CHAPTER XVI

THE CRANK MECHANISM AND LUBRICATION

1. The crank mechanism

The requirements on the crank mechanism of an air-cooled engine are identical with those in a water-cooled engine. A rigid crankshaft, free from torsional vibration, is required together with high mechanical efficiency and low weight of the crank-mechanism.

The crankshaft must be checked for torsional vibration by calculation; this is outside the scope of this book and reference may be made to abundant literature on this subject. In engines with a short crankshaft for four-cylinder in-line engines or V eight cylinder engines, a crankshaft mounted in properly dimensioned bearings is sufficiently strong and rigid.

Table 42

Average Dimensions of Crankshafts

	Main Shaft Bearing		Crank Pin Bearing		Web		Cylinder spacing
	diameter	length	diameter	length	width	thickness	
In-line petrol	0·7	0·6	0·6	0·6	0·9	0·5	1·12
engines	0·8	0·4	0·7	0·45	1·2	0·2	1·35
Petrol engines V	0·75	0·45	0·6	0·55	0·9	0·45	1·2
	0·85	0·60	0·7	0·4	1·2	0·2	1·6
In-line oil engines	0·7	0·65	0·65	0·6	1·0	0·4	1·25
	0·8	0·55	0·7	0·45	1·3	0·25	1·4
Oil engines V	0·7	0·5	0·7	0·6	1·0	0·35	1·45
	0·8	0·6	0·75	0·4	1·35	0·2	1·65

All dimensions are given as multiples of bore D

Table 42 shows the distances between cylinder centres, bearings dimensions and crankshaft web dimensions of petrol and compression-ignition engines. The diameters of main bearing journals and crank pins are restricted by

circumferential velocity and considerations of design, a current requirement affecting crank pins (to quote an example) being the possibility of removing the connecting rod through the cylinder bore. A certain increase in crank pin diameter may be attained through splitting the big-end of the connecting rod obliquely.

An advantage inherent to air-cooled engines is the fact that after detaching the cylinder, the crankcase mouth opening is increased by double the spigot thickness, this leaving a larger diameter for dismantling the connecting rod. For this reason a horizontally split big-end can be used in air-cooled engines with crank pins of large diameter. A horizontally split big end is simpler to manufacture and stiffer for the same weight than one split obliquely.

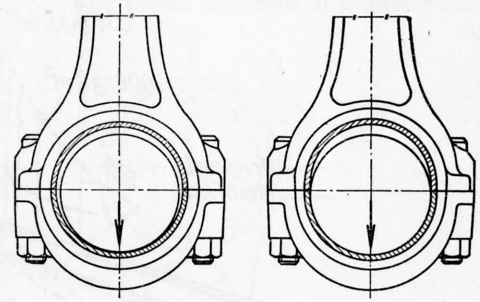

FIG. 319. Distortion of the connecting rod due to the accelerating forces of the piston.
The left hand picture illustrates the shape of the connecting rod at low speed, the right hand one shows the deformation of the big end bearing due to large accelerating forces at high speeds.

The dimensions of compression-ignition engine crank pins do not vary much from those used in petrol engines. The increased bearing load is dealt with by the use of bearing materials which stand up to higher pressures, such as lead-bronze. The data in the table refers to plain bearings. With roller bearings certain improvements may be effected.

In the construction of air-cooled engines the V and the flat layout have gained favour as the crankshaft of such types is comparatively rigid and the counterbalance weight relieves the bearing (especially the centre bearing) although it also increases crankshaft weight while reducing the natural frequency of torsional vibration. The balancing of revolving masses need not be perfect in such a case and it depends on the dimensions and properties of the bearing carrying the greatest load.

The radius at the junction of the webs and pins should be large in order to prevent local concentration of stress, minimum radii of 0·025 to 0·04 D being advisable.

In order to secure the best utilization of the crankshaft constructional material highly stressed sections should be avoided. By the rolling in of material at the transition between pin and web the fatigue resistance of the crankshaft can be increased by up to 60%. The concentration of material around the mouths of oil-ways in crankshaft journals also increases fatigue resistance considerably.

