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.

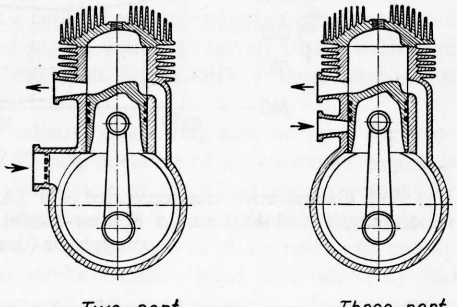


FIG. 331. A two-port and three-port two-stroke engine.

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.

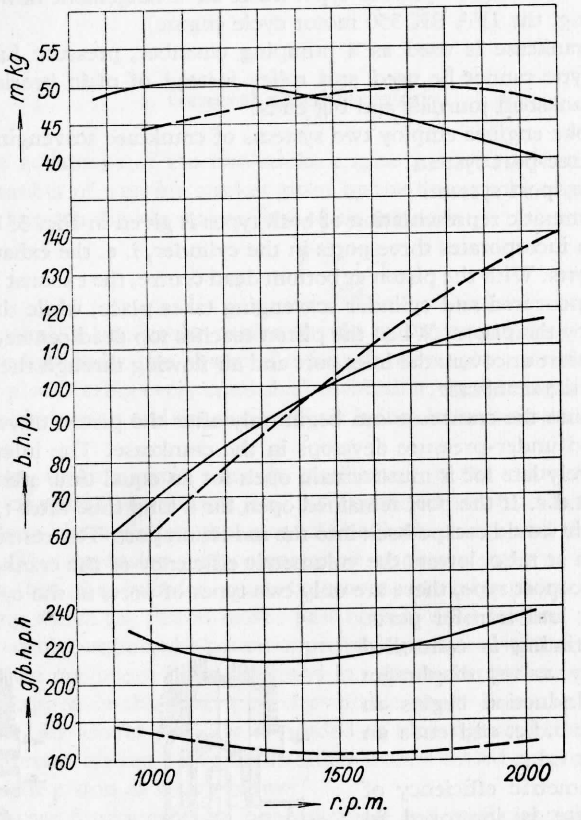


FIG. 332. Comparative characteristics of a JAZ 204 engine with a Roots blower as supercharger (full line) and a Krauss-Maffei engine equipped with a centrifugal compressor (dash line).

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.

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.

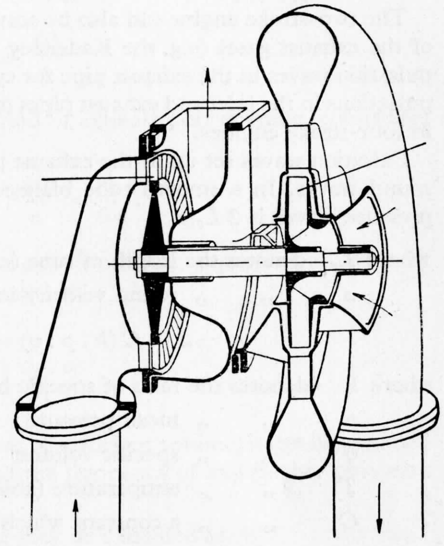


FIG. 333. Exhaust gas turbo-supercharger.

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

$$t = \frac{2\,L_e}{a} \tag{100}$$

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.

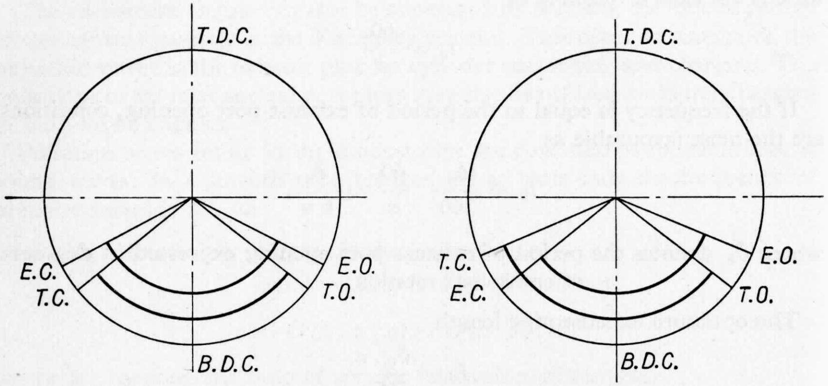


FIG. 334. Symmetric (left hand) and asymmetric (right hand) valve gear of a two-stroke engine.

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

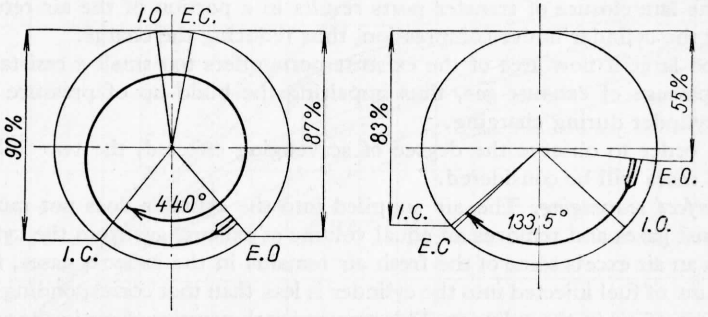


FIG. 335. Difference in timing of a two-stroke and four-stroke engine.

