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 cm², flow section area through the valve a cm² 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-

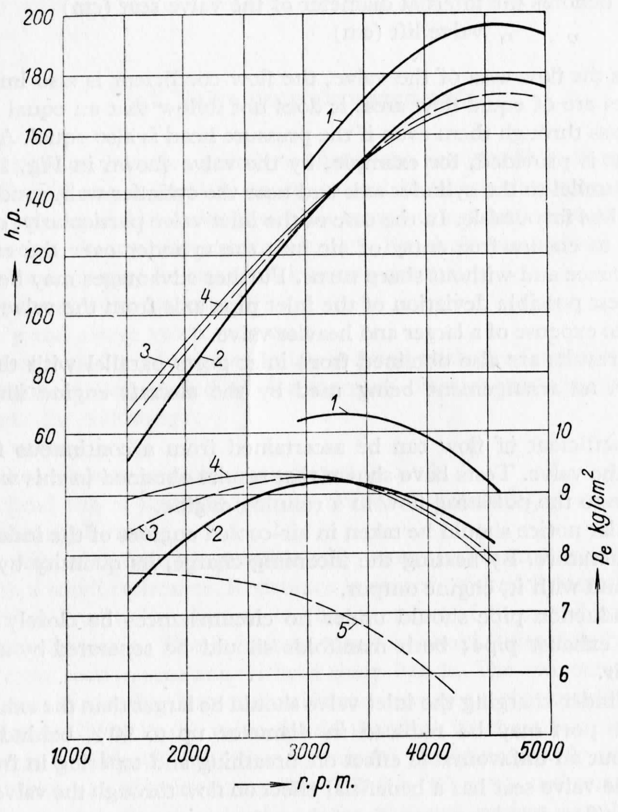


FIG. 266. Influence of valve timing upon the performance of an eight-cylinder engine, bore 84 mm, stroke 98·5 mm with different camshafts according to Table 39. Curve 1 represents engine performance with alcohol fuel.

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

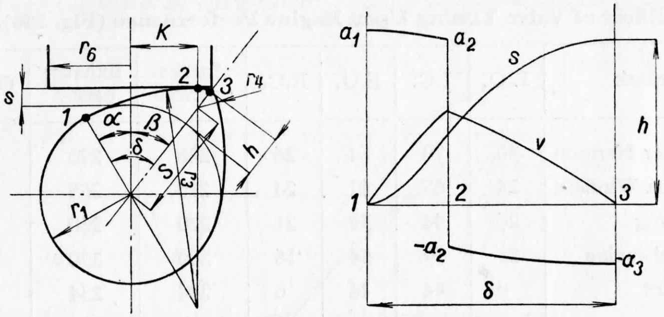


FIG. 267. Harmonic cam.

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.

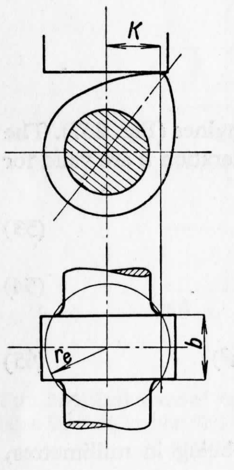


FIG. 268. Minimum diameter of the harmonic cam follower.

This position is indicated in Fig. 268 and the relative minimum radius r_e is

$$r_e = \sqrt{K^2 + \left(\frac{b}{2}\right)^2} \tag{59}$$

where $K = (r_3 - r_1) \,.\, \sin \alpha_2$

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 $\overline{0_1\, 0_3}$ and by drawing a chord parallel with the

tappet axis through centre 0_3. The intersection *1* of this chord with the semicircle indicates the velocity and acceleration of the tappet, since

$$v = \omega \,.\, \overline{0_1\, 1} \text{ (m/s)}; \quad a = \omega^2 \,.\, \overline{0_3\, 1} \text{ (m/s}^2\text{)}$$

When the velocity and acceleration at cam nose are to be found (Fig. 269), the semi-circle is drawn about the line joining centres $\overline{0_1 0_4}$. The chord

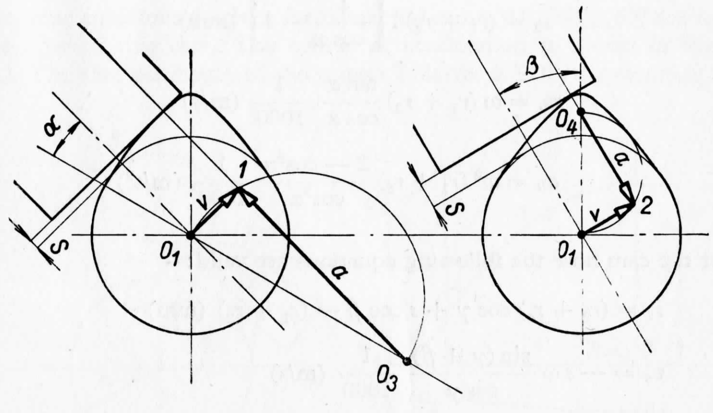


FIG. 269. Graphical determination of lift, velocity and acceleration at the flank and summit of the cam.

parallel with the tappet axis passing through point 0_4, intersects the semicircle at point *2* and then

$$v = \omega \,.\, \overline{0_1\, 2} \text{ (m/s)}; \quad a = \omega^2 \,.\, \overline{0_4\, 2} \text{ (m/s}^2\text{)}$$

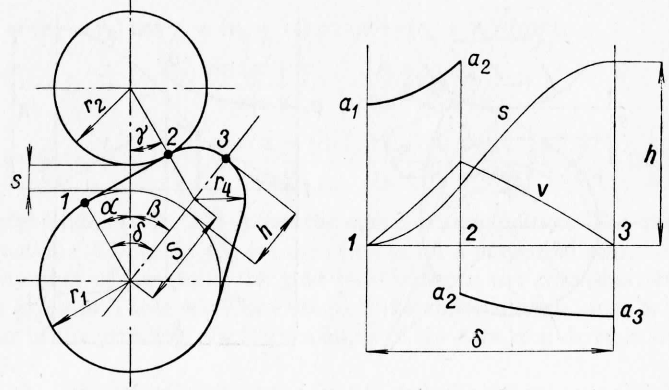


FIG. 270. Tangential cam.

