AUTOMOBILE ENGINEERING

VOLUME IV

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RIGID SIX-WHEELERS

BY

KENNETH T. ARTER, B.Sc.(Eng.) A.M.I.A.E.

LUBRICATION

BY

MAJOR A. GARRARD, W.Ex.

BEARINGS

BY

MAJOR A. GARRARD, W.Ex.

CRANKSHAFTS AND FLYWHEELS

BY

H. KERR THOMAS, M.I.Mech.E., M.I.A.E.

AUTOMOBILE ENGINEERING

A PRACTICAL AND AUTHORITATIVE WORK FOR AUTOMOBILE ENGINEERS, DESIGNERS, AND STUDENTS

EDITED BY

H. KERR THOMAS

M.I.MECH.E., M.I.A.E.

VOLUME IV

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PREFACE

Section XIV of this volume is devoted to rigid six-wheelers, a type which has somewhat unexpectedly developed for general commercial vehicle purposes from what was originally designed as a special vehicle for use over rough country. In the larger and heavier classes, both for goods and passenger haulage, the additional axle with reduced and better distributed wheel loads, is a logical development of increasing importance, and the mechanical features of the type are of great interest.

Section XV covers the subject of general lubrication, now a good deal better understood than formerly. Unquestionably the great improvement in longevity of high-speed engines is due, in very large measure, to the refinements in the general lubrication system which have been gradually developed from the hit and miss methods of the past. Much of the present-day practice is the outcome of experience with aircraft engines, the exacting requirements of which have brought about many of the features which are now in common use.

In Section XVI the general subject of bearings is dealt with, and while ball and roller bearings are the product of specialists, the plain bearings of engines embody a wide range of problems demanding the closest study if satisfactory performance is to be obtained.

The editor feels that some explanation is needed for the somewhat unconventional treatment accorded in Section XVII to the important subject of crankshafts. Mathematical methods, as can be seen from a study of Mr. Morris' comprehensive work on the subject, *The*

Strength of Shafts in Vibration (Camb. Univ. Press), become of so abstruse a nature as to be beyond the abilities of most engineers, and more empirical methods had perforce to be followed if the book was to be of general use to the student. The main principles have, however, been sufficiently explained for a work of this kind, and clearly show the disturbing factors and the general lines which must be followed to obtain good results.

The editor desires to acknowledge the permission, readily accorded to him, to reproduce Figs. 17 to 23 in this section from Vol. VIII of the *Proceedings of the Institution of Automobile Engineers*.

H. K. T.

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RIGID SIX-WHEELERS

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SECTION XIV

RIGID SIX-WHEELERS

THE "rigid" six-wheeler has developed to a large extent during the last ten years as a result of the tendency to increase the loads carried by road vehicles. The term "rigid" has been applied to distinguish it from the "articulated," or trailer, type. Usually, the front axle is of orthodox design, but the frame is supported at the rear by two pairs of wheels, arranged close together at each side, the side members of the frame being continuous from front to rear (see Fig. 20).

A more accurate description would be "three-axle vehicle," since many two-axle designs have twin wheels at the rear, and are, therefore, in effect, "six-wheelers." The latter term is, however, more usual.

Early Development. The Goodyear Tyre and Rubber Company of America was originally responsible for active development of the modern rigid six-wheeler, and actually constructed a vehicle in 1918 with the object of testing the principle. At that time great difficulty was being experienced in the evolution of pneumatic tyre equipment for heavy loads. Twin tyres were undesirable on account of their great width, and, although single tyres as large as 48 in. × 12 in. were tried they were not then satisfactory.

The only apparent solution was to replace these large tyres by two smaller ones in tandem, and a chassis was converted by the use of two worm-drive axles spaced apart by the rear springs, which were inverted from the usual position.

In the course of the next few years this form of bogie construction achieved considerable popularity, with the result that many designs were produced without sufficient attention to the problems involved. As a consequence of numerous failures the rigid six-wheeler fell into disfavour in America.

In 1923 the French Renault Company constructed a six-wheeler for use in the Sahara desert, with a useful load of 1½ tons. The drive was taken from the gear-box to an auxiliary two-speed box mounted above the foremost or leading* axle of the bogie, thence by spur reduction gearing to the rear of the bevel drive of that axle. The rearmost or trailing axle was provided with a second bevel gear, coupled to the first by a shaft supported on flexible couplings.

The performance of this vehicle under the arduous desert conditions demonstrated that the utility of the six-wheeler was not restricted to the carriage of heavy loads.

Equalization of Axle Movement. It should be noted that in all these early designs some form of truck or bogie was used; this usually resembled one of the several arrangements commonly used on railway rolling stock. The advantage of this construction will be seen from page 709, which shows the two methods diagrammatically. Clearly, when the axles are attached independently to the frame (Fig. 1A), if the wheels pass over an irregularity in the road, of height h, the frame will be raised through distance d. With the bogie construction the axles are linked together by some form of equalizing bar (Fig. 1B) attached at its centre to the frame, with the result that the vertical movement of the latter is halved.

Advantages of the Rigid Six-wheeler. The following are the principal advantages usually claimed over the four-wheeler—

* The terms "leading" and "trailing" will be used throughout this section to distinguish between the two axles.

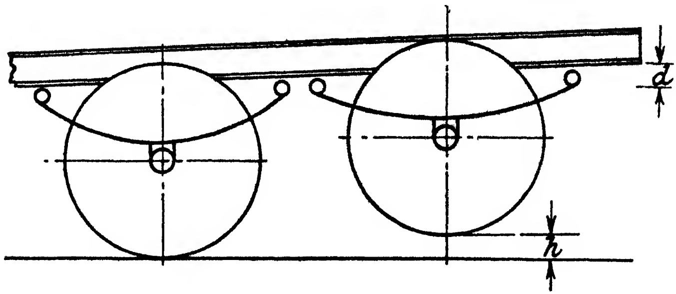


FIG. 1A

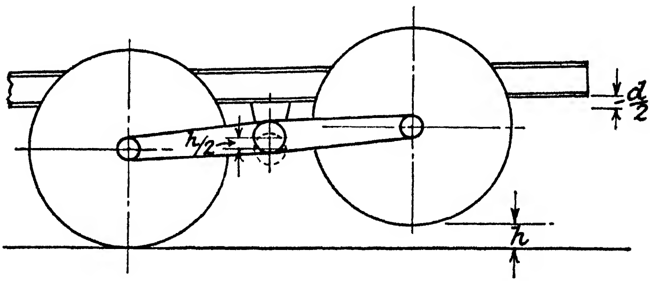


FIG. 1B

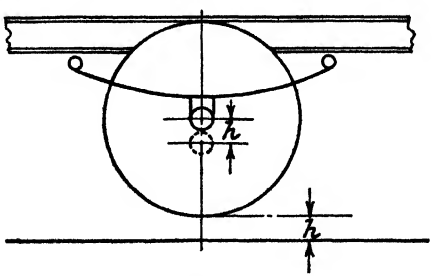


FIG. 1C

1. Increased riding comfort.
2. Reduced impact on the road.
3. Ability to carry greater loads.
4. Cheaper to operate.
5. Freedom from skidding and wheel slip.
6. Ability to cross difficult country.

1. It has been shown that the use of some form of bogie is desirable; in fact, no vehicle is built without an equalizing device between the two rear axles. Comparing Fig. 1B with 1C, which represents a single axle, it is evident that the effect of the bogie is to halve the amplitude and double the frequency of the vertical movements of the frame.

The marked difference in the riding of the four- and six-wheeler is shown in Fig. 2, which is a reproduction of records of vertical movements on two omnibuses. The instrument used in obtaining these records comprises an oil-damped pendulum attached to a stylo, beneath which a strip of paper is moved at uniform speed. The inertia of the pendulum causes the stylo to trace a curve, of which the amplitude is approximately proportional to the vertical accelerations of the instrument itself.

The tendency of the six-wheeler to convert the road humps into a high-frequency ripple is obviously a step towards the ideal suspension which would entirely eliminate movements perpendicular to the road. Further improvement could be gained by equalizing the load between three rear axles, thus reducing the vertical movements to a quarter of the amplitude prevailing on a four-wheeler. This would, however, introduce complications which hardly appear justified.

2. The effect of a wheel striking an irregularity in the road is to accelerate the axle in a vertical direction; a portion of the energy thus applied is dissipated in the

friction of the road springs, but the greater part is expended when the wheels return into contact with the road, and it is the resultant impact which tends to destroy the even surface.

The U.S.A. Bureau of Public Roads has carried out

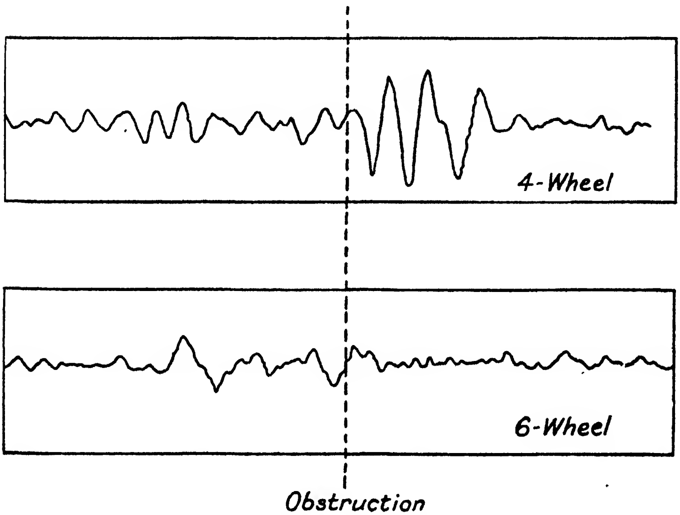


Fig. 2

an investigation into the destructive effects of various types of vehicle, and the published results show that the unsprung component of impact reaction of a six-wheeler is about one-half that of a similar four-wheeler. The measured stress in the road surface was also about one-half, provided that the bogie wheels were at least 36 in. apart. A slight reduction was effected up to a spacing of 42 in., but no advantage was gained by further separating the wheels. It was also noted that the rolling resistance was decreased as a result of the lesser road deformation.