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

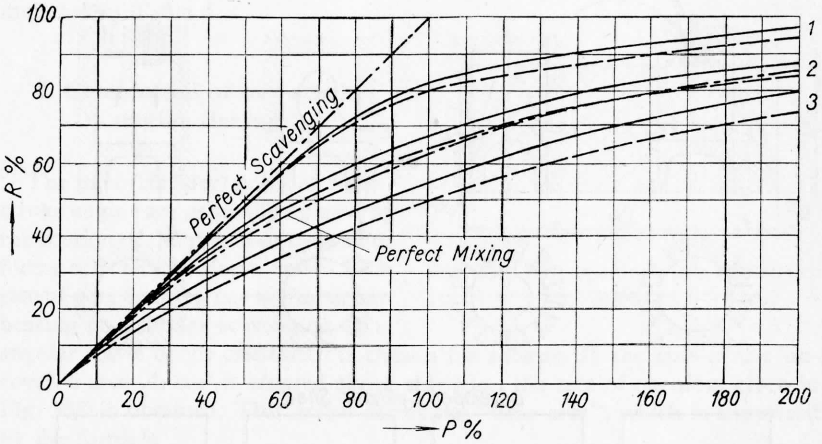


FIG. 336. Different methods of scavenging two-stroke engines.

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

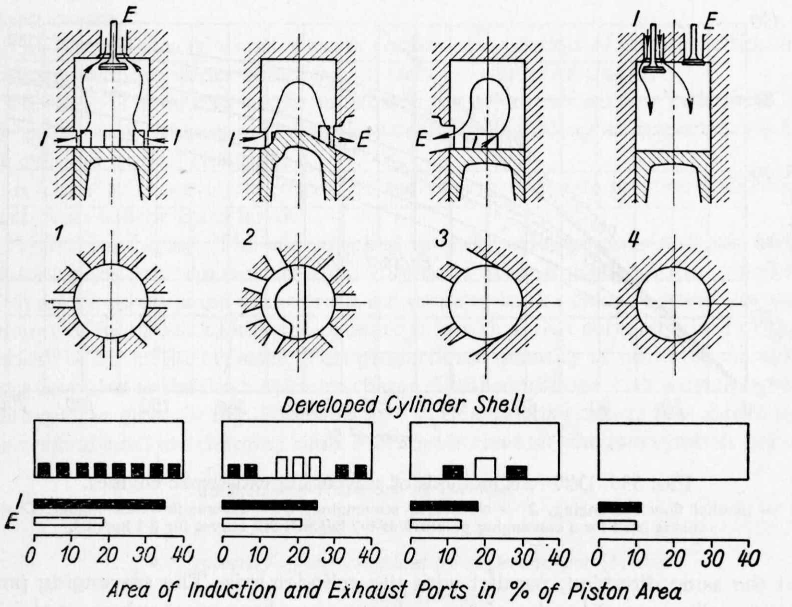


FIG. 337. Comparison of flow cross-section areas for different methods of scavenging.

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

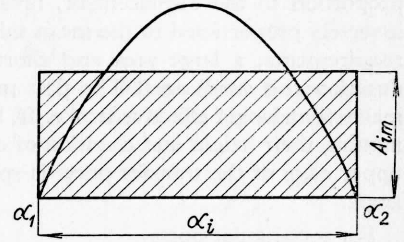


FIG. 338. Port openings in a two-stroke engine.

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

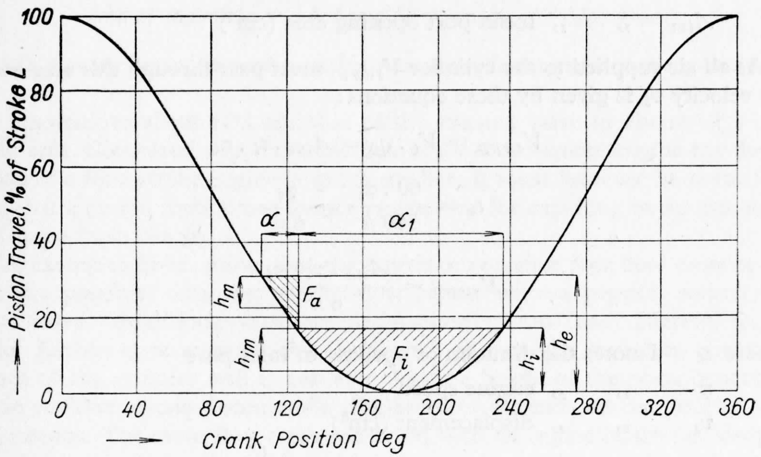


FIG. 339. Restriction of flow area by piston travel at bottom dead centre.

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 $\%^\circ$ (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.

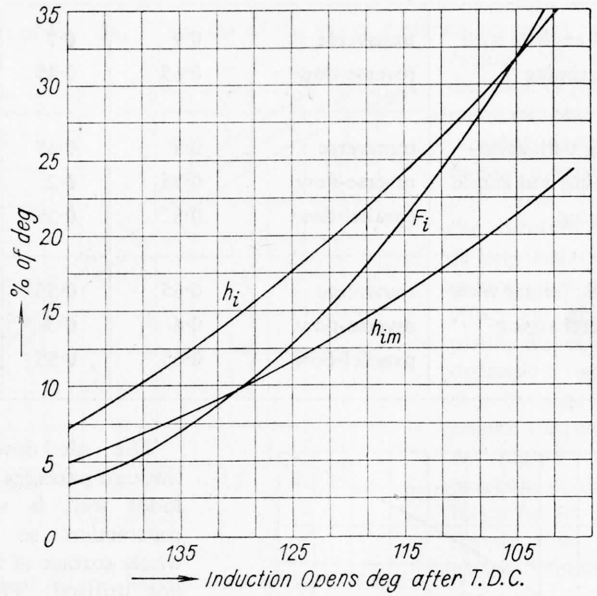


FIG. 340. Nomogram for the calculation of the F_i area valid for a connecting rod length of 4.5 R. The values of h_i and h_{im} are expressed in % of the stroke.