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:

$$s_b = (r_1 + r_2) \cdot \left[\frac{1}{\cos\alpha} - 1\right] \text{ (mm)} \tag{60}$$

$$v_b = \omega\,(r_1 + r_2)\,\frac{\tan\alpha}{\cos\alpha}\,\frac{1}{1000} \text{ (m/s)} \tag{61}$$

$$a_b = \omega^2\,(r_1 + r_2)\,\frac{2 - \cos^2\alpha}{\cos^3\alpha}\,\frac{1}{1000} \text{ (m/s}^2) \tag{62}$$

For the cam nose the following equations are valid:

$$s_s = (r_2 + r_4)\cos\gamma + s\cos\beta - (r_1 + r_2) \text{ (mm)} \tag{63}$$

$$v_s = -\,s\,\omega\,\frac{\sin(\gamma + \beta)}{\cos\gamma}\,\frac{1}{1000} \text{ (m/s)} \tag{64}$$

$$a_s = -\,s\,\omega^2\left[\frac{s}{r_2 + r_4}\,\frac{\cos^2\beta}{\cos^3\gamma} + \frac{\cos(\gamma + \beta)}{\cos\gamma}\right]\frac{1}{1000} \text{ (m/s}^2) \tag{65}$$

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):

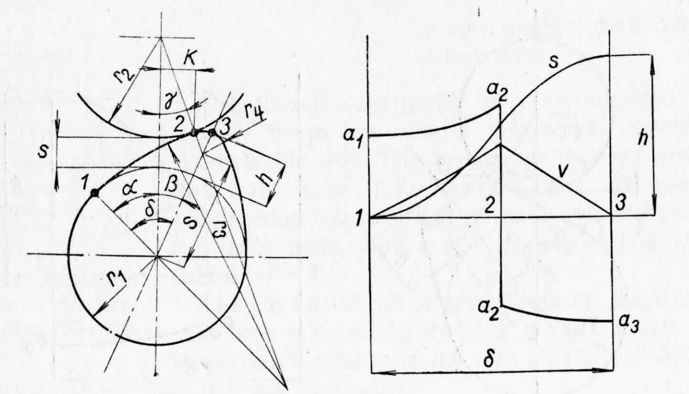


FIG. 271. Cam with convex flank.

$$s_b = (r_3 + r_2)\cos\gamma - (r_3 - r_1)\cos\alpha - (r_1 + r_2) \text{ (mm)} \tag{66}$$

$$v_b = \omega\,(r_3 - r_1)\,\frac{\sin(\alpha - \gamma)}{\cos\gamma}\,\frac{1}{1000} \text{ (m/s)} \tag{67}$$

$$a_b = \omega^2\,(r_3 - r_1) \cdot \left[\frac{r_3 - r_1}{r_3 + r_2}\,\frac{\cos^2\alpha}{\cos^3\gamma} - \frac{\cos(\alpha - \gamma)}{\cos\gamma}\right] \cdot \frac{1}{1000} \text{ (m/s}^2) \tag{68}$$

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).

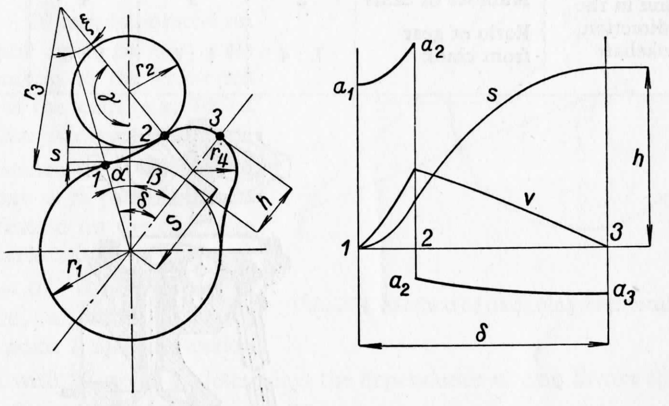


FIG. 272. Cam with concave flank.

Concave flank cams are found on the cam rings of radial engines and the following equations (Fig. 272) are applicable:

$$s_b = (r_3 - r_2)\cos\gamma + (r_1 + r_3)\cos\alpha - (r_1 + r_2) \text{ (mm)} \tag{69}$$

$$v_b = -\,(r_1 + r_3) \cdot \omega\,\frac{\sin(\alpha + \gamma)}{\cos\gamma} \cdot \frac{1}{1000} \text{ (m/s)} \tag{70}$$

$$a_b = -\,(r_1 + r_3)\,\omega^2\left[\frac{\cos(\alpha + \gamma)}{\cos\gamma} + \frac{r_1 + r_3}{r_3 - r_2}\,\frac{\cos^2\alpha}{\cos^3\gamma}\right] \cdot \frac{1}{1000} \text{ (m/s}^2) \tag{71}$$

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

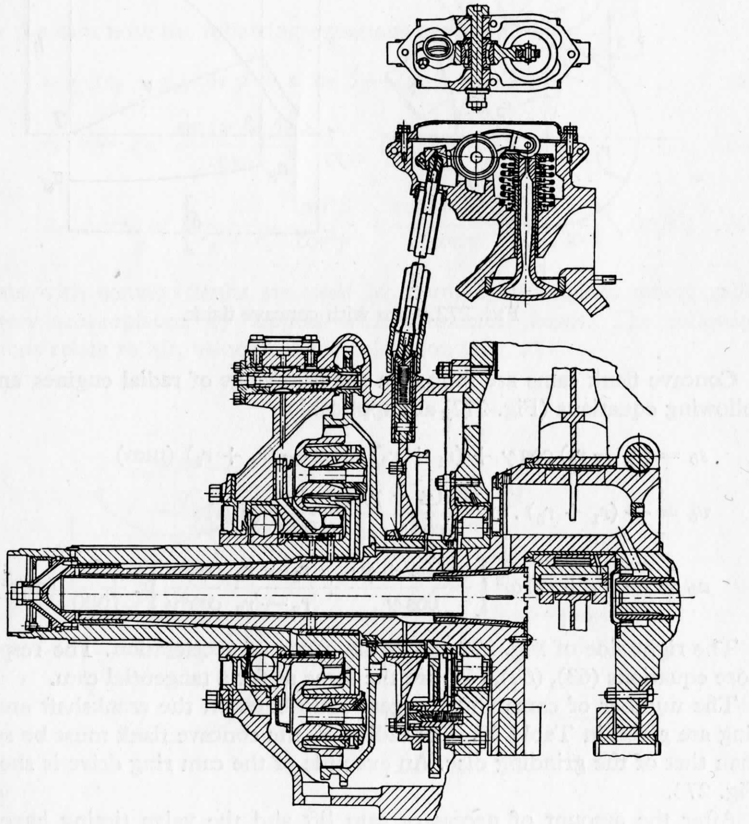


FIG. 273. Camshaft drive at the Wright Cyclone aircraft engine.