Bronze bearings require a hard surface for the main journals and the journals of compression-ignition engine crankshafts are today hardened mainly by the application of direct induction heating. Up to 500 to 550 Brinell hardness can be obtained.

The connecting rod is regarded as a partly reciprocating and partly revolving mass. Its weight therefore must be specified with utmost care. The big end must be sufficiently stiff to prevent distortion through accelerating forces. In highly stressed engines a certain amount of big end distortion is always

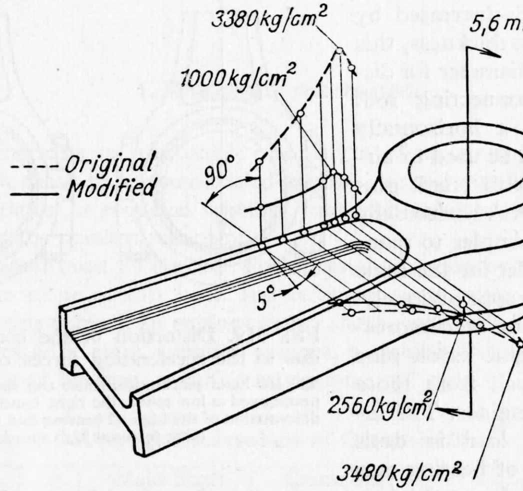

FIG. 320. Stresses in the connecting rod of an aircraft engine.

reckoned with and a slight bearing clearance at right angles to the connecting rod axis is provided in order to prevent the bearing "gripping" the pin or seizure occuring near the split of the bush. The deformation of a bearing by accelerating forces is shown in Fig. 319. Big end bearings are generally bushed with thin walled sleeves. The same observations made about cylinder bolts apply to connecting rod bolts: their axes should be located at the geometric centre of the seating surface of the big end and cap. If this is not the case, the bolts are exposed to bending stress when tightened and the connecting rod is distorted. This is particularly important with light forked connecting rods.

Side thrust at the joint can be taken up by a step in the joint face. In such a case the joint faces are formed with mating straight or curved-wall splines which transmit lateral forces, and make the use of inclined split big ends possible even with large diameter crank pins.

The small end eye mass is reciprocating and must therefore be as light as possible. The correct shaping of the joint between the small end eye and connecting rod shank is important. The stress at this point is greatest and fractures are known to originate from it. Fig. 320 shows the stress at this point of the

connecting rod as determined by actual measurments. The reduction in stress attained by the modification shown in Fig. 321, (right) is marked by full line. The use of a single radius between the connecting rod shank and small end eye (left) which is often used by designers for its greater convenience, is poor design.

The pistons of an air-cooled engine have considerable influence on engine noise. It is important to keep the cold piston clearance at a minimum

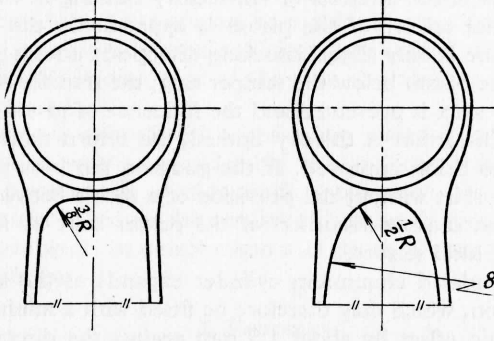

FIG. 321. Modification of the connecting rod providing an increase of strength.

and to reduce its variation as the engine warms up. This may be done by:

1. compensating for piston expansion;
2. flexible pistons with split skirts;
3. use of aluminium cylinders.

Expansion of the piston is usually compensated for by cast-in struts of a material with a small coefficient of expansion; as an example of this type, the Bohnalite-Nelson piston with invar struts may be quoted. Today cast-in rings in the region of the scraper ring are often used and these do not permit the expansion of this section of the piston.