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 β = 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

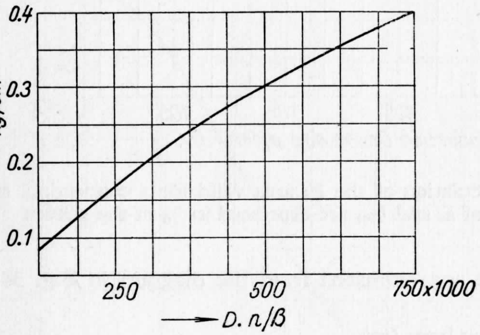


FIG. 341. Chart for the calculation of scavenging pressure.

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.

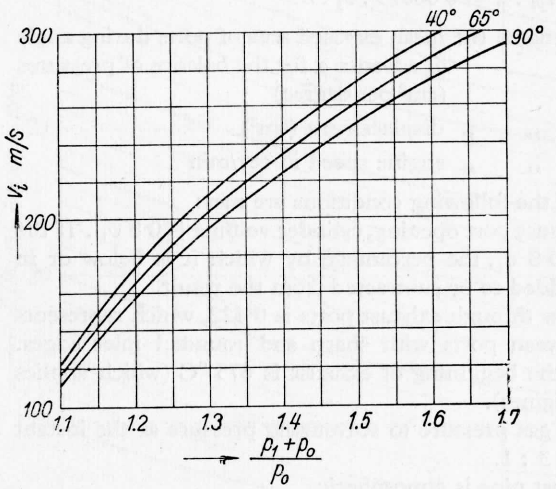


FIG. 342. Chart for the determination of the v_i/S ratio from the pressure ratio $(p_1 + p_0)/p_0$ for different temperatures of the scavenging air.

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.

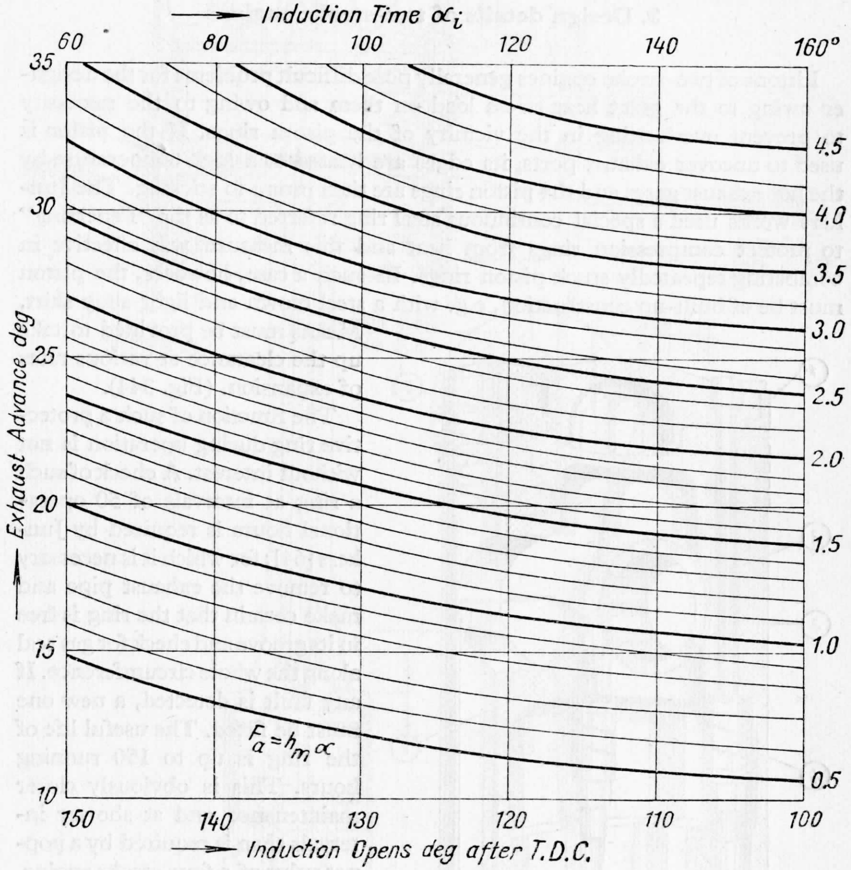


FIG. 343. Chart for the determination of the exhaust advance.

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.

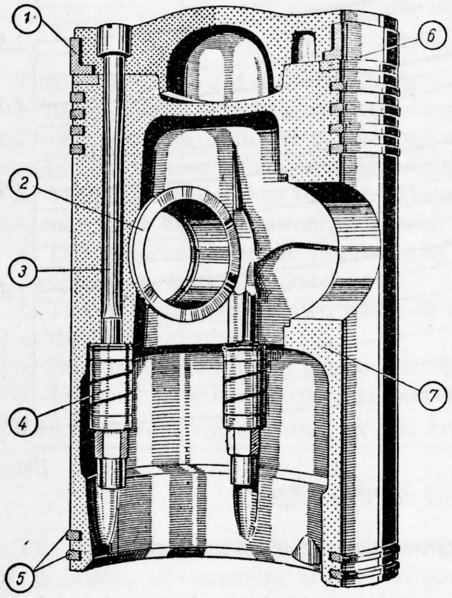


FIG. 344. Piston of the two-stroke Junkers aircraft oil engine.

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.