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.

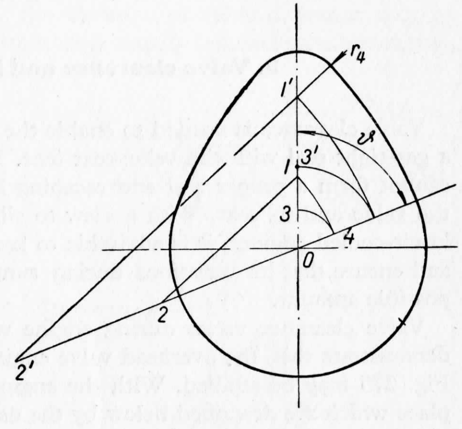


FIG. 274. Method of designing harmonic cams.

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

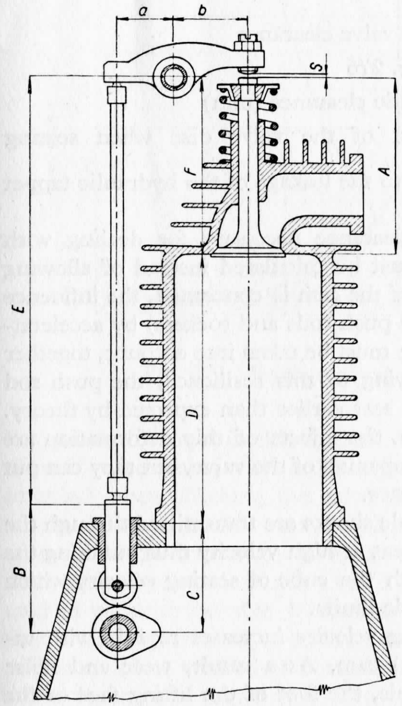
$$L_3 = C\alpha_c\,(t_c - t_n) + D\alpha_d\,(t_d - t_n) + F\alpha_f\,(t_f - t_n) \quad \text{(mm)}$$

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,



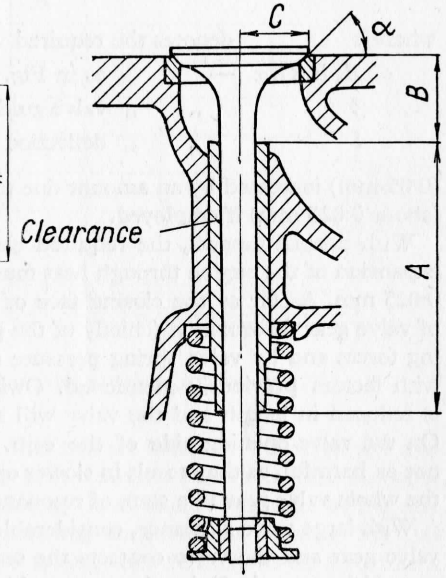


FIG. 276. Jammed position of valve in the guide.

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

$$s = \frac{(0{\cdot}5\,A + B)\,.\,\tan\alpha + C}{A}\,J + I,$$

where s denotes the required valve clearance

A, B, C, α — as in Fig. 276

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

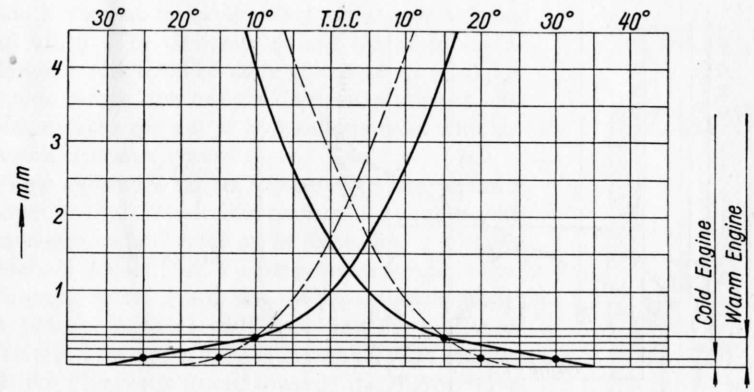


Fig. 277. Variation of valve timing with valve clearance.

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

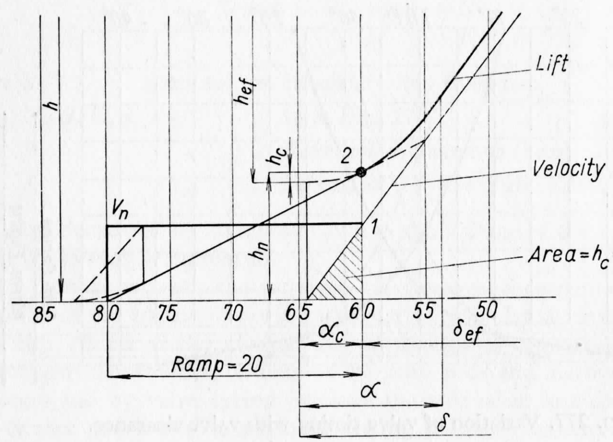


FIG. 278. Design of the cam ramp.

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

$$\delta_e = \delta - \alpha_c \tag{73}$$

$$\sin \alpha_c = \frac{\dfrac{360}{2\pi} \cdot v_r}{r_3 - r_1}, \tag{74}$$

$$r_3 - r_1 = \frac{s^2 - (s - h_e)^2}{2\,(s - h_e - s \cdot \cos \delta)} \tag{75}$$

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.

