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KNOWLEDGE GAINED FROM PRACTICAL EXPERIENCE IN THE DESIGNING OF AIRCRAFT ENGINES

By Oskar Kurtz

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I. INTRODUCTION

In recent aircraft development, the demands made on the power plant offer the engine designer a special incentive accompanied, however, by difficult problems. The greater cruising speed, which requires a slender fuselage for reducing the drag, also makes it necessary to reduce the size of the power plant and to utilize the piston displacement to a greater extent than hitherto. These goals necessitate the use of higher working loads and new basic shapes. The meeting of these requirements necessitates, above all, a constructive knowledge and experience, but also a constant improvement in the materials and in their treatment. The discovery of new basic shapes for the engine members does not progress, however, so rapidly as required for the problems of development. The designing of new aircraft engines must therefore be based largely on knowledge acquired by practical experience. Progress can no longer be made by jumps, but must be laboriously accomplished by the collaboration of educated and experienced scientists and engineers.

If Germany is behind other countries in some aspects of aircraft-engine design, this is due to technical and economic obstacles to our development since the war. Even our automobile industry was for several years in a similar situation. Just as the latter, however, finally succeeded in reaching the status of the other countries, German air-

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*"Konstruktionserfahrungen beim Bau von Luftfahrzeugmotoren." Z.F.M., Dec. 14, 1932, pp. 691-701, and Dec. 28, 1932, pp. 721-730. A lecture delivered before the Wissenschaftliche Gesellschaft für Luftfahrt in Berlin, June 24, 1932.

craft-engine designers will soon bridge the remaining gap.

While the general lines of development of aircraft engines depend on the status of the science, the most dearly bought lessons from experience, especially in the development of structural members, rightly remain the intellectual property of the factories. But just because the development of aircraft engines is difficult and takes much time, it is desirable to publish occasionally some of the results obtained for the use of all aircraft-engine designers, all the more because the present economic situation makes cooperation desirable in aviation.

It would be of especial interest and utility if the aircraft-engine works would themselves make original reports regarding their experiences. There are reasons, however, which require no explanation why it would not be possible for the factories to make direct reports of their experiences. For this reason the writer, as a member of a neutral research institute, was commissioned to report some of the experiences in question. On behalf of the D.V.L. (Deutsche Versuchsanstalt für Luftfahrt) the writer thanks the aircraft-engine works and the German Luft Hansa for so generously furnishing the data which made this report possible. Of course all the knowledge gained in recent years cannot be treated exhaustively in a single lecture. Hence only a few of the most important points will be discussed.

II. GENERAL INFORMATION ACQUIRED FROM THE CONSTRUCTION OF RECENT AIRCRAFT ENGINES

1. Water-Cooled Engines with Cylinders in Line

During recent years noteworthy progress has been made in increasing the power and reliability of water-cooled in-line engines. Figure 1 represents the fundamental structure of these engines. Here the carburetor arrangement and the distribution of the combustible mixture is made conspicuous, because these greatly affect the design and working characteristics of all the structural types.

In the conventional 6- and 12-cylinder vertical engines (fig. 1,1) two carburetors are now generally used for one row of cylinders. A portion of the intake air flows through the channels of the crankcase, which has double side walls. Thus the intake air is preheated and the oil in the crankcase cooled. A special advantage of this type of engine is its reliability without an oil radiator which, with its additional weight and drag, is always an undesirable adjunct to the power plant. This type of engine requires no oil radiator even for increased power (with compression ratios up to 7.3:1). It will pay to take advantage of this fact in new designs, also because the double side walls increase the rigidity of the crankcase. If structural reasons exclude the use of the intake air for cooling the oil, some of the relative wind can be conducted through the channels in the crankcase. A disadvantage of this type is the difficulty of obtaining, with only two carburetors, a uniform distribution of the mixture for the row of six cylinders.

This type was further developed by the addition of a blower before which an induction carburetor is placed (fig. 1,2). Important advantages of such engines are a uniform mixture distribution and quiet running, but an oil radiator is necessary, unless some of the intake air or relative wind is led through the crankcase. If an oil radiator is used, the mixture must be preheated by the cooling water or by the exhaust gases. The mixture blower, generally in the form of a low-pressure compressor for a compression ratio of about 1.2, can be developed, with relatively small alterations (i.e., by increasing the revolution speed and enlarging the rotor wheel), as a medium-pressure blower for full-pressure altitudes up to about 4,000 m (13,123 ft.). Up to this altitude, according to previous experience, no control clutch nor step gear is necessary for operating the blower.

The engine with declutchable supercharger is of a type similar to that used in automobiles (fig. 1,3). With the blower in gear, the carburetors work as pressure carburetors, the normal air inlet being automatically closed. These engines can run temporarily with a high power excess (about 25 percent), but they do not equal the mixture-blower engines in the excellence of the mixture distribution in normal operation. The extent to which supercharged engines will be used in aviation, must depend on their development.

Altitude engines for full-pressure altitudes above 4,000 m (13,123 ft.) are equipped with disconnectible superchargers and control clutches or step gears, pressure carburetors being generally used (fig. 1,4). Multistage centrifugal or Roots blowers are being developed as preliminary compressors for such altitudes. Further information is not yet available, since no such altitude engines have yet been placed in actual service.

It is known that, despite the most careful adjustment, the mixture distribution in large in-line engines of the conventional type needs to be improved. How great the numerical differences may be, is shown by several investigations conducted by Oestrich in the D.V.L. at the suggestion of the writer. Figure 2 shows the experiments with a conventional six-cylinder engine at full throttle with one carburetor. The lower line represents the mean content of unburned $CO + H_2$,* as determined with a Siemens exhaust-gas tester; the middle line, the exhaust-gas temperatures; and the upper line, the maximum combustion pressures measured with the D.V.L. maximum-pressure gage. The inequalities in the mixture distribution are shown by the differences in the unburned constituents in the individual cylinders. The similarly fluctuating peak pressures (for reasons which will be explained) are not conclusive for judging the distribution of the mixture. These conditions are still worse in a 12-cylinder engine with two carburetors for each row of cylinders, in which the outer cylinders receive the poorest mixture (fig. 3). Here the differences in the maximum pressures are not great. Unfortunately no engine with a mixture blower was available for a comparative investigation. The experiments, however, corroborate the practical experience that special measures are necessary in large engines for improving the mixture distribution, either by mixture blowers or later by fuel injection. With a mixture blower, the use of a third carburetor for a row of six cylinders is probably superfluous.

The present status of development in Germany is illustrated by the following examples of recent German water-cooled in-line engines.

Figure 4 represents the 12-cylinder Junkers engine L 88 with spur-gear drive, which is made in the individual-cylinder type with enclosed gears and a continuous gear housing. The carburetors are mounted on the outside

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*No guaranty for the absolute correctness of the measured values can be given, since the investigation of the accuracy of the instrument has not yet been finished.

for the sake of easy accessibility. A noteworthy innovation is the Junkers internal shock absorber in the spur-gear drive, which, without increasing the weight, almost completely damps the detrimental rotary-oscillation resonance. Since the internal damping simultaneously renders it possible to get along with smaller crankshaft dimensions, it provides in-line engines with such a favorable control of the vibrational conditions, that it will pay to make use of it in new designs. The L 88 engine has also been successfully used with an oil clutch and remote propeller drive in the Junkers large airplane G 38.

The 12-cylinder engine BMW IX is provided with a mixture blower and a suction carburetor. Originally the mixture was led through a special pipe from the blower to each row of cylinders. This arrangement yielded no satisfactory distribution, however, since pressure differences occurred in the two pipes, which could not be eliminated even by joining them. A uniform distribution was obtained in the most recent type (fig. 5), in which a pressure pipe from the blower divides into two delivery pipes for each row of cylinders. This engine is of the individual-cylinder type, with two valves and open control mechanism, and is characterized by its simple design, ease of maintenance and low cost of production.

The Daimler F 2 engine (fig. 6) has a disconnectible supercharger. The four-valve cylinders of each row are held together by a light-metal cap. This engine was not designed simply for aircraft, but is also used on motor boats. With the supercharger running, the power can be raised from 800 hp to 1,000 hp (i.e., 25 per cent) for a period of 30 minutes.

Our modern water-cooled engines are still built with relatively large piston displacements and corresponding dimensions which are detrimental for installation in many airplanes. Several other countries, which, after the war, had considerably more funds to use in engine development and also the knowledge acquired in racing, now have an advantage over us in this respect and make engines of smaller dimensions which can be more favorably installed. It is to be hoped, however, that our factories will soon be able to overtake them.

2. Air-Cooled In-Line Engines

The development of air-cooled in-line engines, which can accomplish noteworthy results in the low-powered class, has reached the same stage in Germany as in other countries. The design of these engines is determined largely by the cooling of the cylinders. Due to the unavoidable irregular heat expansion in operation, the development of these engines has hitherto been successful only in the individual-cylinder type. Foreign attempts to produce air-cooled in-line engines in the block type have proved unsuccessful, and designers have returned to individual vertical cylinders. The cooling of four-cylinder rows by the relative wind has been satisfactorily accomplished, even on high-speed airplanes up to 25 hp per liter (61 cu.in.) of piston displacement. On the contrary the six-cylinder in-line engines cooled by the relative wind met with no permanent success.* The advantages for visibility and mounting resulted in air-cooled in-line engines being made chiefly with the cylinders inverted.

The Hirth HM 60 (fig. 7) is the smallest German in-line engine. Its design is noteworthy in various respects. A Hirth built-up shaft is used for the crankshaft drive, with roller bearings for the revolving parts and needle bearings for the oscillating parts. Thus it is possible to lubricate the bearings with fresh oil and to conduct part of the intake air through the inside of the crankcase. In addition to an effective cooling of the crankcase and bearings, a very small oil consumption of 0.5 to 1g/hp/hr. is effected by these measures, to which the small clearance of the Bohnalite piston may contribute. Moreover it is thus easier to control the temperature of the crankcase, since it is subjected to only slight temperature variations between the hot and cold state of the engine. These principles of construction have thus far yielded good results in practical flight tests.

The four-cylinder in-line engine Argus As 8 (fig. 8), the winner of the 1930 European circuit flight, has a greater piston displacement and power. This engine is

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* In the meantime the Italian six-cylinder engine Colombo S 63 was used successfully in the European circuit flight of 1932. This engine demonstrated the possibility of cooling rows of six cylinders at high flight speeds. (Compare O. Kurtz, "Die Flugmotoren des Europa-Rundflugs 1932," Z.F.M., Vol. 23, 1932, pp. 577-581.)

characterized by its simplicity. The combustion air is taken directly from the surrounding atmosphere. The mixture is heated by the exhaust gases before it enters the carburetor. Since the crankshaft is provided with plain bearings, lubrication by continuous oil circulation with a dry sump is employed. The upper crankcase sump serves as an oil holder and is cooled by the relative wind. In the further development to greater powers this arrangement was nevertheless abandoned, and a specially cooled oil container was provided for avoiding excessive temperatures. The power of the Argus As 8 R engine was raised from 80 to 160 hp, at which the victory in the 1932 European circuit flight was won. This power corresponds to a volumetric efficiency of 25 hp per liter (0.41 hp/cu.in.), which is high for air-cooled aircraft engines.

Doubling the As 8 produced the 8-cylinder As 10, with the cylinder rows placed at an angle of 90 degrees to each other (fig. 9). Since many parts of the As 8 are used, the mixture delivery, crankshaft bearings and lubrication remain the same. Even the circulation of the cooling air is fundamentally the same as in the As 8. The previous flight tests and the preliminary tests made in the D.V.L. for the installation of the As 10 as a pressure engine showed that there were no difficulties in the way of using the relative wind for cooling even the 8-cylinder two-row in-line engine.

Aside from an experimental engine, high-powered, air-cooled, in-line engines have not yet been built in Germany. America and Italy have developed air-cooled 12-cylinder, in-line engines, but the control of their temperature by the relative wind has not proved satisfactory. Artificial cooling of the cylinders by a blower driven by the engine gives good results, since only a small amount of energy is absorbed by a suitable blower and the requisite pipes.

An experimental example of this type is the 12-cylinder Daimler F 3 engine (fig. 10). Although only one was made, this engine is interesting as a step in development, since the knowledge gained from experimenting with it constitutes the basis for its further development. The cooling is effected by six centrifugal blowers between the cylinder rows, whose common shaft is driven at the rear of the engine. In the D.V.L. experiments the cooling air was so uniformly distributed along the cylinders with the aid of guide plates that the cylinder temperatures averaged 300° C. (572° F.) at

N_e = 600 hp and n = 1,820 r.p.m., that is, not much above the permissible limit for continuous operation according to previous tests. The blower absorbed about 36 hp or about 6 percent of the engine power. In further developments a different arrangement of the blowers will doubtless be employed.

3. Air-Cooled Radial Engines

In air-cooled radial engines the design is not influenced so much by the mixture delivery and the blower arrangement as in in-line engines. In small five-cylinder radial engines, a satisfactory mixture distribution with good preliminary heating of the mixture is obtained with one carburetor (BMW Xa, S.H.13a, Salmson AD 9). The somewhat larger 7-cylinder sporting engines require two carburetors to obtain a uniform distribution (S.H.14, S.H. 14a). In engines of medium and high power three arrangements are now used (fig. 11).

The mixture-distribution spiral, as used in the old type Jupiter engine, long gave satisfaction, but has been superseded by the mixture blower. Mixture blowers and suction carburetors are now regularly used in high-power radial engines. The blower and its driving gear, which will be considered later, have been developed into reliable structural units.

For altitude engines up to full-pressure altitudes of 4,000 m (13,123 ft.), the mixture blowers can be converted into superchargers by suitable alterations (Cf. II, 1). Higher full-pressure altitudes can be reached with comparative ease simply by a two-stage arrangement, by installing a relatively small auxiliary blower with a disengaging gear in front of the mixture blower. In this case the carburetor is between the auxiliary compressor and the mixture blower, which forms the second stage. Whether a cooler for the air is required after the first stage will depend on the compression ratio and has not yet been determined. No experiments with this arrangement have been tried in Germany.

Among the recent smaller radial engines, the BMW Xa and the Siemens S.H.13a are well known. The S.H.14a (fig. 12) is a further development of the 7-cylinder S.H.14, for the purpose of increasing the power and reducing the weight and resistance. With considerably reduced weight (128 kg (282.2 lb.) instead of 146 kg (322 lb.)) and over-all diameter (93 cm (36.6 in.) instead of 96 cm (37.8 in.)) the power was increased from 110 to 160 hp and a power loading of 0.8 kg/hp (1.76 lb./hp) (very good for an engine of this size) was attained. The cylinder heads were shrunk on to reduce head resistance. The weight was reduced by making the crankcase of cast elektron which has yielded good results in these dimensions, and by the use of a lead-bronze journal bearing instead of a ball bearing for the master connecting rod. The latter measure and the improved shape of the cylinder head enabled a reduction in the diameter.