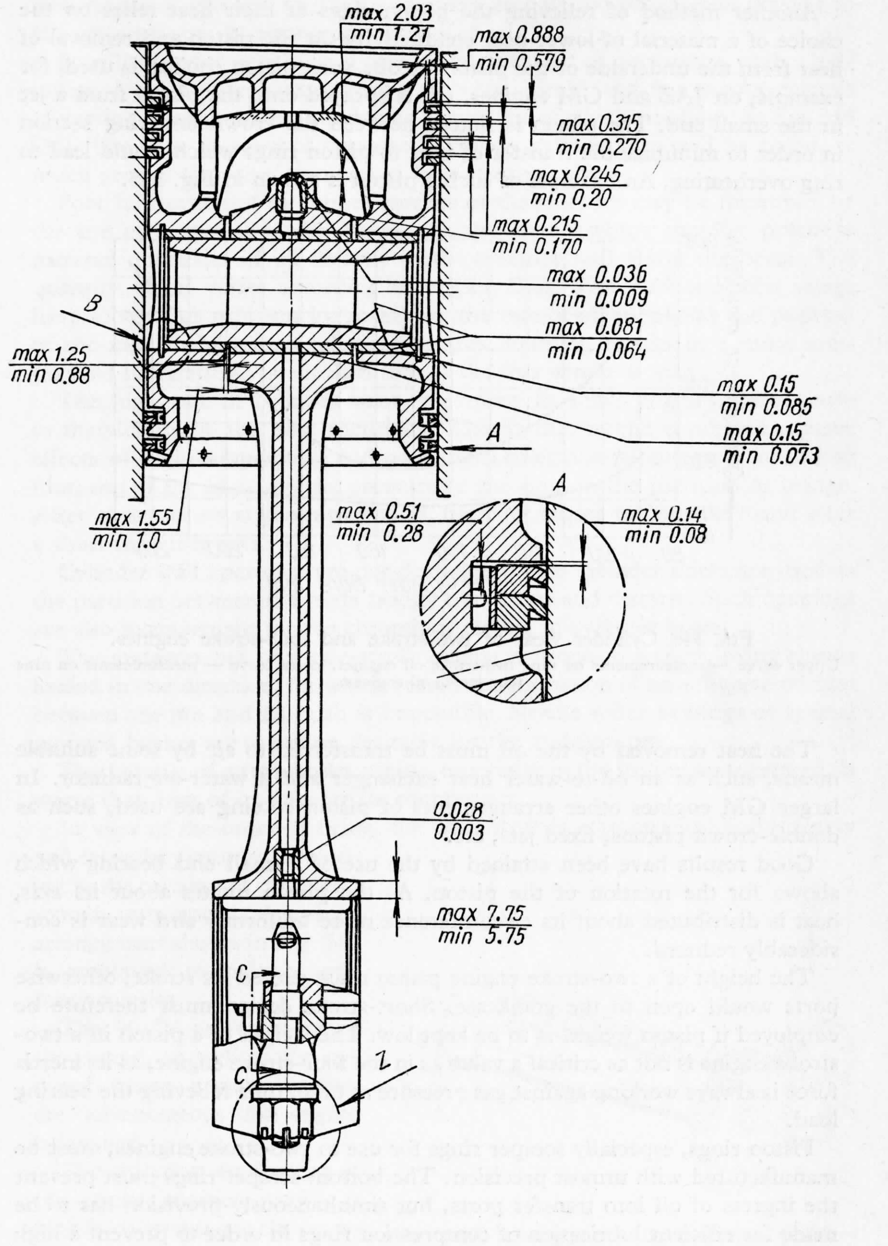


FIG. 345. Piston of the Soviet two-stroke JAZ 204 engine.

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.

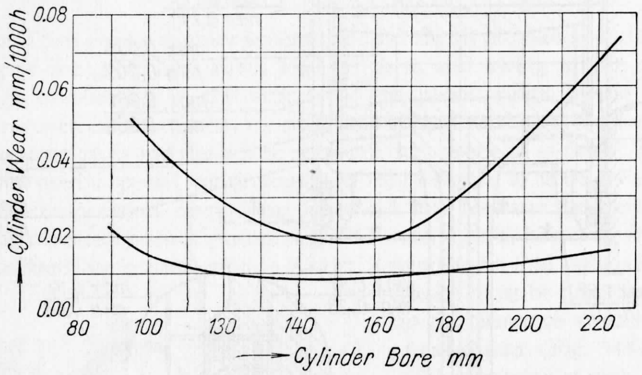


FIG. 346. Cylinder wear of two-stroke and four-stroke engines.
Upper curve — measurements on nine two-stroke oil engines, lower curve — measurements on nine four-stroke oil engines.

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.

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 re-

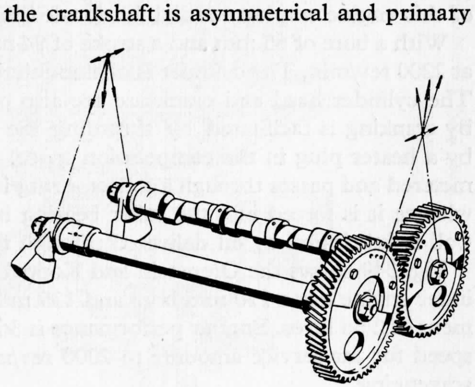


FIG. 347. Arrangement for balancing primary couples in a two-stroke engine.

quired. 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.