3. The value of six-wheelers in prolonging the useful life of road surfaces has led in many countries to preferential treatment, and they are permitted by law to carry a greater gross weight. The following are the maximum weights permitted in Great Britain at the present time—

	Goods Vehicles	Omnibuses
Total weight (2 axles)	12 tons	10 tons
Total weight (3 axles)	19 tons	12 tons
Maximum weight on any one axle (2 axles)	8 tons	$6\frac{2}{3}$ tons
Maximum weight on any one axle (3 axles)	$7\frac{1}{2}$ tons	$4\frac{1}{2}$ ton

At the same time the smoother riding renders a six-wheeled lorry useful for the carriage of bulky and fragile articles, whilst the additional comfort attracts passengers in the case of an omnibus.

4. At the present time the cost of operating a commercial vehicle is influenced largely by overhead charges, i.e. wages, interest on capital cost, etc. It is usually conceded that in this respect the large six-wheeled lorry has some advantage over the greater number of smaller four-wheelers required to carry the same total load, provided, of course, that it is kept fully employed.

5. Since the bogie has the effect of reducing vertical accelerations of the axles, it follows that there will be much less tendency for them to leave the ground. The beneficial effect of this in reducing skidding and side-slip is very marked. It will be shown later that, if no differential gear is fitted between the axles, wheel spin is also minimized.

It can also be demonstrated that, in turning, the six-wheeler frequently has an additional advantage over the twin-tyre four-wheeler. In applying the principle of Ackerman steering to the former, it is usual to arrange for the projections of the axes of the front wheels to

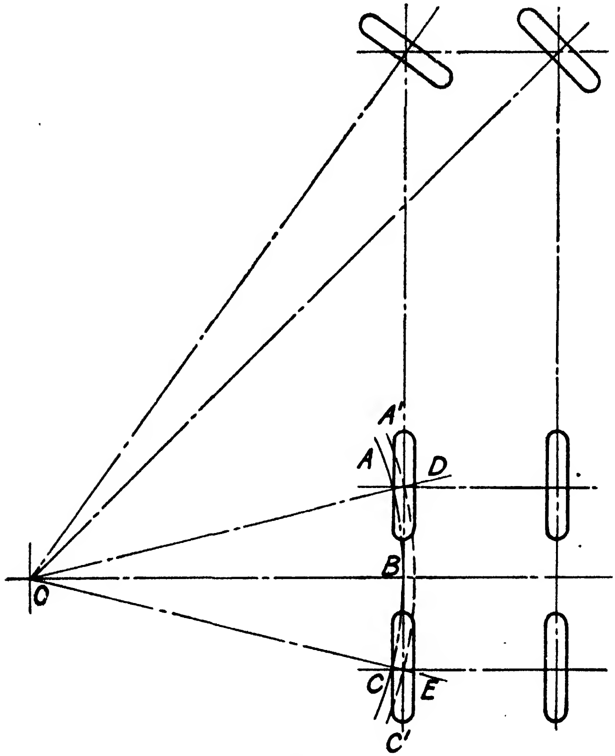


Fig. 3

intersect a line *OB* parallel to the rear axles and equidistant from them, as shown at *O* in Fig. 3. It is evident that the rear wheels are constrained to follow the circular path *A′ D E C′* whilst themselves rotating in the plane *D B E*, so that their axes if projected would not pass through O, the centre about which the vehicle is turning. The resultant slip is easily calculated.

Let $DE = 3{\cdot}5$ ft., which is about the usual figure for all except the largest six-wheelers. The minimum value of OB will be approximately 26 ft.

$$OD = \sqrt{OB^2 + BD^2} = \sqrt{26^2 + 1{\cdot}75^2}$$
$$= 26{\cdot}06 \text{ ft. approx.}$$

The slip of each wheel during a turn through 180° will therefore be—

$$\pi\ (26{\cdot}06 - 26{\cdot}0) \text{ ft., or about } 2\tfrac{1}{4}\ in.$$

If both wheels were coupled together on a single axle, their spacing apart would be (say) 10 in., supposing 8 in. pneumatic tyres to be used.

The corresponding slip must, therefore, be—

$$10\ \pi = 31{\cdot}4\ in.$$

For this reason it is preferable to avoid the use of twin tyres on six-wheelers wherever possible. They are chiefly employed on War Department vehicles, as they facilitate the attachment of non-skid chains.

6. Having considered the above advantages it is not difficult to appreciate the superiority of the six-wheeler with two driving axles in traversing difficult country. In the first place it is possible to concentrate a greater proportion of the total weight over the driving wheels, so that adhesion is improved. Provided that both wheels on one axle or two on the same side of the vehicle are capable of driving, and that no differential gear is fitted between the axles, wheel spin cannot occur. Further, the bogie construction permits greater displacement of the axles relative to the frame than can be tolerated on a four-wheeler.

About the time when the Renault machine was attracting world-wide interest in its journey across the

Sahara desert, the British War Department was searching for a lorry which would be both attractive to the private user and at the same time capable of traversing rough ground. The track vehicle was obviously unsuitable for commercial purposes, and the possibilities of the rigid six-wheeler were quickly recognized. A special design, now known as the W.D. Type, was evolved and patented, and since 1926 users have been encouraged by a subsidy to purchase six-wheel lorries of approved manufacture. In the event of war, of course, all subsidy vehicles would be commandeered by the War Office.

The W.D. Six-wheeler. The construction of the War Office design is shown in Fig. 4. It was evolved with the following objects in view*—

(*a*) That the suspension of the driving axles should be such that the exertion of tractive effort should not disturb the distribution of weight over the four driving wheels.

(*b*) That the suspension should permit of free articulation of the axles without distortion of the springs.

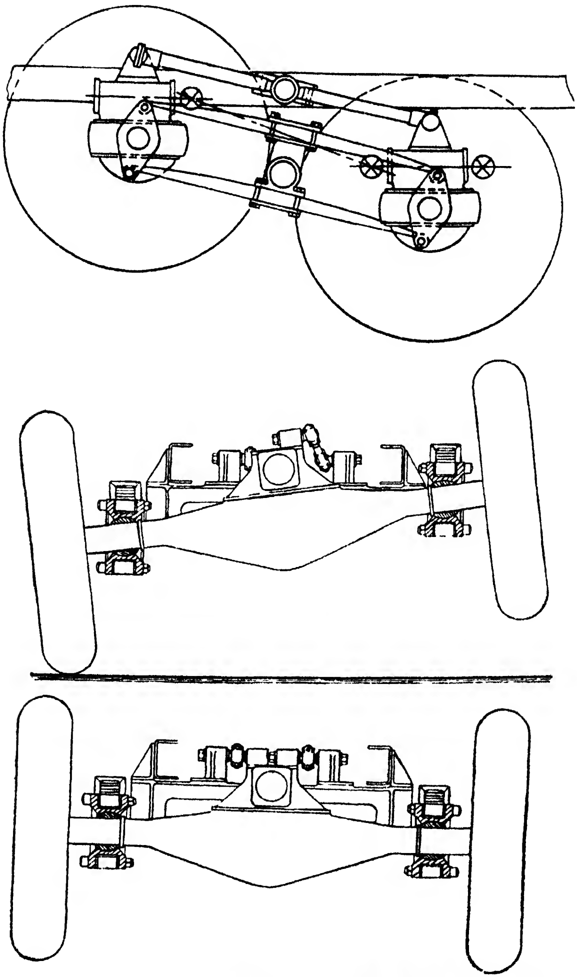
(*c*) High available tractive effort.

(*d*) Low intensity of pressure between wheels and ground.

It will be seen that the road springs, which transmit the tractive effort to the frame, are duplicated on each side. At each end they are pin-jointed to a special bearing on the axle tube, whilst they are clipped in the centre to a trunnion bearing. Torque reaction is taken to the frame by the horizontal links above the axles. It will be shown later that this arrangement ensures equal load distribution over the axles.

Provision is made for the spherical bearing to slide along the axle tube towards the adjacent wheel so that

* Captain (now Lt.-Col.) C. H. Kuhne, "Load Carrying Vehicles in the Army," *Proc. I.Mech.E.*, 1927, p. 221, and "Military Transport Vehicles," *Proc. I.A.E.*, vol. xxii, p. 140.



(*The Institution of Automobile Engineers*)

FIG. 4. DETAILS OF THE W.D. SIX-WHEELER

free articulation can take place without twisting of the springs.

Both axles are driven by worm gearing. Owing to the action of the suspension system the worm shafts remain parallel under all conditions of relative movement (Fig. 5). No differential gear is used between the axles, but as the shaft is short the angular movements are large, and a special universal joint, capable of working up to 26°, is employed.