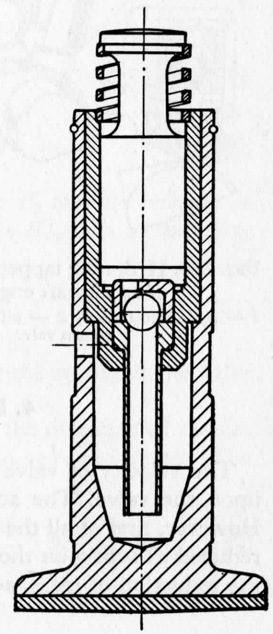


FIG. 279. Cross section of the hydraulic tappet.

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.

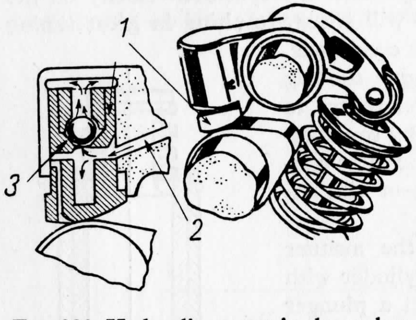


FIG. 280. Hydraulic tappet in the rocker of an aircraft engine.

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 $= \dfrac{W}{g}$ (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:

$$P_1 = \frac{a\,.\,x\,.\cos\beta}{g}\left[\frac{W + \frac{1}{2}W_1}{x^2\,.\cos^2\beta} + W_2 + \frac{W_1}{2x^2} + \frac{W_3 k_1^2}{y_1^2}\right] \text{ (kg)} \tag{83}$$

If $\beta = 0$, then the equations are simplified to

$$P_2 = \frac{a}{g}\left(W + W_1 + W_2 x^2 + \frac{W_3 k_1^2}{y_2^2}\right) \text{ (kg)} \tag{84}$$

$$P_1 = \frac{a\,.\,x}{g}\left(\frac{W + W_1}{x^2} + W_2 + \frac{W_3 k_1^2}{y_1^2}\right) \text{ (kg)} \tag{85}$$

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/s²)
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.

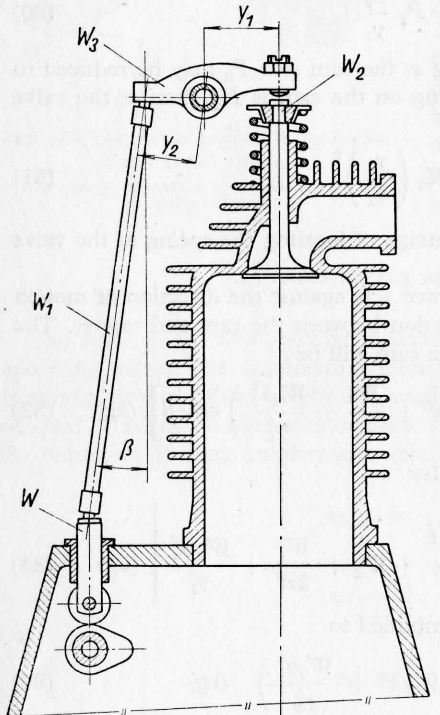


FIG. 281. Illustration of valve gear.

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/cm²)

A_e denotes the exhaust valve area (cm²)

Beside the force of acceleration, the force of valve springs also acts upon the inlet and exhaust cams

$$P_s = (S + K \,.\, s \,.\, x \,.\, \cos\beta) \,.\, \cdot x \,.\, \cos\beta \quad \text{(kg)}$$

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/cm²)
A_e „ „ area of exhaust valve (cm²)

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

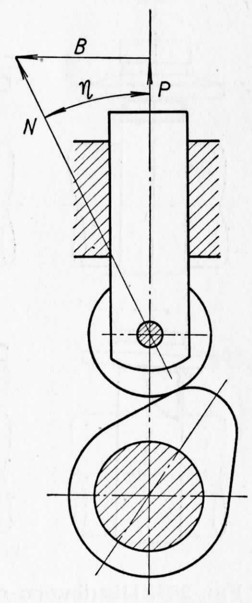


FIG. 282. Side thrust on the tappet guide.

In order to keep side thrust on the tappet guide low, $\eta > 30°$ should be avoided. The specific pressure at the contact of the cam with follower, given for two convex surfaces as

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{r_3 + r_2}{r_2 \cdot r_3}} \quad (\text{kg/cm}^2) \tag{90}$$

one concave and one convex face

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{r_3 - r_2}{r_2 \cdot r_3}} \quad (\text{kg/cm}^2) \tag{91}$$

one flat and one convex surface

$$p = 610 \sqrt{\frac{N}{b} \cdot \frac{1}{r_2}} \quad (\text{kg/cm}^2) \tag{92}$$

where b denotes the cam flank width in cm

r_2 and r_3 denote the curvature radii in cm.

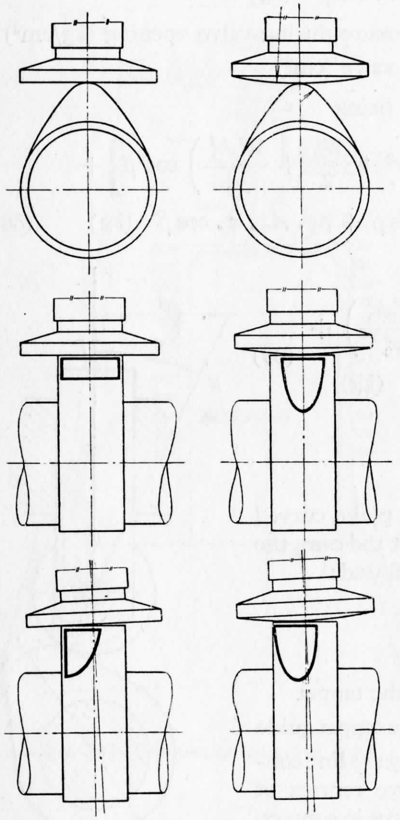


FIG. 283. Distribution of pressures between cam and tappet in the case of a flat and convex follower with the tappet axis not normal to the cam axis.