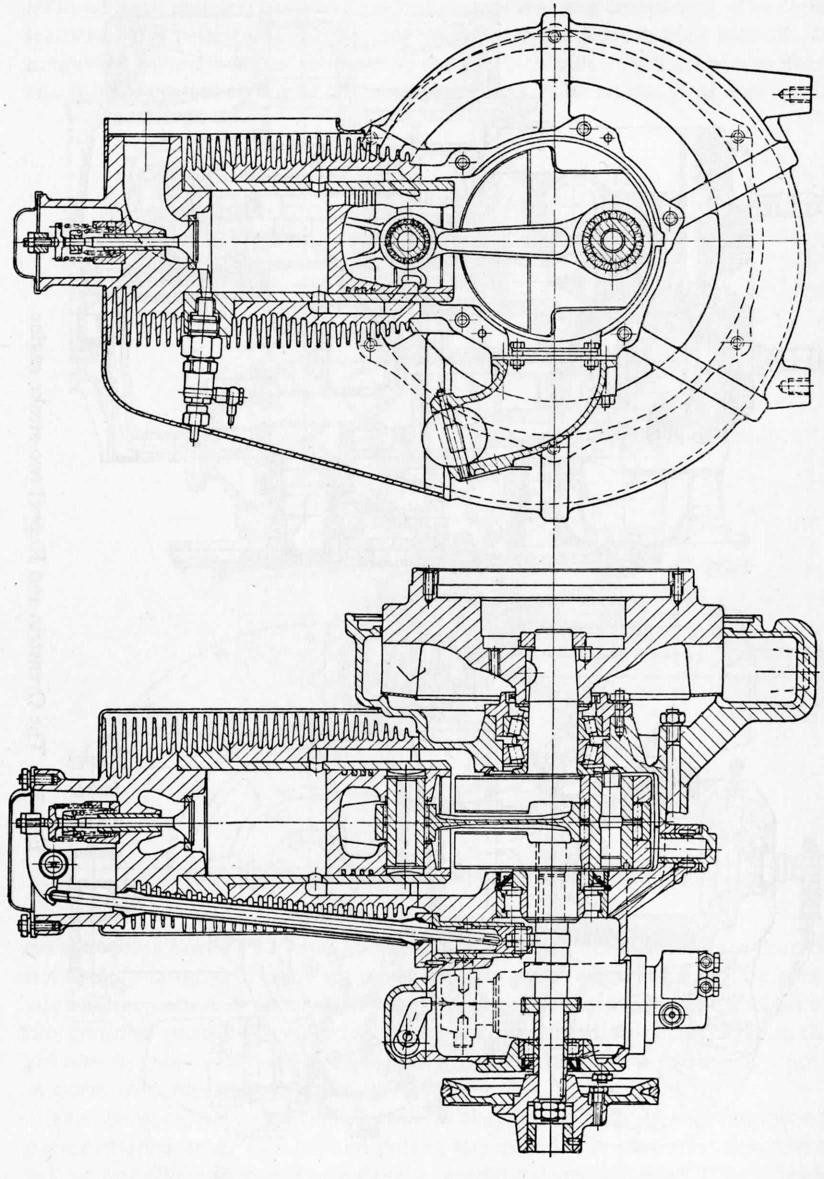


FIG. 348. Transverse and longitudinal section of the two-stroke Stihl engine.

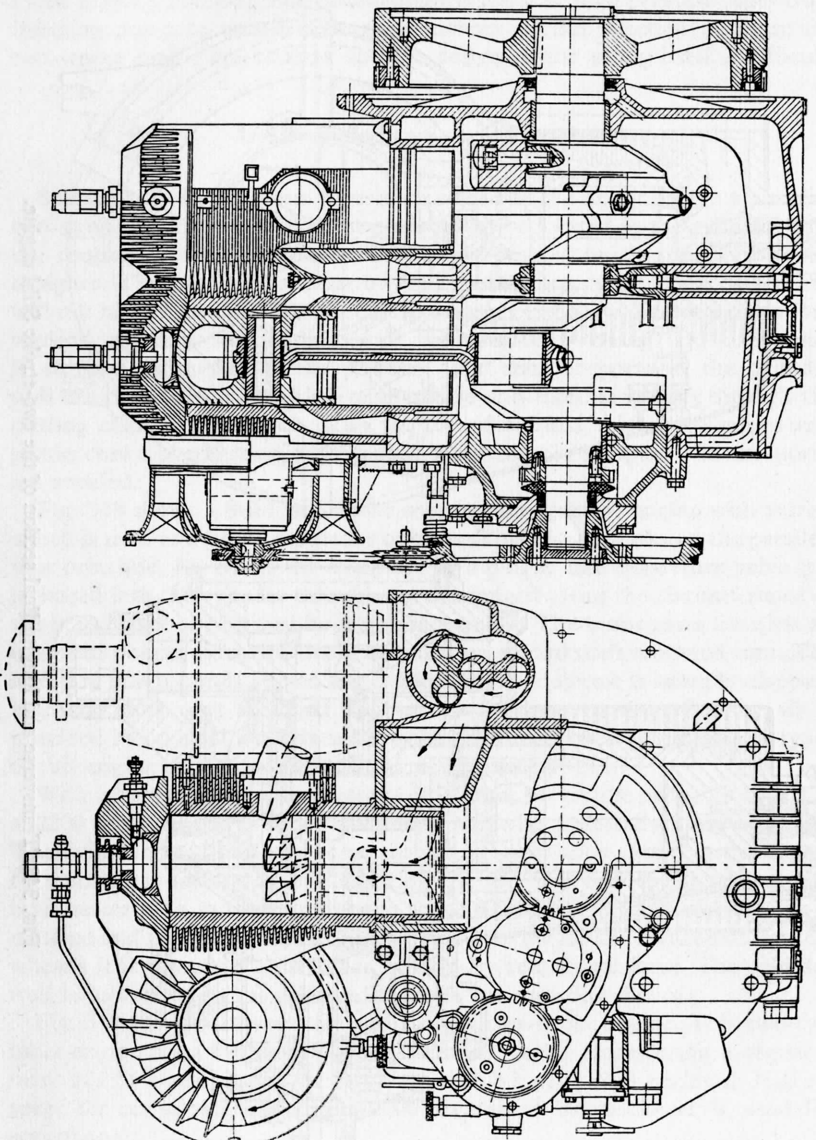


FIG. 349. The Orenstein and Koppel two-stroke engine.

