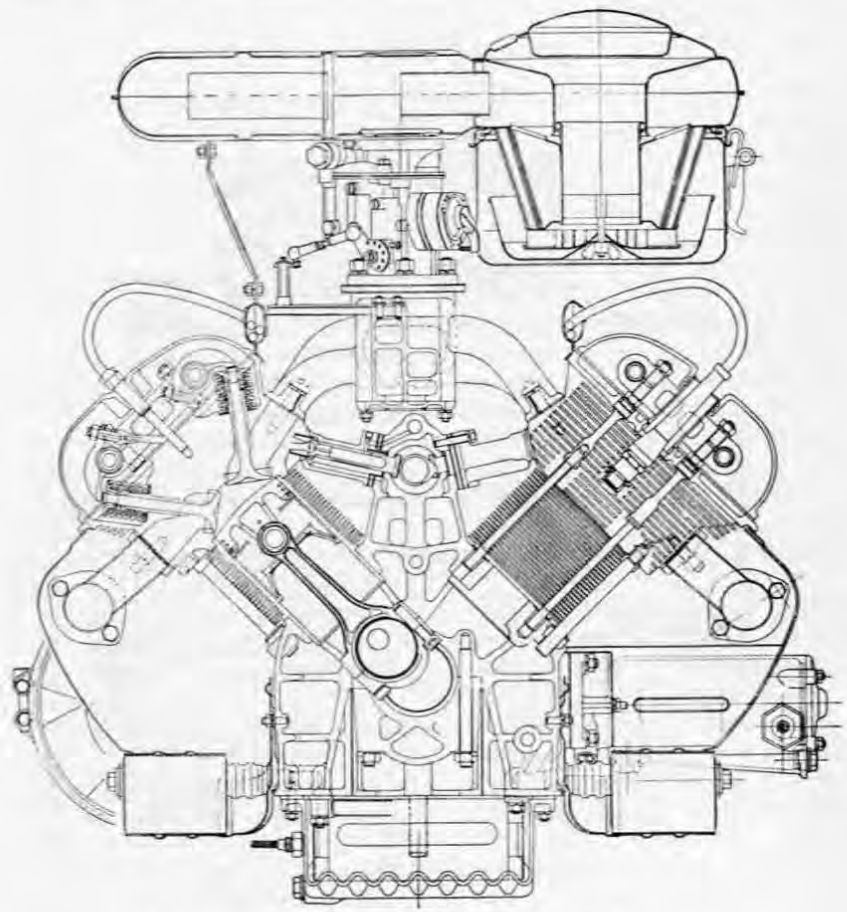


FIG. 388. Transverse section of the Tatra 603 F engine.

from atmosphere direct to the exhaust valve ensures an uniform temperature of the whole cylinder circumference. The cast iron cylinders have machined fins with a small spacing of 3·4 mm, the mean fin thickness being 1 mm. A bellows-type thermostat controls engine cooling by operating the air-flap in the outlet ducting behind the fan. Fresh air is also drawn through two oil coolers mounted on the engine sides. The general layout of this engine is depicted in Figs. 388 and 389.

The combustion chambers of this engine are hemispherical and widely inclined valves are operated by short push rods from a camshaft mounted high in the crankcase. The weight of all valve gear component parts is very low, thus

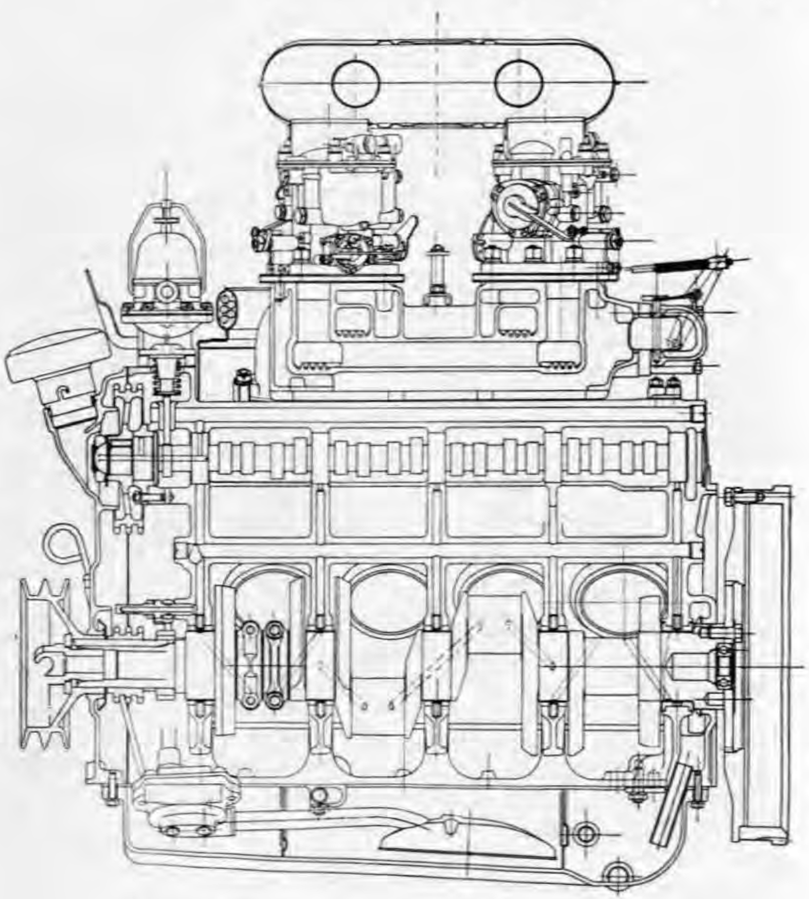


FIG. 339. Longitudinal section of the Tatra 603 F engine.

reducing the stress on the valve gear at high engine speeds. Volumetric efficiency is also high owing to small cylinder dimensions and to the large valves employed. As the valves are in a plane perpendicular to that of the crankshaft, the cooling air flow area is sufficiently large. The cooling surface area of the cylinder head is 2130 cm², compared with the 1190 cm² area of the

Tatra 87 engine with identical bore diameter. The cylinder surface cooling area is 1575 cm².

The close grouping of cylinders in each bank (with centre-to-centre distances of only 100 mm) results in an engine of very small overall length. Neither in weight nor in dimensions does this engine exceed the corresponding values for the Tatra 600 engine, yet performance is 60% higher. Sparking plug accessibility is exceptional. The inlet manifold is an aluminium casting and is heated by exhaust gases.

The aluminium alloy crankcase is open from below to facilitate crankshaft mounting and the opening is covered by a sheet steel sump. Crankcase rigidity has been enhanced by an upward crankcase wall extension for the high camshaft position. The four-throw crankshaft with throws 90° apart (the cylinder banks enclose an angle of 90°) is supported in five plain composition bearings. Thin-walled bearings with lead-bronze bushings are used for the big ends. Balancing of primary reciprocating masses is achieved by bobweights integral with the crankshaft.

A gear type oil pump is driven from the crankshaft by worm gears together with the ignition distributor. The oil flows through a passage drilled in the crankcase wall to and from the oil coolers mounted on both sides of the crankcase. As the cooler attachment bolts are drilled, no separate tubing is required and no separate oil unions need be attended to during engine dismantling. Cooling air is drawn through the oil coolers, making the cooling effect independent of vehicle speed. Oil from the coolers flows through multi-edge cleaners to lubricated parts. Coolers and cleaners are protected against damage in winter by by-pass valves. Oil pressure is indicated by a dashboard light.

The crank mechanism is very light in order to reduce frictional losses at high engine speeds. Piston skirts are cut-away in order to provide space for the bobweights. High engine speeds are made possible by the light crank assembly.

The compression ratio has been raised to 12 : 1 for racing purposes and with alcohol fuel a performance of 200 b.h.p. was reached at 7500 rev/min. The engine can go up to 8000 rev/min with the following modifications: stronger valve springs, racing camshaft and a new induction manifold for the two twin choke carburettors.

Other parts remained as in standard engines. Development work on this engine is continuing and improved performance may be expected.

16. The Vespa Engine

This is a typical two-stroke twin cylinder engine with both bore and stroke of 63 mm. The engine displacement is 394 cm³ and performance is 14 b.h.p. at 4350 rev/min. A maximum torque of 19·5 ft lb (2·7 mkg) is produced. The timing of mixture entering the crankcase of this engine is not controlled by the bottom edge of the piston skirt but by ports in the flywheel and an unsymmetric timing diagram is thus obtained. A Solex 26 AHCD carburettor is fitted. The radial fan is V belt driven and its positioning is apparent in Fig. 390. The engine is mounted in the rear of the vehicle.

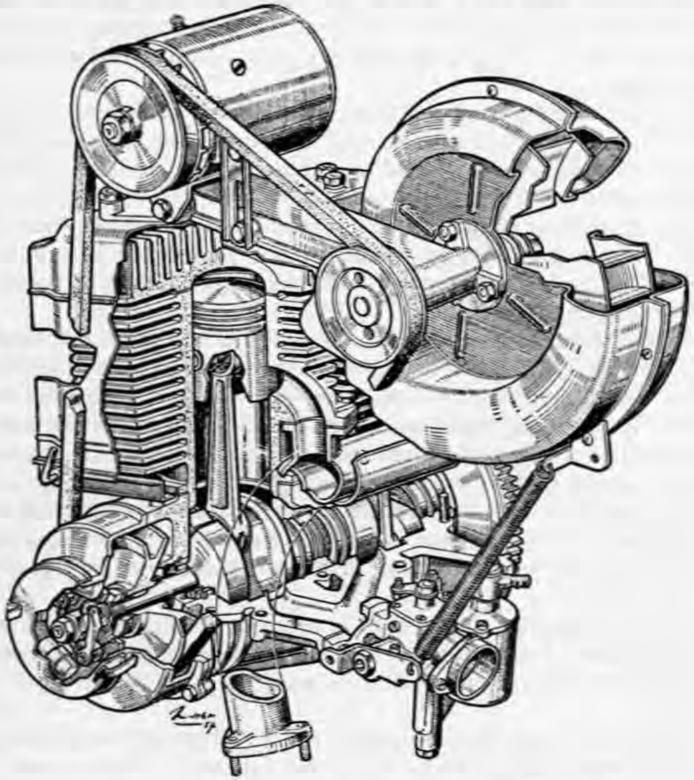


FIG. 390. Sectional view of the two-stroke Vespa engine.

B. COMPRESSION-IGNITION ENGINES

1. The ČKD Engine

The 6V 145-ADR engine was the result of the first effort of the ČKD works to produce an air-cooled heavy duty engine of large cylinder dimensions. Previously air-cooled engines with bores exceeding 140 mm had been designed for aircraft and tank propulsion. The ČKD engine is, however, intended for

railway and tractor use. The engine in question is composed of standard parts which are common for a six- and twelve-cylinder layout. By the additional use of superchargers, engines of four different and regularly stepped performances

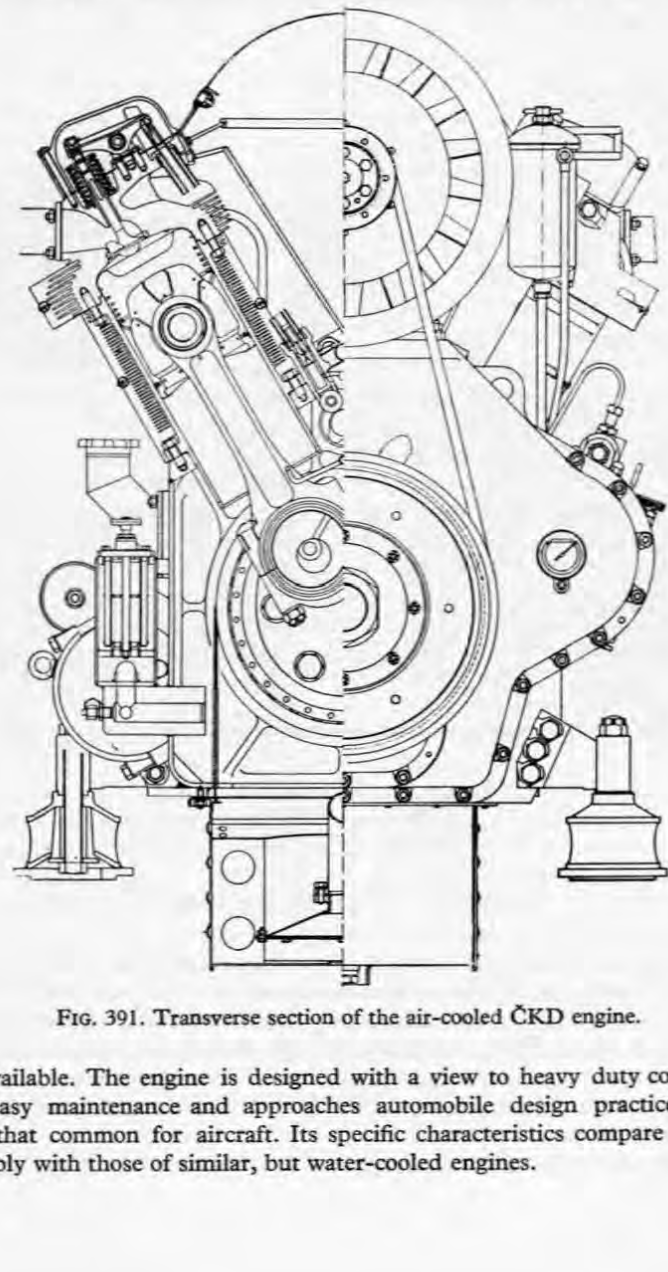


FIG. 391. Transverse section of the air-cooled ČKD engine.

are available. The engine is designed with a view to heavy duty conditions and easy maintenance and approaches automobile design practice rather than that common for aircraft. Its specific characteristics compare very favourably with those of similar, but water-cooled engines.