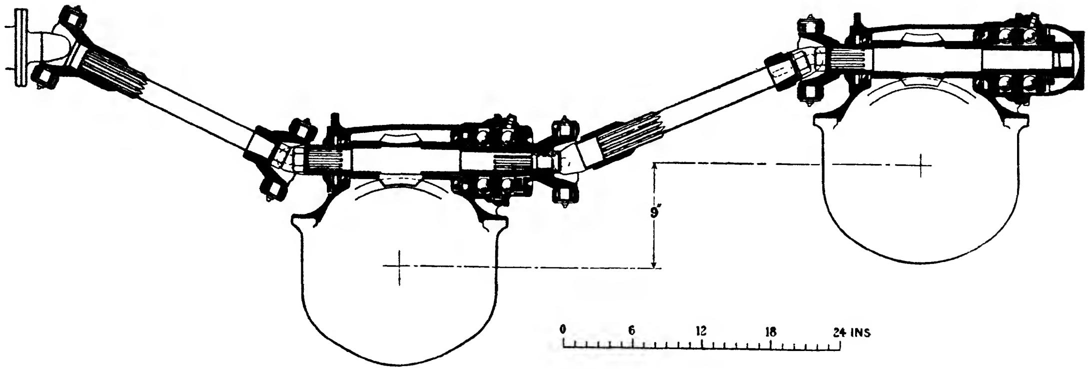
Fig. 6 shows one of these vehicles crossing a deep ditch. The limit of articulation has been passed, but the leading axle of the bogie is still driving, and the absence of spinning of the trailing wheels is clearly evident.

Limitations of the Six-wheeler. Although the rigid six-wheeler has great advantages in cross-country work and the carriage of heavy loads, the total number of these vehicles in use is small compared to the four-wheeler. It is evident that the duplication of driving axles must incur some additional dead-weight, and it is sometimes argued that this must have an adverse effect on fuel consumption. In the case of vehicles up to, say, 10 tons gross weight* operating on main roads, this objection is probably justified. An omnibus company operating four- and six-wheeled double-deckers, of approximately 9½ and 11½ tons gross weight respectively, found little difference in cost per ton-mile between the two types.

The duplication of wearing parts is also open to criticism, both from the point of view of cost and possibility of breakdown. This disadvantage also would seem to apply more particularly to the lighter types.

It may, therefore, be concluded that the bogie construction is suitable for commercial vehicles exceeding a gross weight of about 10 tons, and for any size of

* The legal limit for an omnibus with 2 axles.



(*Institution of Mechanical Engineers*)

FIG. 5. SHOWING PARALLELISM OF WORM SHAFTS

vehicle which is required to traverse rough country. Naturally, the case is somewhat different when the owner is in receipt of the War Office subsidy.

The Bogie. The arrangement of the bogie and its means of attachment to the main frame or structure are of great importance. Their functions are—

(Institution of Automobile Engineers)

FIG. 6. A VEHICLE CROSSING A DEEP DITCH

1. To equalize the distribution of load between the axles.
2. To accommodate displacements of the axles relative to the frame and one another.
3. To resist side thrust on the axles; and
4. In some cases to transmit reactions set up by the driving torque.

As the wheels ride over the varying surface of the road, they must be capable of relative movements, as shown in Fig. 7. At the same time the wheels on each side must be inter-connected in some way to equalize the loads. There are many ways of accomplishing this, and Fig. 8 shows some typical arrangements diagrammatically.

In Type 1 the wheels are attached to the extremities of a rocking beam, pivoted at its centre to a single leaf

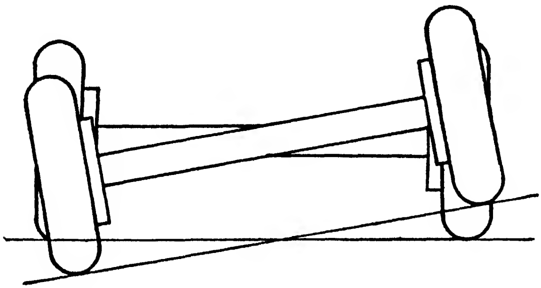


FIG. 7

spring. This is employed in the T.T.A. equipment for the conversion of four-wheelers.

In Type 2, used on the Büssing, the spring itself becomes the rocking beam.

The W.D. design, with two parallel springs mounted on a trunnion, has already been described (Fig. 4). Alternatively, each spring may be mounted on a separate trunnion (Fig. 8, No. 3).

In a number of designs, chiefly conversion outfits, the wheels are provided with separate springs, and the equalizing linkage is attached to the frame. Type 4, employed by Berliet, is one arrangement. The rocking bar has to be amply proportioned, and this may have an adverse influence on the length of the springs, in which case a long beam (Type 5) might be employed.

Type 6 has been used by Tilling Stevens. It has the advantage that the long member is only in tension, and is not subjected to bending stresses. It is obviously

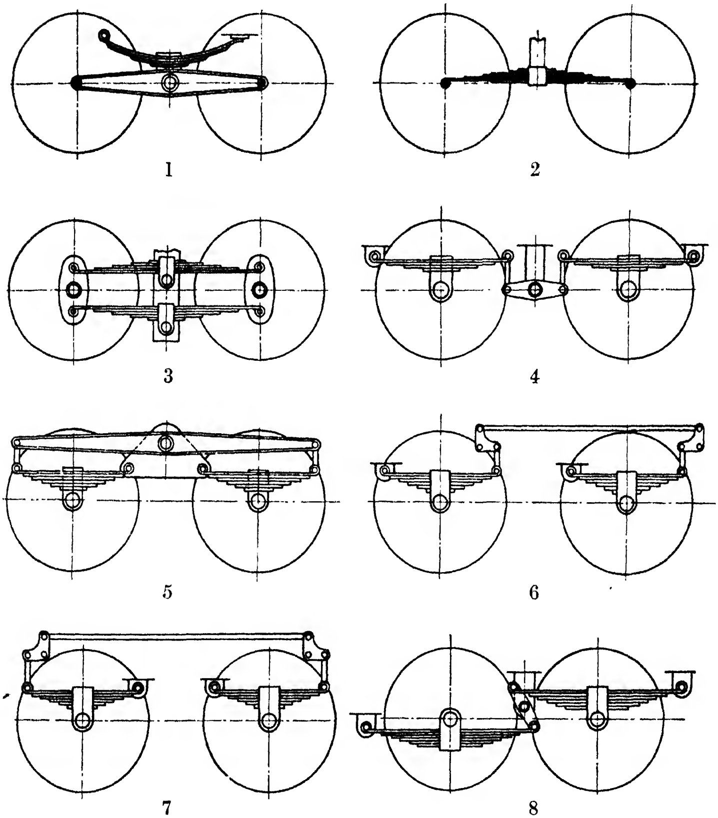


Fig. 8

preferable to No. 7, which involves a long compression member.

Type 8, the Baico conversion, permits a slightly longer spring than the two foregoing designs, although it is not always convenient for the fixed ends of the springs to be at different levels.

Several of these bogie arrangements are common in railway practice, notably Nos. 2, 4, and 7. In every case when the vehicle is standing on a level road the dead weight must be equally distributed between the axles. It is desirable for this equality to be preserved under running conditions as well.

Wheel Centres. The minimum centre distance of the bogie axles is obviously limited by the diameter of the wheels, which is, in turn, usually determined by the size of tyre required to support the load. Considerations of wheel slip and tyre scrub call for a close spacing of the wheels. On the other hand, this is not always mechanically convenient, as it restricts the size of the road springs, and may cause excessive angularity in the cardan shaft between the axles.

It has been shown that when the vehicle is turning, the imperfection in tracking of the bogie wheels causes slip in the plane of rotation (page 713), but this is comparatively slight. There is also a tendency for them to slide in a direction at right angles to that of the slip. This is usually termed "tyre scrub."

In Fig. 9 the bogie is turning about centre O. It will usually be found that the true turning radius is OB_1, which bisects at right angles a line joining the wheel centres.

Suppose the vehicle to turn through angle B_1OB_2. It will be sufficient to consider the wheels on one side, those in the vertical plane through XB_1. Clearly, as the bogie travels from B_1 to B_2, they will turn about centre B into the plane through AXB_2, as indicated by the arrows. Thus, each wheel is subjected to a sliding action at right angles to the plane in which it rotates.

If d is the centre distance of the axles, the amount of scrub experienced by each wheel will be $\frac{d}{2} \times \angle B_1OB_2$.

Taking d equal to 3·5 ft. as before, the theoretical scrub caused by turning through 180° will be

$$1{\cdot}75 \times \pi = 5\tfrac{1}{2} \text{ ft.}$$

Although for this reason it is desirable to arrange

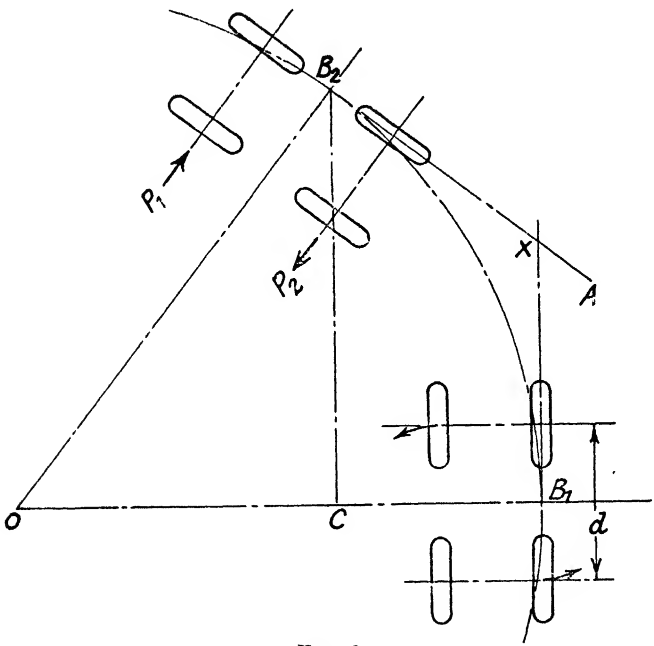


FIG. 9

the bogie wheels reasonably close together, wheel scrub is not so serious as would at first appear. This is by reason of the great lateral flexibility of pneumatic tyres. When a six-wheeler is driven slowly over a hard level surface with the steering locked over, it can be clearly seen that the tyres flex sideways as they come into contact with the ground, so that the scrubbing action is usually very slight.