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 follower 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/s²) at maximum lift s_3 (cm)

a_2 „ „ minimum deceleration at the valve in m/s² 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/mm²),

G denotes the modulus of rigidity (shear) = 8200 kg/mm²,
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.

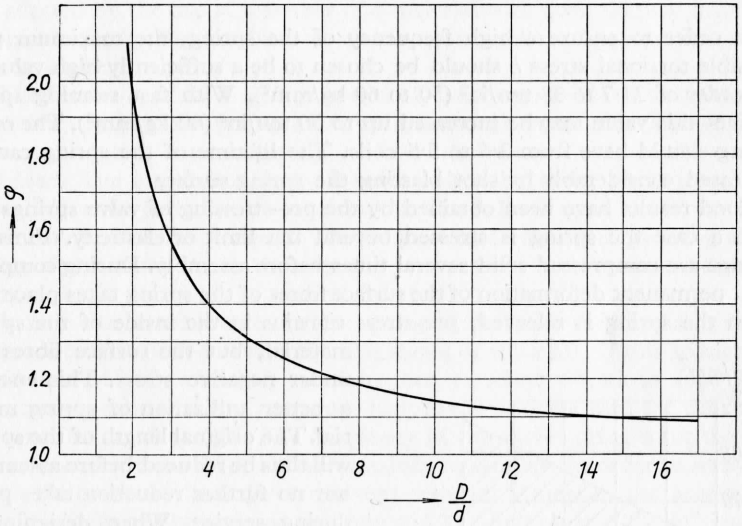


Fig. 284. Correction coefficient for the calculation of cylindrical springs.

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

$$n = 485\,140 \frac{d}{i \,.\, r^2} \quad \text{per min} \tag{99}$$

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).

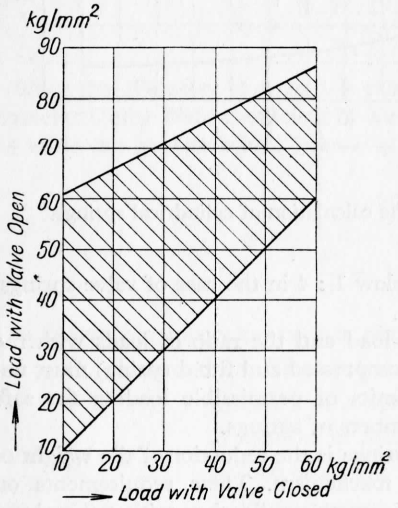


FIG. 285. Recommended stresses in valve springs.

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.

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.

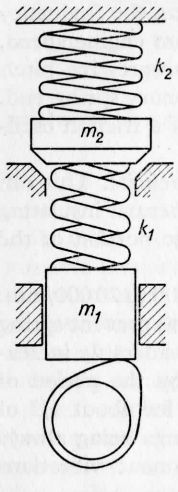


FIG. 286. Schematic representation of a resilient valve gear.

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.

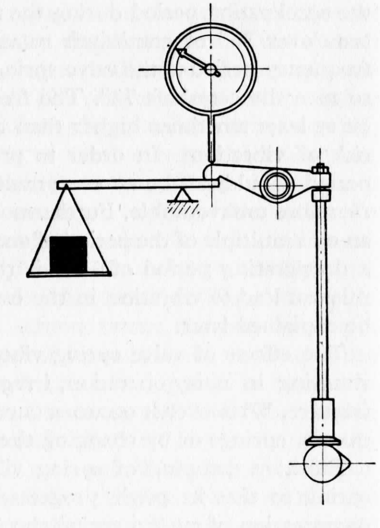


FIG. 287. Measurement of the elastic deformation of the valve gear by static load.

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.

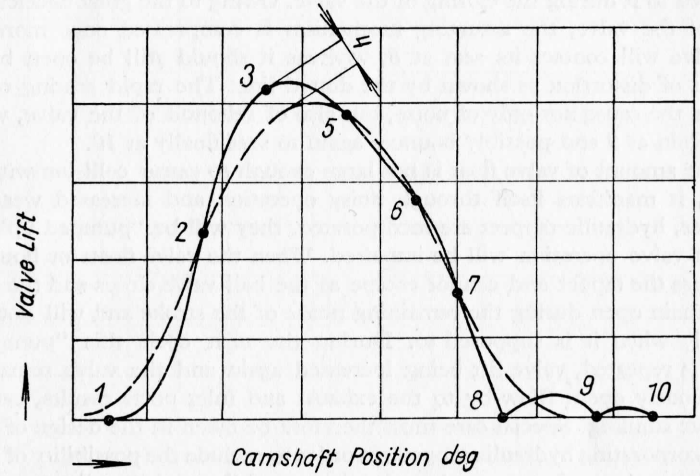


FIG. 288. Theoretical (dash line) and real (full line) course of the valve lift showing the effect of valve gear resilience.

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.

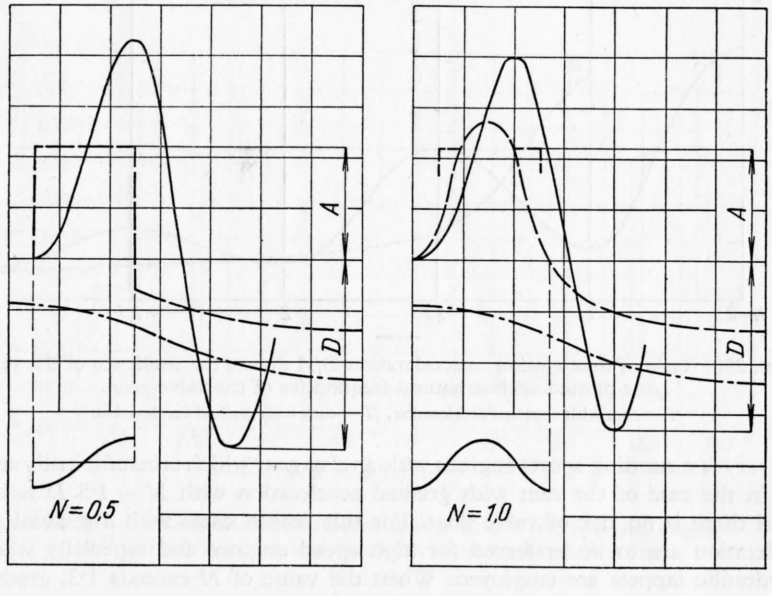


FIG. 289. Theoretical (dash line) and real (full line) course of acceleration of a cam with constant acceleration (left hand) and with a continuous course of acceleration (right hand). The dash and dot line represents the valve spring load.