The bore of this engine being 145 mm and stroke 180 mm, the displacement is 17·8 litres for the six-cylinder unit. A continuous performance of 150 b.h.p. is given at 1400 rev/min with a possible peak of 180 b.h.p. at the same speed. The performance of the supercharged engine is 220 b.h.p. The engine weighs 2866 lb (1300 kg).

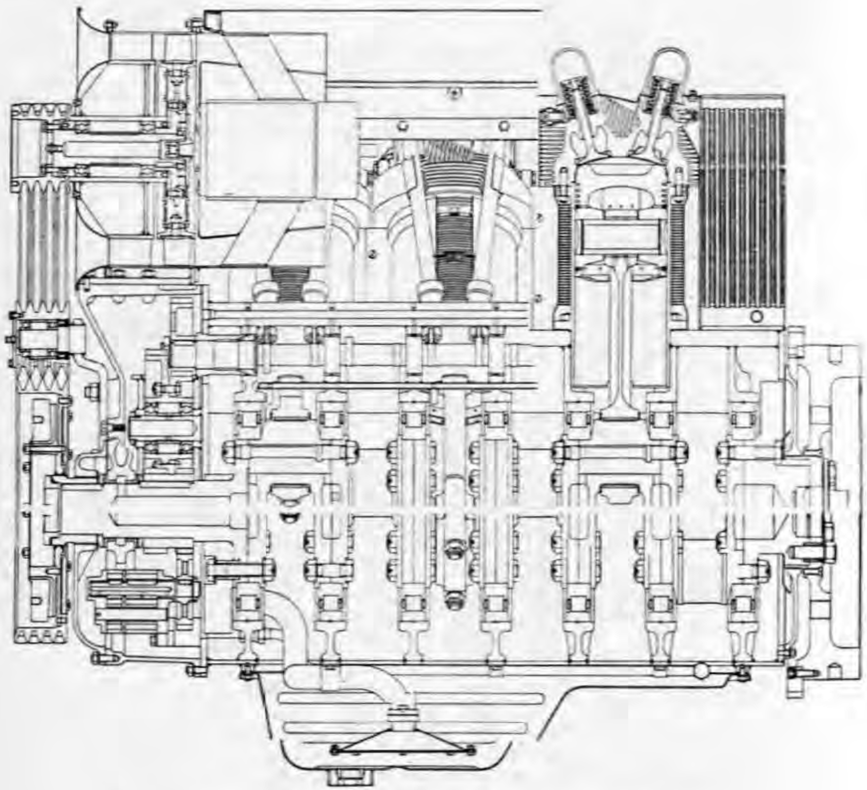


FIG. 392. Longitudinal section of the air-cooled ČKD engine.

The crankcase is welded plate steel with cast steel partitions. The aluminium pistons are cooled by oil sprays directed against the underside of the crowns by fixed nozzles (Fig. 391). The connecting rod is fully machined with bearings cast in lead-bronze. The crankpins of 100 mm diameter are surface hardened. The built-up crankshaft is housed in seven roller main bearings, the front journal being fitted with a pulley for the fan belt drive incorporating a rubber damper of torsional vibrations.

A single camshaft, driven by spur gears from the front end of the crankshaft,

is employed. The injector pump attached to the side of the engine is readily accessible. The cylinder head is of aluminium alloy and it carries two slightly inclined valves. Note the finning of the exhaust valve area which is apparent in the longitudinal section of the engine (Fig. 392). The injector nozzles are located sideways on the edge of the combustion chamber which is formed by a cavity in the piston crown. In a later version of the engine the nozzle axis almost coincides with that of the cylinder. The steel cylinder is nitrided and the head is attached by bolts and a flange without a gasket, thus simplifying dismantling.

The V arrangement which is unusual for six-cylinder engines gives small overall dimensions and enables the standardization of parts for the twelve-cylinder unit. The direct-injection combustion chamber makes for low fuel consumption and a clean exhaust.

The engine is started either by a 24 V 15 b.h.p. electric starter or by an auxiliary air cooled petrol engine. Cooling air for the engine is supplied by an axial fan at the rate of 88 ft^3 (2·5 m^3) and at 110 to 120 mm water column pressure to the spaces between cylinders. The relatively wide spacing of cylinders has made possible the use of extensive finning of the cylinders (8700cm^2) and cylinder heads (7800 cm^2). The uniform distribution of temperature through the cylinders is aided by a quickly detachable cowling. A complete cylinder unit weighs 55 lb (25 kg), the cylinder barrel weighing 30·8 lb (14 kg) and the head without accessories 17·6 lb (8 kg).

The twelve-cylinder unit using the same basic component parts has also two banks of cylinders inclined by 60°. The engine displacement is 35·6 litres and constant performance 300 b.h.p. at 1400 rev/min with a maximum of 330 b.h.p. This performance is increased to 450 b.h.p. by the addition of a supercharger. Overall engine dimensions: width 1190 mm, height 1320 mm, length 1910 mm, weight 4365 lb (1980 kg).

2. The Continental Engine

A compression-ignition engine with a performance of 750 b.h.p. has been developed by the Continental works to complete its large range of air-cooled petrol engines. This twelve-cylinder unit has a bore and stroke of 146.1 mm like the petrol engines. The engine is supercharged by two exhaust turbochargers and its technical data are given in Table 47 (type AVDS-1790). Cylinder banks are arranged at an angle of 90°. The turbo-charged engine can be uprated to 850 b.h.p. and it is automatically compensated for loss of performance at high altitude or through high inlet air temperature. The chief advantage of the compression-ignition engine is indicated in this instance by a fuel consumption 40% lower than that of a similar petrol engine.

The cooling air is provided by two axial fans. This arrangement makes for a uniform distribution of air flow over the cylinders and the shortest possible exit route of the warm air from the armoured engine compartment. The bullet-proof grilles offer a considerable resistance to air flow and the total fan input is therefore almost double normal requirements. The cooling fans consume 110 b.h.p. at maximum engine speed. The axial fans have a diameter of 25·9 in (660 mm) and they rotate at twice crankshaft speed. The cylinder head temperature with an output of 750 b.h.p. is only 175 °C, the pressure head at the cylinders being 150 mm water column. Such a low cylinder head temperature is conducive to freedom from injection nozzle troubles.

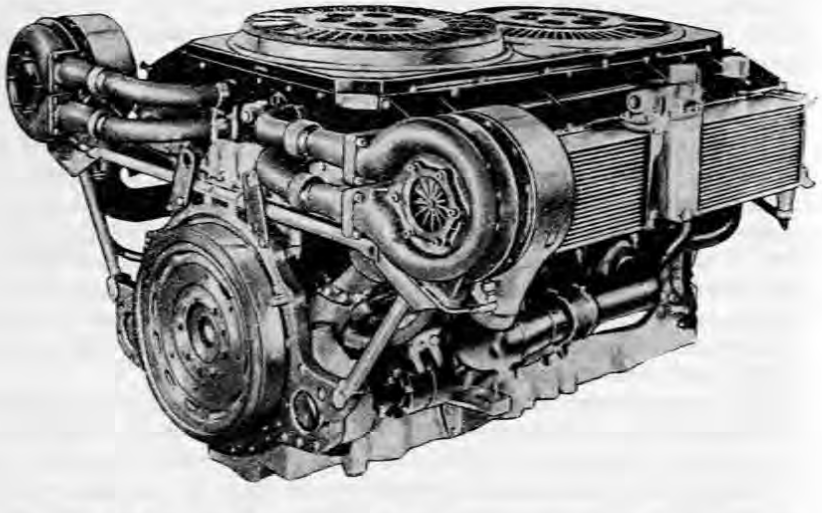


FIG. 393. View of the Continental 750 b.h.p. compression-ignition engine using turbo-chargers.

The pistons are cooled by oil sprays directed at the crown interior face from fixed nozzles from a separate oil pump supplying 75 litres of oil per minute, with thermostatic control set for 60 °C. The lubrication pump has an output of 190 litres per minute with an 80 μ full-flow filter in the circuit.

Injection pump and turbo-charger oil is cleaned separately by a finer, 25 μ filter. The fuel is supplied by the fuel delivery pump at a pressure of 25 lb/in^2 (1.75 kg/cm^2) at a rate of 750 litres per hour, and running on JP-4 fuel or kerosene is therefore possible.

The maximum m.e.p. occurs between 1600 and 1800 rev/min and amounts to 1564 to 1791 lb/in^2 (110 to 126 kg/cm^2). The crankcase is therefore extremely robust and forms a strong foundation for plain bearings which are 4·25 in (108 mm) in diameter and 1·9 in (50 mm) in length. The lead-bronze big end bearings have a diameter of 3·74 in (95·25 mm) and a length of 1·34 in (34·5 mm). An extremely rigid crankshaft is used with webs 1·5 in (40 mm) wide and reciprocating masses are balanced to 85 %. Cylinder centres are 8·97 in (228·6 mm) apart, i.e. 1·56 D. The general layout of the engine is shown in Fig. 393.

3. The Deutz Engine

Deutz engines are being produced in a standardized range comprising single to four-cylinder units as well as six- and eight-cylinder units. The technical data of the range are given in Table 48. The basic cylinder of the range has a bore of 110 mm and a stroke of 140 mm. The swirl chamber of this engine has been mentioned previously and is apparent in the transverse section of

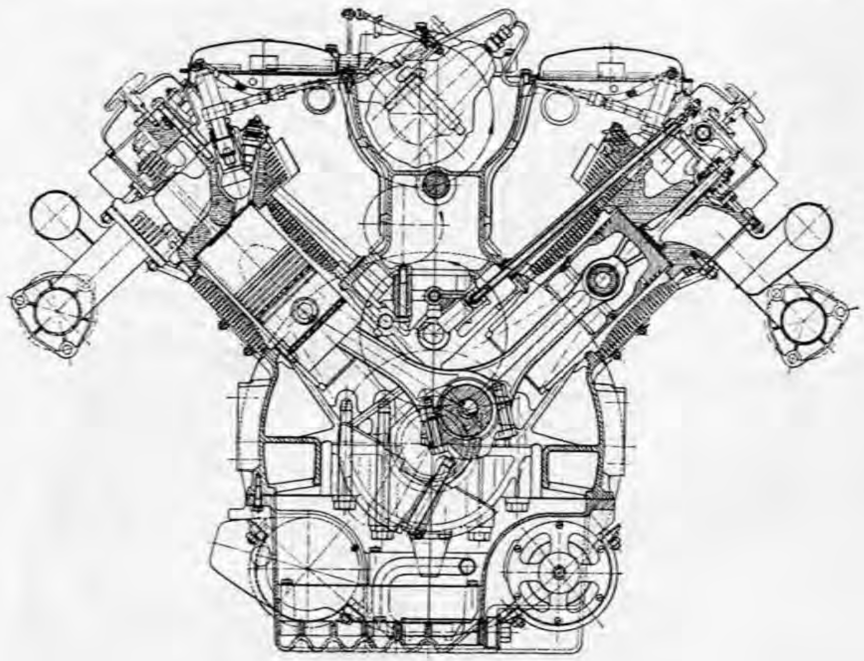


FIG. 394. Transverse section of the eight-cylinder Deutz engine.

the eight-cylinder engine in Fig. 394. An electric glow plug is fitted in the chamber to facilitate starting. The cylinder head was designed with a view to the possibility of conversion for running on substitute fuels. Figs. 216 and 217 show a section through the four-cylinder unit.