In practice the imperfection in tracking appears to be immaterial as long as the ratio

$$\frac{\text{distance between centres of driving axles}}{\text{distance, front axle to point midway between driving axles}}$$

does not exceed 0·32, and the minimum turning radius is not unduly small relatively to the effective wheelbase.

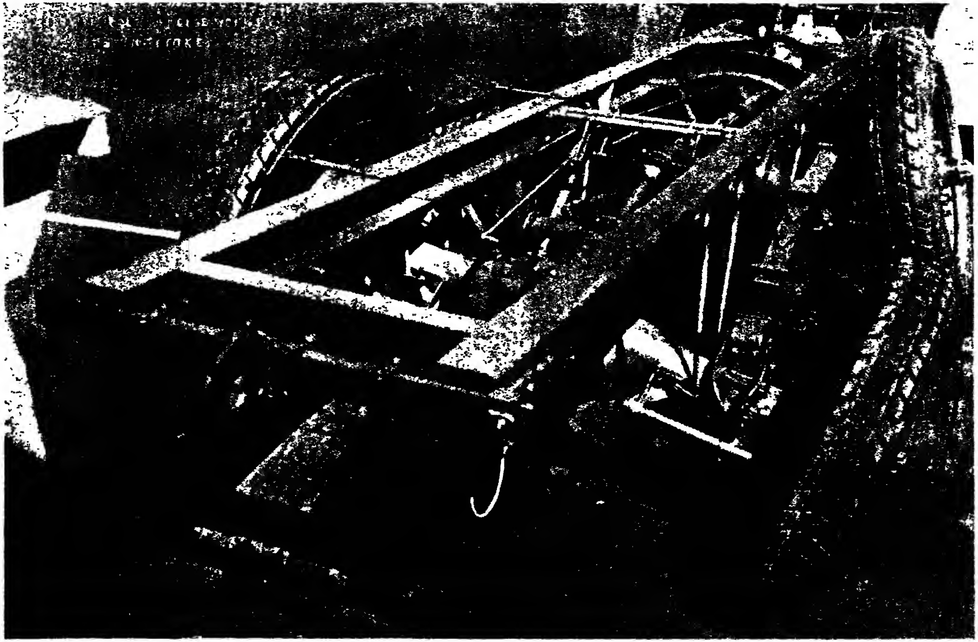
Articulation of Axles. Except on the smoothest roads, the vertical deflections of the bogie wheels will seldom coincide, with the result illustrated in Fig. 7. From Fig. 8 it will be apparent that two methods of attaching the springs to the frame and axles are usually employed:

(*a*) the spring may be clipped at its centre to the trunnion, in which case the extremities are pinned or shackled to the axles, or

(*b*) the axle or equalizing beam may be clipped to the centre of the spring, and the extremities of the latter shackled to a frame member.

In both cases tilting of the axles must cause twisting of the road springs unless special provision is made, as in the W.D. design with the sliding ball arrangement. The springs of four-wheeled vehicles are almost invariably subjected to this, but it is more important when a bogie is used for two reasons. The length of spring resisting torsion may be only about half that available on the four-wheeler, as, for example, in Types 2, 3, and 4 (Fig. 8); also on cross-country machines the deflections may be much greater.

Instead of a sliding ball the Thornycroft employs a "gimbals" arrangement (Fig. 10). In the case of omnibuses and lorries intended for service on properly surfaced roads the only provision is usually to allow for a limited "end float" of the spring eyes on the shackles,



J. I. Thornycroft and Co., Ltd.

Fig. 10. Thornycroft Six-Wheeler, showing "Gimbals" Attachment of Springs

so that the springs are relieved of a great deal of side thrust, and are subjected only to torsion.

Side thrust on the springs will be found to occur on the majority of designs when the vehicle is turning. Reverting to Fig. 9, it will be apparent that the friction between the tyres and road will tend to oppose the rotation of the bogie about B_2. Consequently the axles must exert a thrust in the direction indicated by P_1 and P_2. The magnitude of these forces is indeterminate on account of the lateral flexibility of the tyres. They can be equalized by fitting radius rods between the axles, or a torque member joining them can have the same effect. In practice, however, special provision is now rarely made, so that the thrust has to be transmitted through the springs to the frame.

Transmission and Final Drive. When both axles of the bogie are used for driving, worm gears are most frequently employed. It is then a simple matter to couple the two wormshafts together by a short shaft. An open shaft with some form of universal joint coupling at each end, as in Fig. 5, is commonly employed. Alternatively, where a telescopic torque tube is fitted between the axles, this may be arranged to enclose the shaft. Bevels are inconvenient unless auxiliary gearing can be incorporated at some point in the transmission, owing to the obvious difficulty of providing a "straight through" drive on the leading axle. Chain drive, where it is permissible, occasions no exceptional complication. The design of the live axles and final drive gearing seldom differs from conventional four-wheel practice, with the exception of a few special types of bogie.

In the case of low-loading double-deck omnibus chassis the practice of locating an underslung worm gear towards one side of the axle has been adopted by several manufacturers (see Fig. 20). This is likely to result in excessive angularity between the wormshaft of the leading axle and the cardan shaft coupling it to the gear-box, since the distance between the two is usually less than where a single rear axle is used. However, it is not essential that the two wormshafts should be parallel to one another. A better disposition is shown in Fig. 11, where they are inclined at equal angles to the inter-axle cardan shaft. If the ends of the latter carry some form of Hooke's joint the velocity ratio will always be unity. At the same time the angle between the gear-box and front wormshaft is much more satisfactory than if both were horizontal.

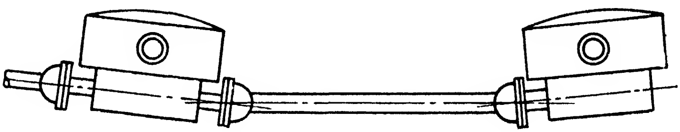


FIG. 11

Torque Reaction. All the bogie arrangements so far considered have the effect of dividing the dead weight on each side equally between the two wheels. This equality is not always preserved when the wheels are driving, as the torque reaction tends to transfer the load from one axle to the other.

In Fig. 12 the axles of the bogie are linked together at each side by two parallel springs, and the latter fixed to a trunnion block pivoted at X. If W is the total weight of the vehicle, and the centre of gravity is situated at distance b behind the front axle, then

$$W = R_t + R_l + R_f$$

and $$Wb = (R_t + R_l) \times L$$

where $L = (a + b)$, the wheelbase

R_t = load on trailing axle

R_l = ,, ,, leading ,,

R_f = ,, ,, front ,,

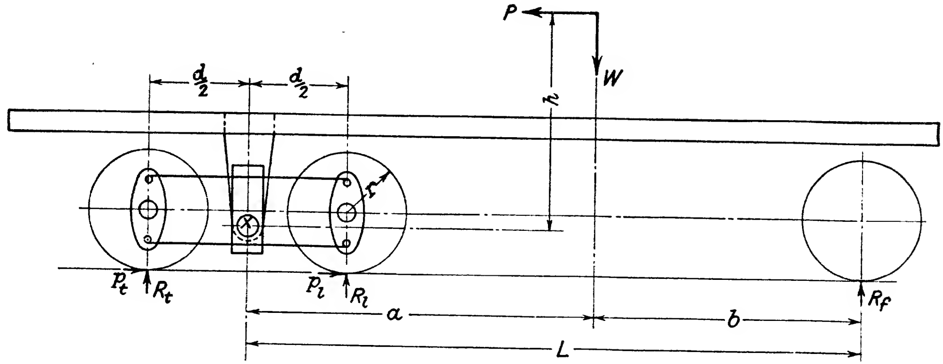


FIG. 12

Suppose the total resistance to motion is P; this must equal the tractive effort exerted at X. P is taken to act at height h above X. Now the torque reaction will tend to rotate the axles in a direction opposed to the rotation of the wheels, with the result that the bogie is likewise constrained to swivel about X. When the vehicle is being driven in a forward direction, as in the diagram, this torque must clearly be opposed by an increase in R_t, i.e. load is transferred from the leading to the trailing axle. At the same time force P has the effect of transferring load

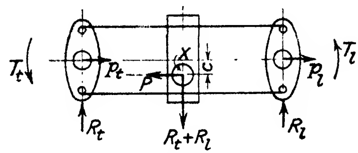


FIG. 13

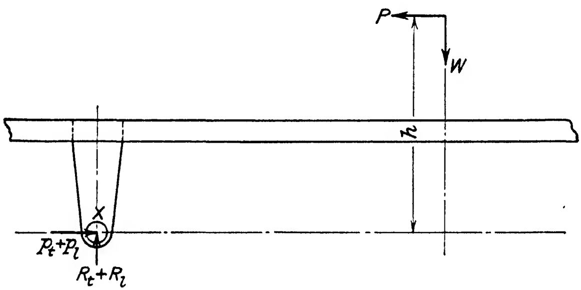


FIG. 14

from the front wheels to the bogie, so that $(R_t + R_l)$ increases proportionately to the tractive effort.