Split-skirt pistons may be fitted into cylinders with a smaller cold clearance. With the expansion of the piston, the split skirt is relatively compressed and seizure is thus prevented. An example of a piston of this type is given in Fig. 322.

The piston clearance must be specified in the case of the air-cooled engine with special care in order to prevent piston slap and seizure. The use of large piston ovality (up to 0·2 mm) is advisable as this provides clearance for the

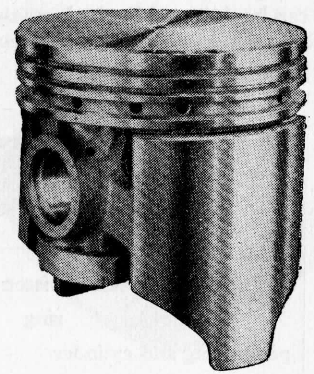

FIG. 322. A modern piston o a high-speed engine.

piston during expansion, especially if steel struts are cast into the piston, acting as bi-metallic elements by changing its elliptic section into a circular one during expansion.

Care should also be taken to provide the correct tolerance between the gudgeon pin and its bosses. A heavy interference fit restricts the expansion of the piston in the direction of the gudgeon pin. Excessive allowance on the other hand, is a cause of knock.

The surface treatment of the piston (graphiting, tinning, anodic oxidation, etc.) is beneficial in the interests of satisfactory running-in with small piston clearances. Partial seizure of the piston is apparent by the knocking noise emitted. If seizure is only slight, knocking disappears after a period.

By slotting the piston below the scraper ring, the transfer of heat from the piston crown to skirt is prevented and the influence of piston crown temperature on skirt distortion is thereby limited, the return flow of oil from the scraper ring also being improved. If the gudgeon pin boss projects directly from the piston skirt without the provision of a recess between them, a thin gudgeon pin can cause a distortion of the piston skirt by its own bending thereby causing local seizure.

A chromium plated aluminium cylinder expands at the same rate as an aluminium piston, which may therefore be fitted with a minimum allowance.

A gudgeon pin offset by about 1·5 mm against the direction of rotation increases the period during which the load on the piston is transferred from one side to the other, thus reducing piston slap at top dead centre.

Piston rings are very important to piston cooling, as heat is transferred from the piston to the cylinder wall mainly through the piston rings. The numerical values of thermal load on the piston rings are given in Table 43. Piston rings are available with normal or extra tangential pre-stress. Rings with high tangential stress begin to vibrate at higher engine speeds. When the ring begins to vibrate owing to the periodical pressure of combustion, the ring ceases to provide a gas seal and fatigue fracture may even occur. Ring vibration can be detected by the increase in blow-by of gases to the crankcase at a certain critical speed. This blow-by should not exceed the value of 0·2 to 1% of the induced volume.

Table 43

Thermal Stresses in Piston Rings and Pistons

Heat transfer between		kcal/m² h °C
piston and side wall of piston ring	upper side	600
	lower side	14,000
piston ring and cylinder		30,000
piston skirt and cylinder		300

Oil control rings prevent piston over-lubrication and remove surplus oil from the cylinder wall. The specific pressure between the piston ring and cylinder, depending on ring stress and the area of the contact surface, is important. The common type of scraper rings has a sealing face 0·75 mm wide and a specific pressure of 49·7 lb/in² (3·5 kg/cm²). Scraper rings produced today have widths of down to 0·1 mm and specific pressures as high as 284 lb/in² (20 kg/cm²), such rings, if provided with sufficient lubrication, give long service life even under arduous conditions. Plentiful lubrication of the cylinder with efficient oil control rings is preferred today.

The oil retained by the ring must be provided with a free passage of flow by arranging a number of holes of sufficient cross section area from the piston ring groove and the leading edge of the ring and also by sufficient notches in U shaped rings.

If the first groove is fitted with a chromium plated ring, wear on the ring itself and cylinder wear is considerably reduced, this being the reason for the recent widespread use of chromium plated rings. Sealing rings should be narrow in order to reduce the amount of running-in required and in order to reduce groove wear. In petrol engines of up to 90 mm bore, a ring width of 2 mm is sufficient.