The eight-cylinder unit has two banks of cylinders at an angle of 90°, resulting in regular firing intervals which make it possible to attain satisfactory engine balancing. The crankcase is extended at its upper extremity to form a cradle in which the injection pump is mounted, the accessibility of the latter from above thus being very good and the rigidity of the former being enhanced. The crankshaft is seated in plain bearings which somewhat increases the overall length of the engine. The timing gear drive is

Specifications of Deutz Engines Table 48

	Number of cylinders	Displacement litres	Performance b.h.p.	Speed rev/min	Maximum torque kgm	Weight kg
F 1 L 514	1	1·33	15	1650	7	330
F 2 L 514	2	2·66	30	1600	14	385
F 3 L 514	3	3·99	45	1600	21	430
F 4 L 514	4	5·32	60	1600	30	475
F 4 L 514	4	5·32	82	2300	30	475
F 6 L 514	6	7·98	120	2250	45	625
F 6 L 614	6	7·98	120	2250	45	725
F 8 L 614	8	10·64	160	2250	60	850

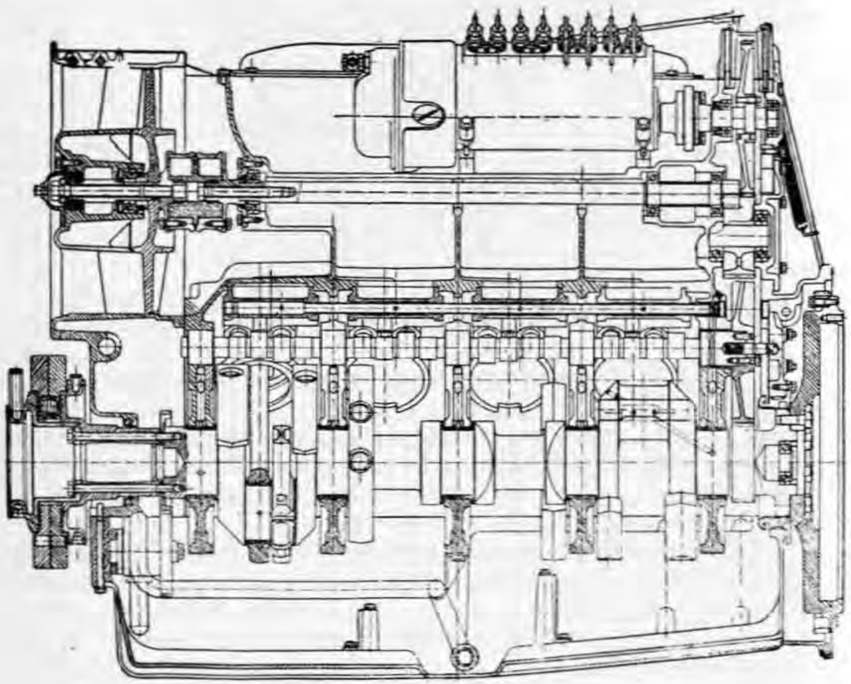


FIG. 395. Longitudinal section of the eight-cylinder Deutz engine.

from the flywheel end i. e. from the vicinity of the nodal point of torsional vibration, which is particularly beneficial for the injection pump drive.

The axial fan is also driven from the flywheel end through gears and a long inclined shaft with flexible coupling. The fan supplies air at a pressure of 210 mm water column and works at an efficiency of 70 %. The consumption of air is 35·3 ft³/b.h.p.h (1 m³ per b.h.p./h). The six-cylinder engine is produced both as an in-line unit and as a V engine (range 64). It is interesting to note that the V engine is heavier by 220 lb (100 kg), the difference being evidently due to the much wider spacing of cylinders.

The connecting rods are mounted side-by-side in pairs on common crankpins (V 8) and inclined big end caps are used. A friction damper of torsional vibrations is fitted to the front end of the crankshaft. Other details are shown in Fig. 395.

4. The Guiberson Engine

This air-cooled nine-cylinder compression-ignition radial engine was originally intended for aircraft use but its characteristics did not prove suitable for that purpose. It did, however, prove suitable as a prime mover for tanks, for which a light air-cooled radial engine was required. The radial engine proved its worth despite its height, which is excessive for use in tanks.

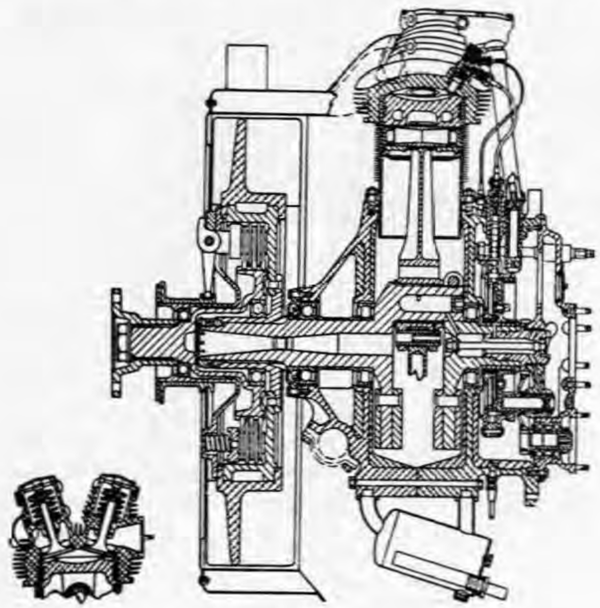


FIG. 396. Section of the Guiberson engine.

The engine has a bore of 130 mm and a stroke of 140 mm, the displacement being 16·74 cm³ and performance 265 b.h.p. at 2260 rev/min, which results in a weight: performance ratio of 2·3 lb/b.h.p. (1·1 kg/b.h.p.), the weight of the engine being 693 lb (290 kg).

Each cylinder is fitted with its own injection pump. The combustion chamber is lentil-shaped and the nozzle is located at its periphery. An axial fan mounted direct on the crankshaft is employed (this being an advantage of the radial engine). The air-cooled Continental radial petrol engine was also employed for the same purpose. The design of the Guiberson engine is shown in Fig. 396.

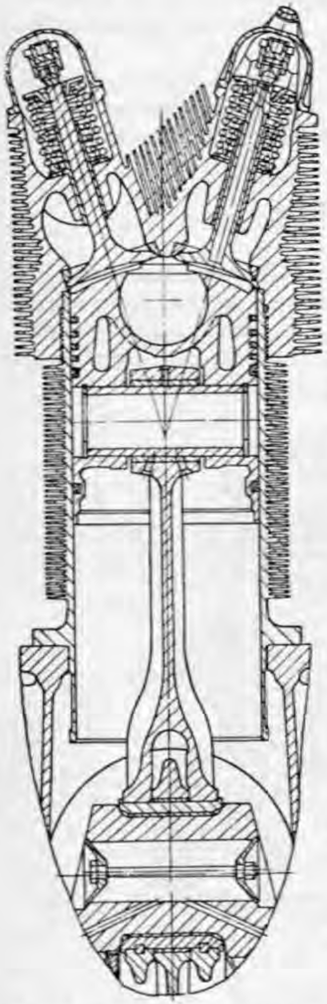


FIG. 397. Section of a cylinder of the MAN compression-ignition engine.

5. The MAN Engine

The 16-cylinder MAN engine is an interesting design of a compact tank engine. Owing to limited space and accessibility the cylinders of the engine are arranged in the unusual H shape. In fact, the unit consists of two superposed flat 8-cylinders with a common crankcase divided vertically in the axis of the crankshaft. The latter are connected by gear wheels.

Three camshafts are fitted on each side of the engine; the top and bottom shaft operate the exhaust valves, while the induction valves of both cylinder banks are actuated by the middle camshaft. Aluminium heads are bolted to the steel cylinders. The head and cylinder fins have a small spacing which is particularly dense in the region of the exhaust valve (see Fig. 397). The forked connecting rods of opposing cylinders are attached to a common crank pin. The general arrangement of the engine is shown in Fig. 398.

The cylinders have a bore of 130 mm, stroke 165 mm, and a compression ratio 15·5 : 1. Fuel is injected directly into a spherical chamber in the piston crown. The performance of this unit with a displacement of 35·1 litres is 690 b.h.p. at 2000 rev/min. The weight of the engine is 3400 lb (1545 kg) which corresponds to a weight: performance ratio of 4·9 lb/b.h.p.

(2·24 kg/b.h.p.). Supercharging by a mechanically driven centrifugal supercharger, which would result in a performance of 900 b.h.p., was also considered.

The development of this engine was not completed.

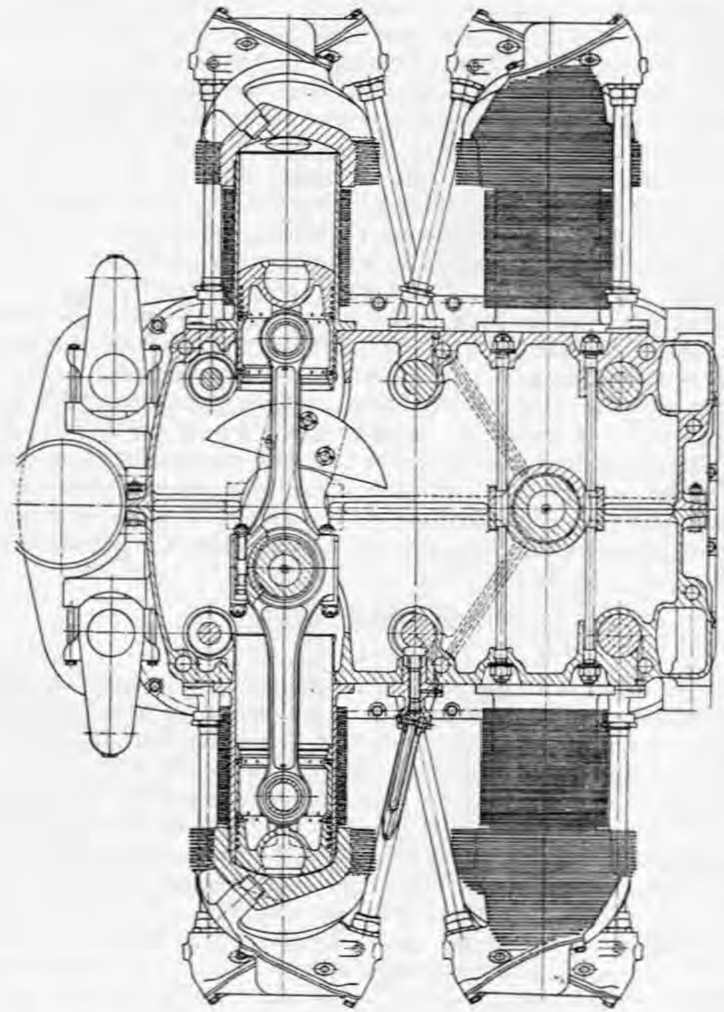


FIG. 398. Transverse section of the sixteen-cylinder MAN engine.

6. The N.A.T.I. Engine

A development of the Soviet Tractor Research Institute (N.A.T.I.) is a two- or four-cylinder unit with compression-ignition. The bore of the two-cylinder DV 13 type engine is 90 mm, stroke 115 mm. The performance of this unit is 14 b.h.p. at 1600 rev/min. The four-cylinder DV 30 unit has a bore of 92 mm and stroke of 120 mm and its performance is 32 b.h.p. at 1600 rev/min and a specific fuel consumption of 0·4 lb/b.h.p. (190 g per b.h.p.h). The four-cylinder engine in its cast-iron version weighs 496 lb (225 kg). A sectional view of the two-cylinder unit is shown in Fig. 399.

The cylinder barrel is an iron casting with cast fins spaced 8 mm apart. The cylinder head is of aluminium with two parallel valves. A small chamber is a free fit in the cylinder head to which it is attached by the injection nozzle carrier. The volume of the injection chamber amounts to 30 % of the total volume of the combustion chamber. Both the exhaust and inlet manifolds are on one side of the head, the other side being occupied by the injection chamber. Cooling air is supplied by radial fans with V belt drive from the crankshaft. One fan with a double impeller is employed for each pair of cylinders, each cylinder being served by a separate section of the impeller. Thus uniform distribution of air over all cylinders is achieved. The main air stream is directed against the cylinder heads. A single axial fan for the four-cylinder engine is being developed with a centrifugal oil-cleaner mounted in the fan hub. The four-cylinder engine requires the supply of 1589 ft^3 (45 m^3) of cooling air per b.h.p. per hour. The maximum temperature of the cylinder head at full load is 210 °C and that of the upper cylinder section 180 °C. Cylinder centre-to-centre distance is 5·5 in (140 mm). An engine with cylinder centres the same distance apart but with a bore increased to 105 mm is being tested.