Let p_t = tractive effort exerted by trailing axle
p_l = ,, ,, ,, ,, leading ,,
r = effective wheel radius

Considering the forces acting on the bogie (Fig. 13), $p_t \times r = T_t$, the torque reaction of the trailing axle,

and
$$p_l r = T_l$$
$$T_t + T_l = T$$
$$p_t + p_l = P$$

Equating moments about X,

$$(T_t + T_l) + R_l \cdot \frac{d}{2} = R_t \cdot \frac{d}{2} + (p_t + p_l) \cdot c$$

$$\text{i.e. } T + R_l \cdot \frac{d}{2} = R_t \cdot \frac{d}{2} + \frac{Tc}{r} \quad . \quad . \quad . \quad . \quad (1)$$

Consider now the forces acting on the frame (Fig. 14) and take moments about the front wheel.

$$(R_t + R_l)\, L = Ph + Wb$$
$$= \frac{Th}{r} + Wb \quad . \quad . \quad . \quad . \quad (2)$$

From Equation 1,

$$R_t - R_l = \frac{2T}{d}\left(1 - \frac{c}{r}\right) \quad . \quad . \quad . \quad . \quad (3)$$

From Equation 2,

$$R_t + R_l = \frac{Th}{rL} + \frac{Wb}{L} \quad . \quad . \quad . \quad . \quad (4)$$

Hence,

$$R_t = \frac{Th}{2rL} + \frac{Wb}{2L} + \frac{T}{dr}(r - c) \quad . \quad . \quad . \quad (5)$$

and

$$R_l = \frac{Th}{2rL} + \frac{Wb}{2L} + \frac{T}{dr}(r - c) \quad . \quad . \quad . \quad (6)$$

Thus the effect of torque T is to transfer a load equal to $\frac{2T}{dr}(r - c)$ from the leading to the trailing axle. If h and T are known, the remaining data are easily obtainable, and the load transferred can be calculated.*

* This method of determining the effect of torque on load distribution will be found developed more fully in the *Automobile Engineer*, May, 1931 ("Rigid Six-wheeled Vehicles," by W. Steeds).

The expression for R_t and R_l is, therefore, composed of three distinct parts. The term $\frac{Wb}{2L}$ expresses the static loading of the axles, $\frac{Th}{rL}$ is the load transferred from the front axle to the rear bogie by the action of the resisting force P, whilst $\frac{T}{dr}(r-c)$ is the transferred load within the bogie itself.

It is now possible to determine the maximum torque which can be developed without wheel spin.

Case I. If a differential is fitted between the axles, T_t must equal T_l. Since R_t is greater than R_l, the limiting factor is clearly the tendency for the leading wheels to slip.

Let μ = coefficient of friction between tyres and road.

Then, when slip is about to commence

$$T_l = \mu \,.\, R_l r = \frac{T}{2}$$

and $R_l = \frac{T}{2\mu r}$ . . . . (7)

Substituting this value in Equation 6, the limiting torque can be found.

Case II. If the torque is not equally divided, the limit may be either the tendency for the leading axle to lift or for both to lift.

In the first case $R_l = 0$, hence $T_l = 0$ and $T_t = T$. Substituting these values in Equations 1 and 2, it will be found that

$$T_{limiting} = \frac{Wbdr}{2L(r-c)-hd} \quad . \quad . \quad . \quad . \quad (8)$$

On the other hand, if both wheels slip, $T_t = \mu r R_t$ and $T_l = \mu r R_l$.

Substituting for R_t and R_l in Equation 2, it will be found that

$$T_{limiting} = \frac{Wb\mu r}{L - \mu h} \quad . \quad . \quad . \quad . \quad . \quad (9)$$

It should be observed that, in the design illustrated, it is advantageous to arrange the trunnion X below the centre line of the axles, as the transference of load is reduced.

A numerical example will show more clearly the magnitude of the torque reaction effect. The following represents a typical case, and approximately represents normal (top gear) running conditions.

$$\begin{aligned} W &= 20{,}000 \text{ lb.} \\ a &= 6 \text{ ft.} \\ b &= 10 \text{ ft.} \\ L &= 16 \text{ ft.} \\ T &= 2{,}500 \text{ lb. ft.} \\ r &= 1{\cdot}5 \text{ ft.} \\ d &= 3{\cdot}5 \text{ ft.} \\ c &= 0{\cdot}5 \text{ ft.} \\ h &= 4{\cdot}0 \text{ ft.} \end{aligned}$$

It will be found that the dead load on each rear axle is 6,250 lb.

From Equations 5 and 6,

$$R_t = 6{,}934 \text{ lb.}$$
$$R_l = 5{,}982 \text{ lb., the transferred load,}$$

being 952 lb., or 7·4 per cent of the total load on the bogie. If the trunnion had been in line with the centres of the wheels ($c = o$), this would have been increased to 1,429 lb., or 11 per cent.

It is now possible to calculate the maximum torque which can be applied to the driving wheels without slipping. If a differential is used between the axles we have seen that the limiting condition is given by

$$R_l = \frac{T}{2\mu r}$$

Let $\mu = 0{\cdot}6$. Then, using the same data as before,

$$R_l = \frac{T}{1{\cdot}2 \times 1{\cdot}5} \text{ (Equation 7)}$$

Substituting in Equation 6,

$$\frac{T}{1{\cdot}2 \times 1{\cdot}5} = \frac{T \times 4{\cdot}0}{2 \times 1{\cdot}5 \times 16} + \frac{20{,}000 \times 10}{2 \times 16} - \frac{T}{3{\cdot}5 \times 1{\cdot}5}$$

$$\text{i.e. } \frac{T}{1{\cdot}5}\left[\frac{1}{1{\cdot}2} - \frac{4}{32} + \frac{1}{3{\cdot}5}\right] = \frac{10^5}{16}$$

Hence $T_{limiting} = 9{,}432$ lb. ft.

If the torque is unequally distributed, and the limiting factor is the tendency of the leading wheels to lift, substitution in Equation 8 shows that

$$T_{limiting} = 58{,}330 \text{ lb. ft.}$$

If both wheels tend to slip,

$$T_{limiting} = 13{,}230 \text{ lb. ft. (Equation 9).}$$

The latter will, therefore, be the deciding factor in the case under consideration. However, such a torque is large compared to the weight of the vehicle, and it may be concluded that in neither case will slip be likely to occur on dry roads.

The magnitude of P, and its height h, will vary according to the conditions prevailing at any given moment. The factors influencing P and h are—

(*a*) Wind resistance, acting at the centre of pressure.

(*b*) Rolling resistance.

(*c*) Inertia, acting at the centre of gravity during acceleration.

(*d*) A component of the weight of the vehicle, acting parallel to the direction of forward motion and through the centre of gravity when ascending a gradient.

However, maximum torque is usually only exerted when starting from rest on a hill, so that, unless a very strong head wind has to be allowed for, item (*a*) can be ignored.

It is interesting to consider one other case, viz. when the entire torque is opposed by the resistance of the driving wheels themselves to motion.

Then $p_t = p_l = O$, and Equations 1 and 2 must be re-written as follows—

$$(T_t + T_l) + R_l \cdot \frac{d}{2} = R_t \cdot \frac{d}{2} \quad . \quad . \quad . \quad . \quad (10)$$

$$(R_t + R_l)\, L = Wb \quad . \quad . \quad . \quad . \quad (11)$$

$$\text{Hence } R_t = \frac{Wb}{2L} + \frac{T}{d} \quad . \quad . \quad . \quad . \quad (12)$$

$$R_l = \frac{Wb}{2L} - \frac{T}{d} \quad . \quad . \quad . \quad . \quad (13)$$

Using the previous data,

$$R_t = 6{,}964 \text{ lb.}$$
$$R_l = 5{,}536 \text{ lb.}$$

This condition is, however, more likely to arise when the lower gears are in use. If T be increased to 10,000 lb. ft., it will be found that—

$$R_t = 9{,}107 \text{ lb.}$$
$$R_l = 3{,}393 \text{ lb.}$$

Transferred load = 5,714 lb., or 45·6 per cent.

Fig. 15 shows a slightly different type, in which the extremities of the springs are pin-jointed as before, but their centres are separately pivoted to a frame member, thus forming a parallel motion linkage. With this

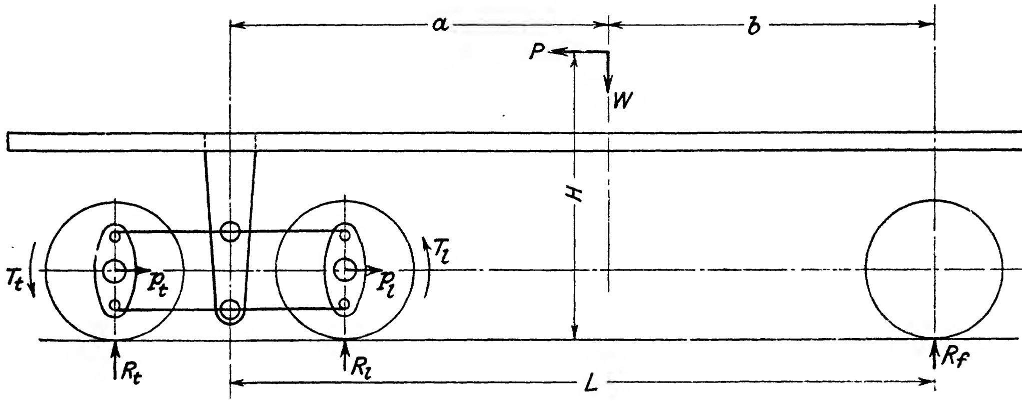


FIG. 15

arrangement the torque reactions can no longer disturb the equality of the vertical reactions, and $R_t = R_l$ under all conditions. By equating torques the expression for R_t and R_l is found to be

$$R_t = R_l = \frac{Wb}{2L} + \frac{TH}{2rL} \quad . \quad . \quad . \quad . \quad (14)$$

This is of the same form as Equations 5 and 6. Substitution of the numerical data already used gives

$$R_t = R_l = 6{,}510 \text{ lb.}$$

Ignoring the unlikely tendency of the front axle to lift, the only limit to the torque is the simultaneous slipping of all four driving wheels, and

$$T_{limiting} = \frac{Wb\mu r}{L - \mu H} \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Taking $\mu = 0{\cdot}6$, as before, the limiting torque is found to be 13,890 lb. ft.