2. Bearings

Plain bearings are usually employed for the crankshaft, as they are simple, small and silent in operation. Petrol engine bearings are most often bushed with white metal (composition, a Babitt) shells. Babbitt alloys are at their best with a content of 86 to 89% of tin. A valuable property of such alloys is the embedding of foreign hard matter brought by oil with resulting protection of the journal surface. Should oil supply to the bearing fail, it "runs", i.e. melts and the shaft is not damaged.

White metal is cast into stiff thin-walled shells. The overall thickness of such a thin wall shell is about 1·5 to 2 mm, the thickness of the bearing metal layer being about 0·5 mm. The bearing metal is cast onto steel strip and scraped down to the prescribed thickness and only then is the strip cut up into sections of the required length, which are then bent into required shape. As a precaution against turning, the shells are fitted with anchor strips bent outward, as shown in Fig. 323. The bearing housing in the crankshaft or connecting rod must be circular and accurately machined. The diameter of such a flexible thin wall bearing cannot be checked, the only possible measurement being that of the shell thickness and length with the half-shell pressed into

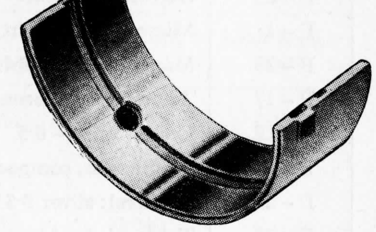

FIG. 323. Shell of a thin-walled bearing.

the semi-circular groove. The rate of wear of such bearings is rapidly reduced by the use of thin Babbitt lining (Fig. 324). Shells with a microscopic layer of 0·05 to 0·12 mm of Babbitt are now being produced and have a reputation of greatly increased life. Three-layer bearings with an intermediate layer of lead-bronze to prevent shaft damage with a run bearing are also produced.

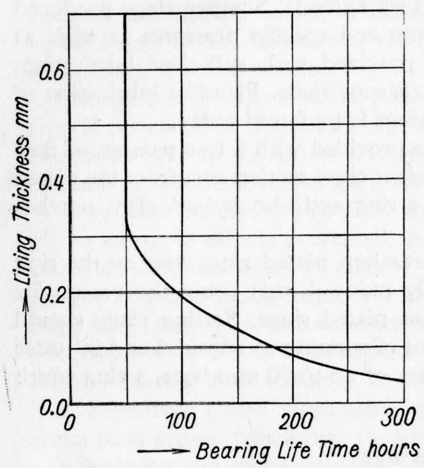

FIG. 324. Service life of a bearing in relation to the thickness of bearing metal.

The normal type of Babbitt bearing will withstand loads of 1280 to 2130 lb/in² (90 to 150 kg/cm²), shells with a skin of about 0·1 mm thickness will resist higher loads. Table 44 gives maximum load figures during laboratory material fatigue tests for an operating life of 100 hours under conditions of ideal lubrication, oil cleanliness, optimum temperature and bearing clearance.

Lead-bronze which will stand up to higher loads than white metal (up to 3550 lb/in² — 250 kg/cm²) is used extensively in compression-ignition engines. The advantages of lead-bronze are more apparent at high temperature as its hardness at 120 °C is 12·6 to 15·8 ton/in² (20 to 25 kg/mm²) (Brinell), whereas with white metal composed mainly of tin, it is only 5 to 7·6 ton/in² (8 to 12 kg/mm²) (Fig. 325) [45].