7. The ONAN Engine

This air-cooled compression-ignition engine is highly commendable for use as a stationary unit as the amount of work involved by valve grinding, decarbonizing and attention to the ignition accessories is slight. Engines of this type are therefore used also for relatively low specific performance.

This engine is of the flat-twin type with bore and stroke of 88·9 mm and a performance at 12 b.h.p. at 1800 rev/min. The majority of components are aluminium castings, which results in an overall weight of only 220 lb (100 kg) for the engine. Overall dimensions of the engine: height 18·6 in (476 mm), width 29·3 in (745 mm), length 24 in (610 mm).

The crankcase is cast integral with the cylinder barrels which are fitted with cast iron liners. The connecting rods, with inclined caps, can be withdrawn through the cylinder bore. Indirect fuel injection is employed with a chamber formed directly in the aluminium head. The camshaft is housed above the crankshaft and incorporates cams for the two injection pumps. The large oil

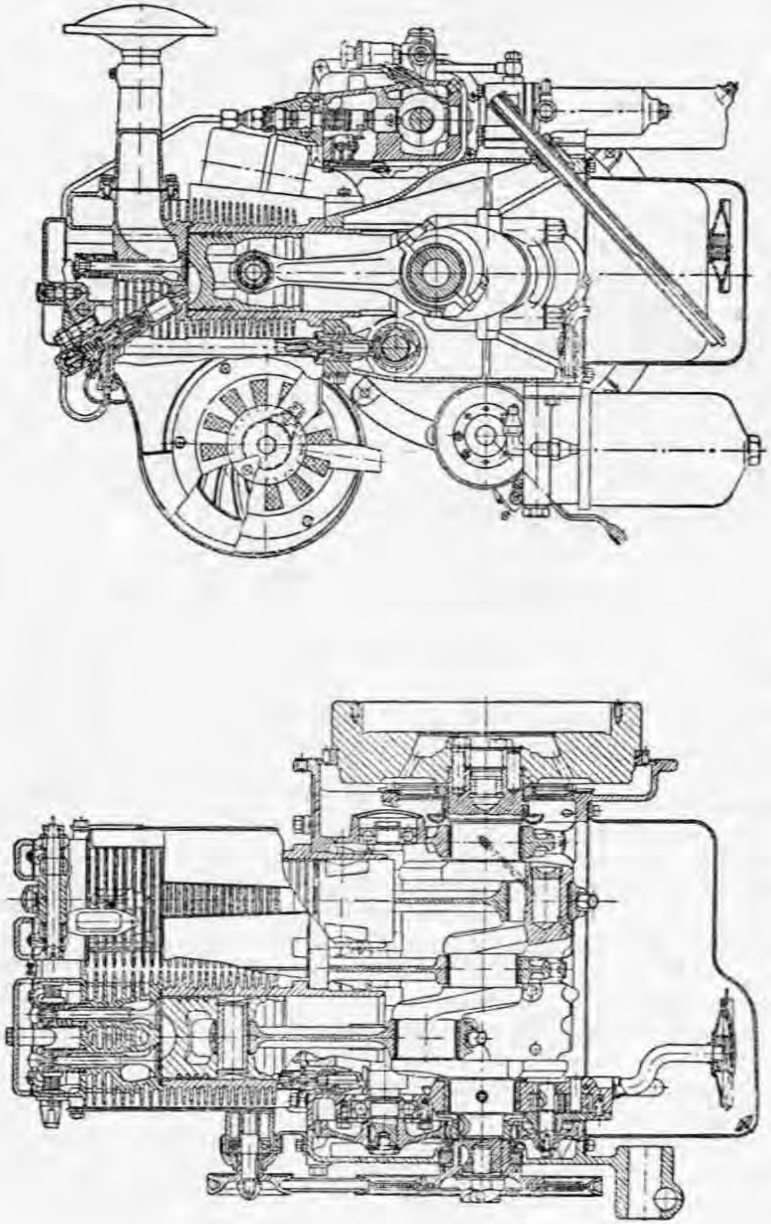


FIG. 399. Longitudinal and transverse sections of the NATI engine.

tank under the engine forms the base of the unit. The large oil cleaner in the tank is worth noting. Air cleaners are oil-filled. This engine is chiefly intended for electric generator units (Fig. 400).

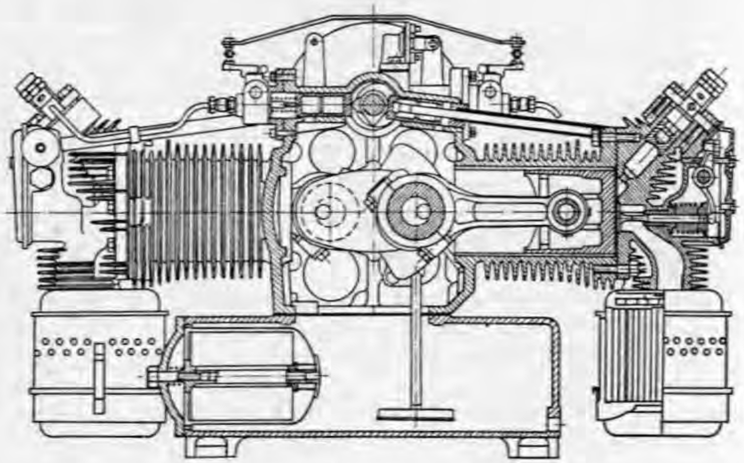


FIG. 400. Transverse section of the Onan engine.

8. Petter Engines

A new range of air-cooled engines in production comprises 1, 2, 3 and 4 cylinder units with a bore of 76·2 mm and stroke 76·2 mm with a displacement of 347·4 cm^3 per cylinder, i.e. 1390 cm^3 for the four-cylinder engine. A performance of 5 b.h.p. at 3000 rev/min per cylinder for continuous running for 12 hours is stated, the mean effective pressure thus being only 61 lb/in^2 (4·32 kg/cm^2). The mean piston speed under such running conditions is only 25 ft/s (7·62 m/s). Specific fuel consumption at maximum output is 0·5 lb/b.h.p.h (233 g/b.h.p.h). The weight to performance ratio of the single cylinder engine is 48 lb/b.h.p. (22 kg/b.h.p.), for two cylinders it is 32·9 lb/b.h.p. (14·9kg/b.h.p.), for three cylinders 28·9 lb/b.h.p. (13·25 kg/b.h.p.) and for the four-cylinder engine 24·7 lb/b.h.p. (11.35 kg/b.h.p).

The crankpins of these engines are hardened and the bearing metal is of lead-bronze. Cylinders are of cast iron with cast fins. The cylinder heads are also of cast iron with parallel valves reaching outward slightly beyond the cylinder wall. The larger inlet valve is shrouded for better turbulence of the incoming charge of air inside the cylinder. The rocker box is of independent construction and it is attached to the cylinder head by two bolts which also attach the bracket of the rocker pivot supported at one end only. One of these bolts also retains the aluminium rocker box cover.

The hemispherical combustion chamber is formed in the piston crown which has cavities for the valves. The upper piston ring is chromium plated, the second and third have oblique faces and the fourth is an oil ring. The fifth piston ring, which also serves for oil control, is below the gudgeon pin boss. Oil for lubrication of the rocker gear is pumped through an outside pipe and the rate of flow is controlled by a needle valve. Valve timing is as follows: IO – 10°, IC – 50°, EO – 45°, EC – 15°.

The injector nozzles are of the twin jet type and are inclined by 30°. The compression ratio is 18·5 : 1 and injection begins 29° before T.D.C.

The cooling air stream is brought to the inlet side of the engine first and the cowling is quickly detachable. A radial fan is employed on the single-cylinder engine, whereas the multi-cylinder types are fitted with axial fans with inlet guide wheels and eight-blade impellers of 7·9 in (203 mm) diameter. The fan is driven by two V belts. The crankshaft-to-blower gearing is 1 : 1·79 in the case of the four-cylinder engine and the static pressure at 5375 rev/min of the fan is only 30 mm water column and the volume of air 1590 m^3 per hour. The input to one blower is 1·15 b.h.p. The sectional view of this engine is in Fig. 401.

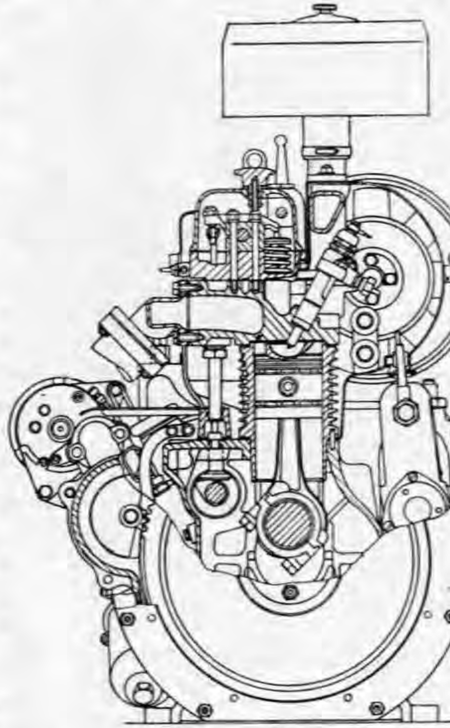


FIG. 401. Transverse section of the Petter engine.

9. The Praga Engine

The Praga Automobile Works manufacture a six-cylinder air-cooled engine for its V 3 S off-the-road truck. This engine is derived from the Tatra range with its 110 mm bore and 130 mm stroke. With a displacement of 7412 cm^3, the performance of the engine is 98 b.h.p. at 2100 rev/min. Maximum torque is 260 ft-lb (36 mkg.) The compression ratio is 16·5 : 1. Valve timing: IO – 4°, IC - 48°, EO - 42°, EC - 10°. The end of fuel injection occurs at 13° to 15° before T.D.C. A diagrammatic sectional view of this engine is shown in Fig. 402.

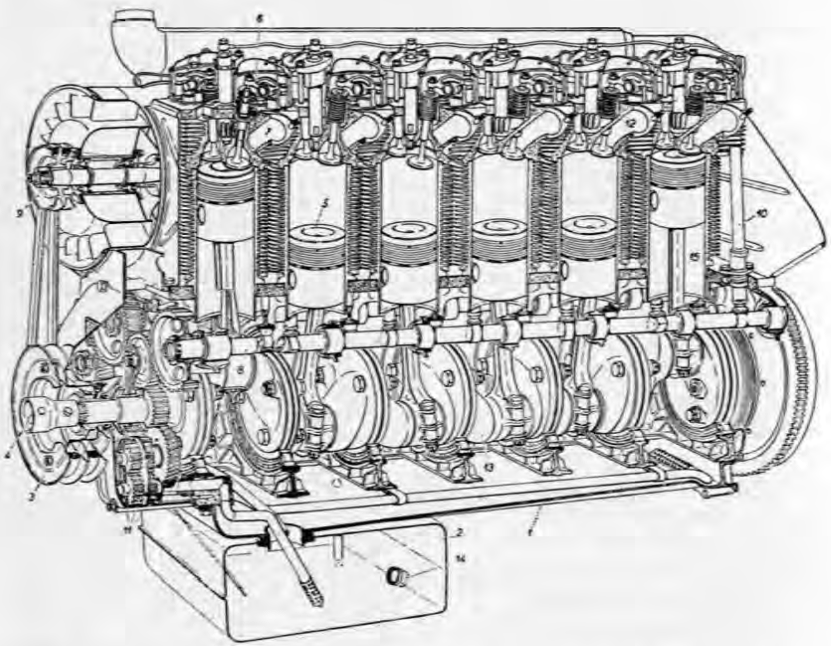


FIG. 402. Sectional view of the Praga engine mounted in the V3S lorries.

10. The Simmering Engine

The general layout of this air-cooled compression-ignition engine is influenced by the fact that it was intended as a prime mover for tanks. In order to keep the engine light and small, the "flattened X" layout was adopted for the sixteen-cylinder engine. With a bore of 135 mm and a stroke of 160 mm, the engine displacement is 36·8 litres. The performance is 750 b.h.p. at 2000 rev/min. The net weight of the engine without the exhaust gas turbine driven supercharger is 4077 lb (1850 kg) and the gross weight 4959 lb (2250 kg).