In the W.D. type previously illustrated, the trunnion axis is usually arranged as close as possible to the level of the wheel centres, with normal spring loading. It can then be shown that the reactions are always equal, and are given by Equations 12 and 14. The limiting torque will be as Equations 9 and 15.

Stresses in Springs Due to Torque Reaction. It should be observed that in some designs of bogie the torque reaction will impose longitudinal forces on the springs, and possibly in the torque resisting members as well. Consider the W.D. design again. Since the springs are attached at each end to a member which is free to rotate on the axle, they are relieved of torque reaction, and have only to transmit the tractive effort of the axles. In Fig. 16 p represents the thrust of the axle; this is opposed by the reaction of the springs, and

$$p = F_2 + F_3$$

The torque reaction T is balanced by the torque arm, and $T = F_1k = pr$. Often $K = r$ approximately, where r is the wheel radius.

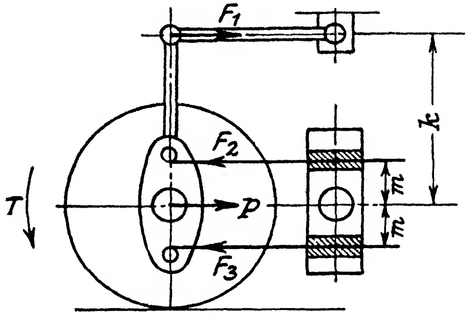


FIG. 16

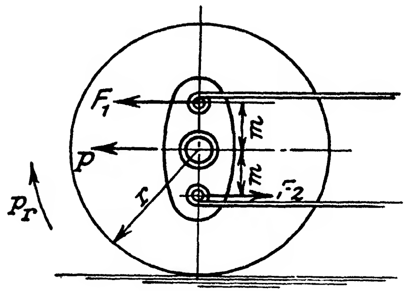


FIG. 17

Then, $F_1 = \dfrac{T}{r} = p$ . . . . . . (16)

$F_2 = F_3 = \dfrac{p}{2}$ . . . . . . (17)

If the torque reaction is taken by the springs (Fig. 17) then, equating torques—

$$F_1 \,.\, m + F_2 \,.\, m = pr$$

Also, $F_1 + p = F_2$ . . . . . . (18)

Let $m = \frac{}{3}$

Then, $F_1 + F_2 = 3p$ . . . . . . (19)

$F_1 - F_2 = -p$ . . . . . . (20)

$\therefore \quad F_1 = p$ . . . . . . (21)

$F_2 = 2p$ . . . . . . (22)

Thus the thrust in the bottom spring is four times as great as before.

Torque Reaction Members. The desirability of providing a special torque resisting member is thus decided by two factors—

1. The design of the bogie in relation to permissible inequality of axle loads, and

2. The ability of the road springs to withstand longitudinal stresses.

On cross-country and military vehicles it is usually considered desirable to relieve the springs of these stresses in view of the other heavy duties they have to perform. In any case the design should be such that the axle loading is not disturbed by the torque reactions, since only two wheels may at times be available for driving.

It will be found that goods and passenger vehicles operating on main roads usually have a fair margin of adhesive weight, so that some disturbance of weight distribution is permissible.

If a torque member is to be used, there are several alternatives to the hinged link already considered. The early Goodyear designs had a telescopic member between the axles, consisting of two tubes, one sliding within the other. This relieved the springs of torque reaction loads, but obviously transferred some of the weight from the leading to the trailing axle.

It is also possible to transfer the reactions of each axle separately to the frame by a torque tube or any similar device used on two-axle vehicles.

Spring Mountings. The purpose of the bogie springs is, of course, to smooth out vertical forces. It has been shown, however, that they may be subjected to loads in two other planes; longitudinal forces set up by

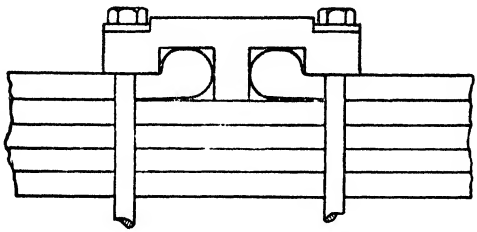


FIG. 18

torque reactions, and transverse or twisting loads when turning.

The attachment of the springs to the frame must, therefore, be very rigid. Messrs. Jonas Woodhead have patented a "divided back-plate" spring shown in Fig. 18. The main leaf is in two parts, and the inside eyes enable the spring to be positively located so that the tendency for it to work loose is minimized. On the Thornycroft (Fig. 10) the springs are secured in box anchorages by wedge blocks retained by a pair of bolts. When a trunnion bearing is used it should be of ample length to resist side thrusts on the axles.

Reference has been made to the twisting of the springs when the axles are tilted up (page 724); the two short springs often employed on six-wheelers are obviously subjected to greater torsion than one long spring. Some improvement is effected by the arrangement shown in Fig. 21 where the springs are attached to extension arms on the axles. As these arms are highly stressed the additional length of spring obtainable is strictly limited.

The Third Differential. It is, of course, usual to provide a differential gear between the off- and near-side wheels of each driving axle. The necessity for a third differential, to divide the engine torque between the two axles of the bogie, has been the subject of much debate, and opinion is still divided. It is obvious that in practice the effective diameters of the tyres on both

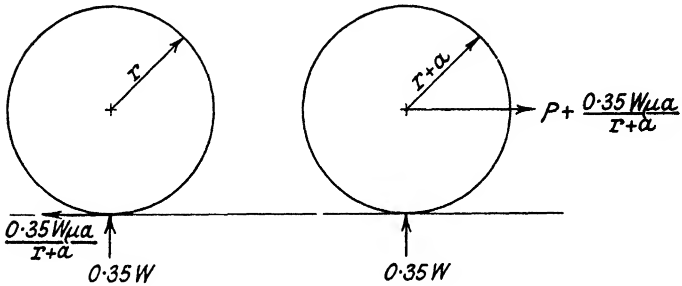


FIG. 19

axles will seldom, if ever, be exactly equal. Tyre pressures frequently differ, and the treads may not wear equally, whilst inequality of load distribution also influences tyre deflection.

Consider a typical case, in which 70 per cent of the total weight of the vehicle is distributed equally between the two rear axles (Fig. 19). Then if W = total weight in lb. each axle will support 0·35 W lb. For simplicity it will be assumed that the weight distribution is not disturbed by the torque reaction.

Let r = rolling radius of wheels on one axle, in feet.

$r + a$ = rolling radius of wheels on the other axle, in feet.

P = tractive resistance of vehicle, lb.

μ = coefficient of friction between tyres and road.

If no differential gear is provided between the axles, it must be assumed that the larger wheels will exert all the driving effort, and the smaller wheels must slip.

Suppose the driving wheels to rotate through 1 radian, then the vehicle will move forward through a distance $(r + a)$, and the smaller wheels will slip by an amount a.

Useful work expended in driving the vehicle

$$= P\,(r + a) \text{ ft. lb.}$$

Additional work expended in dragging the smaller wheels $= 0{\cdot}35\,W\mu a$ ft. lb.

$$\text{Total effort} = P + \frac{0{\cdot}35\,W\mu a}{r + a} \text{ lb.}$$

The magnitude of the "drag," $\dfrac{0{\cdot}35\,W\mu a}{r + a}$ can be better appreciated by a numerical example.

Let $P = 40$ lb. per ton

$\mu = 0{\cdot}6$

$a = 0{\cdot}01\,r.$

$$\text{Then useful effort} = \frac{40\,W}{2240} = 0{\cdot}0179\,W \text{ lb.}$$

$$\text{Drag} = \frac{0{\cdot}35\,W \times 0{\cdot}6 \times 0{\cdot}01\,r}{1{\cdot}01\,r} = 0{\cdot}00208\,W \text{ lb.}$$

Thus it appears that a difference in rolling radius of 1 per cent has increased the tractive resistance by

$$\frac{0{\cdot}208}{0{\cdot}0179} = 11{\cdot}6 \text{ per cent.}$$

Fortunately the effect of a small difference in radii is not so critical as would appear from the foregoing analysis, owing to the flexibility of rubber tyres. It is probable that, as the tyre drives against the road surface, some compression of the material approaching the point of contact takes place, and this has the effect of reducing the rolling radius in proportion to the torque applied. Consequently the tyres themselves have a tendency to equalize their radii, and, therefore, the torque transmitted by them.

The actual magnitude of the drag caused by unequal wheel radii has been investigated by the measurement of tractive resistance. In one test* the drawbar resistance of a loaded six-wheeler was measured under various conditions. It was found that when the tyre pressures of the leading and trailing axles were 30 lb. and 110 lb. per sq. in. respectively, the tractive resistance was 9 per cent greater than with equal pressures in all tyres. Starting from rest this was increased to 12 per cent.

Another investigator† estimated the increase of drag of the bogie alone by rolling it down a hill, with and without a differential. It was ascertained that a 40 lb. difference of tyre pressure was the maximum that could be passed by casual inspection of the tyres, and under these conditions the rolling resistance of the bogie was increased by 27 per cent without the differential.

Unfortunately, neither of these experiments was carried out with the wheels actually driving, so that they only apply when the vehicle is "coasting." Furthermore, the differences of tyre pressure are exceptional, and would not be tolerated by any organized transport concern; apart from any question of wheel slip the partially deflated tyres would quickly be ruined by such treatment. If it be conceded that the possible

* G. Watson, *Proc. I.A.E.*, vol. xxii, p. 185.