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

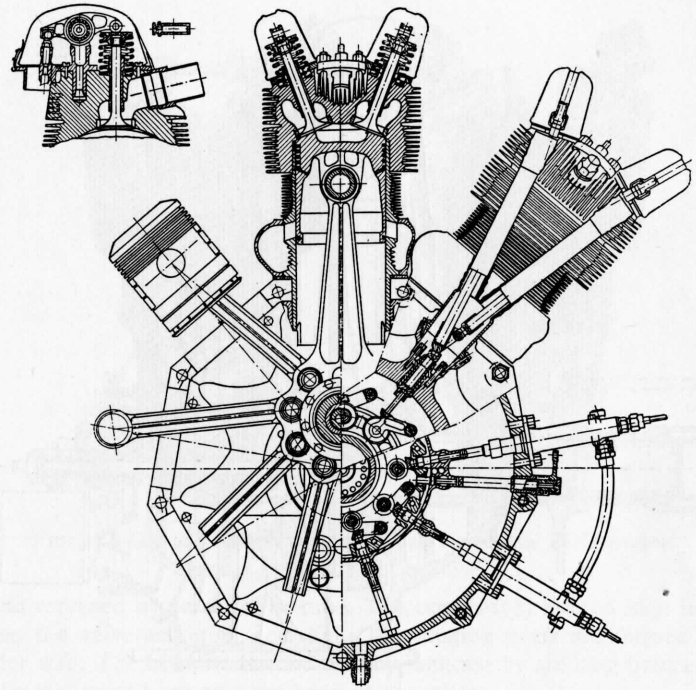


FIG. 350. Transverse section of the two-stroke ZOD engine.

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.

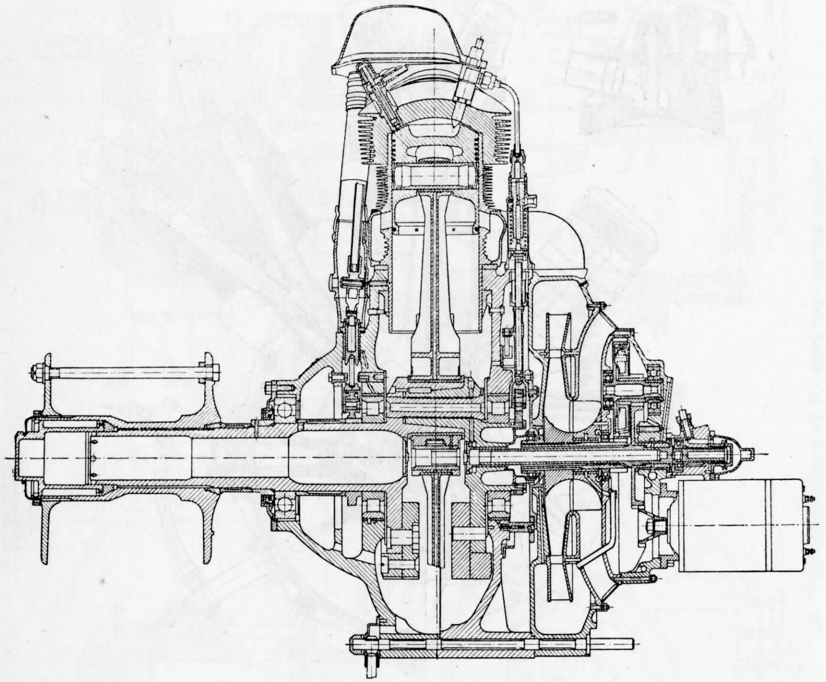


FIG. 351. Longitudinal section of the two-stroke ZOD engine.

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

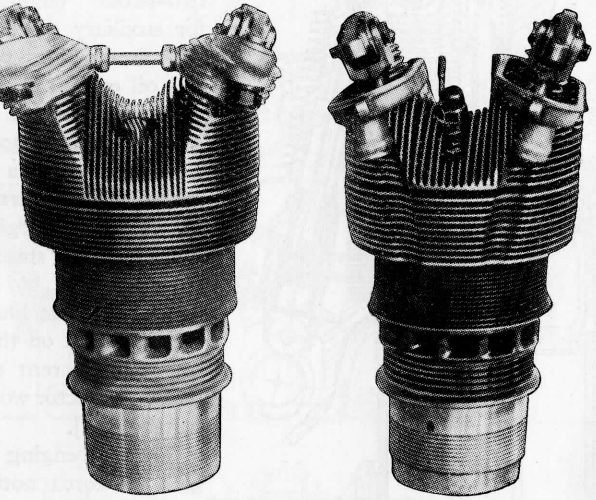


FIG. 352. Old and new design of the cylinder of the ZOD engine.

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.