It is apparent from the sectional drawing of the engine shown in Fig. 403, that each crankpin bears a master rod with three connecting rods. Cast lead-bronze is used for the 3·5 in (90 mm) diameter big end bearing. The crankpins are hardened. Steel cylinders are screwed into aluminium cylinder heads and

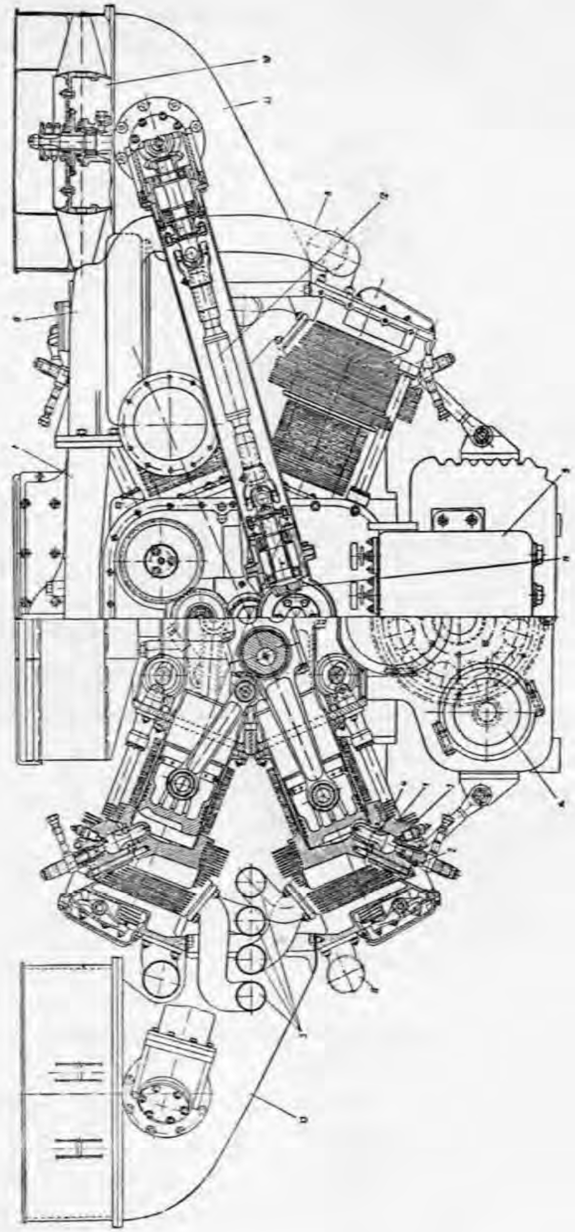


FIG. 403. Transverse section of the Simmering engine.

attached to the crankcase by flanges and bolts. There are two valves in each cylinder head (shown in the longitudinal section in Fig. 404), the inlet valve port being of 60 mm diameter, the exhaust 56 mm. Valve lift is equal (10·5 mm) for both valves. The cylinder centre lines are 8·5 in (215 mm) apart.

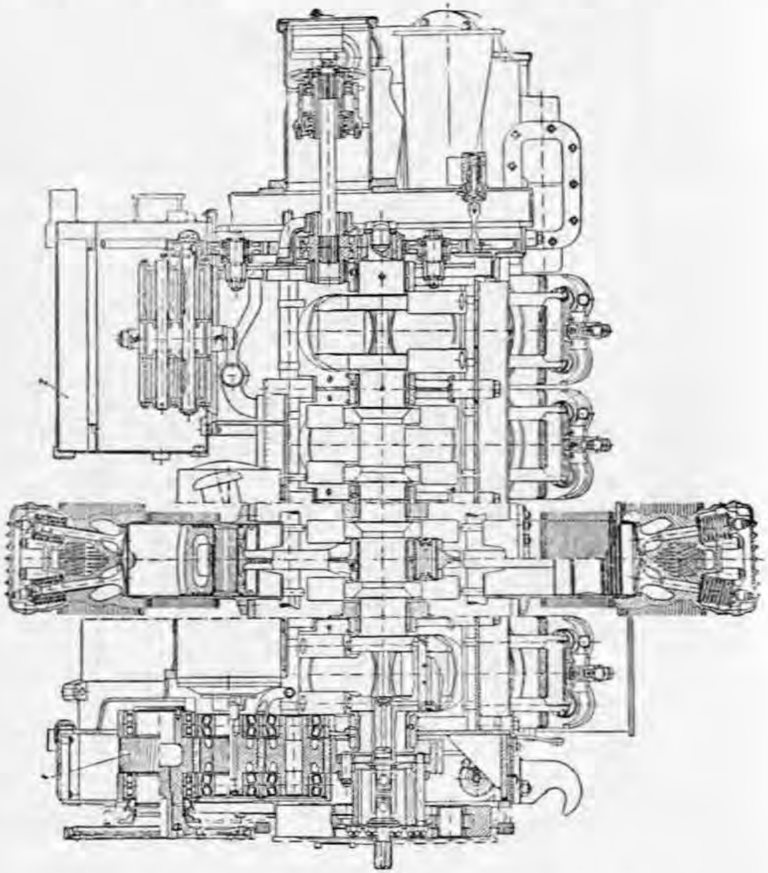


FIG. 404. Longitudinal section of the Simmering engine.

The engine is supercharged by two exhaust gas turbine driven superchargers and valve timing overlap is therefore as much as 135°. Valve timing: IO – 62°, IC – 35°, EO – 45°, EC – 72°. Two fans are used to draw air over the engine and their positioning has been subjected to the specific space limitations of an armoured vehicle. Transmission to both fans is through bevel gears and each has a performance of 212 ft^3 (6 m^3).

A special chamber for indirect fuel injection is employed in the design as is apparent in the transverse section of the engine. Glow plugs are used to facilitate starting. The compression ratio is 14·5 : 1 and the specific fuel consumption is 0·40 to 0·49 lb/b.h.p. h (200 to 225 g/b.h.p. h). The rate of fuel supply per stroke decreases with increasing engine speed and thus suitable torque characteristics are obtained and cooling is aided at maximum engine speed. The engine dimensions are: height 35·4 in (900mm), length 45·2 in (1150 mm), width over cylinder heads 53 in (1350 mm), overall width 106 in (2700 mm). In order to reduce gearbox size and weight, the intermediate gear between the engine and gearbox increases transmission revolutions so that an engine speed of 2000 rev/min corresponds to a transmission speed of 3000 rev/min.

11. The S.L.M. Engines

The Schweizerische Lokomotiv- und Maschinenfabrik produces a standardized range of air-cooled engines with 110 mm bore and 140 mm stroke. The production programme includes engines in various layouts from single-cylinder to four-cylinder types and four, eight and twelve-cylinder units. Units with up to and including six cylinders are of in-line design, eight- and twelve-cylinder engines are of the flat or V type.

The eight-cylinder engine of 10·64 litres has a continuous performance of 120 b.h.p. at 2000 rev/min. Direct fuel injection is employed, the combustion chamber shape being shown in Figs. 405 and 406. The cylinder head is of cast iron and a copper gasket separates it from the cylinder. The crankshaft runs in plain aluminium bearings.

Air is drawn into the cylinders from the ducting

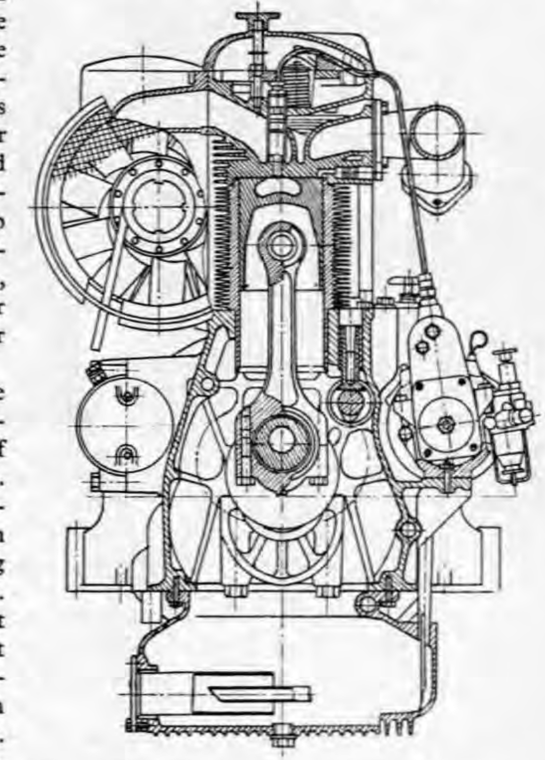


FIG. 405. Transverse section of the S.L.M. engine.

between the fan and engine, thus the slight supercharging effect of the fan is utilized. The oil cooler is mounted in this pressurized zone.

A notable feature of S.L.M. flat engines is the close spacing of cylinders which are only 5·9 in (150 mm) apart, despite the use of plain bearings; this amounts to 1·36 D with a 110 mm bore. Axial fans with two impellers are also used for the flat engines.

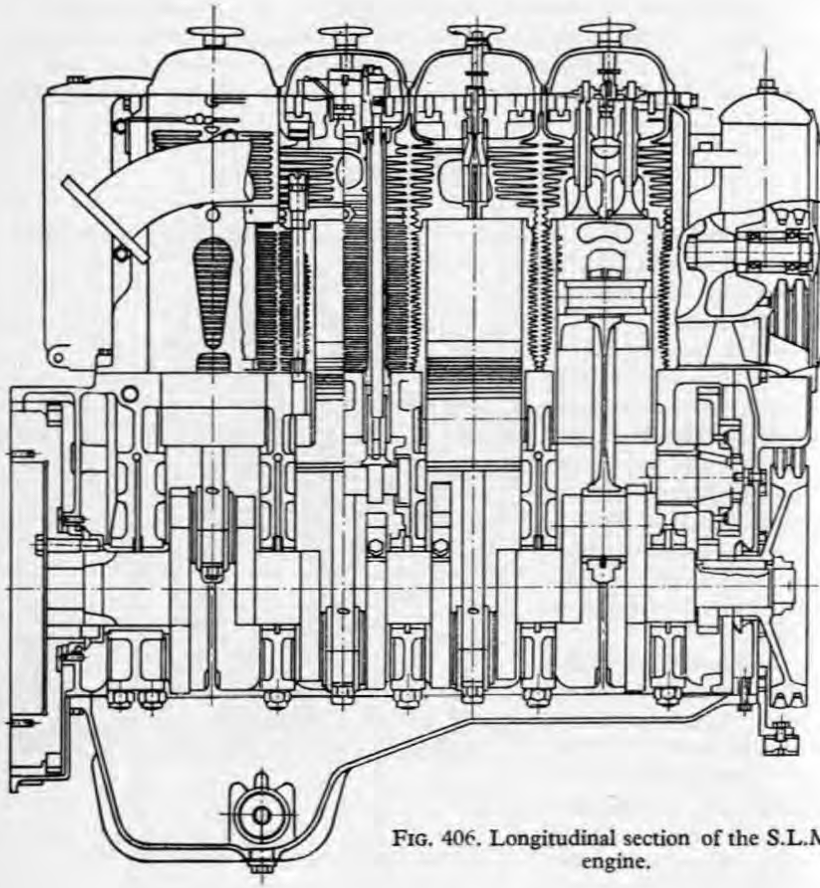


FIG. 406. Longitudinal section of the S.L.M. engine.