We are no longer behind other countries in the development of large air-cooled engines. Two recent engines of this class are the S.H.20 developed by Siemens and the Hornet built under license by the B.M.W., the two engines having about the same power with somewhat different piston displacements. Both engines combine the structural principles which have proved successful for this power class, namely, forged duralumin crankcases, open steel cylinder liners with shrunk-on cast light-metal heads, two valves and mixture blowers.

The largest German radial engine is the As 7 developed by Argus with a maximum power of about 800 hp (fig. 13). This engine, which was first built for experimental purposes, exhibits all the characteristics of a present-day radial engine designed for the minimum head resistance. The design of the four-valve cylinder with housed-in control mechanism is adapted to high thermal loading and the attainment of the minimum drag with individual cylinder housings. For this purpose the intake and exhaust passages have an oval form, even the pipes being correspondingly shaped. The crankcase consists of two forged elektron parts.

III. DEVELOPMENT OF A FEW SUBASSEMBLIES

1. Cylinder Types

The cylinder, together with the piston and valves, is the part most highly stressed by the heat of combustion. Since its development requires much experience, factories are loath to abandon a successful cylinder type for one that has not been well tested.

Hitherto in Germany, closed steel cylinders with open valve mechanism and welded-on guides for two valves were chiefly used on water-cooled engines. This type has now been well developed for high cylinder performances. Figure 14 shows two such cylinder forms, an older form 1 and the latest form 2, which is based on the following experiences. In the transition to the higher cylinder performances, the arched cylinder head of the original form became warped as a result of the increased thermal loading. This caused the valves to lose their tightness, so that the seats of the exhaust valves were warped by the overheating. The warping of the valve seats was also partly due to leanness of the mixture, which could not always be avoided, despite careful carburetor adjustment. An effective remedy was provided by a substantial reenforcement of the cylinder head. Moreover, for lengthening the life of the cylinder, valve seats were made of chrome-tungsten steel for the exhaust valves and the valve stems were also strengthened. The rectilinear vertical welding of the water jacket did not prove to be sufficiently reliable in continuous commercial use. Frequent leaks from deformation and corrosion necessitated a reenforcement of the sheet-metal jacket, the lower part of which is now made with many corrugations and an oblique welded seam to allow for mechanical deformation and heat expansion. The jacket lid is cupped and separately welded. Corrosion-proofing is considered elsewhere (reference 1). The cylinder is provided with many fins for more rapid heat transfer. Moreover the flange at the foot of the cylinder is considerably reenforced, in order to increase its resistance to bending in operation and thus prevent failure of the flange and water jacket. This type is now sufficiently reliable. The limit of the attainable mean pressure with this type for maximum cylinder outputs of 67 hp is 9 to 9.5 kg/cm^2 (128 to 135 lb./sq.in.) at a compression ratio of 7.3. It is determined chiefly by the thermal stressing of the relatively large valves.

Another solution (Junkers L 88) for the closed steel cylinder is shown in figure 15,1. Here also the valve chambers with guides are welded to the liner machined from a single piece. The valve mechanism is protected by a cover, and the cylinders are held together in a block by the valve-mechanism housing, as in the Fiat type. The cylinder output is 67 to 70 horsepower.

For higher piston loads a type was developed with flanged light-metal head, four valves and pressed-in valve seats (fig. 15,2). The construction of this cylinder, although only experimental, indicates the present trend.

The light-metal cylinder head, the division of the valves and the favorable shape of the combustion chamber render it possible to lower the detonation limit and raise the thermal load of the cylinder.

For the F2 engine, Daimler developed a design with four valves, similar to an earlier one by Packard (fig. 16,1). The liner, with its head, is machined from a single steel block. The steel valve guides, channels and water jackets are welded on. The head is relatively rigid, despite its thinness. With a bore of 165 mm (6.5 in.) this cylinder yields 70 hp in normal operation with p_e = 8.5 kg/cm² (120.9 lb./sq.in.) and 83 hp with a supercharger and p_e = 9.9 kg/cm² (140.8 lb./sq.in.

A similar design has been developed as a Diesel cylinder with precombustion chamber (fig. 16,2). The cooled precombustion chamber is between the valves in the middle, with the injection nozzle above it. The cylinder head is reenforced, in order to withstand the high pressures. The central position of the precombustion chamber is made possible by the four-valve design, though this necessitates two camshafts. Even the valve sections had to be reduced somewhat, though this does not greatly affect the cylinder charging, since the throttling in the intake pipe is eliminated.

The cylinder material is usually chrome steel with a tensile strength of about 100 kg/cm² (1,422.4 lb./sq.in.). A reduction in the wear was expected from the use of nitrided liners, but results thus far have been to the contrary. While nitrided cylinders have thus far worked well on water-cooled engines, they have not been satisfactory on air-cooled engines and have worn out the piston rings rapidly. Hence the results of further experiments must be awaited.

The cylinders of air-cooled engines have passed through various stages of development. After long research, in which the Siemens Company did pioneer work through the investigations by Gosslau, two designs were evolved, which are now universally employed and which withstand the thermal loading of the air-cooled cylinder:

a) The open steel liner with light-metal cylinder head screwed on hot and shrunk on. This relatively expensive type is standard for large engines whose cylinders have a high thermal loading and whose air resistance must be kept small. Cast cylinder heads are now used, which are cheaper than forged ones, with their fins and channels being machined out of a single piece. The Y alloy, or some similar heat-resisting light metal, is a suitable material.

b) For moderate thermal loading, i.e., in smaller engines which must be cheaply produced, the cylinder head is attached to the liner by screw bolts, with a soft copper gasket for tightness. With this method of attachment the heat flow is not so good as with the shrunk-on head, though good results have recently been obtained (As 8 R engine) even at quite high thermal loadings.

Figure 17 shows comparative views of present-day cylinder types. The Jupiter type with closed liner (fig. 17,1) is no longer made. Its advantages consisted in a certain lack of sensitiveness under heavy loads. The Jupiter engine, which was originally built as a high-powered engine for fast airplanes, is now used on relatively slow commercial airplanes. Under poor cooling conditions (e.g., when used as a pusher engine in tandem arrangement), the cylinder heads sometimes become warped and then by overheating cause carburetor back fires. In order to lengthen the life of these cylinders, exhaust-valve seats of chrome-tungsten steel were pressed in.

A still greater improvement is the design developed and tested by Siemens with a screwed-on light-metal head and conical shrunk ring, which is secured by a nut and insures a tight fit of the contact surfaces (fig. 17,2). The shrunk ring was found not to be absolutely necessary, as shown in two other designs (fig. 17,3 and 4). Hence the bottom fin of the light-metal head is reenforced.

Figures 17,4 and 5 are successful designs of the screwed-on type for small engines. These afford a comparison of the valve-seat fastenings of light-metal heads. For the valve seats, which are mostly made of aluminum bronze, three methods of fastening have given good results, namely, the slightly conical shrunk-in seat, the pressed-in seat, and the screwed-in seat which is now preferred.

Two cylinder designs of the S.H.20 engine are shown in figure 18:

1. Former design with shrunk ring;

2. New design without shrunk ring, but with push-rod housing and valve-mechanism compensation of the Siemens type.

The latter type offers less head resistance than the other.

With respect to the thermal expansion of an air-cooled cylinder, provision must be made for balancing the valve clearance, for which special cams as well as devices similar to the Jupiter design with Invar-steel push rods for linking the cylinder head and rocker bracket have proved satisfactory.

2. Blower Driving Gears

On the basis of the knowledge that mixture blowers belong to the essential equipment of large aircraft engines, such blowers and their driving gears have been undergoing development for several years, so that considerable knowledge has been acquired concerning them. Two methods were first investigated. In America the high-speed blower with small rotor diameter and special driving gear with overrunning clutch has been developed. In Germany and elsewhere mixture distributers revolving at crankshaft speed were built. These are structurally simpler, but require rotors of larger diameter. This type, which is still in use, has not proved entirely satisfactory, disturbances in the mixture distribution having occurred in many engines, so that the desired performances were not attained. The mixture blowers are now made universally in the form of high-speed centrifugal blowers.

The structural difficulties lie in the designing of reliable step-up gears of 1:7 and 1:10.5 in a restricted space, in protecting from vibrations and shocks the highly stressed gears and the light rotor made of steel, duralumin, or elektron, and in making the blower housing impervious to the oil in the crankcase. The peripheral velocity of the rotors varies between 120 and 200 m/s (394 and 656 ft./sec.) at 13,000 to 20,000 r.p.m. The transmission of vibrations to the blower gearing can now be satisfactorily prevented in various ways. According to experience with radial engines in which no great torsional vibrations occur, overrunning plate clutches or flexible drives suffice and are reliable. On the other hand, these simple damping or flexible linkages have not proved feasible for vertical engines, due to the excessive wear and tear. The torsional vibrations require flexible clutches, which can absorb more energy without excessive wear. The influence of the blower drive on the torsional vibrations in in-line engines has not been fully determined. The natural vibration frequency of the crankshaft drive is reduced by the mass of the highly stepped-up and energy-absorbing rotor. Probably the blower with its flexible clutch has a damping effect on the vibrations. The oil seal requires still further improvement in new designs.

Figure 19 shows two examples of blower drives for radial engines with different types of clutches. In one engine it is a simple spring-loaded overrunning plate clutch which limits the turning moment, while the gearing in the other contains a damping device which absorbs the peaks of the turning moment. The overrunning plate clutch has been used in many types of drives and has given satisfaction. The clutch must be adjusted to the correct turning moment, since the engine speed varies with excessive sliding friction. The flexible blower drive has done well in long experimental operation. In the blower drives, a large rotor shaft is necessary in order to avoid deflections and to insure smooth operation of the gearing. In both driving gears, the oil seal is effected by long flowback spirals and splash rings.

While mixture blowers and medium-pressure compressors are geared to the engine, a disengaging clutch is necessary in the case of superchargers and displacement blowers for high pressures, combined in special cases with a locking device. Figure 20 represents the supercharger driving gear and clutch of the Daimler F2 engine. Since this supercharger runs only for brief periods, it can be driven by a planetary gear directly from the crankshaft, whereby, in the disengaged condition, the rotor revolves at the speed of the crankshaft. The blower is engaged by retarding the gear casing by means of a band brake actuated by oil pressure. A plate clutch operated by oil pressure was developed by Argus for driving a Roots-type displacement blower (fig. 21). Oil delivered under pressure by a special pump, when the clutch is thrown in, acts on two pistons which press the plates together by means of levers. The clutch revolves at the speed of the crankshaft, since the gearing is in the blower casing. Figure 22 is another example of such a blower clutch, which belongs to the As 10 engine. In this clutch the movable cylinder presses the plates together without the help of levers. Noteworthy is the compact design which was developed from experience with previous clutches.

3. Reduction Gears

Reduction gears are more often used, as engine speeds are increased. In Germany the Farman bevel gears are used for radial engines, while spur gears are preferred for in-line engines, due to advantages for installing.

The Farman gear (built under license by the B.M.W.) is now very reliable. In the development of this gear there were several difficulties to be overcome, due chiefly to lack of rigidity of the individual parts and to the mounting of the planetary pinions, which are especially sensitive with respect to lubrication. The planetary pinions were originally mounted on bronze bushings with white-metal linings. For the high and uncontrollable loads, which may occur if all three pinions do not transmit the turning moment uniformly, the white-metal bearings have not proved satisfactory for large vertical engines despite the most thorough lubrication. The life of these bearings was short, and trouble sometimes resulted from overheating. The planetary pinions were then mounted on rollers which ran on hard intermediate bushings, thus eliminating all troubles with in-line engines. In radial engines, as in many other cases, the results have been different. Here the white-metal bushings have worked well. Many of these gears have been used on Jupiter engines with satisfactory results.

Figure 23 shows two radial-engine gears, an older form with a reduction ratio of 2:1 for the Jupiter engine (1), and another with 1.6:1 for a more recent radial engine. The new type is more compact, the plate carrying the pins for the planetary pinions, with wedge-shaped outline, being pressed on to the driving shaft. The new type was necessitated by the development of the shaft with a flange for attaching the hub. Moreover, it makes it possible to change the reduction ratio by changing the pinion plate, without exchanging the whole gear. Here the planetary pinions are mounted with lead-bronze bushings on hardened pins. The forward bearing of the driving shaft is designed according to the most recent information. Hitherto, roller self-aligning bearings were used for the axial support of the driving shaft. These bearings can be readily installed and simultaneously absorb the transverse forces and the longitudinal thrust of the propeller. The slight axial play of these bearings, which increases somewhat with use, is transmitted to the shaft and causes variations in the propeller speed when, for example, load variations are produced by gusts. Although this problem has not been entirely solved, roller bearings for absorbing the transverse forces and longitudinal bearings for absorbing the thrust have been installed on the new engines. Furthermore, the new gears have been simplified by eliminating the ring, which has proved superfluous for the uniform distribution of the forces to the pins of the planetary-pinion plate. The use of flanged shafts with centering splines, instead of conical shafts, facilitates the mounting and replacing of the hubs.

Roller bearings for the planetary pinions have been installed in the gears of in-line engines (fig. 24). The new type (fig. 24,2) has been further developed in the same direction as in radial engines. The conical shaft in the older gear (fig. 24,1) necessitated very careful polishing of the hub, in order to obtain a smooth fit for the cone. Despite this measure, the cone was galled by the working of the hub, which made it difficult to remove the hub and led to the introduction of a thin shim of a copper-zinc alloy between the cone and the hub. Here also the flanged shaft promises to be an improvement.

Spur gears are just coming into use in Germany for in-line engines. Figure 25 shows an example in which the connection between crankshaft and driving pinion, elastic spring clutch and spur teeth, is noteworthy.

IV. SPECIAL KNOWLEDGE ACQUIRED FROM EXPERIENCE IN DESIGNING VARIOUS ENGINE PARTS

1. Pistons

Among the parts of an aircraft engine the thermally and mechanically heavily loaded piston is a structural member whose design often presents considerable difficulty. Until recently the skirt-type piston with closed skirt was chiefly used for water-cooled in-line engines, while slipper-type pistons, which are lighter and produce less friction, were principally used for air-cooled radial engines. Slipper-type pistons are now being successfully used in water-cooled engines, while skirt-type pistons are again coming into use in air-cooled engines, although in an improved form. The reason for abandoning slipper-type pistons in air-cooled engines is partly on account of deficient oil-tightness due to the necessarily great piston clearance. It has often been found necessary to add an oil-scraper ring at the bottom which is possible only with a skirt-type piston. Bohnalite pistons with Invar strips have thus far been used only in the Hirth HM 60 engine. Whether constructional and weight considerations will permit its wider introduction, which, due to the small piston clearance, might increase the effective power, must yet be determined by comprehensive research.