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

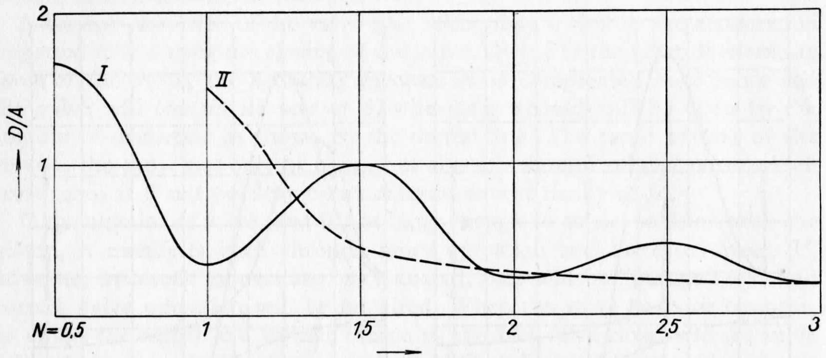


FIG. 290. Ratio of deceleration to acceleration D/A due to the resilience of the valve gear plotted against natural frequencies of the valve gear.
I — cam with constant acceleration, *II* — cam with gradual acceleration.

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 $1{\cdot}3$, 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.

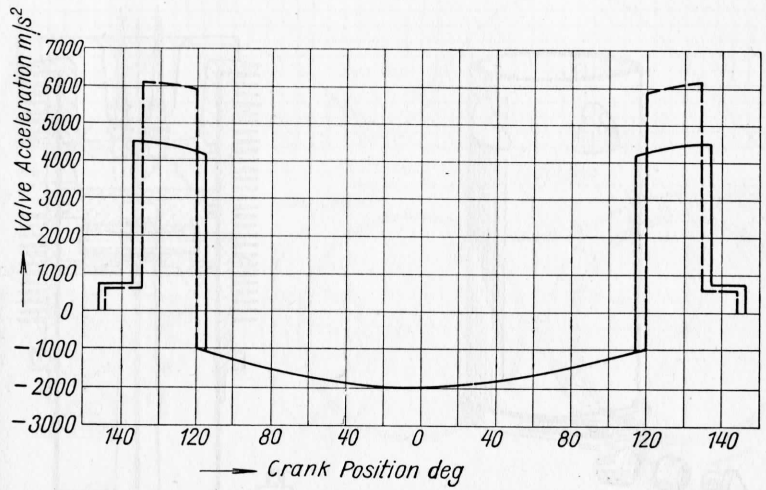


FIG. 291. Theoretical course of acceleration in a T 603 sporting engine at 7,000 rev/min, original (dash line) and modified (full line).

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 $1{\cdot}5$ 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

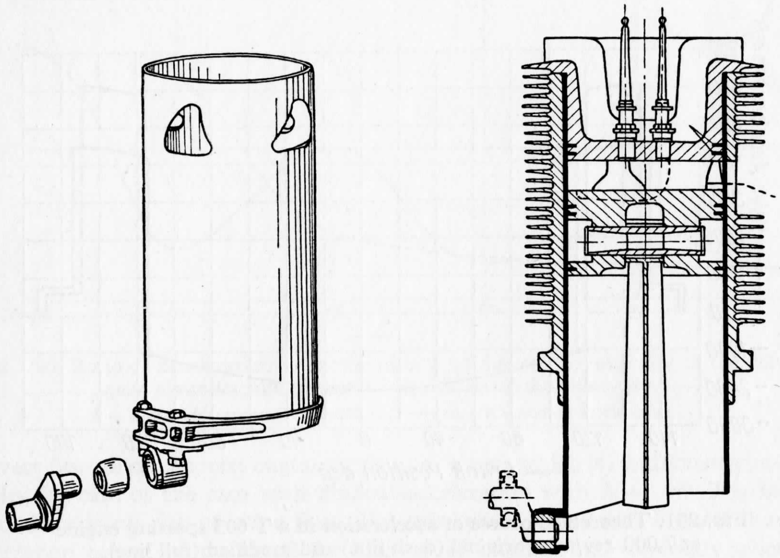


FIG. 292. Actuation of the Burt sleeve valve.