Table 44

Maximum Bearing Load for a 100 hours Fatigue Test

Type of bearing	Material and thickness of lining mm	Load kg/cm²
F – 1	Tin-base Babbitt – 0·5	120
F – 23	High-lead Babbitt – 0·5	155
F – 1	Micro – tin Babbitt – 0·1	180
F – 23	Micro – lead Babbitt – 0·1	180
F – 17	Trimetal: lead bronze 0·63, Babbitt 0·15	180
F – 12	Lead bronze – 0·5	245
F – 81	Aluminium, compact (without bush)	385
F – 2 1	Trimetal: silver 0·33 + lead-indium 0·025	520
F – 77	Trimetal: lead bronze 0·3 — 0·6 + + lead-tin-copper 0·025	520

With the use of lead-bronze for bearing material, the journal must be hardened in order to prevent a rapid rate of wear. The journal must moreover be rigid to prevent whip with the accompanying overloading of the bearing at its edges. Foreign matter is not easily embedded in lead-bronze and efficient means of oil filtration must be provided. Recommended clearances for lead-bronze bearings are given in Table 45.

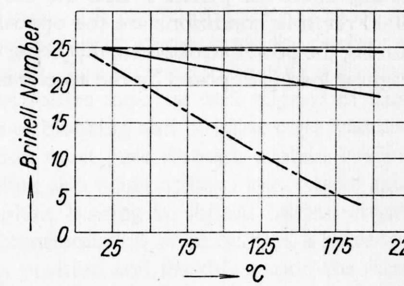

FIG. 325. Influence of temperature on the strength of bearing metals.
Dash line — silver, full line — lead bronze, dash and dot line — lead bearing metal.

Aluminium bearings have suitable frictional properties and are used either as full bushes or cast onto a steel shell. The high heat conductivity of aluminium assists the removal of heat from the bearing. The requisities for aluminium bearings are a hard journal of at least 450 Brinell and clean oil, maximum advisable loads being 3550 to 4980 lb/in² (250 to 350 kg/cm²). Abrasive foreign matter is absorbed by aluminium better than by bronze. The clearance must be sufficient to prevent restraint of the journal with subsequent seizure.

Silver bearings have excellent characteristics and withstand high loads. They are preferred for big end bearings in radial engines. They are excellent in respect of fatigue resistance and freedom from corrosion and their high heat conductivity is valuable. Their properties are further enhanced by the use of a top layer of lead and indium, 0·025 mm thick.

Oil must always be supplied to plain bearings on the relieved side. The loaded bearing side must not incorporate oil distribution grooves, which considerably reduce bearing capacity. Oil grooves should be avoided where possible. Oil is fed to big end bearings to the frontal side of the crankpin

Table 45

Recommended Bearing Clearances

Bearing	Babbitt mm	Lead bronze mm
Main bearings: basic clearance up to 50 mm dia	0·050	0·060
for each further 25 mm	0·020	0·025
Crank pin bearings:		
basic clearance up to 50 mm dia	0·040	0·050
for each further 25 mm	0·015	0·020

viewed from the direction of rotation. The variation of bearing pressure in the big end bearing with varying crankshaft speeds is shown in Fig. 326. At 2000 rev/min the maximum loads imposed on the bearing by gas pressure acting upon the piston *F* and the load by accelerating forces *I* is small. At 5000 rev/min conditions are the opposite, the load transmitted to the bearing during the power stroke is small, (being cancelled by centrifugal force) while the highest load is imposed by the accelerating force *I* during the induction stroke.