Figure 26 compares the skirt-type pistons now in use and shows that, in the large sizes, the structural designs are similar. The piston-pin bosses are supported both above and below by a strong rib. We have learned to make the piston heads of large-bore, in-line engines quite thick in the middle and to insure gradual transitions between the head and the skirt. Even the web above the top piston ring was made thicker, in order to protect the first ring, which is generally rather dry, from the action of the flame. The number of rings varies, but is never less than four: two compression rings and two oil-scraper rings. One of the scraper rings, which usually have the form of U rings with sharp scraping edges and oil holes or slots, is generally located below the piston-pin bosses. The use of two scraper rings is necessitated by the large piston clearance, which, on large modern, water-cooled engines, is between 0.4 and 0.5 mm (0.016 and 0.020 in.) for bores of about 160 mm (6.3 in.) and between 0.75 and 1 mm (0.03 and 0.04 in.) on air-cooled engines. This number of rings is also necessary when the cylinder liners ovalize. The piston pins are generally secured by "Seeger rings," though "mushroom safeties" are also used. When Seeger rings are used, the portion of the web outside the groove must not be too thin, since otherwise the light metal might break at this point. Figure 27 shows a new form of skirt-type piston for a large radial engine.

As piston materials, heat-resisting, cast, light-metal alloys containing nickel and copper, as well as chill-cast Y metal, have been successfully used. Pistons forged from Y metal have also been tried, but their running characteristics were not so good as those of the cast pistons. Careful testing of the ring clearances in the grooves is always desirable, because tight rings soon cause seizing of the piston; though this is also a matter of lubrication. No definite rules can yet be made regarding the number of rings and the best shape for their ends. This must also be determined by systematic experimentation.

The good results obtained with slipper-type pistons in air-cooled Jupiter engines encouraged their introduction into a few water-cooled in-line engines, which, however, had bores of 160 mm (6.3 in.) instead of 146 mm (5.75 in.).

The results obtained over a long period of operation with these pistons have yielded considerable information regarding their design for large cylinders. On the whole, the slipper-type pistons have favorably affected the running characteristics of the engines and even caused them to run quieter as a result of the reduction in the inertia forces. Longer service, however, developed unforeseen disturbances. As shown in figure 28,1 and 2, these pistons were made in two forms, with one and with 2 supporting ribs for the piston-pin bosses, the single-rib type being somewhat lighter. Since the length of the piston pin and therefore its bending stresses are smaller in slipper-type pistons, the wall thickness of the pins was somewhat reduced, thus permitting a bending load of about 3,100 kg/cm² (44,093 lb./sq.in.), that is, a value which had proved sufficient in other engines in long operation. Although the computed bending strength was sufficient, the pins were not sufficiently resistant to ovalization due to the thinness of their walls. This ovalization resulted in transverse breaks in the piston-pin bosses (fig. 29) and longitudinal breaks in the piston pin itself. These breaks occurred only on single-ribbed pistons, in which the piston-pin boss supported in the middle, is somewhat yielding, while the considerably more rigid bosses of the two-ribbed pistons did not break. Comparative breaking tests with a series of different piston types and with piston pins of different wall thickness indicated that the two-ribbed slipper-type piston had the same strength as the skirt-type piston. These results showed that, in future, the piston designer must take into account not only the bending strength, but also the ovalization of the piston pin, and the fact that the piston-pin bosses should be supported as rigidly as possible.

From this it does not follow that only the two-ribbed design with its correspondingly greater weight is the only reliable one for large slipper-type pistons, but it appears structurally possible, even for large bores, to endow the lighter single-ribbed type with sufficient strength by a suitable design. The otherwise good results and the advantages of the slipper-type piston recommend the further development of this type.

2. Valves

Improvements have recently been made in the design, material, and cooling of the valves. While the form of plate valves used in automobile engines predominated a few years ago, the tulip or cup shape now predominates, at least for exhaust valves. Intermediate forms have also been successfully used. The longer transition radii of the tulip valve favor the strength and the heat flow to the stem but, on the other hand, the valve guide cannot descend so far as in the case of the plate valve. Figure 30 shows a few of the present valve types.

1. Plate intake valve for water-cooled engines.

2. Tulip valve for exhaust of water-cooled engines and for intake and exhaust of air-cooled engines.

3. Valve with salt cooling.

4. Valve for small engines.

The fitting of the spring retainer is always fundamentally the same. The split conical sleeve has generally proved to be the safest. Exhaust valves for highly stressed engines generally have hollow stems of larger diameter. In valves with high thermal loading, good results have been obtained by cooling with salts of various composition, which are introduced into the stem in the liquid state, the stem then being tightly closed by a pressed-in plug (fig. 30,3). The heat exchange is facilitated by evaporation in the hot lower part of the stem and by condensation in the upper part, so that the temperature is lowered. Mercury is no longer used for cooling the valves, because it is rather heavy and volatilizes too easily. On the other hand, good results have recently been obtained with motor trucks by inserting copper plugs into the hollow stems. Copper plugs will probably be good to use in aircraft engines also, but further investigations should first be made.

Exhaust valves are now made chiefly from austenitic steels WF 100 or KE 965, which have high heat resistivity and have yielded good results. The operating characteristics of these valves require, however, special material for the guides and seats. "Kuprodur" (copper-duralumin) bronze has worked well for valve guides, while various aluminum bronzes have been tried in the light-metal valve seats, and tungsten steel and chrome-tungsten steel in the steel cylinder heads.

Since austenitic steel cannot be made hard enough, valves made from this material require caps of case-hardened steel. Even case-hardened surfaces do not always withstand high stresses, but are indented or cracked. Recently, therefore, small plates of hard metal, like "Widia" steel (carboloy), as pressure surfaces, have been rolled into the valve caps with good results.

3. Bearings

The short life of the big-end bearing of the connecting rod was until recently one of the chief causes of the short life of various aircraft engines. The original white-metal bearings could not long withstand the high surface pressures which ran up to 150 kg/cm² (2,134 lb./sq.in.). For this reason, it was recommended not to exceed 80 to 90 kg/cm² (1,138 to 1,280 lb./sq.in.) in long continuous operation. The fear of not being able to withstand the high surface pressures, with bearings whose dimensions are limited by structural conditions, led to the installation of roller bearings, the construction of which, with the use of case-hardened crankshafts, often considerably delayed the development of the engines. The reliability of the bearings has been considerably increased, however, by recent improvements in design and in materials. Figure 31 compares the big-end connecting-rod bearings of present radial and in-line engines. The most frequent failures in journal bearings begin with the cracking of the white-metal lining. When the cracks grow so large as to destroy the bond between the lining and the chill casting and cause the latter to crumble, reliability of operation is endangered. On the other hand, bearings with single cracks, which do not change, may, under some conditions, last a long time. In some of the in-line engines the injuries to the bearings were due chiefly to distortions of the connecting rods, which were too weak to prevent deformation at the transition from the shank to the big end of the connecting rod and at the connecting-rod cap. The connecting rods were distorted at these points by the inertia forces at the upper dead center. These distortions caused excessive stresses and the formation of cracks in the white-metal liners. Improvement was effected by reenforcing the rods at the weak points. This experience shows that distortions of white-metal bearings must be avoided.

A great improvement was effected by the use of lead-bronze bearings, the development of which was begun several years ago. The good results obtained with these bearings in various engines over a long period of time make it probable that they will come into more general use in aircraft engines as well as in other engines. Lead bronze is an alloy with a casting temperature of about 1,100° C. (2,012° F.) and a composition of about 66 percent copper, 33 percent lead and 1 per cent iron. Of course the copper and lead content can be varied. The best composition for the particular case must first be determined by the producers. The chief advantage of lead-bronze bearings is the great strength of the castings, which prevents their rupture, even when somewhat distorted, and gives them good running characteristics, light wear, high loading capacity and a certain insensitiveness even with poor lubrication. In the making of lead-bronze bearings, the following points have been found to require special attention.

1. The shaft material.- Despite the lower strength of the lead bronze in comparison with the shaft material, shafts of soft-tempered steel cause a somewhat greater wear than hard-tempered shafts. Soft shafts score easily at first, though they improve somewhat from continued running, as they become polished.

2. Bearing clearance.- This must be greater than for white-metal bearings. Moreover, hardened crankpins require somewhat more clearance than soft ones. Definite statements cannot yet be made regarding the requisite bearing clearance, since it depends on the size and stresses of the bearings and must be accordingly determined for new engine types by further experimentation.

3. Lubrication.- Lead-bronze bearings require plenty of oil and indeed, more than white-metal bearings. The oil must be distributed evenly, through suitable grooves, over the whole surface of the bearings. The oil film is therefore relatively transient and must be frequently renewed.

Thus far the following results have been obtained. A BMW Xa engine with lead-bronze bearings was run for a total of 406 hours without showing any wear worth mentioning. In a BMW Va engine of the D.L.H. (Deutsche Luft Hansa), which, after running 203 hours, was discarded for other reasons, a wear of only 0.02 mm (0.0008 in.) was measured in one direction and 0.05 mm (0.002 in.) in the direction perpendicular to the latter. The shaft and bearing liner were somewhat roughened. In Hornet engines, after running 325 hours, a wear of the crankpin of 0.07 mm (0.0028 in.) was measured, while the lead-bronze bushings showed no measurable wear. Presumably this greater wear of the Hornet crankpin is attributable to the lower strength of the material.

Another advantage of the lead-bronze bearings consists in their good behavior in the absence of proper lubrication. If the oil delivery is interrupted, the higher temperature causes lead to separate from the alloy and to form a film which lessens the wear of the bearing. Destruction of the bearing by squeezing out, as in the case of white-metal, is thus avoided. Nothing conclusive, however, can yet be said regarding the behavior of lead bronze under the various loading conditions occurring in aircraft engines. According to tests up to date it may be assumed that lead-bronze bearings have a higher loading capacity and probably a better heat conductivity, due to their copper content, than white-metal bearings. It is not yet known, however, just where the limits lie.

The following facts have also been learned concerning journal bearings. Direct casting of the small end of the connecting rod from white metal (especially Hoyt metal), which is common in the construction of automobiles, has yielded good results with various engines. Torque-stand tests by the D.V.L. (Deutsche Versuchsanstalt für Luftfahrt) for 100 hours on a BMW Va engine with thermite bearing metal, an alloy consisting essentially of lead with additions of antimony, tin, nickel and cadmium, showed that this metal also works satisfactorily under certain conditions. These bearings were cast by the centrifugal method.

A test with needle bearings for the connecting rods of a BMW Va engine, which was extended with good results over a running time of 120 hours on the torque stand and on an airplane, showed that needle bearings may also be reliable for this purpose. They require case-hardened journals, however.

Roller bearings are used with both solid and split connecting rods. The rollers generally bear directly on the hardened crankpins and roller races of the connecting rods. In split connecting rods the joints are provided with rounded indentations to prevent the rollers from running over, a device which has worked well. Although roller bearings are now very reliable, there is a disadvantage in the costly production of the hardened crankshafts, in which there are relatively many rejections, and in the difficulty of obtaining uniform hardness in all the crankpins. Figure 32 shows worn places from roller bearings on a case-hardened crankshaft, such as are sometimes produced on this type. Even here improvements have recently been made in the finishing. Simpler production is effected by using built-up crankshafts of the Hirth type. It is now regarded as a rule in designing not to make the length of the rollers greater than their diameter, though longer rollers have given good results with a suitable design of the bearings. Special care must always be exercised in making the roller race in the hardened rod, it being important to round the edges of the oil holes and lubrication grooves well, since it has been found that the crumbling of the hardened layer begins at these points and then leads to the destruction of the race.

While forked connecting rods with journal bearings are much used on double-bank engines in other countries, articulated rods are commonly used in Germany, for which rollers were originally used in the bearings of the articulated rods. Since the race of the articulated connecting-rod bearing likewise had to be hardened, stresses were at first produced by the hardening process, as was found by cutting the rods in two. These stresses caused breaks in the bearings and consequently in the connecting rods, the occurrence of which was sometimes facilitated by the oil holes. The effect of the bearing design on the stress is shown in figure 33, which shows the results of a contraction-expansion test made by the static-test division of the D.V.L. on an articulated connecting-rod bearing with rollers and journal under operating load. An increase of about 30 per cent is produced in the stress by the local overloading in the roller bearing as compared with the journal bearing, though with a less favorable relative position of the rollers. On the basis of these tests, the design was improved by reducing the stress on the master connecting rod by reenforcing the articulated connecting-rod bearing, which was now left unhardened, and by using a journal bearing for the articulated connecting rod. This type is now in use, with no failures reported as yet.

4. Crankshafts

The general construction principles for the crankshafts of aircraft engines are now well known, after the torsional-vibration breaks of a few years ago afforded the occasion for a thorough investigation. The true fatigue strengths of crankshafts in torsion are still unknown, but this question is now an object of research.

While previously only the very best chrome-nickel steels were used for crankshafts, tests were recently made by the D.V.L. with manganese-silicon steels containing no nickel and hardly any chromium. The static breaking strengths were between 97 and 140 kg/cm^2 (1,380 and 1,991 lb./sq.in.). Nine such crankshafts were tested in 100-hour endurance runs in BMW Va engines under full load, with only two failures. They may therefore be regarded as suitable for aircraft engines under certain stress conditions, as well as for motor-truck engines.

The new structural principles for the crankshafts of in-line engines are exemplified by the As 10 crankshaft with counterweights and 90° spacing of the crank arms (fig. 34). For attaining a high natural vibration frequency, this crankshaft is made very large, especially the front end with flange for attaching the propeller hub. The crankshaft of the Hirth HM 60 engine (fig. 35) is a built-up shaft with all roller bearings. This design has also given good results and has behaved as well as a one-piece shaft with respect to vibrations. This design makes it possible to cut gear teeth on the pins for driving small camshafts. The Hirth design is also favorable for larger engines, when roller bearings are required and when it is not desired to caseharden the whole shaft. Figure 36 shows the design for a 12-cylinder in-line engine which is kept relatively light by making the crankshaft of like members, each consisting of a web and pin.

Figure 37 compares different crankpin types and their lubricating systems. Pressed-in caps are not generally used, since they must be removed for inspecting the shafts and then pressed in anew (1). Removable caps are now preferred in order to facilitate the removal of the dirty oil. Besides threaded caps (3), conical caps secured by bolts (2) are also used. Even the latter have not been entirely satisfactory, since the bolts when too small, are sometimes bent by the centrifugal force, so that the conical caps tend to wear and sometimes break. Good results have been obtained with elektron insets (5), which serve only for conducting the oil. The small tubes much used in motor-truck engines have also been recently introduced. A hollow space in the crankpin, however, always has the advantage that the dirt can settle and that some oil is present when the engine is started.