12. The Tatra 111 Engine

This twelve-cylinder air-cooled compression-ignition engine has a bore and stroke of 110 mm and 130 mm respectively and a displacement of 14 825 cm³. The cylinders are arranged in two banks at an angle of 75°. This com-

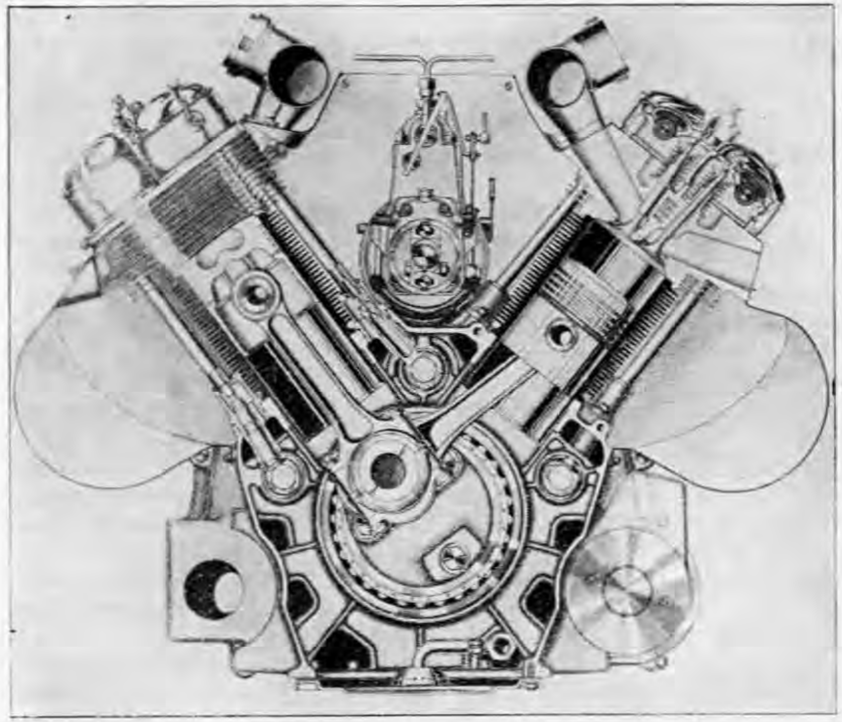


FIG. 407. Transverse section of the Tatra 111 oil engine.

paratively wide angle was chosen in order to provide sufficient space for the injection pump between both banks and also to make it possible to use the same angle for eight- and twelve-cylinder engines of the same basic range.

This twelve-cylinder engine is well balanced, as an in-line six-cylinder engine is inherently well balanced and the angle at which two such units are mounted is of no consequence. The firing intervals are however not regularly timed 60° apart, but 75° and 45°. As their succession is very rapid in the case of the twelve-cylinder engine, this slight irregularity is no disadvantage.

Similar conditions exist in the eight-cylinder layout in which, however, a secondary couple of both the four-cylinder in-line banks remains unbalanced. A crankshaft with crankpins spaced 180° must therefore be employed.

Direct fuel injection is used in the case of the Tatra 111 engine. The original hemispherical combustion chamber has been replaced by one of toroidal shape with high turbulence (Fig. 407). A maximum performance of 220 b.h.p. is supplied at 2250 rev/min.

The version fitted in 10-ton trucks produces 175 b.h.p. at 1800 rev/min and that used in railcars 160 b.h.p. at 1500 rev/min. The dry weight of the complete engine is 2173 lb (986 kg).

The firing order is 1-8-5-10-3-7-6-11-2-9-4-12. A compression ratio of 17 : 1 is employed. The injection pump with a fixed end-of-injection setting is fitted with 8 mm diameter plungers. A five-jet injection nozzle is employed, the jet diameter being 0·25 mm.

In order to facilitate starting, an injection timing override control is fitted and the injection pump drive incorporates a special dog clutch, permitting the pump to be cranked with the engine at rest – an arrangement which facilitates starting at very low temperature. The injection (opening) pressure is adjusted to 2844 lb/in^2 (200 kg/cm^2).

The six-throw crankshaft is mounted in seven oversize roller bearings. As is apparent in the longitudinal section of Fig. 408, oil enters the crankshaft via the plain bearing at the front of the engine. As the pressure valve is in the last section of the crankshaft (adjacent to the flywheel), a large volume of oil is constantly flowing through the crankshaft, not permitting the temperature of oil reaching the last big-end bearing to rise excessively. Plain bearings are employed for the big-end with a lead-bronze lining of the steel shells. Oil is brought under pressure through passages drilled in the crankcase to the bearings of the camshafts from where it flows through the valve lifters, push rods and rockers to the valve. It returns from the cylinder heads through the push rod covers to the crankcase.

Two gear pumps are employed in the dry sump lubrication system, one, the scavenging pump, transferring the oil from the sump under the flywheel to the oil tank, the other, lubricating pump, drawing the oil from the oil tank and forcing it to the lubricated parts.

The timing gear comprises three camshafts. They and the fuel injection pump are driven from the crankshaft by spur gears. The engine is cooled by two axial fans for which the drive was originally arranged from the camshafts by roller chain and torsion bar, but which are belt driven from the front end of the crankshaft in the last modified type. The V belt also drives the reciprocating compressor mounted on the front engine cover. In the new version of the engine, dynamos are mounted in the hubs of the axial fans; their dashboard charging lights also signal the correct operation of the fans.

The induction manifold on the original engine was on the outward side of the cylinder banks, but on new engines the exhaust manifold faces outward. This modification resulted in a uniform distribution of temperature over the cylinder heads which are of aluminium alloy.

The tunnel-type crankcase is of cast iron. The engine is overhung on a flange at the flywheel end which connects it with the gearbox.

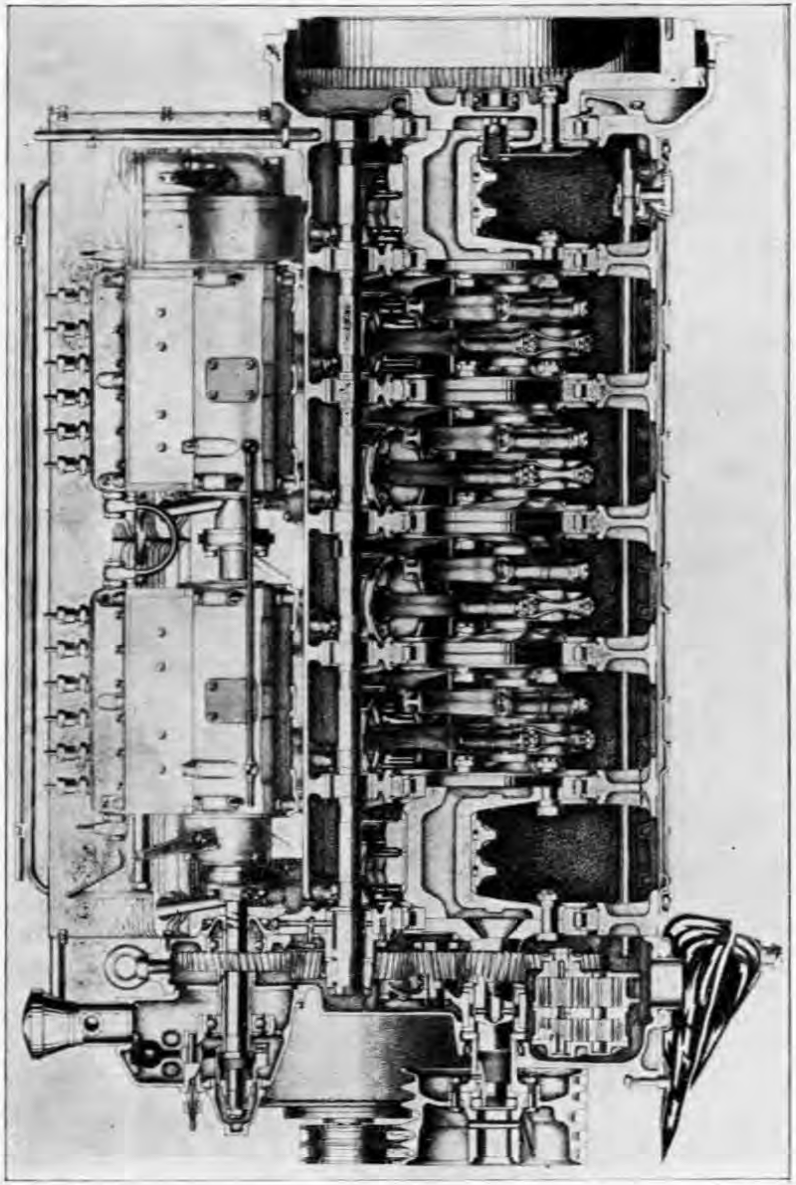


Fig. 408. Longitudinal section of the Tatra 111 oil engine.

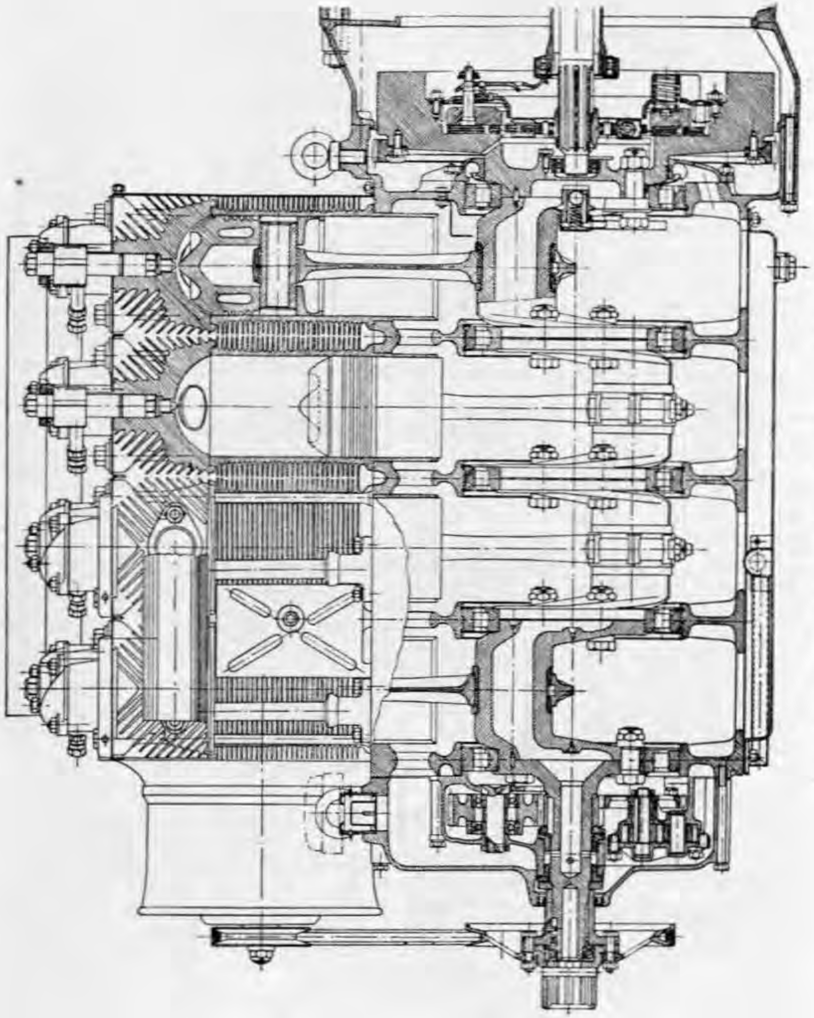


FIG. 409. Longitudinal section of the Tatra 114 engine.

13. The Tatra 114 Engine

This in-line four-cylinder engine is derived from the twelve-cylinder Tatra 111 engine the basic parts of which are used. Complete cylinders, valve gear and crank mechanism are interchangeable with the larger unit. With a bore of 110 mm and a stroke of 130 mm, the displacement is 4940 cm^3 and the

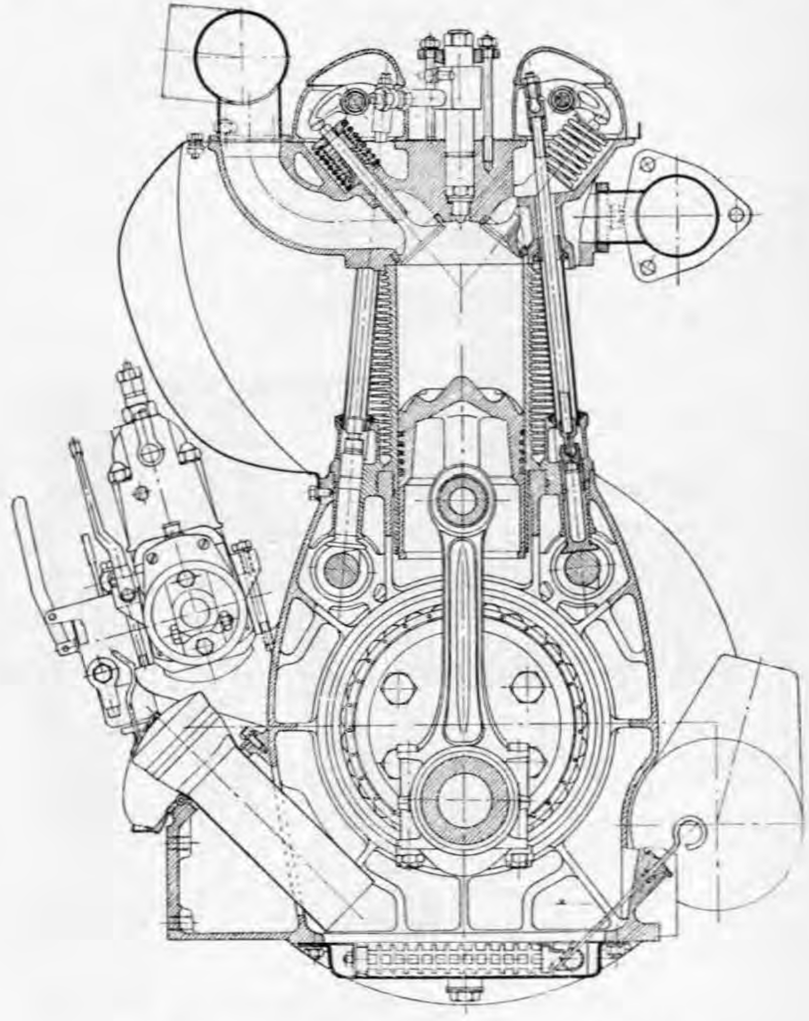


FIG. 410. Transverse section of the Tatra 114 engine.

performance at 2000 rev/min is 65 b.h.p. The weight of the engine including accessories is 1113 lb (505 kg).