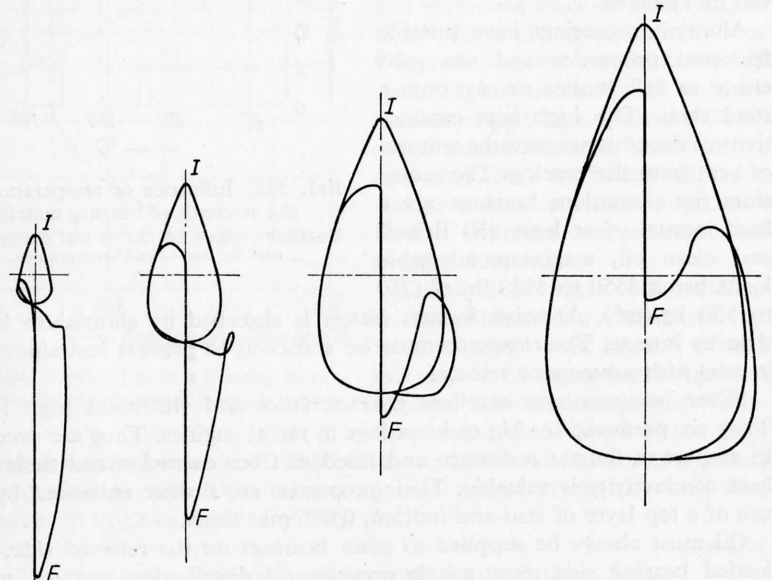

FIG. 326. Variation of load of connecting rod bearing with engine speed.
I — accelerating force, *F* — force developed by gas pressure upon piston. The diagrams are drawn for speeds of 2, 3, 4 and 5 thousand rev/min respectively.

Rolling bearings reduce friction and increase mechanical efficiency. They do not require the same degree of lubrication as plain bearings, but they are noisier in operation. Owing to smaller frictional loss, oil temperature is kept lower, so that the necessity for an oil cooler is sometimes obviated. A built-up crankshaft is however necessitated by the use of rolling bearings and this is more costly.

The built-up crankshaft scores on account of the relatively simple production of its small parts and because in the event of damage to a single journal, the whole crankshaft need not be scrapped, since the damaged part can be replaced. A built-up crankshaft is also noted for its strong internal damping of torsional vibration.

During assembly all bearings must pass through the first outer bearing ring in the crankcase, this having the disadvantage of limiting the choice of roller size which cannot be made according to the requirements set by the size of the inner and outer bearing rings in order to keep the bearing allowance low. Allowance must be specified in accordance with production tolerances of the bearing surface of the outer bearing ring, in order to guarantee the passage of the caged races of rollers fitted on the shaft. The increase in bearing allowance results in increased running noise. The rollers must be well aligned in their cage as misaligned rollers are the cause of blocking and possible cage fracture.

The oil for connecting rod lubrication must pass through a plain bearing on its way into the crankshaft, this bearing also being utilized to act as an axial crankshaft bearing. The centre of a plain bearing is shifted during engine running in order to make possible the accommodation of an oil film. The centre of a rolling bearing does not change its position and for this reason the combination of both types of bearing on a single shaft is inadvisable and for the same reason the bearing through which oil is supplied is provided with a large radial clearance in order not to transmit high loads and to avoid possible seizure.

The route of the oil passing from the flywheel end to the remote cylinder is long, resulting in an oil pressure loss and oil warming. In order to prevent failures of this last bearing, its correct lubrication must be ensured. If a pressure relief valve is incorporated in the crankshaft at the flywheel end (Tatra 111 etc., see Fig. 408), large quantities of oil flow through the shaft, cool oil is supplied to the remote bearing under pressure unreduced by the valve.

When roller bearings are used on a shaft which is not quite rigid, distortion of the journals resulting in outer edge overload on rollers must be considered. It is obvious that if clearance between rollers and bearing rings diminishes, by the inclination of the journal, to nil or less, the rollers or rings will be damaged. Long rollers are therefore not suitable and roller edges must be rounded off.

In motor cycle engines which use built-up crankshafts and one-piece connecting rods, big end rolling bearings are convenient. Split rolling bearings do exist, but they require a high standard of manufacture. For high bearings loads and high speed operation, caged roller bearings must be used. Roller bearings have a very low rate of wear and make short engine dimensions possible.

3. Lubrication

Cylinder temperature is always higher in air-cooled engines and a greater amount of heat is, therefore, transferred to oil. Should oil temperature be above 100 °C, an oil cooler must be incorporated. In high-performance water-cooled engines the use of an oil cooler is also necessary, for if an oil-to-water heat exchanger is employed, nothing is gained as heat lost to water must ultimately be transferred to the surrounding air so that the water radiator must