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

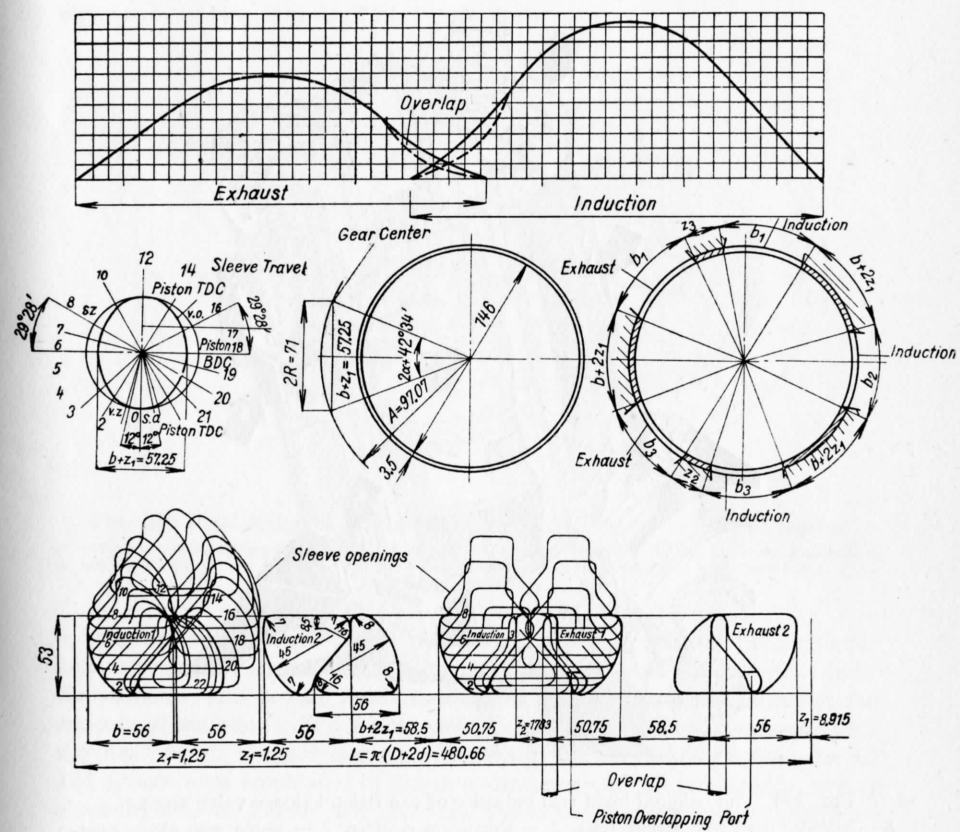


FIG. 293. Bristol sleeve valve gear.

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

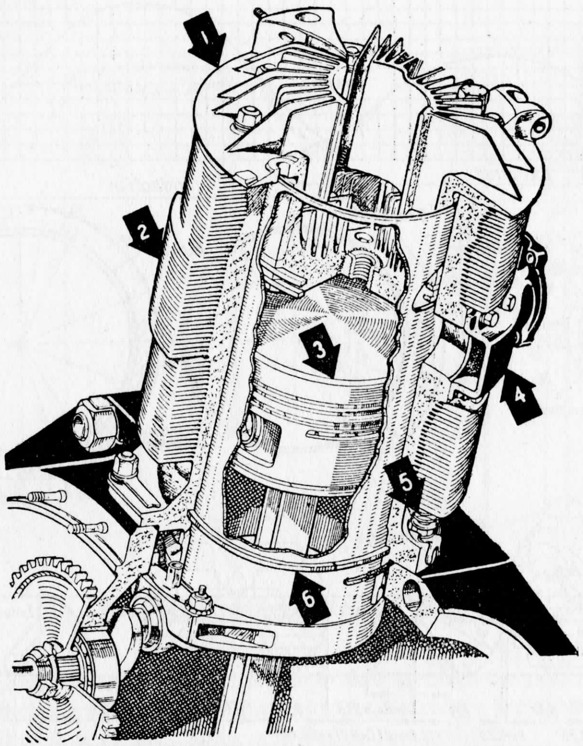


FIG. 294. The original head and cylinder of the Bristol sleeve valve engine.
1 — cylinder head with sealing rings, *2* — aluminium cylinder, *3* — piston with piston rings, *4* — induction pipe, *5* — securing bolt of cylinder, *6* — scraper ring of the sleeve valve.

integral casting with a cooling area of 539 in^2 (3480 cm^2) 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 in^2 (3750 cm^2) 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 in^2 (950 cm^2) in cooling area, total cooling area per head being now 728 in^2 (4700 cm^2). 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 in^2 (5000 cm^2).

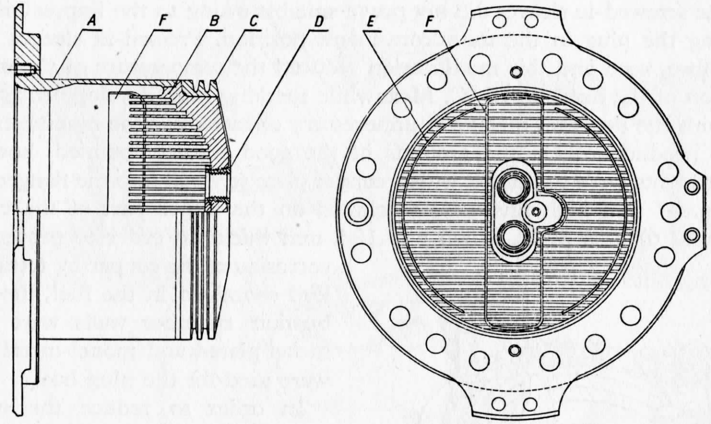


FIG. 295. Final design of the cylinder head of the Bristol sleeve valve engine.
A — steel sleeve with flange, *B* — finned base plate of a copper-chromium alloy, *C* — cast iron holder of sealing rings, *D* — Monel metal spark plug quill, *E* — anti-vibration stiffener of fins, *F* — channel and air 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 in^2 (650 cm^2) and the overall cooling area being 670 in^2 (3850 cm^2). 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 in^2 (3180 cm^2) and the overall cooling area 1005 in^2 (6150 cm^2). 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.

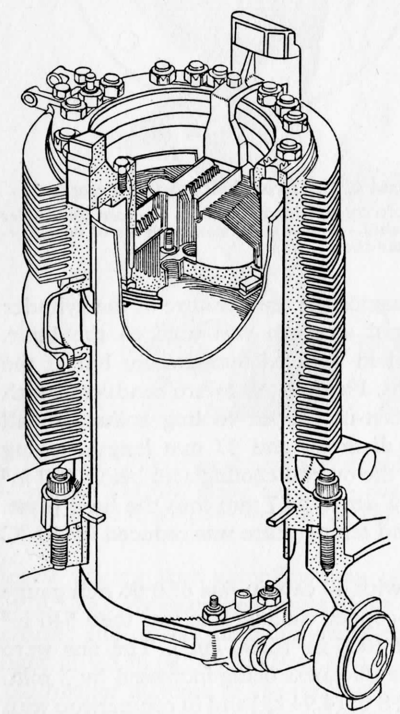


FIG. 296. Final design of the cylinder of the Bristol sleeve valve engine.