In the built-up crankshafts of radial engines, various types are used side by side. Cone connections are still much used in small radial engines (fig. 38,1), but are no longer used in new designs. Clamping connections are preferred in large engines on account of their simplicity (fig. 38,2). The torque is transmitted by friction due to clamping and by keyways machined in the crankpins. Of the two designs shown, with two keyways and with one keyway, respectively, the former has not proved entirely satisfactory, because, despite the most accurate fit, it was not always possible to obtain a uniform bearing over the whole circumference of the pin. On the introduction of the two keys the lower part of the pin did not bear perfectly, so that the friction joint was not perfect. On the other hand, the single-key connection, which produces a uniform bearing of the pin over its entire circumference, has given good satisfaction and is in use on many engines.

Figure 39 shows two other designs, a tapered-shaft connection (1) and a Hirth shaft with roller bearings (2). The latter design has, for the articulated connecting-rod pins and for the piston pins, needle bearings which have given very satisfactory results for oscillating bearings under heavy loads.

A broken shaft is shown in figure 40. The occurrence of the break, to which the weakness of the material doubtless contributed, was facilitated by too sharp a transition from the pin to the cheek. Notch effects should therefore be avoided by gradual cross-sectional transitions, especially on highly stressed shafts.

The Hirth type of spur gear has been introduced with good results for a power-transmitting and centering connection. It is used not only for built-up crankshafts, but also, as already mentioned, for connecting the hub to the flange of the crankshaft or driving shaft and for coupling drive gears. The spur gear centers satisfactorily, so that dowels can be dispensed with. Figure 41 shows one design of the Hirth hub with a spur gear.

5. Crankcases

Until now the best material for the crankcases of large in-line engines has been some cast aluminum alloy, while cast elektron has been found the best for small air-cooled in-line and radial engines. The designs had to be adapted, however, to the nature of elektron, and much expensive research was necessary until no further rejections had to be made. According to previous experience with smaller aircraft engines, cast elektron is just as reliable as cast aluminum, if due allowance is made for its special properties, sensitiveness to notch and edge stresses and smaller Young's modulus and if good transition shapes are used. On large engines the results obtained with elektron are not yet satisfactory, and further research will be necessary until all difficulties are overcome. It cannot yet be definitely stated as to whether it is possible to effect any considerable saving in weight by substituting elektron for aluminum in large castings, provided all rules of construction are adhered to. For small and lightly stressed castings, however, it is considered reliable and saves weight.

On large in-line engines of the current type, the crankshaft is located on the line of division between the lower and upper halves of the crankcase and is largely supported by the lower half (fig. 42). In the first example of this crankcase, which was divided through the middle of the crankshaft and provided with double walls, cracks developed in the bearing bridges while the engine was running. These cracks started from the hollow groove at the seat of the nut for the anchor bolt and ran almost perpendicularly to the direction of pressure (fig. 42,1). Similarly located cracks developed in various engine types, including 6-cylinder engines. No satisfactory explanation was possible, since even expansion measurements afforded no solution. Presumably the cracks were promoted by distortions of the rigid bearing bridges connected with the weak crankcase walls and by notch effects on the hollow groove. The new model (fig. 42,2) was improved by substituting stud bolts for through bolts, by avoiding notch effects, by strengthening the crankcase walls, and by a better adaptation of the bearing bridges for the transmission of stresses. No cracks developed after these changes.

Figure 43 shows two other designs for crankcases of in-line engines, namely, a three-piece crankcase of cast elektron and a two-piece crankcase of cast aluminum. Both designs have thus far given good results.

In new designs the lower part of the crankcase no longer serves as a support, but is simply a light pan. The upper part alone supports the crankshaft bearings and extends below the middle of the crankshaft, in order to offer sufficient resistance against distortion. Cast-on lugs for suspending the engine have been eliminated, in order to leave sufficient freedom for mounting different engines. For this purpose, special surfaces are provided on the crankcase, to which different engine bearers can be fitted.

In large radial engines the central parts of the crankcase are generally made of drop-forged duralumin or lautal and machined all over, or only on the inside. This crankcase type has given good satisfaction, but is expensive. Recently researches have been begun with a view to using pressed elektron for the central parts of the crankcase, but no definite results have yet been obtained. Figure 44 shows the crankcase of a large radial engine with forged-duralumin middle pieces and cast-elektron connecting parts.

V. AIRCRAFT DIESEL ENGINES

It must be remembered that aircraft Diesel engines have but recently appeared, so that only provisional statements can be made regarding them. The desirability of promoting the further development of Diesel engines does not need to be stressed here, since their chief advantages, small fuel consumption and reduction of fire hazards, are well known. No final decision has yet been reached as to which one of the many types of Diesel can best be adapted to aircraft. The two-stroke-cycle engine is the most promising of the Diesel types. Good results have been obtained with this engine by Junkers and others. Two-stroke Diesel engines generally work with direct injection, which has the advantage of minimum fuel consumption, but the disadvantage of very high combustion pressures. Four-stroke Diesel engines have been successfully used on aircraft not only with direct injection, but also with indirect injection (precombustion chamber, after chamber), which indeed yields lower maximum combustion pressures, but not so low fuel consumption. Opinions regarding the suitability of the different methods are still divided.

Both direct and indirect injection will probably be used concurrently and one or the other will be preferred according to the purpose served. It may be said, however, that, in addition to increasing the revolution speed with respect to weight and driving-gear stresses, the lowering of the maximum pressures is one of the most important considerations in the development of Diesel engines, and that operation methods with low maximum pressures are the most promising.

A Packard Diesel engine with direct injection was tested in the D.V.L. Maximum pressures of 80 kg/cm² (1,138 lb./sq.in.) were measured. Among the structural features of this engine, regarding the success of which in endurance tests we have no information, the elastic attachment of the counterweights to the crankshaft and the flexible coupling between the propeller hub and the crankshaft are worthy of note. Vibration measurements yielded such a low natural vibration frequency of $n_e = 3,400/\text{min.}$, that the maximum dangerous critical revolution speed of the 4.5 order is 700 r.p.m., that is, below the revolution speed necessary for flight. This favorably low vibration frequency was due not only to the flexible coupling but above all to the elastic suspension of the counterweights.

Experiments with other four-stroke Diesel engines likewise show that the vibration problem should receive special attention, since greater vibration amplitudes occur than in carburetor engines, due to the high pressures in the lower harmonics.

Figure 45 shows the fundamental design of a 12-cylinder precombustion-chamber Daimler Diesel engine of about 700 hp, which is still in course of development. The natural location of the injection pump is here between the cylinder rows. This makes it possible to reduce to a minimum the differences in the length of the fuel delivery pipes. The design of the engine is thus simplified by leaving the intercylinder spaces comparatively free.

While four-stroke aircraft Diesel engines are fundamentally similar to carburetor engines and accordingly have about the same outside dimensions, Junkers followed his own ideas in the development of the two-stroke opposed-piston engine. Since Gasterstadt published a report concerning the development of the Jumo-4 engine a few years ago, the principles of the Junkers opposed-piston type are well known. In the meantime this engine has been successfully used on aircraft and systematically developed with increase in power, reduction in weight, and improvement in its operating characteristics. In this development the following lessons were learned.

It was found expedient to fasten the cylinder liners at the bottom (fig. 46). The liner is held in the crankcase by a flange and annular nut in such a way as to leave it free to expand. This fastening is simpler and more durable than the theoretically better fastening in the middle, which permits expansion toward both ends.

The mounting of the driving gears, which connect the two crankshafts, has passed through several stages of development. As shown in figure 47,1, the driving gears were originally fastened to rotating shafts whose ball bearings were fixed in the crankcase. With this arrangement the bearings in the crankcase frequently wore unevenly. The next step was to mount the gears on stationery shafts, which were fastened in the crankcase in cylindrical bushings (fig. 47,2). Since even this arrangement did not prove entirely satisfactory, the shaft was finally secured in the crankcase by tapered bushings which remedied the difficulty. During the development it was found that the gears could be made from 10 to 20 mm (0.4 to 0.79 in.) narrower without exceeding the stress limits.

Weight was saved by improving the crankshaft mounting (fig. 48). Originally special roller races were mounted on the shaft. This indeed enabled the use of soft shafts, but their shape and weight were detrimental, because the bearings had to be covered and the intervening spaces filled out with two-part rings. The weight was reduced by mounting the rollers directly on the shaft, though this arrangement made it necessary for the shaft to be case-hardened.

There were a few other measures which helped to reduce the weight of the Jumo-4 engine. The crankcase pans, which were first made of silumin, were made of elektron with suitable changes in the design. On the other hand in low-stressed housings for scavenging blowers, it was possible to substitute elektron for silumin without changing the design.

During the development investigations, only a few examples of which have been given, the power of the engine was raised from 520 to 800 hp and the weight was reduced from 805 to 730 kg (1,775 to 1,609 lb.). The resulting weight per horsepower of only 0.9 kg/hp (1.98 lb./hp) had never before been obtained in an aircraft Diesel engine.

VI. SUMMARY

This article describes briefly some of the recent improvements in aircraft engine design and calls attention to the importance of the analysis of the knowledge gained from practical experience for the further development of aircraft engines. Since this report describes the present status of German aircraft-engine design, it also gives a brief survey of the products of German aircraft-engine works. The comparison of various designs shows how much systematic research work was done and what strenuous endeavors German manufacturers have made in order to keep pace with the development. It might be shown that we have equaled or exceeded in many fields, if not in all, the progress made in other countries, notwithstanding our shortness of funds. This is all the more important, because the funds available for engine development are much less in Germany than in other countries, where military interests are the deciding factor.

In all development work on aircraft engines, it is necessary to investigate and test systematically a series of individual designs until they meet the requirements. It is often impossible to avoid expensive duplication of the work when the same problems are being investigated in different places. The purpose of the periodical reports, which the manufacturers and the German Luft Hansa have made in a praiseworthy manner, is to avoid at least part of the duplication. It is, however, also intended to give an insight into the work of the designer, which is often too little appreciated, but which is only second in importance to the research work. Lastly, it is shown how important the cooperation of the engine mechanics is in collecting the practical flight data. It is only through the cooperation of the designer, mechanic, and investigator that the German aircraft engine can become an efficient part of the aircraft and be able to compete successfully in the markets of the world.

REMARKS ON THE LECTURES OF KURTZ AND BRUCKMANN
(reference 1)

W. Stieber.- A flexible coupling is inadequate for driving a blower when, due to the probable maximum accelerations, the forces to be transmitted become excessive, which is almost always the case. An overrunning clutch must be provided, which must be protected from overheating, in large engines, by adequate lubrication.

Cracks develop in the crankcase at the webs, because two structural elements, which serve entirely different purposes and are differently distorted, are joined together. The separation of the bearing caps from the crankcase in the inner bearings (2-6) is suggested as the best remedy.

Valve springs should not be stressed to more than 3,000 kg/cm^2 (42,671 lb./sq.in.) and should be damped to prevent fatigue failures.

In roller bearings, accurate guiding of the rollers is essential. Even normal cage clearance permits oblique adjustment and occasions overrunning. Roughness of the surfaces is an indication of overrunning. Roller bearings are very sensitive at high peripheral speeds.

Needle bearings can be used only at low peripheral speeds. The attainment of the greatest possible directional moments is essential in the oblique position of the needles. The best ratio between the needle diameter, needle length, and pin diameter has not yet been determined, but the ratio 1:6:6 is probably a good one.

The Hydrodynamic Theory of the Journal Bearing, a prospective publication by the D.V.L. as a research number, may not include all the phenomena of journal bearings. The success of Michell's axial-thrust bearing shows that its consideration is valuable. Fundamental requirements are suitable oil holes and the elimination of unsuitable grooves.

Special tests were made of white-metal journal bearings up to $p = 70$ atm. and $v = 20$ m/s (65.6 ft./sec.), and running conditions without appreciable wear were attained. The chief causes of wear are elastic distortion of the bearing parts and solid particles in the oil. One of the causes is the grinding effect of bronze crystals which increases with the revolution speed. This effect can be avoided by a special design of the bearings. Distortion can be diminished by using short bearings of larger diameter. This is confirmed by American experiments on the loading capacity of short bearings with l/d = 0.52 and 0.37 (reference 3). A well-designed journal bearing is superior to other bearings for high loading at a high peripheral velocity. Its development, which, until recently, has received but little attention, should be one of the immediate problems for research.

O. Kurtz (concluding remarks).- In Germany various engine types have had blower couplings, which have given satisfactory results and which are capable of safely absorbing the acceleration forces.

The characteristics of roller and needle bearings, especially at high revolution speeds, as described by Dr. Stieber, are well known. The development difficulties have, however, already been overcome. Satisfactory reliability of operation has been attained even with needle bearings for connecting rods under relatively high loading. In new designs there is a predominant endeavor to use journal bearings.

The separation of the crankshaft bearing cover from the lower half of the crankcase has already been largely accomplished in recent German designs.

Written communication: The Zschopauer Motorenwerke I. S. Rasmussen A.G. beg to state that Stieber's criticism of the needle bearing is contrary to their practical experience. The Zschopauer Motorenwerke use needle bearings at the maximum specific stresses and revolution speeds for the connecting-rod bearings of their two-stroke engines, both standard and racing, because they have been found to be the most reliable bearings for connecting rods.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics

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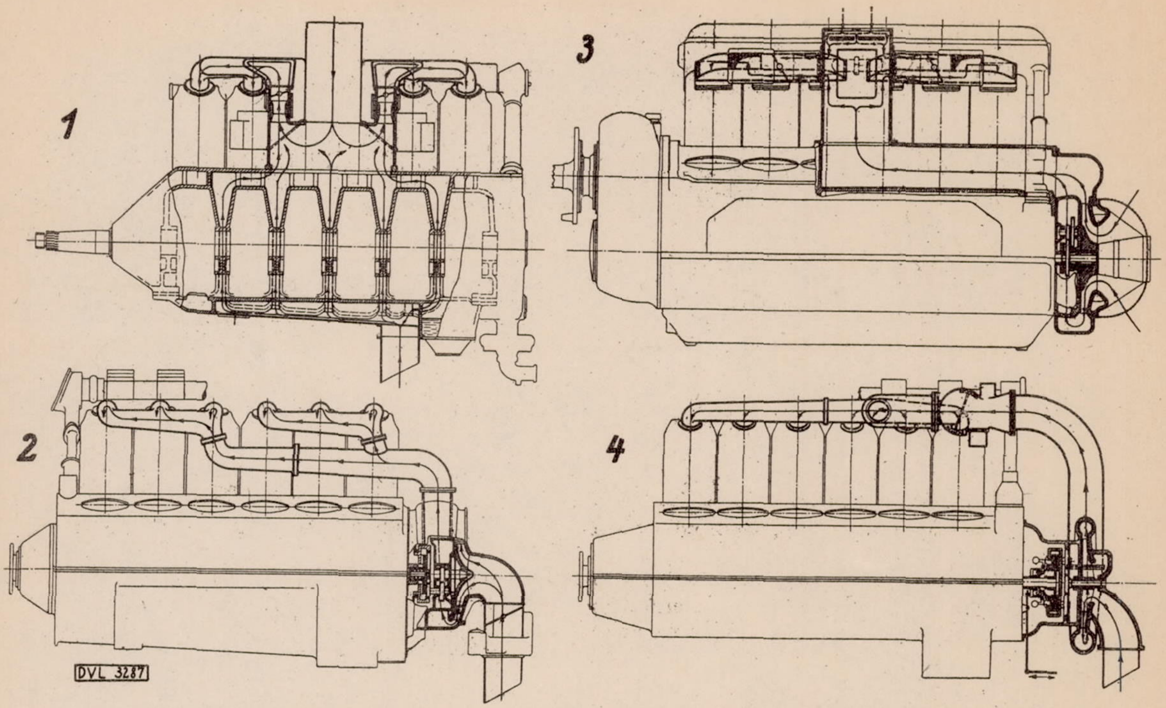


Figure 1.-Delivery of combustible mixture in water-cooled in-line engines.