The tunnel crankcase is of cast iron and the crankshaft is seated in five roller bearings.

The cylinders are spaced, as in the V engines, 6·1 in (155 mm) and thus sufficient room is available for the big-end bearings (Fig. 409). A single oil pump driven by gears from the front end of the crankshaft, is used in the wet sump lubrication system.

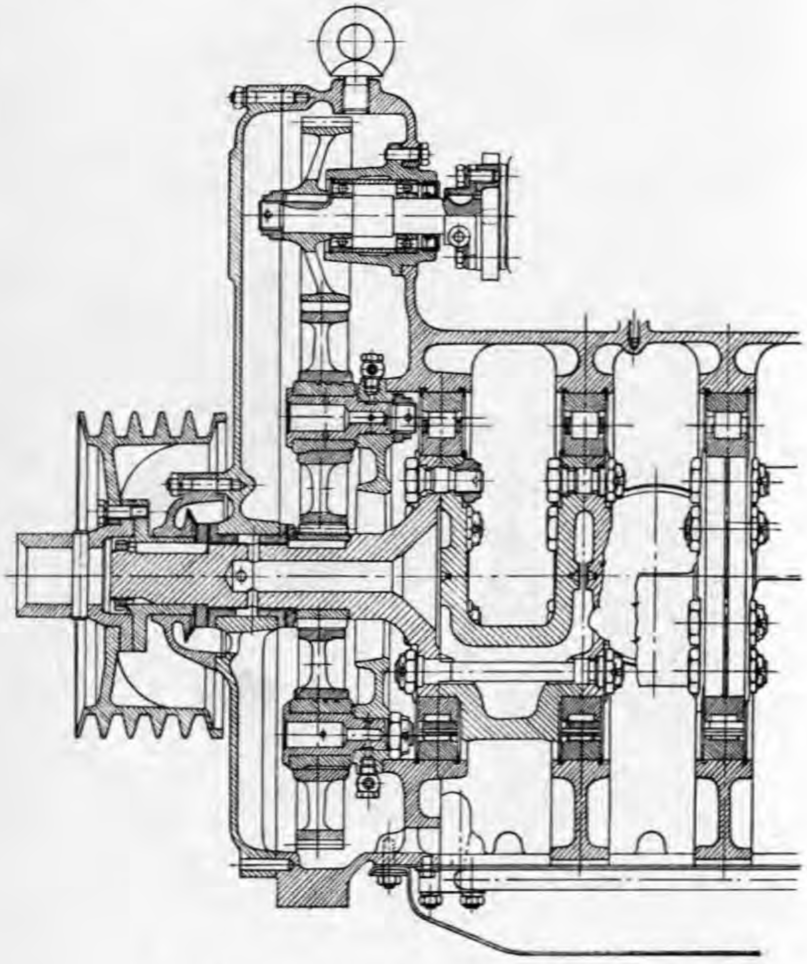


FIG. 411. Longitudinal section of the Tatra 116 engine.

Cooling is by an axial fan for which the drive is taken off the front end of the crankshaft and transmitted by a V belt. The injection pump is mounted on the same side of the engine as the fan. Fig. 410 shows a transverse section of this engine in its older form.

14. The Tatra 116 Engine

This six-cylinder engine has a bore of 120 mm and stroke of 150 mm. The cylinders are arranged in three horizontally opposed pairs (this being a flat engine) and each bank is served by its own axial fan driven from the crankshaft through V belts (Figs. 265, 411). The displacement of this engine is 10 200 cm^3 and its performance 130 b.h.p. at 2000 rev/min. The dry engine weighs 1543 lb (700 kg).

The one-piece tunnel crankcase is an iron casting. The valve gear of each bank is actuated by a camshaft driven by a toothed gear from the crankshaft. The overhead valves are parallel and the injector is located between them. Return oil flows back to the crankcase from the cylinder heads through the push rod covers. The valve gear is lubricated in the usual manner through hollow push rods and drilled rockers which supply oil to the flattened ball rocker pads.

Fuel is injected directly into the toroidal cavity in the piston crown. The injection pump is attached to the upper side of the crankcase and is driven – as are the camshafts – by gear wheels from the front end of the crankshaft. A gear-type oil pump is employed and it is driven from the front end of one camshaft. The supply of oil is stored in the crankcase sump. Oil is cleaned by a multi-edge filter and a pressure warning light is provided on the dashboard.

Both inlet and exhaust manifolds are upswept against the direction of cooling air flow. The six-throw crankshaft is supported in seven roller bearings. Large openings in the crankcase base permit the withdrawal of connecting rods. Cylinder centres are 7·81 in (200 mm) apart. The engine is designed with a view to mounting it under the frame of a seven-ton truck.

15. The New Tatra Range

The new range of Tatra compression-ignition engines has a bore of 120 and stroke of 130 mm. Units with 1, 2, 4, 6, 8 and 12 cylinders are being produced. The technical data of these engines are given in Table 49. The main difference between the new range and the old lies in the use of a single camshaft for timing both in-line and V engines, the use of a single fan also for V engines and the control of cooling by a variation of fan speed by means of a fluid coupling in the fan drive.

The cylinder centre lines are 6·1 in (155 mm) apart, i.e. 1·29 D. Parallel valves are employed in the aluminium cylinder head and their plane is at a considera-

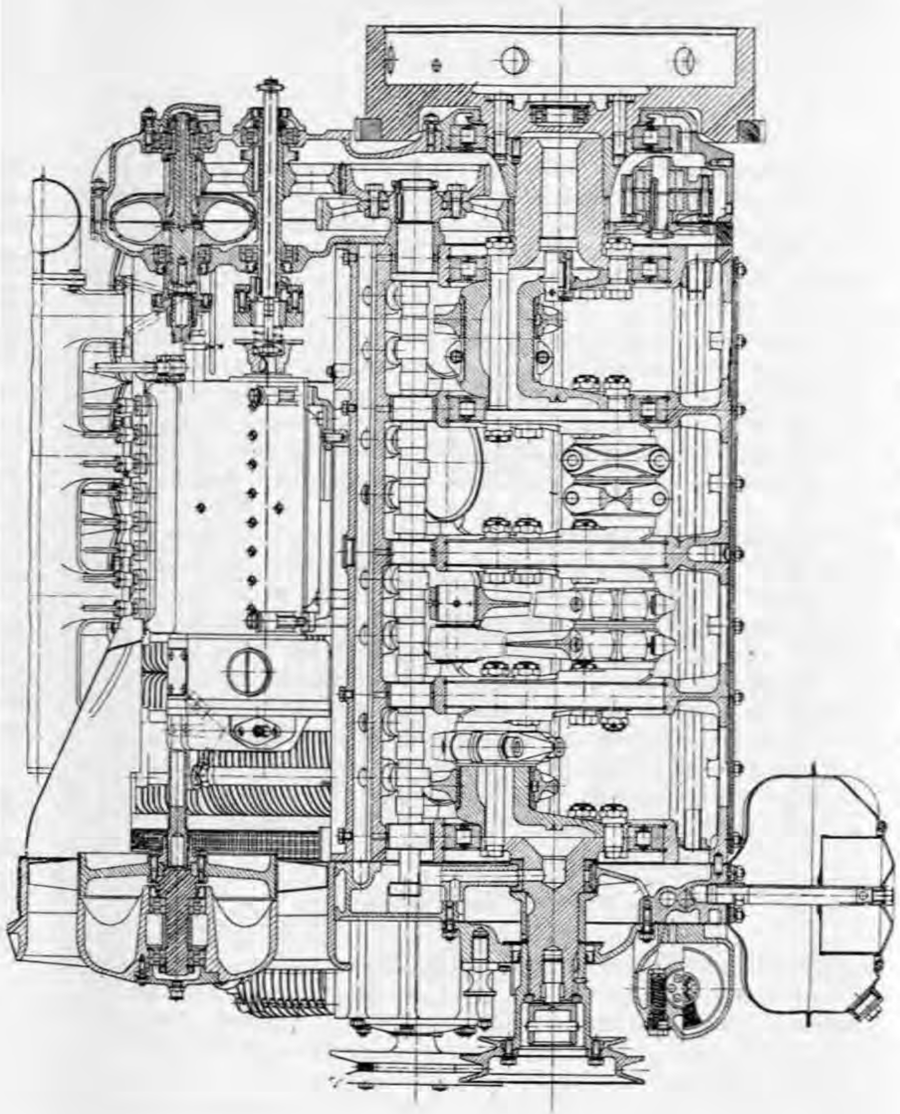


FIG. 412. Longitudinal section of the Tatra 928 engine.

Table 49

Air-Cooled 120 mm Bore Tatra Engines

Type	924	926	928	930
Number of cylinders	4	6V	8V	12V
Bore, mm	120	120	120	120
Stroke, mm	130	130	130	130
Engine displacement, cm^3	5,876	8,814	11,752	17,628
Compression ratio	16·5	16·5	16·5	16·5
Performance, b.h.p.	80	132	180	270
Speed, rev/min	2,000	2,000	2,000	2,000
Unit performance, b.h.p./litre	13·6	15·0	15·3	15·3
M.e.p., kg/cm^2	6·1	6·75	6·9	6·9
Mean piston velocity, m/s	8·7	8·7	8·7	8·7
Weight of dry engine, kg	450	560	595	900
Unit weight, kg/b.h.p.	5·6	4·25	3·3	3·3
Length of engine, mm	970	911	1,066	1,163
Width of engine, mm	654	958	958	958
Height of engine, mm	970	994	994	994

ble inclination from the crankshaft axis which arrangement has the advantage of a sufficient air flow area despite the close grouping of the cylinders. Both exhaust and inlet pipes in the cylinder head are kept short. The injection nozzle is almost concentric with the cylinder and a central combustion chamber in the piston crown is employed. The injection pump, camshaft and blower are driven from the flywheel end of the crankshaft, where torsional vibration of the crankshaft is at its lowest. The fan is driven through gears and a fluid coupling which regulates the speed of the fan. In a cold engine the coupling is not filled with oil and slip occurs. As the engine warms up, a thermostat placed in the stream of cooling air coming from the cylinders opens the oil inlet valve to the coupling, which becomes filled with oil and the slip is reduced to a low percentage. The fan then supplies the full amount of cooling air. This fluid-coupling control is reliable in operation and no wear occurs.

The rear end of the crankshaft also drives the dual oil pump, one section of which pumps the oil from the flywheel end of the crankcase to the oil tank in the front part of the engine and the other supplies oil under pressure from the tank, through a cooler and cleaner to the engine. A dog clutch is incorporated in the fuel injection pump drive permitting turning of the injection pump while the engine is stationary. When starting from cold it is thus possible to inject, prior to turning the engine, a sufficient amount of fuel oil into the cylinders to dilute the oil on the cylinder walls and thus to provide more favourable conditions for a cold start. The arrangement is apparent in the longitudinal section of the Tatra 928 engine in Fig. 412. The transverse section through this engine (Fig. 413) shows the arrangement of the injection pump and blower drive.

The crankshaft is built-up from steel castings and is housed in roller bearings. The tunnel-type crankcase forms a rigid foundation for the two banks of cylinders which are arranged at an angle of 75°. The same basic parts are employed for the derived six-cylinder engine (Fig. 414) and the twelve-cylinder unit (Fig. 415). It is intended to fit exhaust gas turbine superchargers to the eight- and twelve-cylinder engines in order to raise their performance. Highly favourable weight to power ratios will thus be obtained. The four-cylinder unit, for operation under extremely unfavourable conditions such as in combined harvesters, is equipped with a rotary gauze cooling air inlet filter which is self-cleaned by compressed air supplied by a reciprocating compressor (Fig. 117).