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 pro-

duction. 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

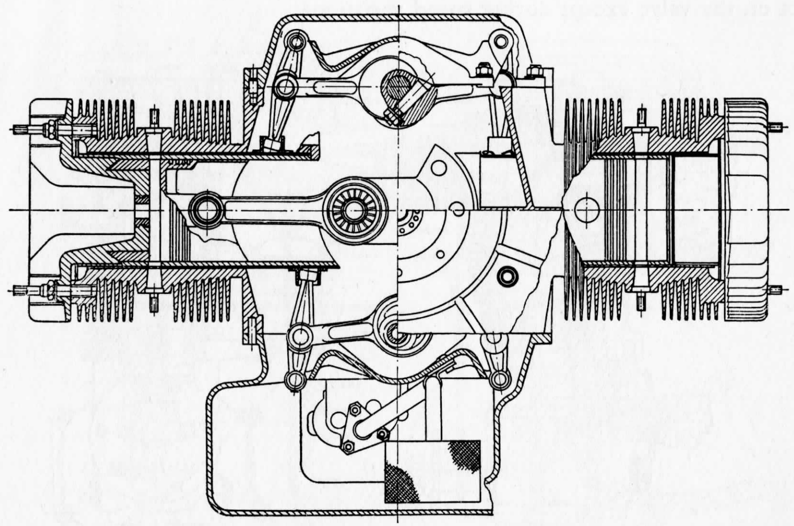


FIG. 297. The Jack and Heinz sleeve valve engine.

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.

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.

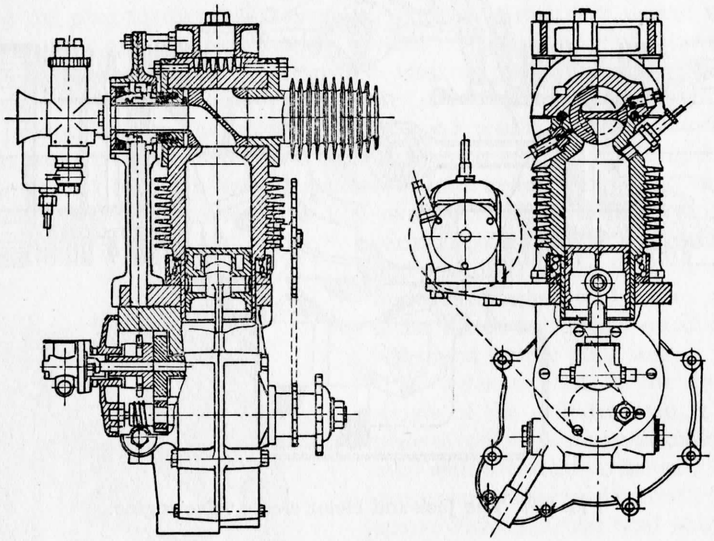


FIG. 298. The Cross sleeve valve engine.

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

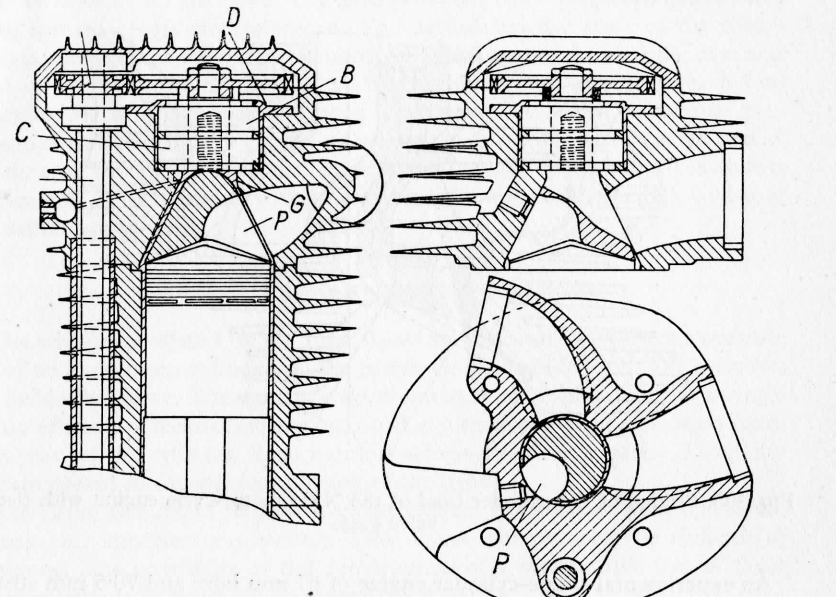


FIG. 299. The Aspin sleeve valve engine.

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.

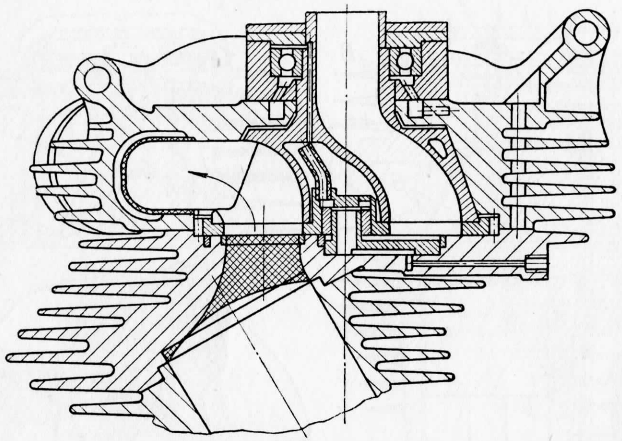


FIG. 300. Section of the cylinder head of the NSU motor-cycle engine with sleeve valve gear.

An experimental single-cylinder engine of 67 mm bore and 70·5 mm stroke with a displacement of 250 cm³ 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

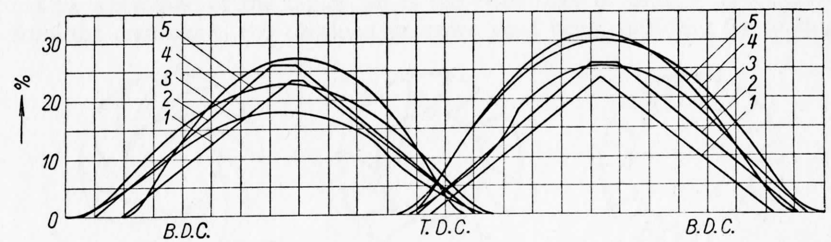


FIG. 301. Comparison of the sleeve valve gear with a poppet valve gear. Rates of port openings are given in percentage of the piston crown area.

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.