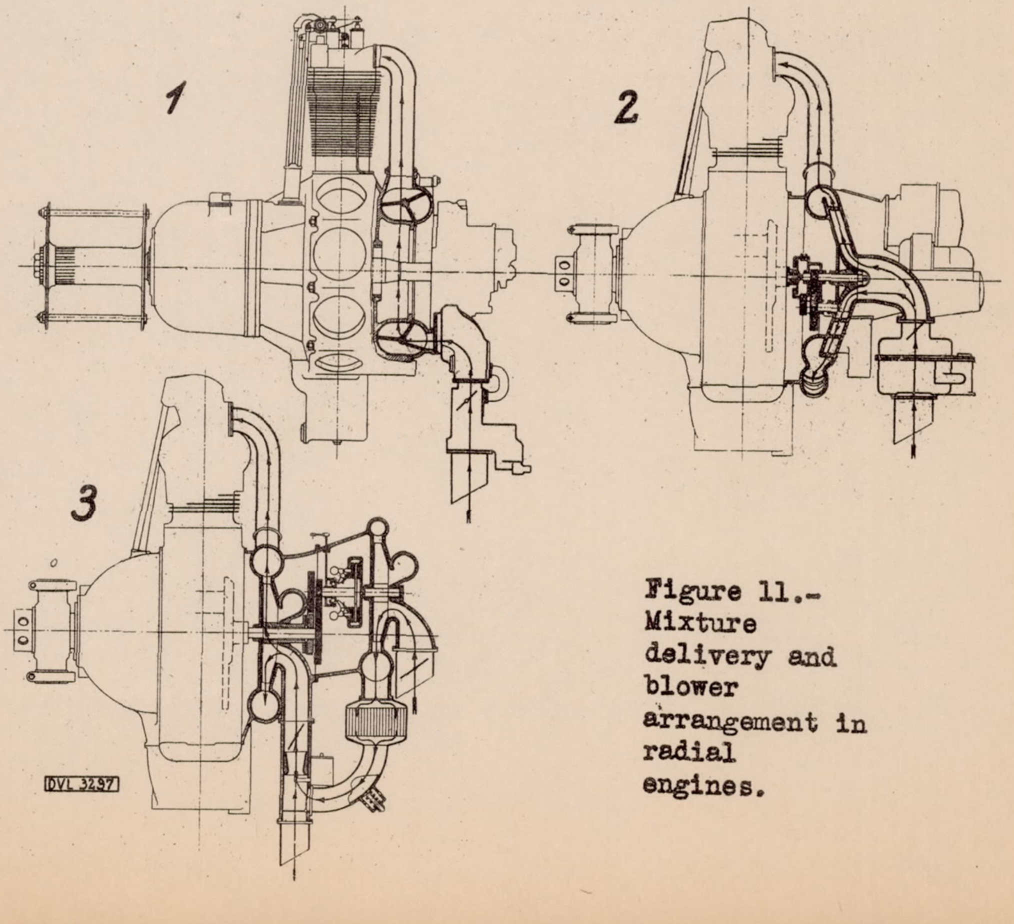


Figure 11.- Mixture delivery and blower arrangement in radial engines.

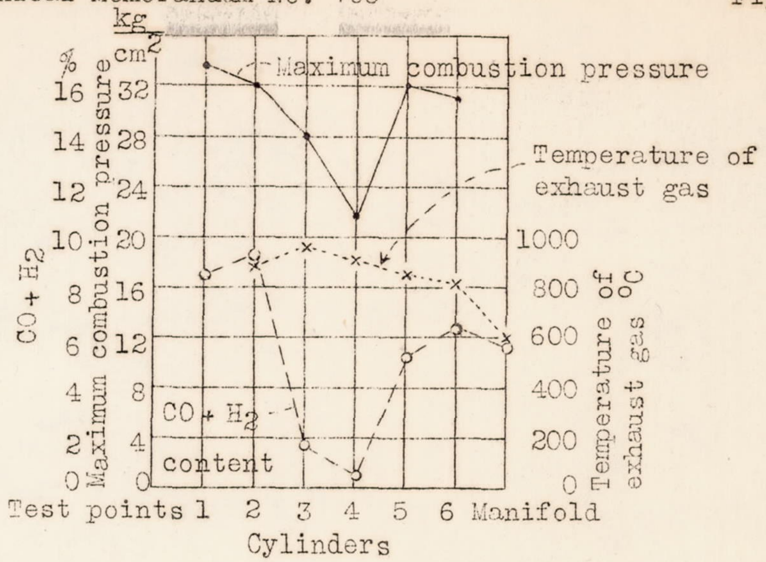


Figure 2.- Mixture distribution in a 6 cylinder in-line engine.

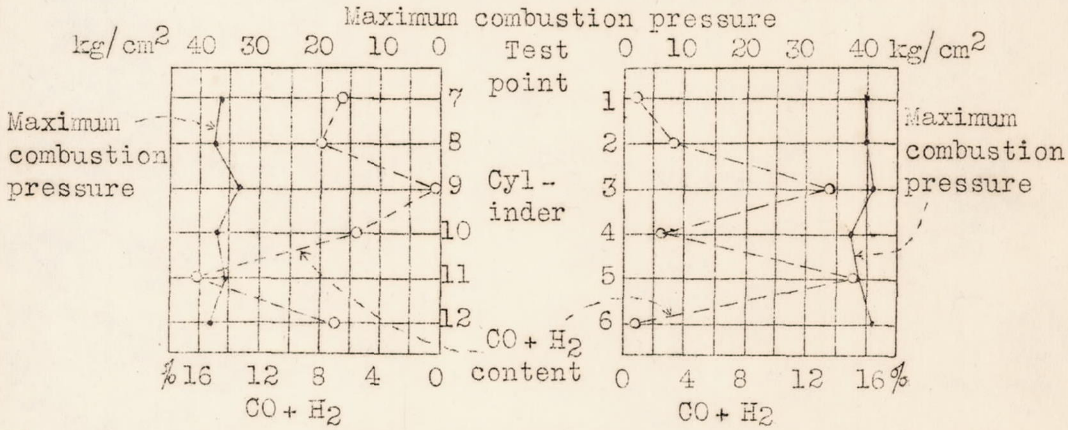


Figure 3.- Mixture distribution in a 12 cylinder in-line engine

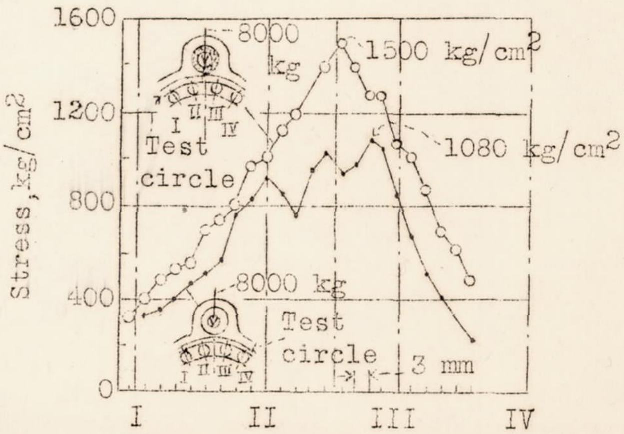


Figure 33.- Stress distribution in web connecting the master and articulated connecting-rod eyes in roller and sleeve bearings of articulated connecting-rod pin.

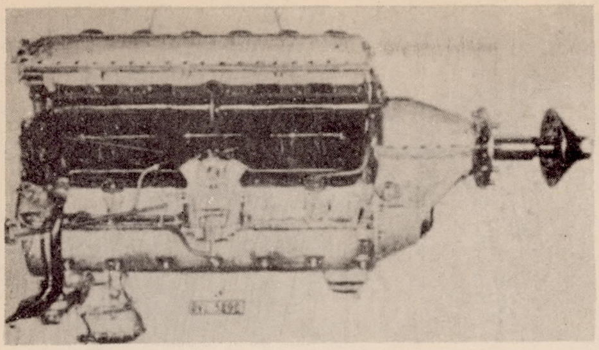


Figure 4.-Junkers L88I aircraft engine.

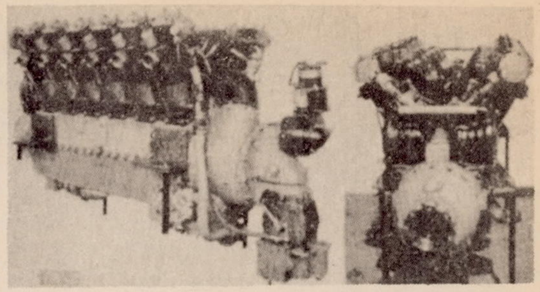


Figure 5.-B.M.W.IXaU aircraft engine.

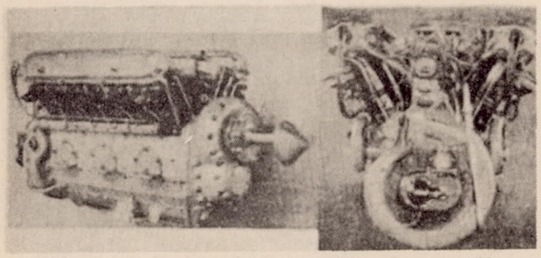


Figure 6.-Daimler F2 aircraft engine.

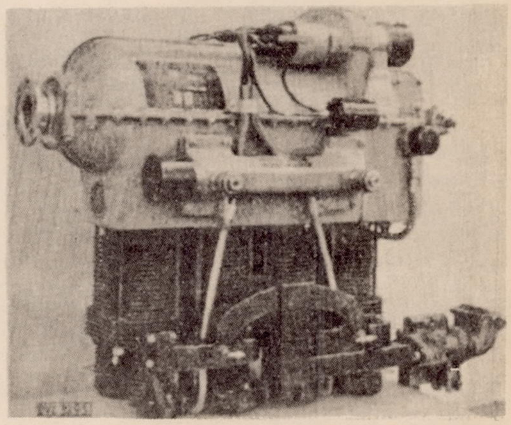


Figure 7.-Hirth H.M.60 aircraft engine.

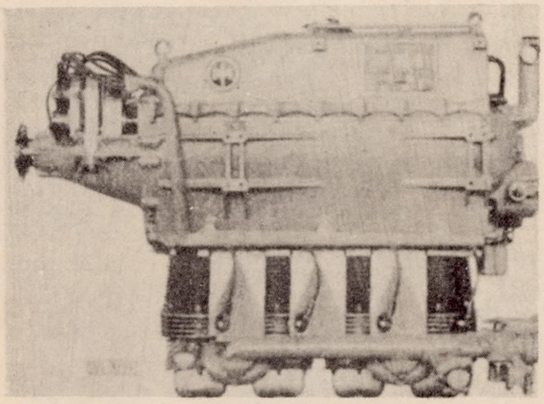


Figure 8.-Argus As8 aircraft engine.

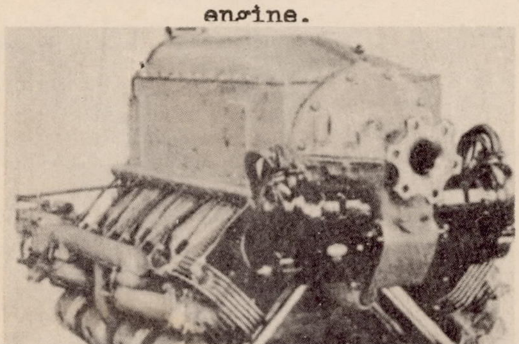


Figure 9.-Argus As10 aircraft engine.

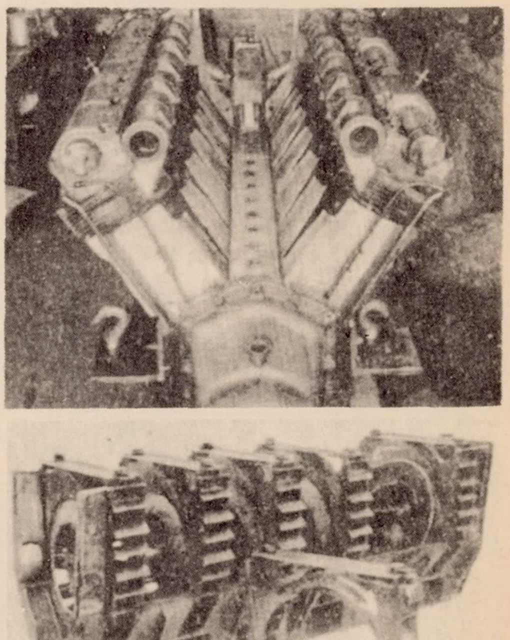


Figure 10.-Daimler F3 aircraft engine.

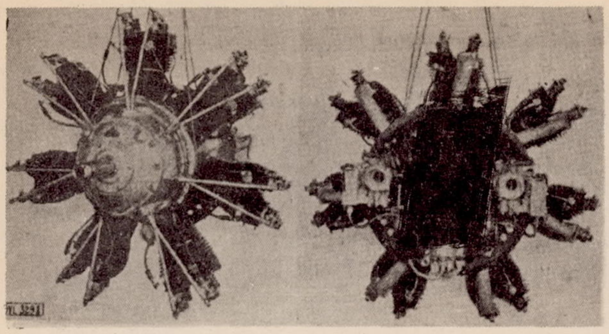


Figure 12.-Siemens S.H.14a aircraft engine.