† G. J. Rackham, "Notes on Bus Chassis Design," *Proc. I.A.E.*, vol. xxii, p. 413.

variation in tyre pressure is unimportant, the only other factor likely to affect the tractive resistance is the transference of load by torque reaction. The numerical example on page 732 shows that in some designs the axle loadings may differ by 10 per cent under normal top gear conditions, whilst under the worst conditions they may exceed 40 per cent. The design of the bogie must, therefore, be taken into consideration in deciding whether a third differential is required.

Where the axles are relieved of torque reaction it is unlikely that the tractive resistance will be noticeably increased in the absence of a differential. It is found in practice that the fuel consumption of omnibuses of this type compares satisfactorily with the equivalent four-wheeler.*

It has been presumed so far that the vehicle is operating on level roads. For cross-country work a differential between the axles is out of the question, since any wheel lifting out of contact with the ground would spin at four times the speed of the rear cardan shaft. On the other hand, if the final drive gears are compelled to rotate together wheel spin cannot occur unless the two driving wheels on one side, or two diagonally opposed wheels, lift off the road simultaneously. The absence of wheel spin is, in fact, quite remarkable. A strictly comparative test between a four- and six-wheeled vehicle, on a gradient varying from one in eight to one in eleven, showed that the wheel spin of the latter was 1½ turns in 250 yds. as compared to four turns on the former.†

When no third differential is used it is theoretically possible for the full engine torque to be applied momentarily to the one axle. This is an extreme case, but it is evident that the final drive components will be subjected

* C. Owen Silvers, *Proc. I.A.E.*, vol xxii, p. 194.
† S. S. Guy, *Proc. I.A.E.*, vol. xxii, p. 184.

to a greater range of stress than when the torque is equally distributed by differential gearing. Thus the additional weight of the latter is to some extent offset by the smaller worm or other gears.

The case for and against the third differential can now be briefly summarized as follows—

For	*Against*
Avoids "drag" set up by unequal tyre radii.	Provided the axle loads are approximately equal under all conditions, small variations of tyre pressure are unimportant.
Equalizes driving loads on wheels and minimizes tyre wear.	Permits wheel spin, and therefore unsuitable for cross-country purposes.
Permits smaller final drive gears.	Additional weight and complication not justified.

Apart from cross-country vehicles, opinions vary as to the relative importance of the above points. So far, no comparable data relating to tyre wear, etc., with and without the differential, have been published, and there is no general agreement, even amongst reputable manufacturers.

Construction of Third Differential. The inter-axle differential is usually small in comparison with those fitted between the wheels, as the torque transmitted is less, although the rotational speed is correspondingly increased.

The most common practice, where worm-geared axles are used, is to accommodate the extra gearing within the worm-shaft housing of the leading axle, as in the A.E.C. design.

The A.E.C. Six-wheel Omnibus Chassis. The A.E.C. "Renown" chassis has been designed for double-deck omnibus bodies, and is, therefore, intended primarily for urban conditions. Fig. 21 shows the arrangement of the bogie. The worm gears are disposed to the near side, beneath a longitudinal seat, so that a low gangway

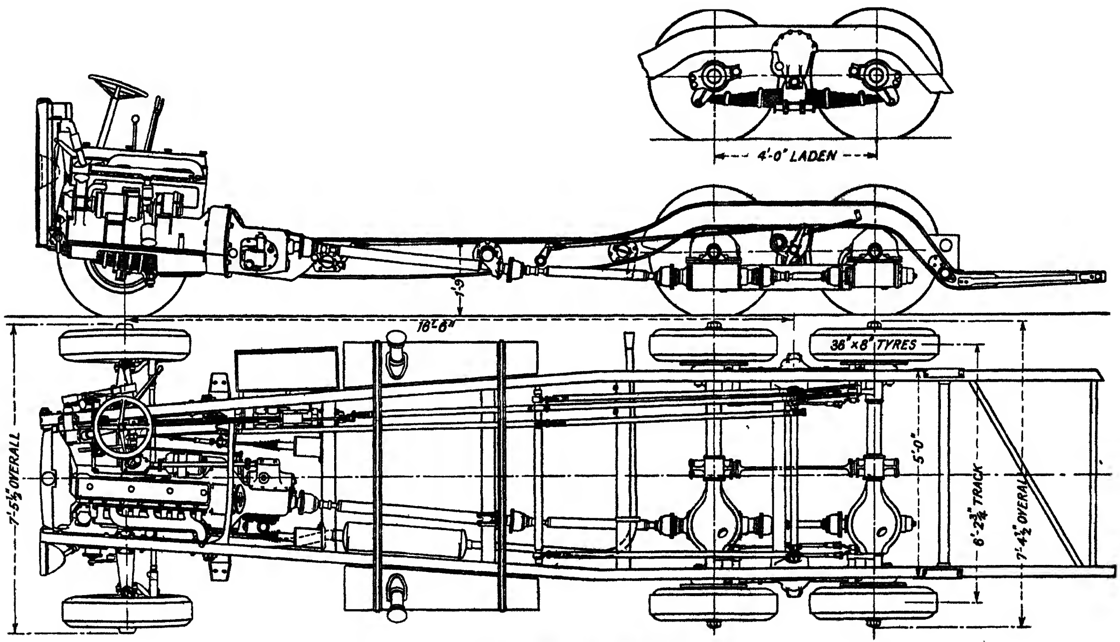
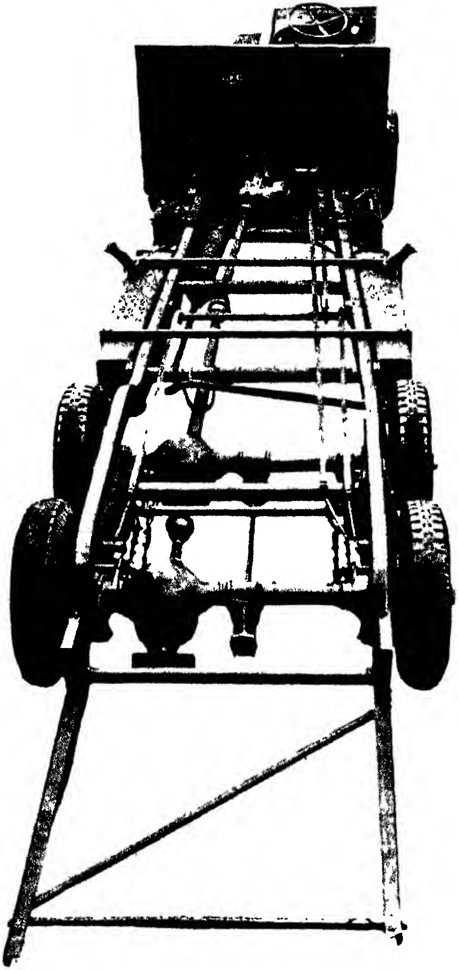


Fig. 20. The A.E.C. Six-wheel Omnibus Chassis



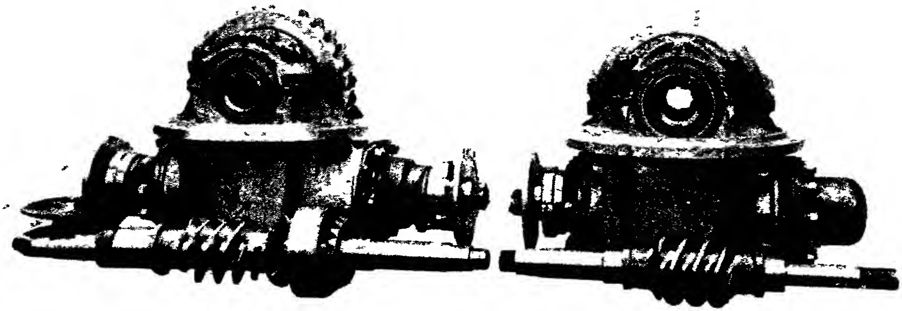
(*A.E.C., Ltd.*)
FIG. 21. A.E.C. "RENOWN" CHASSIS

can be provided in the centre. The axle tubes are coupled by a spring steel torque blade, after the manner of the early Goodyear design, mounted in rubber at one end and free to rotate and slide in a long bearing at the other. This blade is of a deep section in the vertical plane, and sufficiently rigid to transfer torque reaction loads from one axle to the other. At the same time it is flexible in the horizontal plane and free to accommodate minor inequalities of axle movement. Articulation is, of course, provided by the sliding bearing.

A single road spring of considerable length is provided at each side (Fig. 20). These are clamped at the centre to a massive bracket suspended beneath the trunnion pin. The latter is, in turn, supported by bushes in a cast steel bracket riveted to the main frame. Grease nipples are available for lubrication of the trunnion. The total load on the bogie is shared by the two trunnion pins, which are in double shear. However, apart from considerations of stress, they are amply proportioned in order to steady the bogie against twisting. The upward movement of both axles is limited by rubber blocks, which come into contact with the lower flange of the frame.