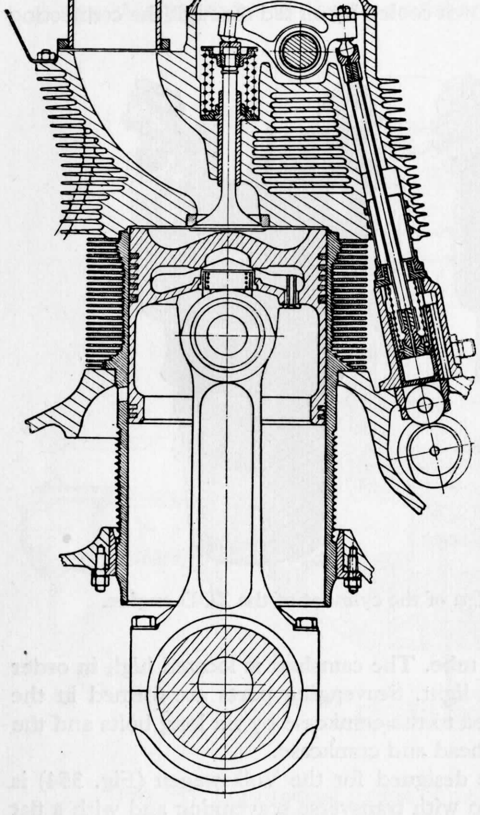
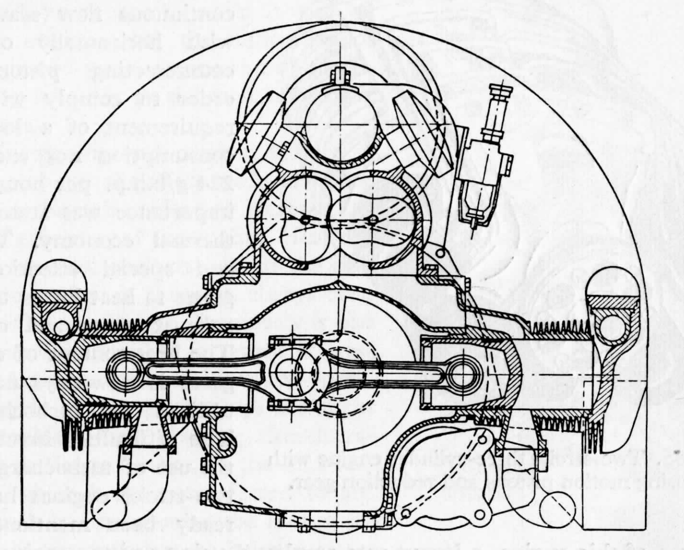


FIG. 353. Design of an air-cooled two-stroke engine of 140 mm bore.

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-



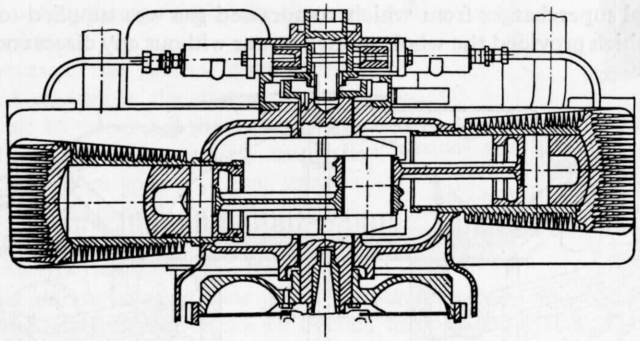


FIG. 354. Design of a two-stroke engine for the VW car.

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.

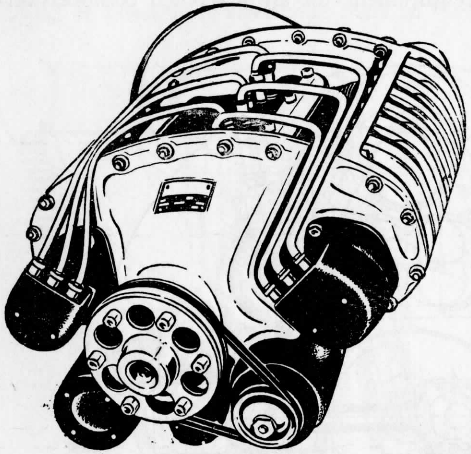


FIG. 355. Two-stroke three-cylinder engine with opposing motion pistons and reduction gear.

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

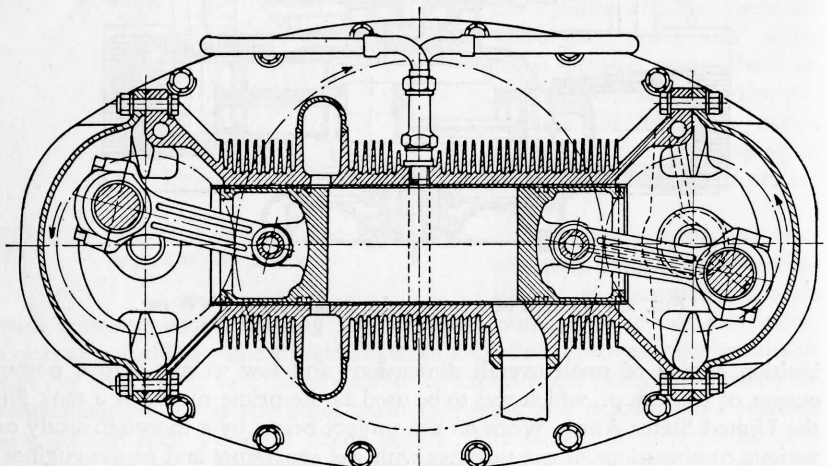


FIG. 356. Transverse section of a two-stroke three-cylinder engine (Culloch).

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.

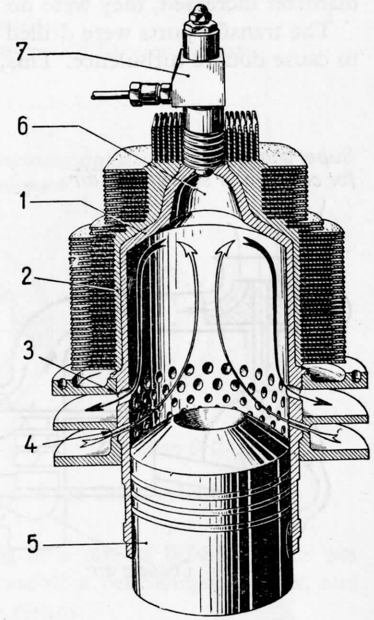


FIG. 357. Cylinder of the air-cooled Barnes and Reinecke engine with reverse flow scavenging.