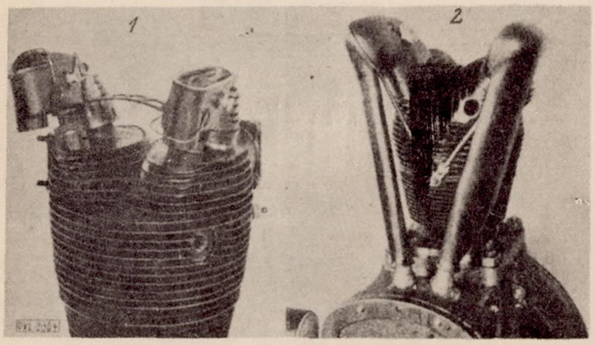


Figure 18.-Two cylinder designs of the Siemens S.H.20 aircraft engine.

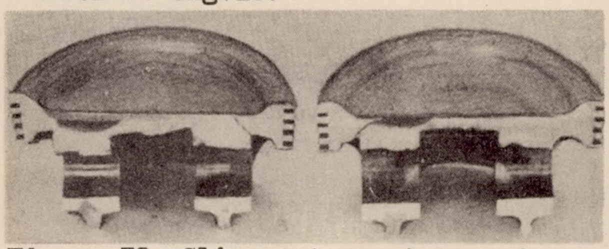


Figure 29.-Slipper-type pistons fractured through piston-pin bosses and piston head.

Figure 34.-Crankshaft of Argus As 10.

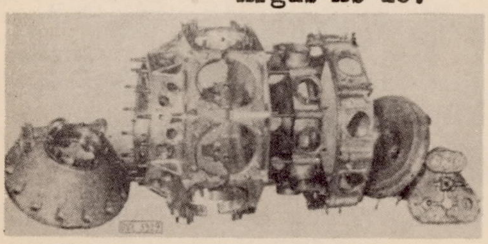


Figure 44.-Crankcase of a radial engine.

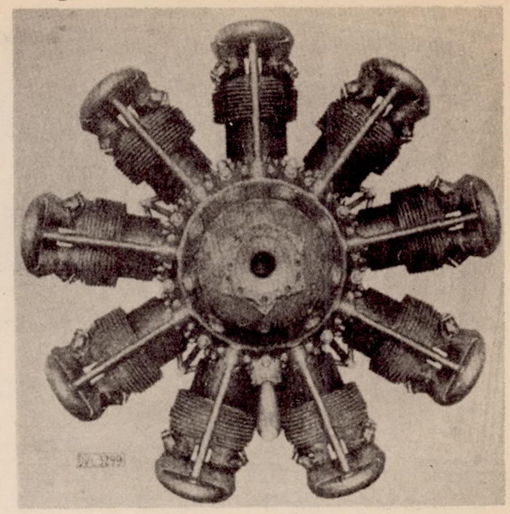


Figure 13.-Argus As7 aircraft engine.

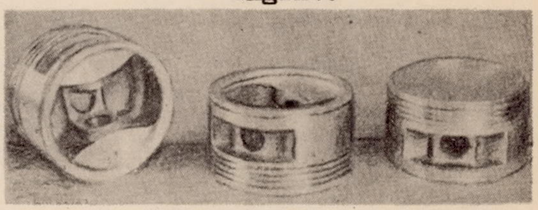


Figure 27.-Skirt-type pistons of Y metal.

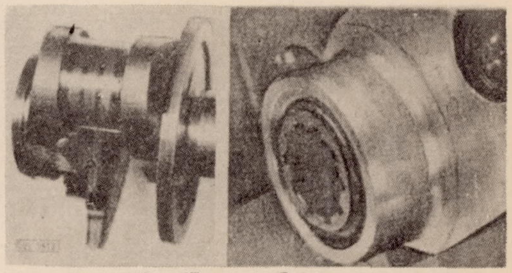


Figure 32.-Worn places on case-hardened crankshaft for roller bearing.

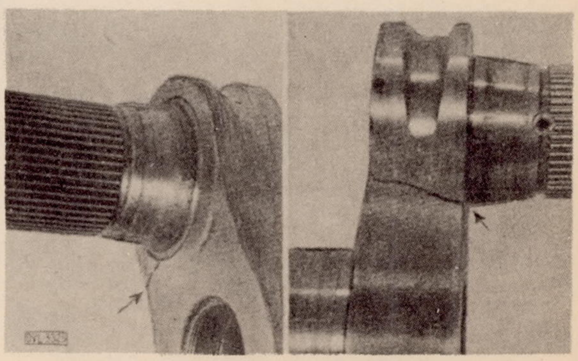


Figure 40.-Fracture of radial engine crankshaft.

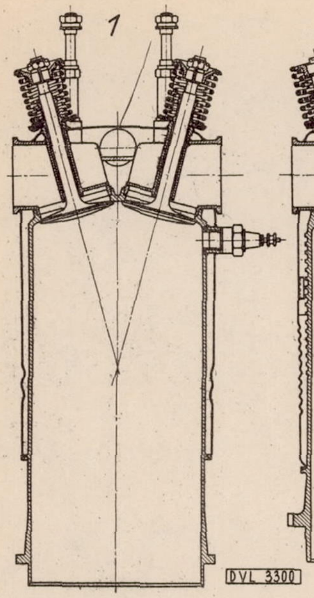


Figure 14.-Old and new cylinder types for water-cooled aircraft engines with open valve gear.

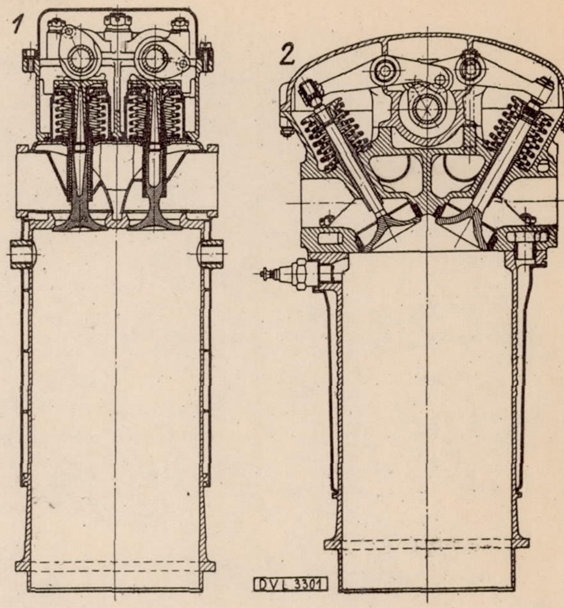


Figure 15.-Cylinder-head types of water-cooled engines.

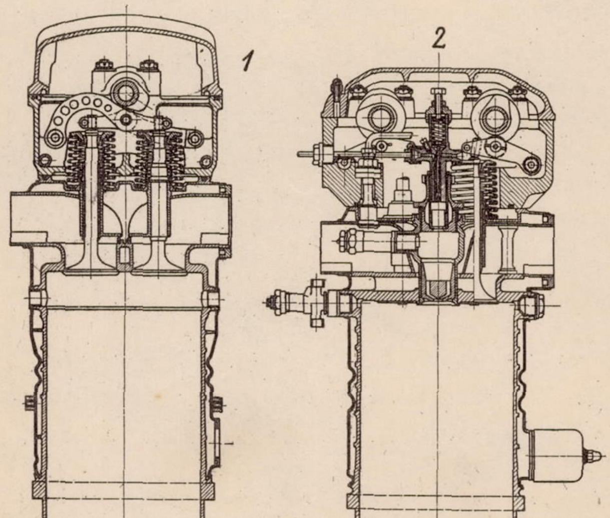


Figure 16.-Cylinder designs for carburetor engine and for Diesel aircraft engine.

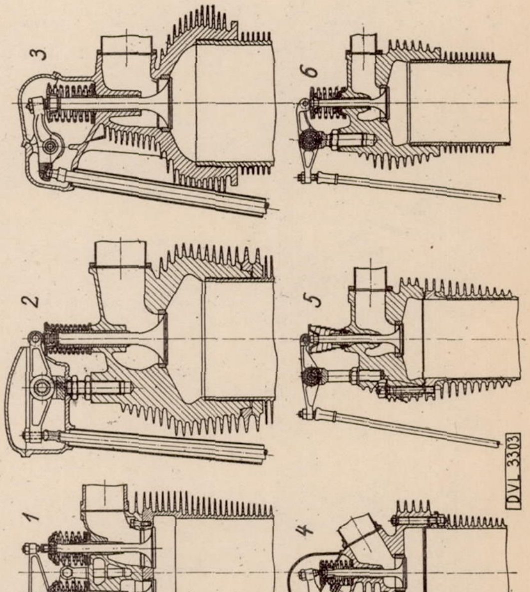


Figure 17. Cylinder-heads of air-cooled engines.

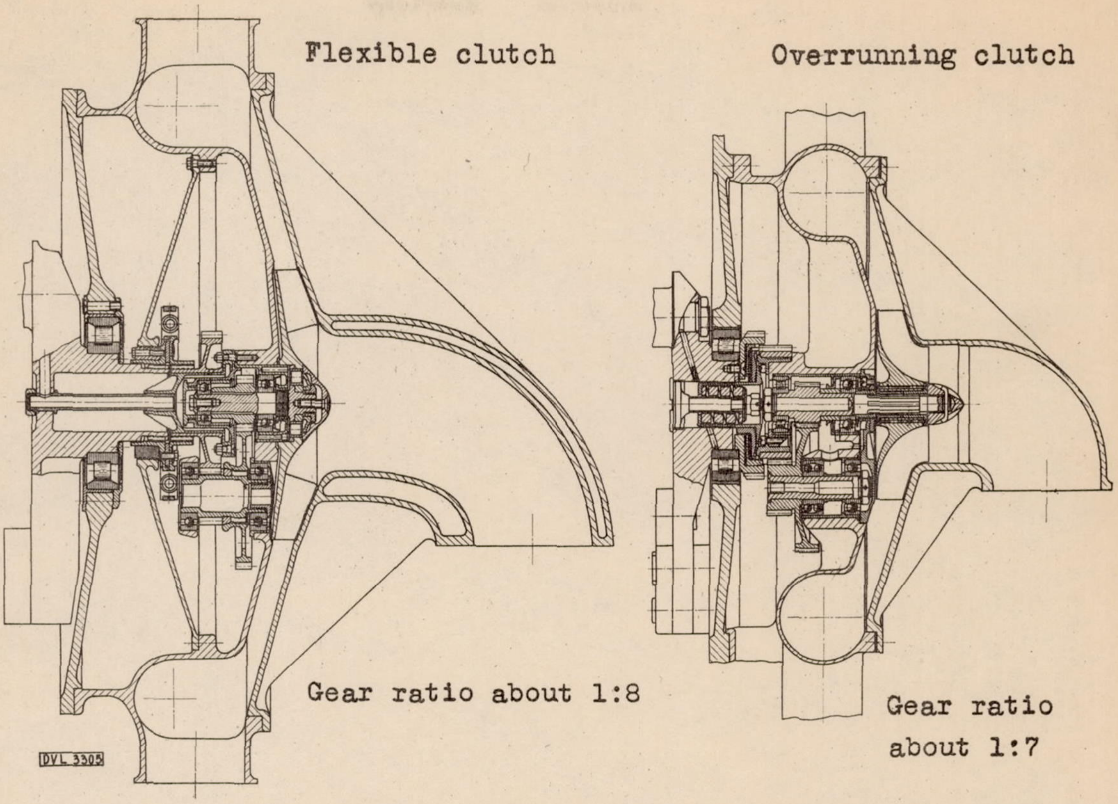
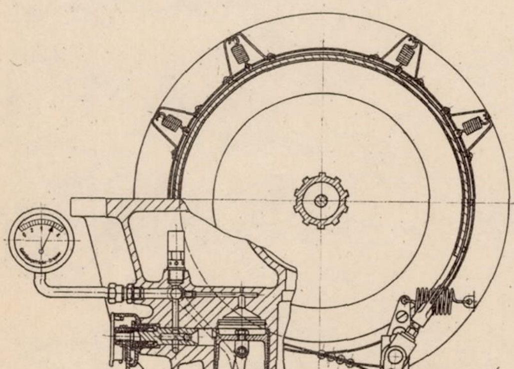
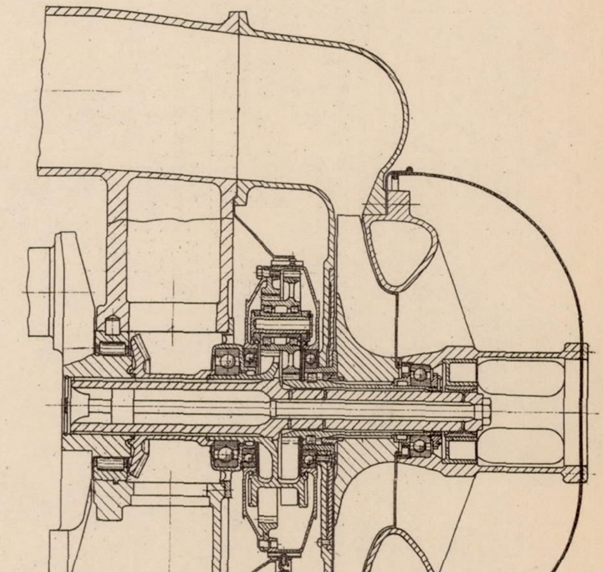


Figure 19.-Blower drives of radial engines.

Figure 20.-Disconnectible supercharger on Daimler F2 engine.





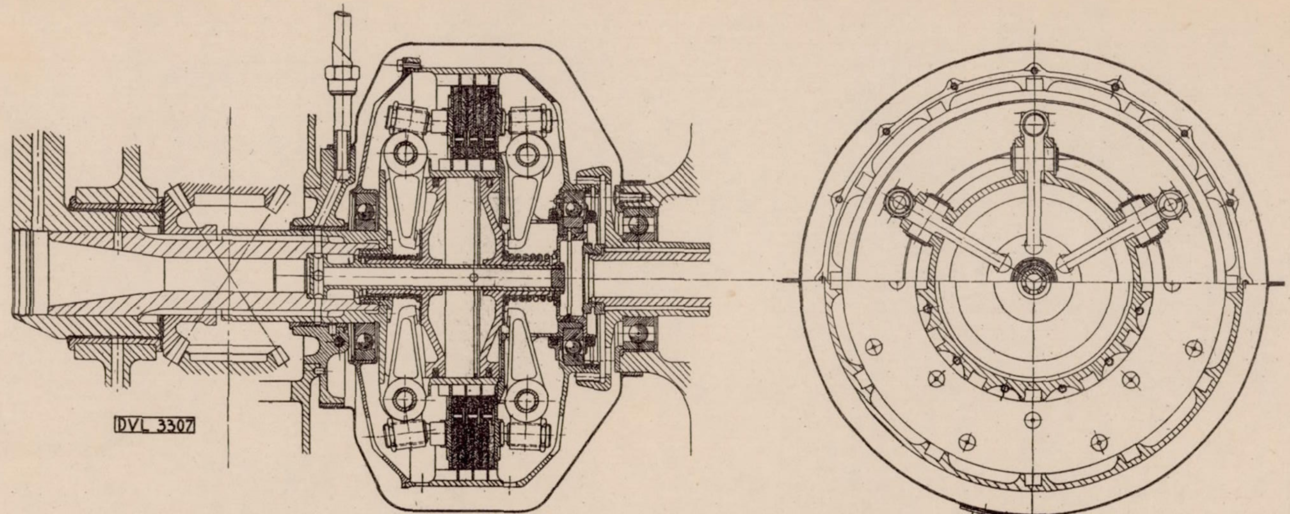


Figure 21.- Plate clutch for Roots-type blower.