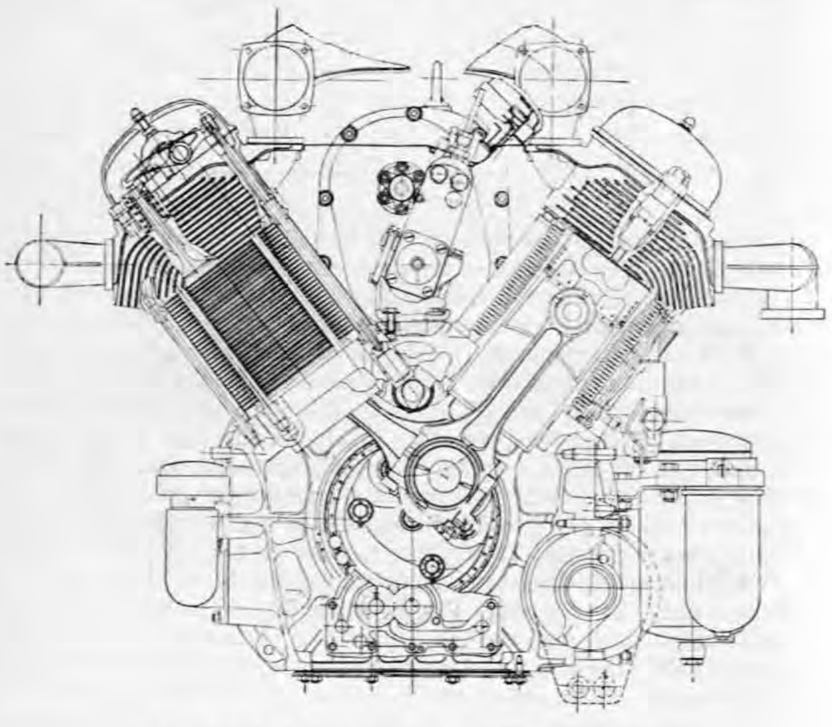


FIG. 413. Transverse section of the Tatra 928 engine.

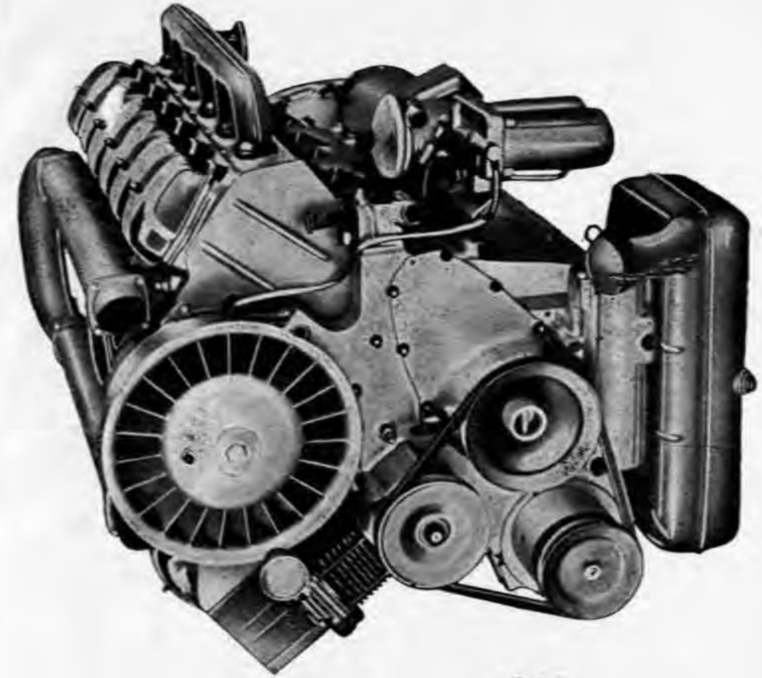


FIG. 415. Twelve-cylinder Tatra 930 engine.

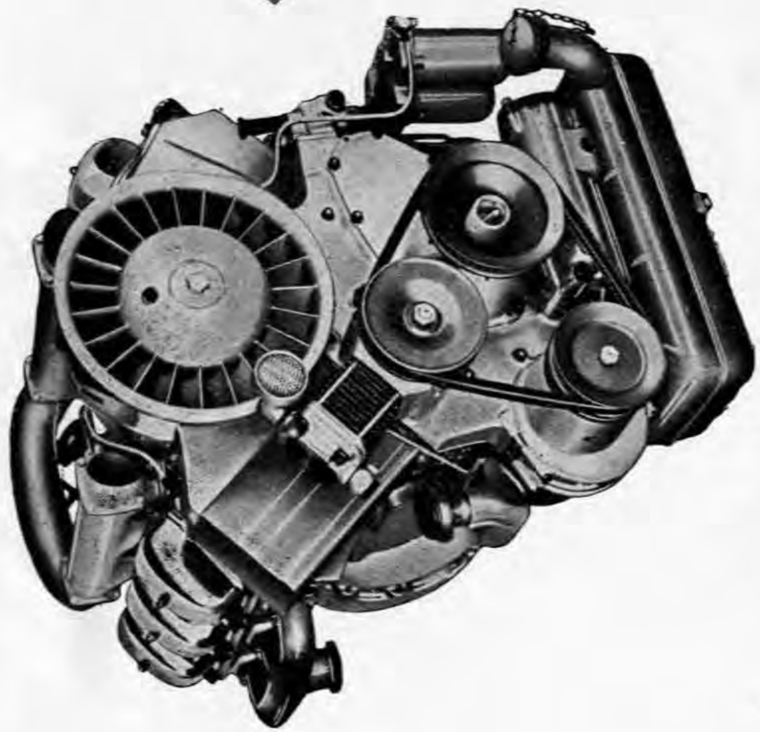
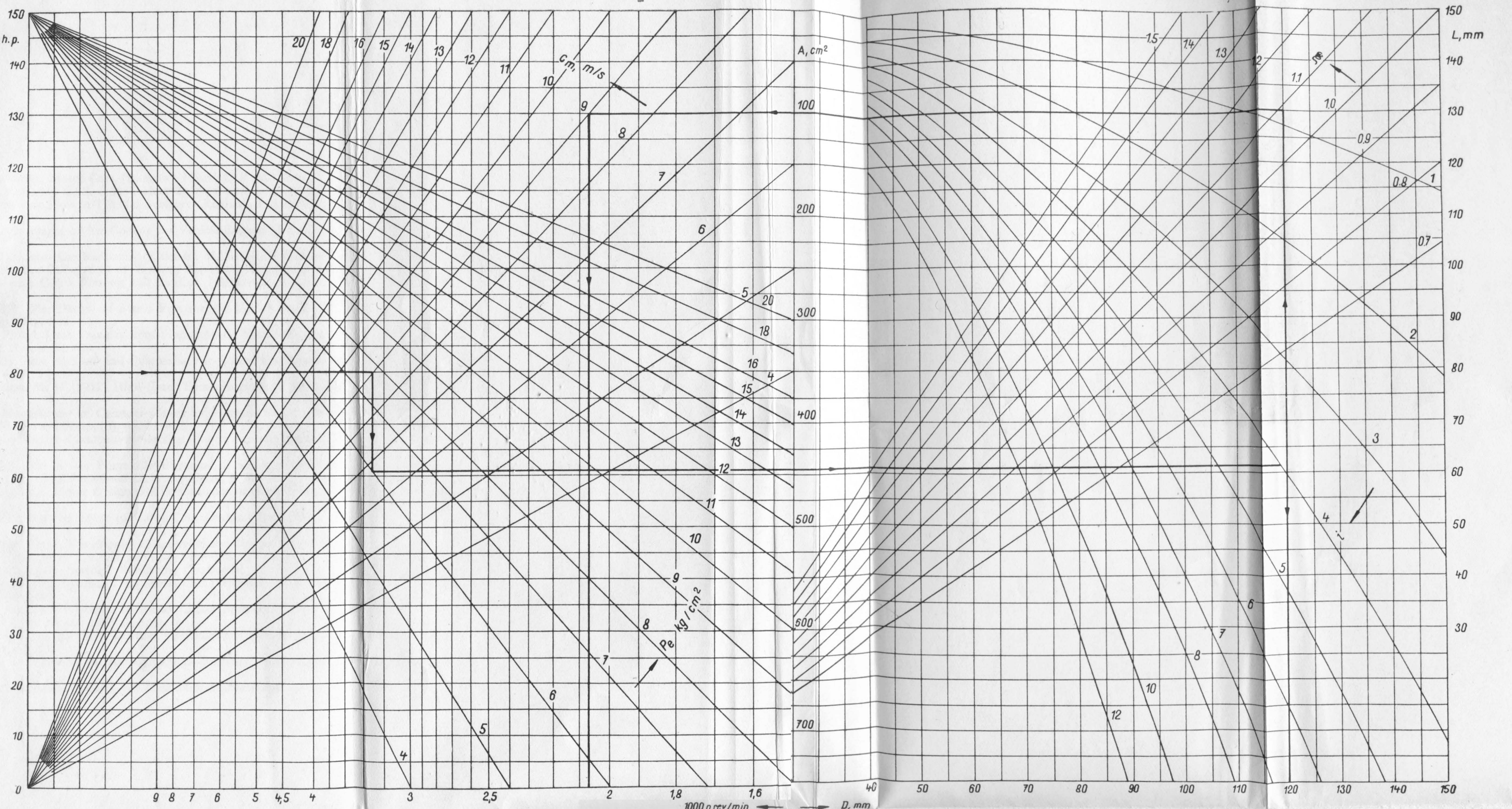


FIG. 414. Six-cylinder Tatra 926 engine.



Nomogram for the Rapid Calculation of Internal Combustion Engines

The nomogram represents the graphical solution of equation 41 written in the modified form

$$\text{b.h.p.} = \frac{A \cdot c_m \cdot p_e}{300}$$

The basic nomogram is complemented by an alignment chart for the solution of the relationship between total piston area on the one hand, and bore, stroke and number of cylinders on the other.

The scales, radiating straight lines and curves of the nomogram are marked by the following symbols for the pertinent variables:

b.h.p. = engine performance,
c_m = mean piston speed, m/s,
p_e = mean effective pressure, kg/sq. cm,
n = engine crankshaft speed, rev/min,
i = number of cylinders,
D = bore, mm,
L = stroke, mm,
ξ = stroke/bore ratio.

Example of use. Let us calculate the dimensions of a compression-ignition engine with the following values given: b.h.p. = 80, c_m = 9.0 m/s, p_e = 6.0 kg/sq. cm, number of cylinders i = 4.

The sequence of operations is shown by the bold line which starts from point 80 on the extreme left vertical scale (b.h.p.). Draw a horizontal line from this point until it intersects the inclined c_m = 9 line. From the intersection draw a vertical line down to the inclined p_e = 6 line. Continue with a horizontal line and read its intersection with the central vertical scale A = 444 sq. cm which is the total effective area of all pistons. Extend the horizontal line and read its intersection with the i = 4 curve as 119 mm for the bore. Select the nearest standard bore, i.e. 120 mm. The intersection of the vertical line drawn through the point D = 120 with the inclined ξ = 1.1 line read on the extreme right hand scale corresponds to stroke L = 132 mm. The stroke/bore ratio ξ = 1.1 has been conveniently selected beforehand. Take the nearest standard stroke of L = 130 mm, and draw a horizontal line through this point until it intersects the inclined c_m = 9 line. The position of this intersection read on the horizontal scale for n corresponds to an engine speed of 2,080 rev/min.

The calculation may be carried out also the other way round, i.e. with the following values given: D = 120 mm, L = 130 mm, n = 2,000 rev/min, and b.h.p. = 80. The course of this calculation is not drawn in the nomogram. The intersection of the vertical n = 2,000 with the horizontal L = 130 determines the value c_m = 8.7 m/s. The intersection of the vertical D = 120 with the curve i = 4 (selected) determines the total piston area A = 452 sq. cm. Draw a vertical line through the intersection of the horizontal b.h.p. = 80 with the inclined c_m = 8.7 line. This vertical intersects the horizontal A = 452 in the point p_e = 6.1 kg/sq. cm.

Dimensions calculated with the aid of the nomogram make possible a rapid orientation during the first phase of the design work. For engines of an output higher than 150 b.h.p. (top of the left hand scale), dimensions are computed likewise, but for one half of the performance and half the number of cylinders.

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