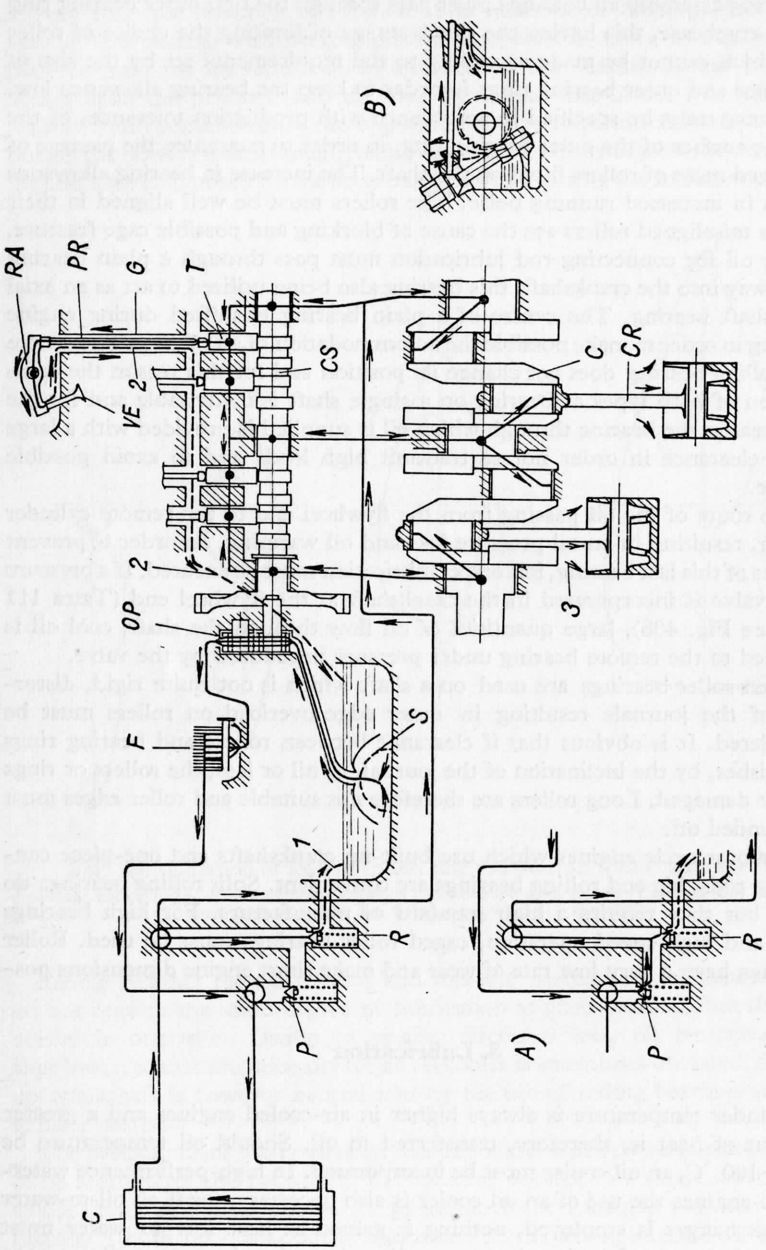

FIG. 327. Lubrication of a Tatraplan engine with an oil sump in the crankcase.
OP — oil pump, *F* — oil filter, *P* — relief valve, *C* — oil cooler, *R* — reducing valve.

be increased in size proportionally. Thus not only is a heat exchanger added, but also an increased radiator surface and some water.

Oil temperature may reach 110 °C and for short spells up to 120 °C. Temperatures under 80 °C are undesirable, for friction loss increases and there is a risk of oil dilution by fuel.