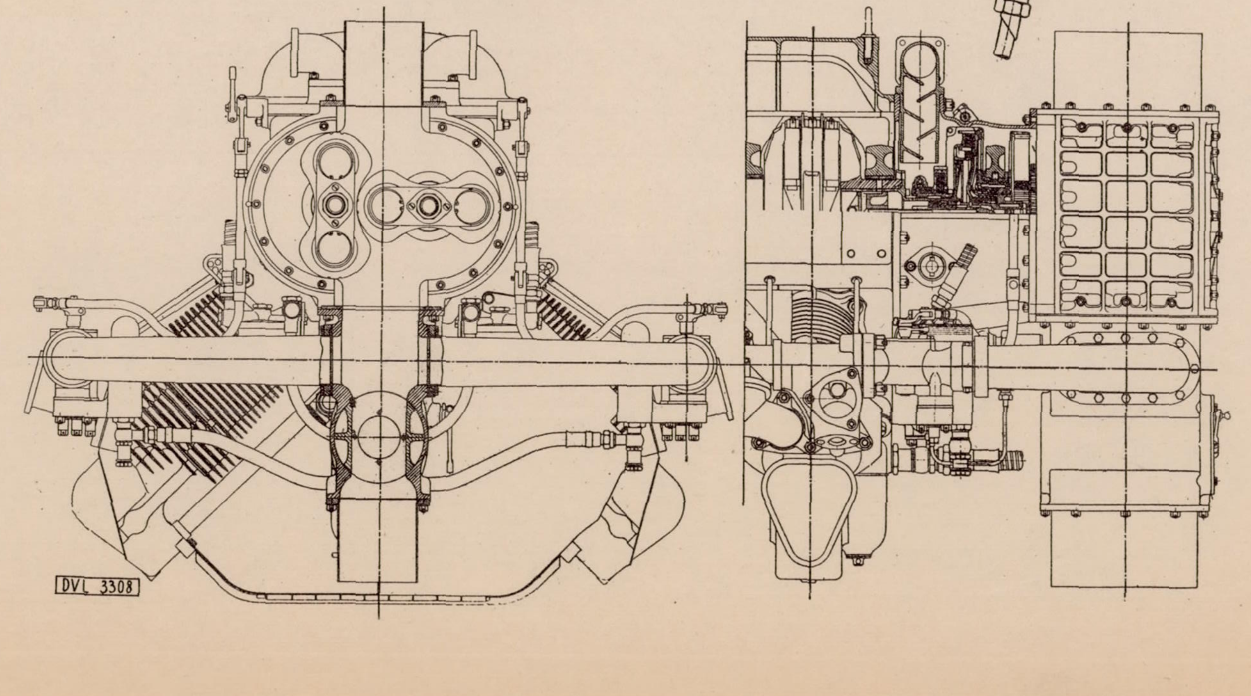
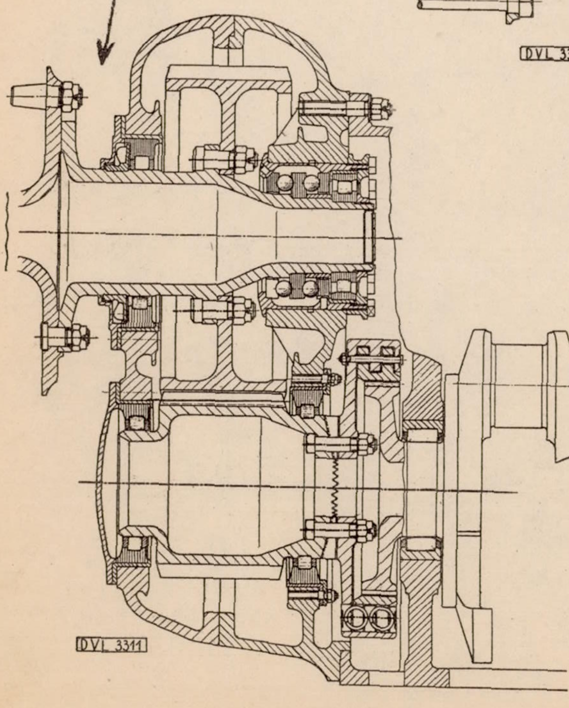


Figure 22.- Roots-type blower on Argus As 10 engine.

Figure 25.- Reduction gear of Daimler F2 engine.



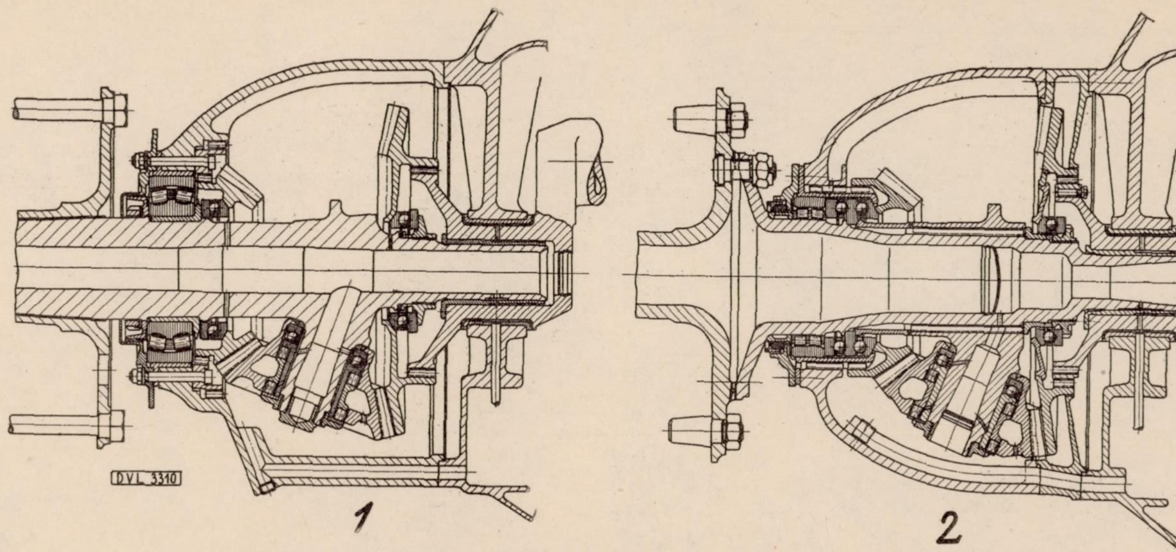


Figure 24.-Reduction gears for in-line engines.

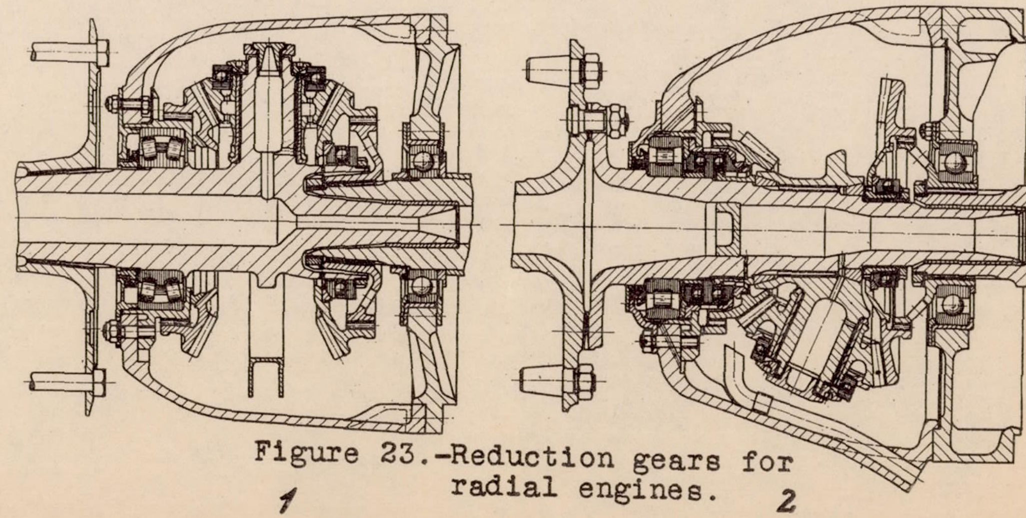


Figure 23.-Reduction gears for radial engines.

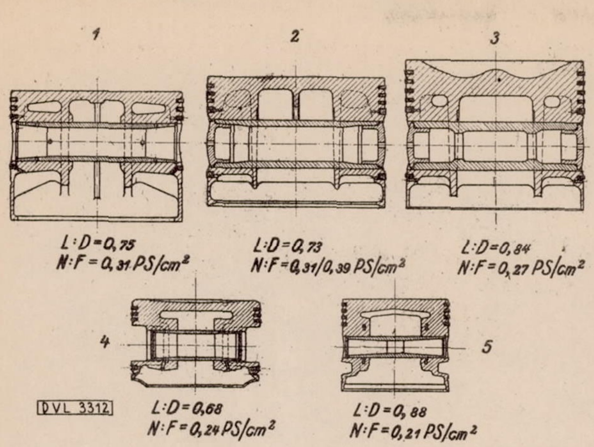


Figure 26.-Skirt-type pistons of aircraft engines.

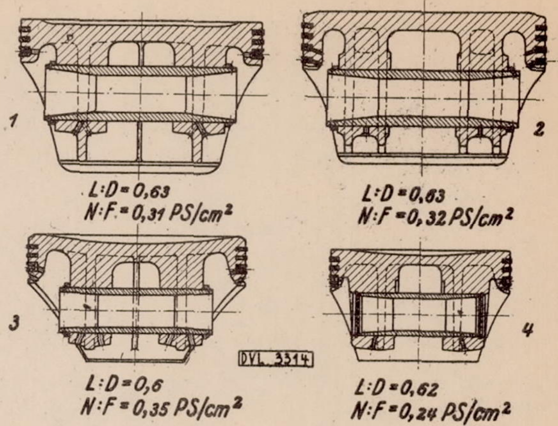


Figure 28.-Slipper-type pistons.

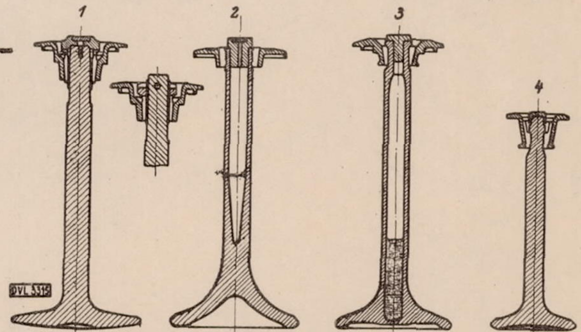


Figure 30.-Aircraft engine valves.

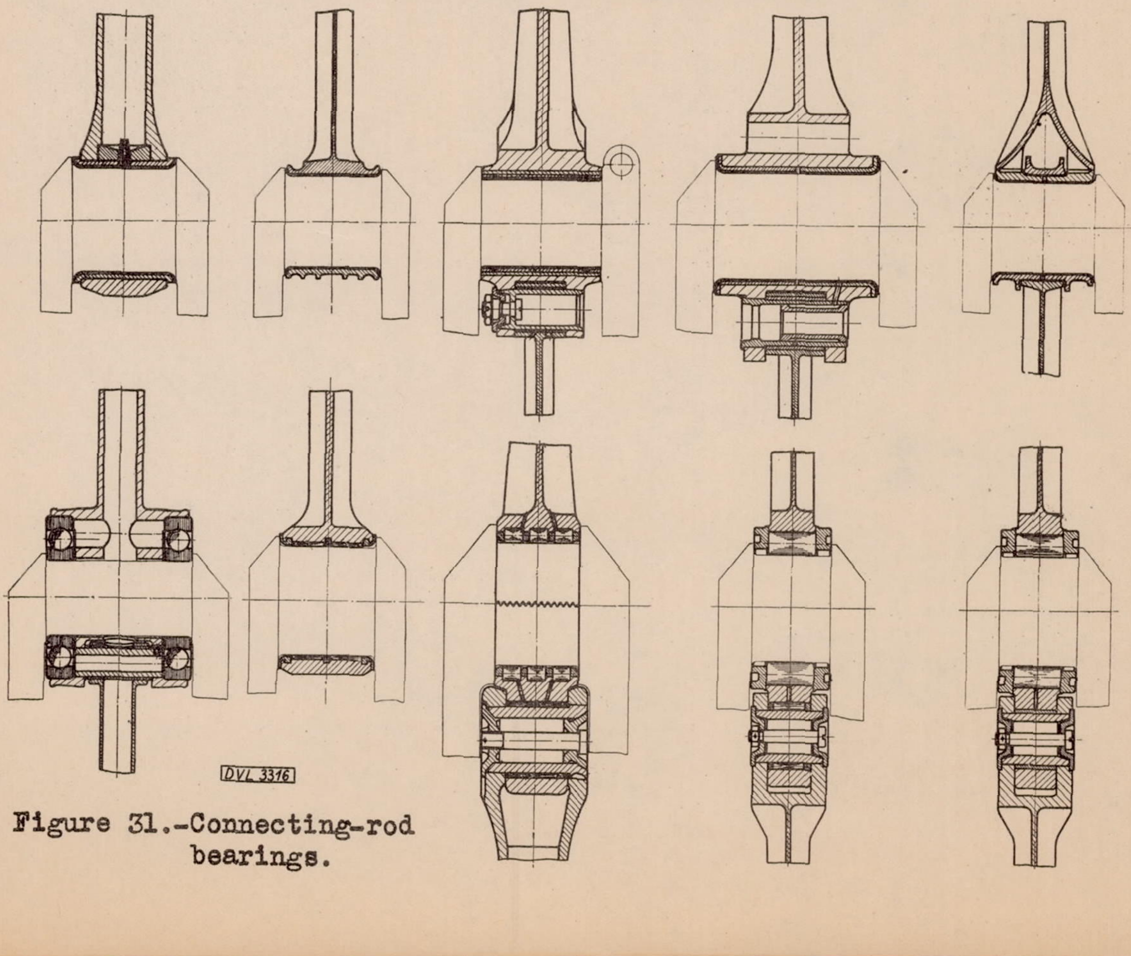


Figure 31.-Connecting-rod bearings.