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

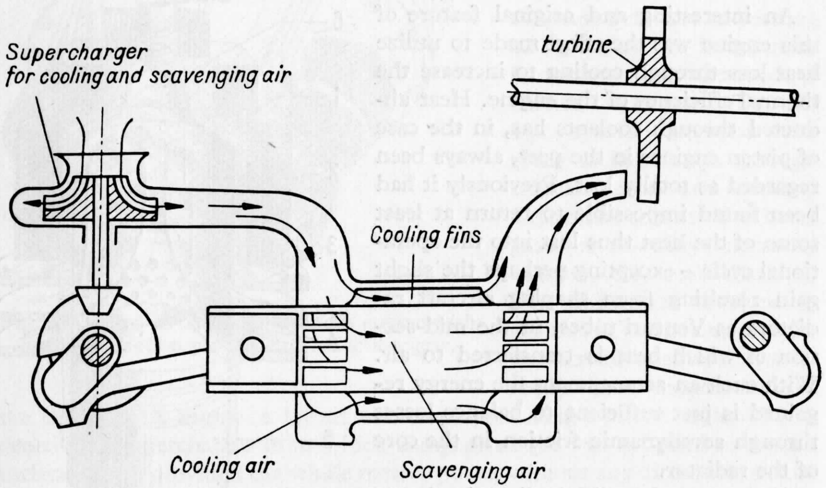


FIG. 358. Arrangement of air supply in a two-stroke aircraft engine.

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

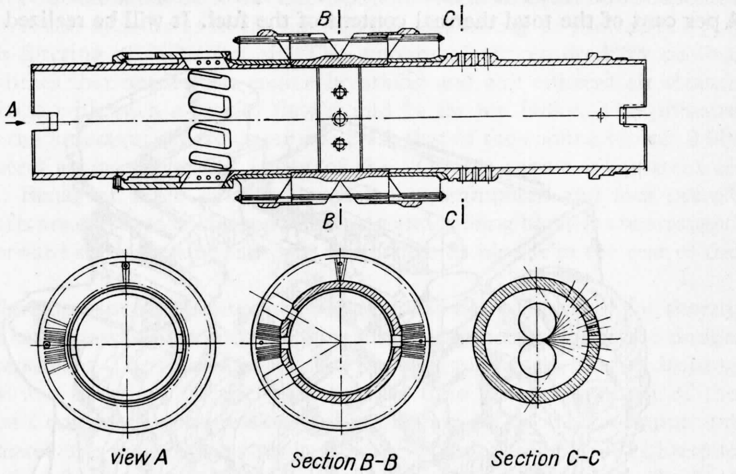


FIG. 359. Original three-piece design of the Orion cylinder.

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

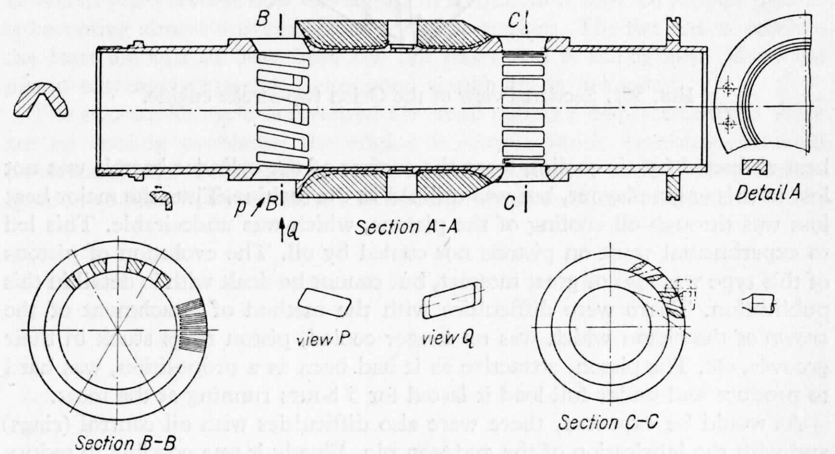


FIG. 360. The final one-piece design of the Orion cylinder.

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

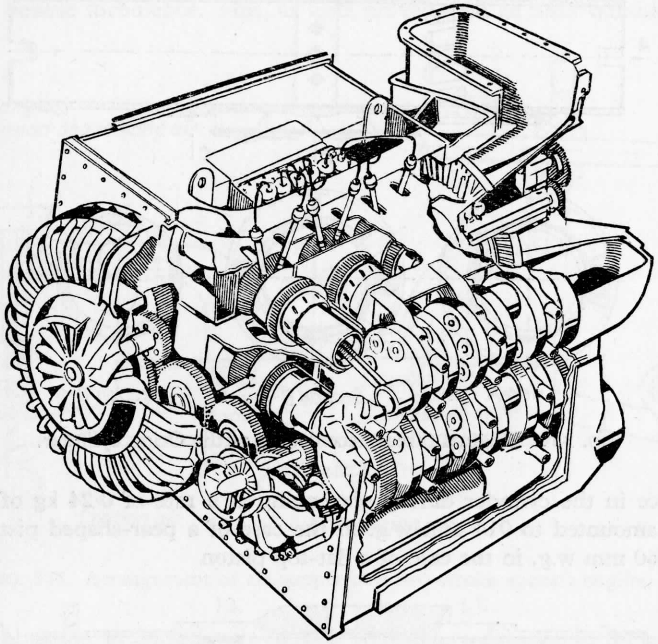


FIG. 361. Sectional view of the Orion two-stroke engine.

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.