Lubricating systems can broadly be divided into those with a "wet sump" and those with a "dry sump". With wet sump lubrication (Fig. 327), the bulk

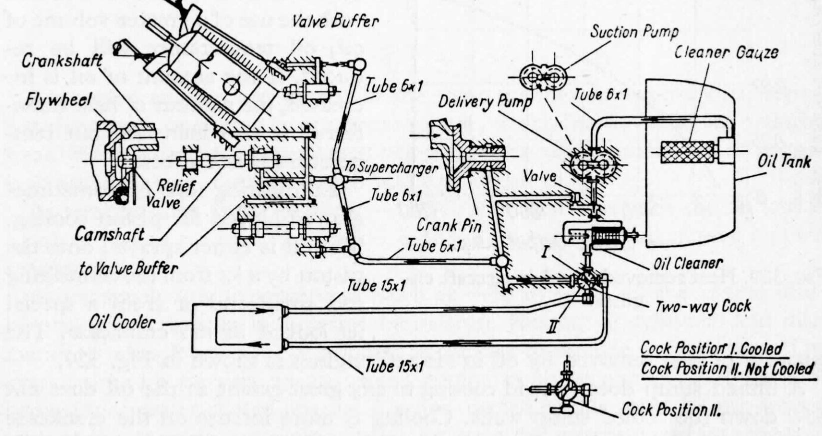

FIG. 328. Lubrication with dry crankcase (Tatra 111). The suction pump delivers the oil from the tank at the flywheel to the oil container.

of oil is in the sump and a single pump circulates the oil. If an oil cooler is employed, it must be interposed between this pump and the bearings. The cooler is then under pressure and must be designed for this pressure. In order to prevent damage to the cooler in winter, when the oil is thick, in a cold engine, a safety by-pass valve is provided in parallel with the cooler and oil passes through this until the oil is sufficiently warmed.

If the engine is intended for off-the-road vehicles in which regular engine running is required even on extreme gradients (up to 60%), the use of a dry-sump system is called for. In such a system the oil supply is in a separate tank remote from the crankcase (Fig. 328). Oil is drawn from the sump and delivered to the tank from the front and rear ends of the crankcase and oil for pressure lubrication of the engine is taken from the tank by the pressure pump. If the tank can be mounted below and in front of or behind the engine, one scavenging pump can be saved and the layout is simplified (Fig. 328). Each one of the scavenging pumps must be of greater capacity than the delivery pump.

If a three-pinion gear pump is used for scavenging, foaming also must be taken into account as oil is delivered to the common return pipe by one pump and air by the other. The oil tank must then be high enough to prevent the loss of foam through the filler orifice. A foam overflow tube to the crankcase may be incorporated. In a lubrication system with only one scavenge pump less oil foam is formed.

An oil flow of 20 to 25 l/b.h.p. h is generally used for petrol engine lubrication and 25 to 35 l/b.h.p. h is specified for compression-ignition engines. In V engines or where roller bearings are employed, the oil flow may be lower. It is an error to assume that with the use of a greater volume of oil, oil temperature will be reduced. If the amount of oil is increased, the amount of heat transferred is also increased but temperature is not affected.

Lubricating oil is sometimes also employed for piston cooling. The oil is either sprayed onto the piston by a jet from the connecting rod small end or from a special jet lodged in the crankcase. The quantity of heat removed by oil in aircraft engines is shown in Fig. 329.

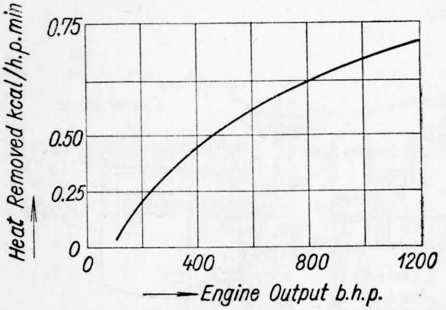

FIG. 329. Heat removed by oil in aircraft engines.

A finned sump does not aid cooling to any great extent as the oil does not flow down the cooled sump walls. Cooling is more intense on the crankcase walls, over which oil flows in a thin film. If cooling by a finned sump is to be efficient, air must be brought to circulate about its cooling area.

Best oil cooling is obtained by the use of a tube-type radiator of sufficient strength and efficiency. Oil should circulate through the cooler several times in order to keep flow velocity in the tubes sufficiently high. Provision must be made for draining the oil out of the cooler. The oil cooler should have approximately the same air resistance as a cylinder in order to utilize air pressure to the utmost.