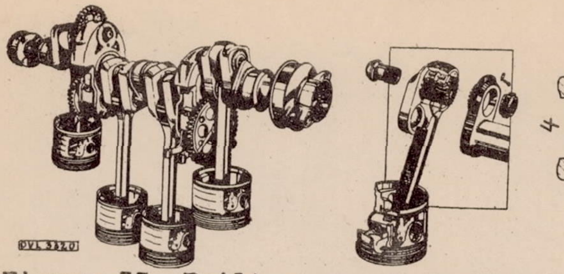


Figure 35.-Built-up crankshaft of Hirth H.M.60 engine.

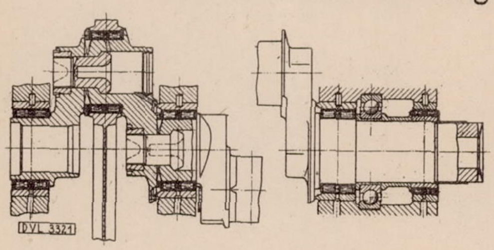


Figure 36.-Built-up Hirth-type crankshaft of a 12-cylinder aircraft engine.

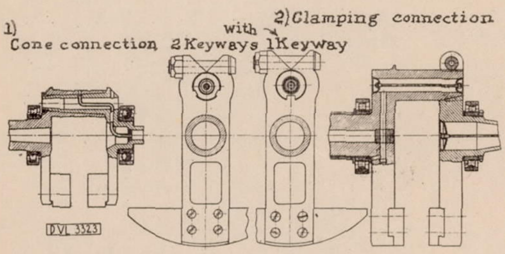


Figure 38.-Built-up crankshafts of radial enginesI.

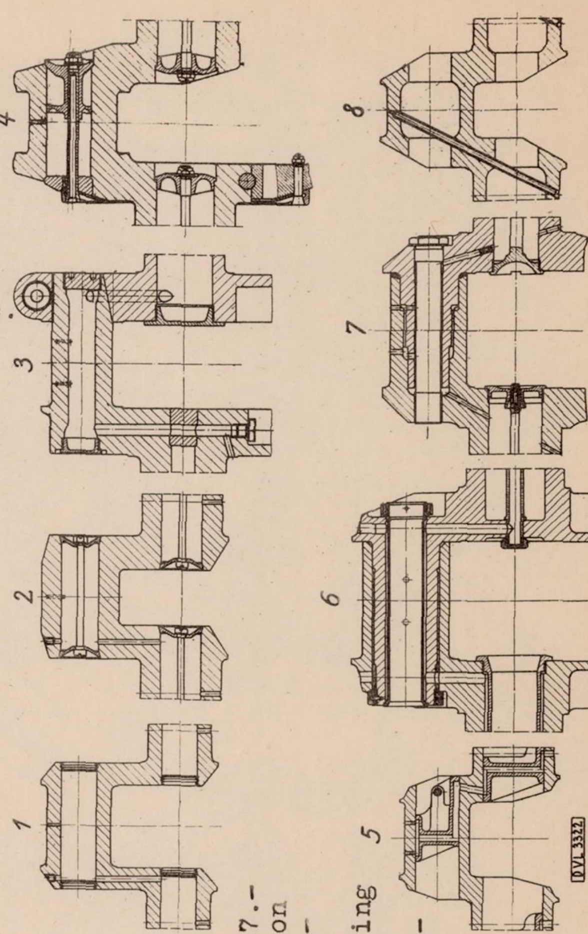
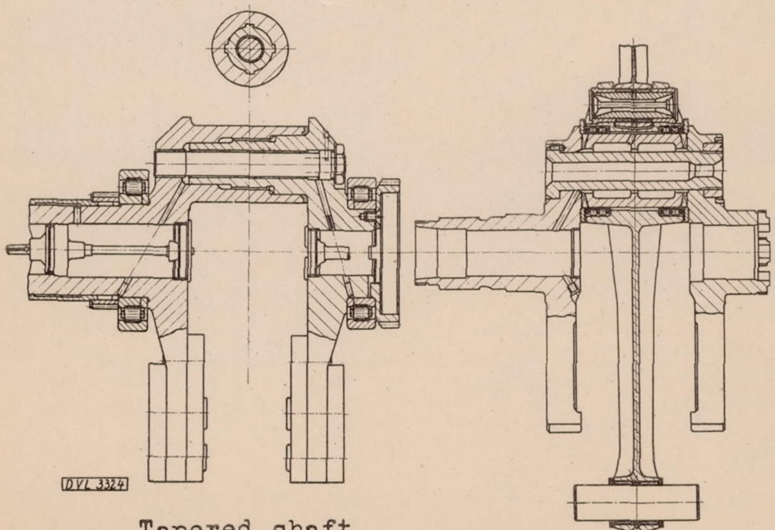


Figure 37.-Comparison of crank-pins and lubricating systems of crank-shafts.



1) Tapered shaft connection

2) Hirth gear

Figure 39.-Built-up crankshafts of radial engines II.

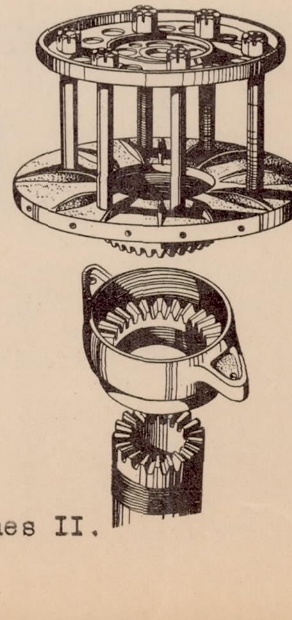


Figure 41.-Propeller hub with spur gear on Hirth H.M.60 engine.

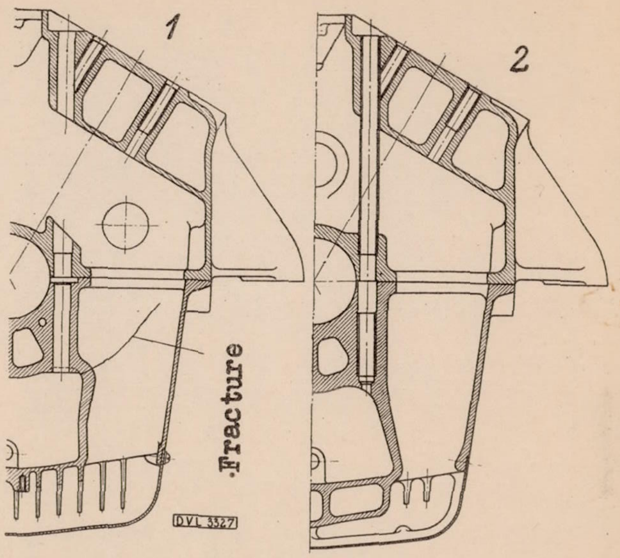


Figure 42.-Crankcase cross sections of 12-cylinder in-line engines I.

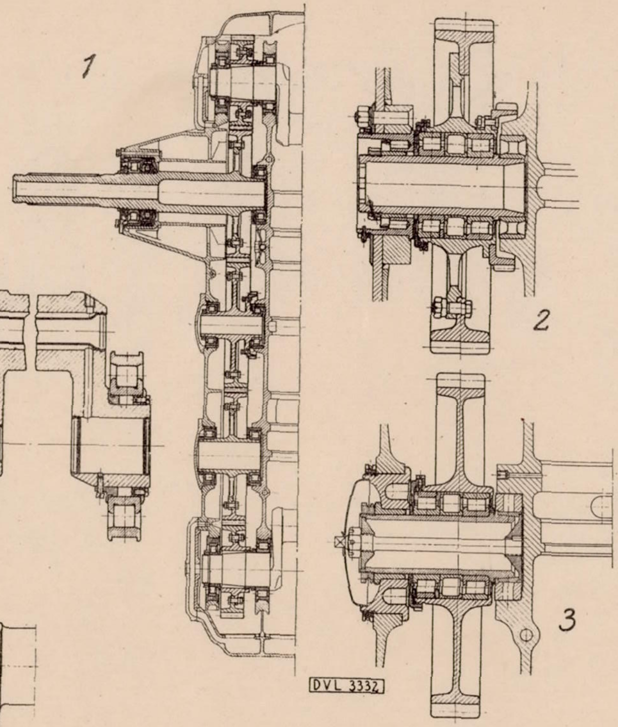


Figure 47.-Driving gears of Jumo-4 Diesel engine.

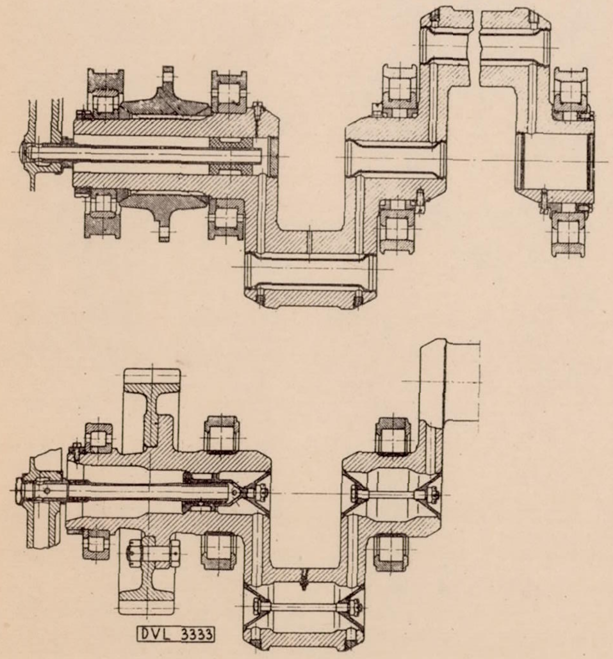


Figure 48.-Crankshaft bearings of Jumo-4 Diesel engine.

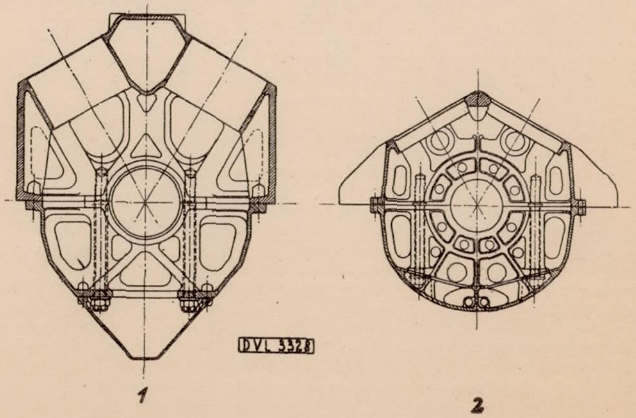


Figure 43.-Crankcase cross sections of 12-cylinder in-line engines II.

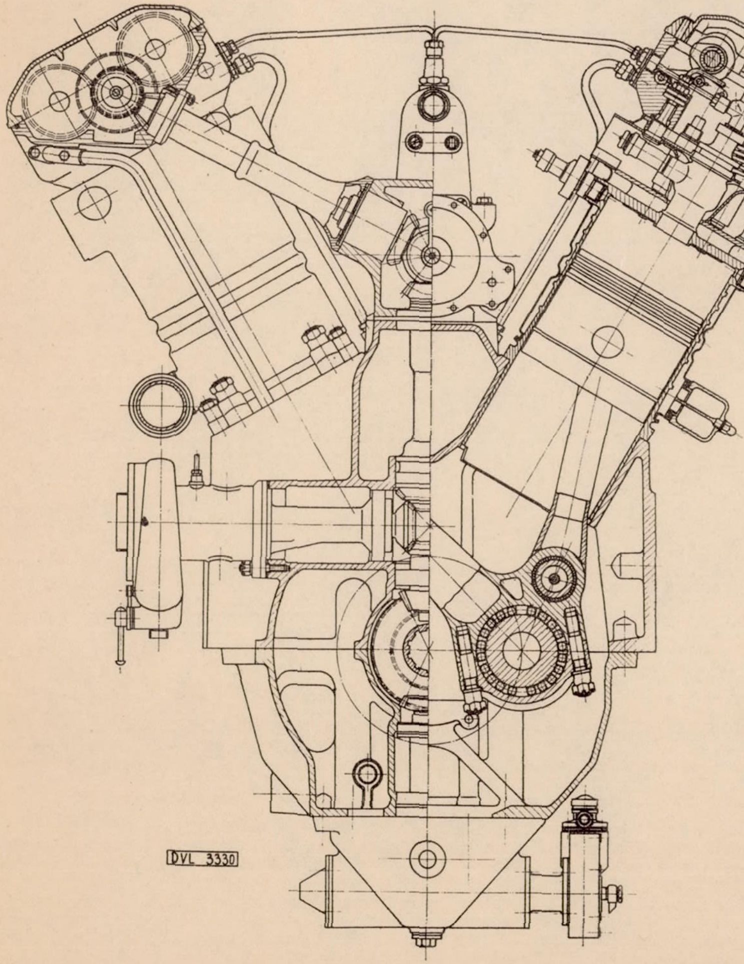


Figure 45.-Design of a 12-cylinder four-stroke Diesel engine.

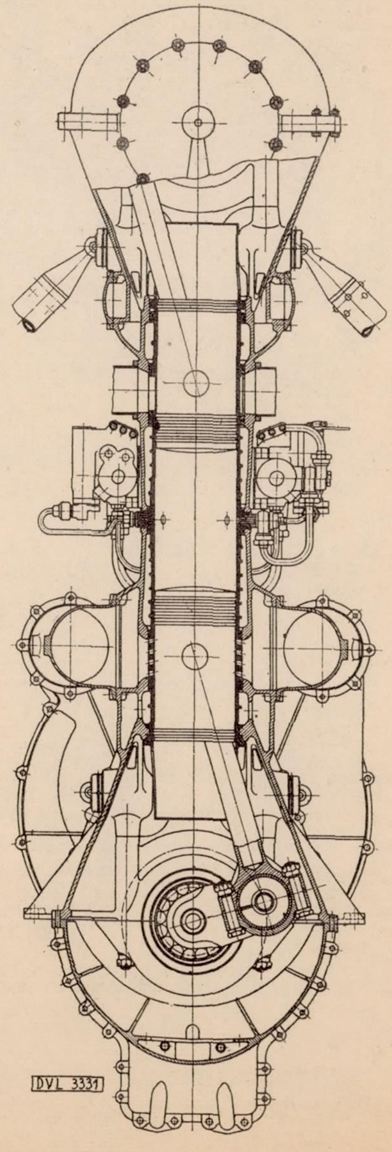


Figure 46.- Fastening of the cylinder liners on the Junkers Jumo-4 Diesel engine.