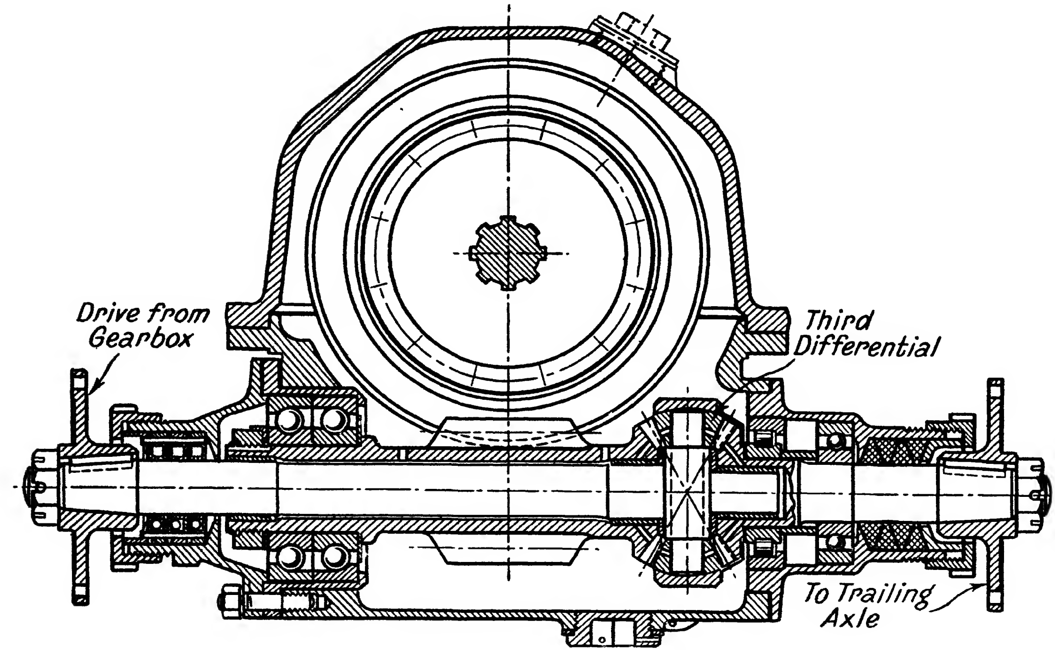
It is evident that, in this design, the driving torque will transfer weight from the leading to the trailing axle. Possibly for this reason a third differential is provided. It is located at the rear of the wormshaft on the leading axle (Fig. 22). The drive is transmitted through a hollow worm to the centre of the bevel differential, and thence back from one side to the worm itself. The other side of the differential distributes the driving torque to the trailing axle (Fig. 23). The arrangement is compact, and avoids the necessity for additional bearings.

The remainder of the axles is of conventional design;



(A E C, Ltd)

FIG. 22. LOCATION OF THIRD DIFFERENTIAL ON WORMSHAFT



(A.E.C. Ltd.)

FIG. 23. DISTRIBUTION OF DRIVING TORQUE

they are of the semi-floating type, and employ taper roller bearings.

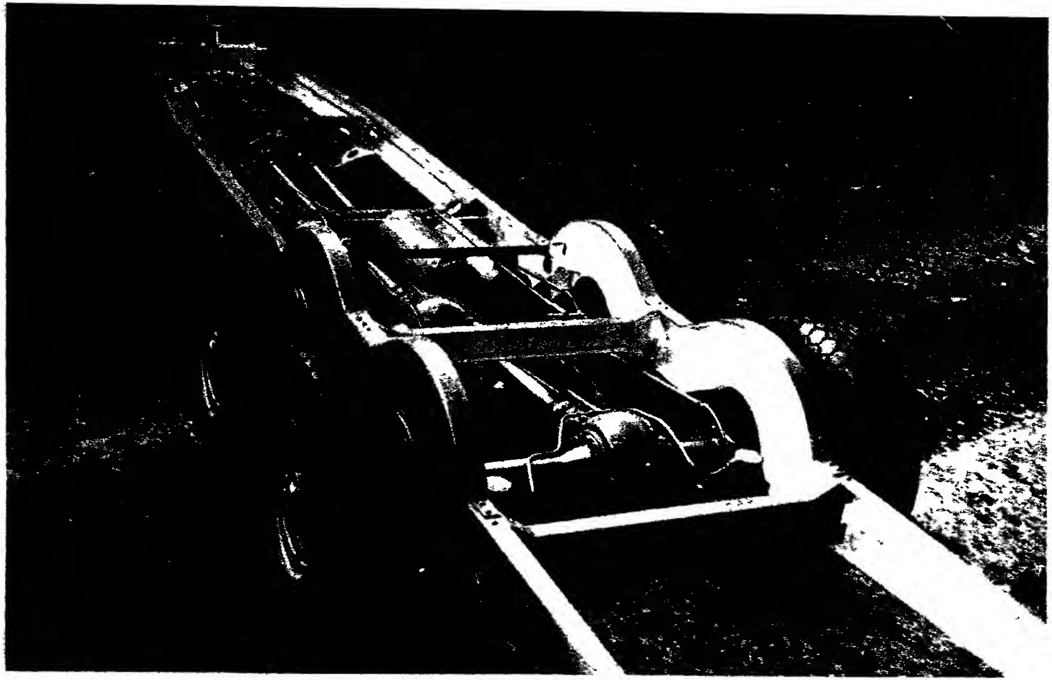
The following are the principal data of the chassis—

	Short Wheelbase	Long Wheelbase
Wheelbase (front axle to centre of bogie)	16 ft. 6 in.	18 ft. 7 in.
Track	6 ft. $2\frac{3}{4}$ in.	6 ft. $2\frac{3}{4}$ in.
Overall length	26 ft. $9\frac{3}{8}$ in.	29 ft. $0\frac{5}{8}$ in.
Chassis weight	4·75 tons	4·83 tons
Engine	6 cyl. 110 × 130 mm. 120 b.h.p.	
Tyres	36 in. × 8 in. single front and twin rear; 38 in. × 9 in. alternative twin rear	

The Büssing Chassis. The Büssing Company, of Brunswick, Germany (now Büssing-Nag A. G.), was the first European concern to manufacture six-wheelers. The design is unusual, two separate cardan shafts being employed. At the rear of the gear-box (Fig. 24) a train of gears distributes the driving torque to a pair of shafts. The rear portion of each drive is encased in a torque tube attached to one of the axles of the bogie, so that torque reactions are transmitted to the frame (Fig. 25). A single road spring of the "progressive" type is clamped at its centre to the frame on each side. No differential gearing is provided between the two transmission shafts. It is claimed for this design that excessive angularity of the shafts, such as may occur with the normal tandem arrangement (Fig. 5), cannot take place.

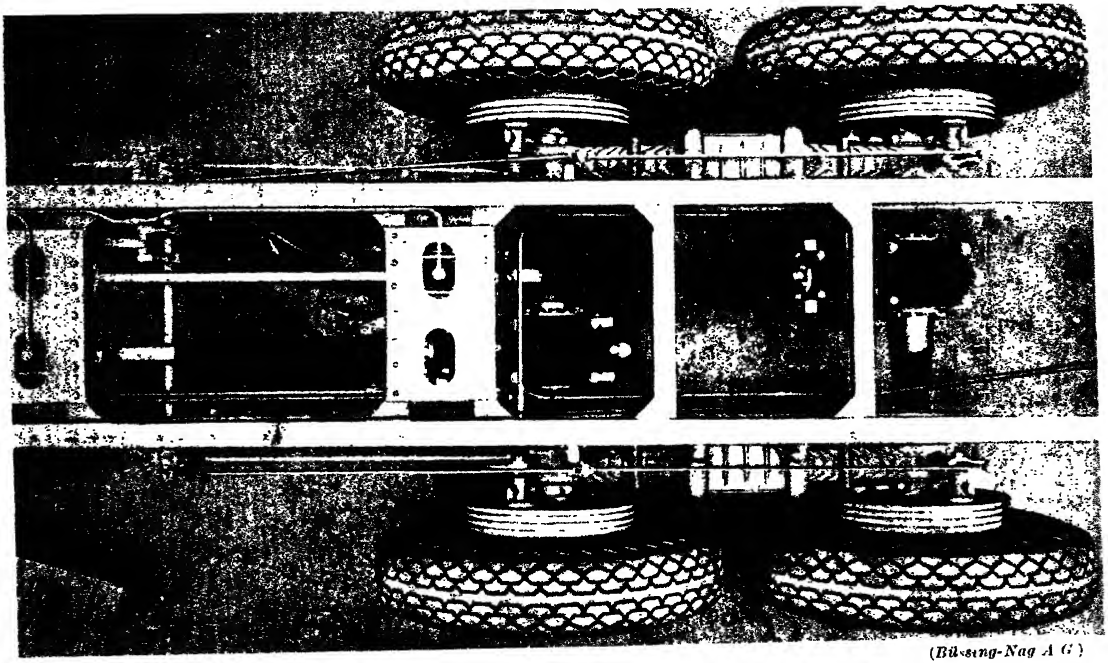
The chassis is of considerable size, and rated to carry a useful load of 9·3 tons with 12·00 in. × 20 in. tyres, or 8·9 tons with 40 in. × 10 in. tyres. The engine is a 6-cylinder unit of 125 mm. bore and 160 mm. stroke. The wheelbase (front axle to centre of bogie) is no less than 20 ft. 2 in., and the overall width 7 ft. 8 in. Such a vehicle would not be legal in this country.

The Scammell "Pioneer" Six-wheeler. This is another



(Büssing-Nag A.G.)

FIG. 24. BUSSING-NAG CHASSIS



(Büssing-Nag A G)

Fig. 25. Bussing Nag Chassis
Plan view showing separate torque tubes

unconventional design employing only one rear axle (Fig. 26). The outer end of each axle shaft carries a pinion located between two idler wheels. Each of the latter drives another gear wheel attached to one of the wheels. The five gears on each side are enclosed in a steel casing, and are supported on ball bearings. As the wheels on either side of the bogie deflect vertically the casing pivots about the centre pinion.

The front axle is also unorthodox, being pivoted transversely at its centre. The vehicle is capable of traversing particularly rough ground (Fig. 27).

Wheelbase (front axle to centre of rear axle)	15 ft. 1 in.
Track	6 ft. 9 in.
Chassis weight	5·5 tons
Useful load	4 to 8 tons according to conditions.
Engine	4 cyl., 5 in. × 5½ in. 85 b.h.p.
Tyres	13·5 in. × 20 in. single

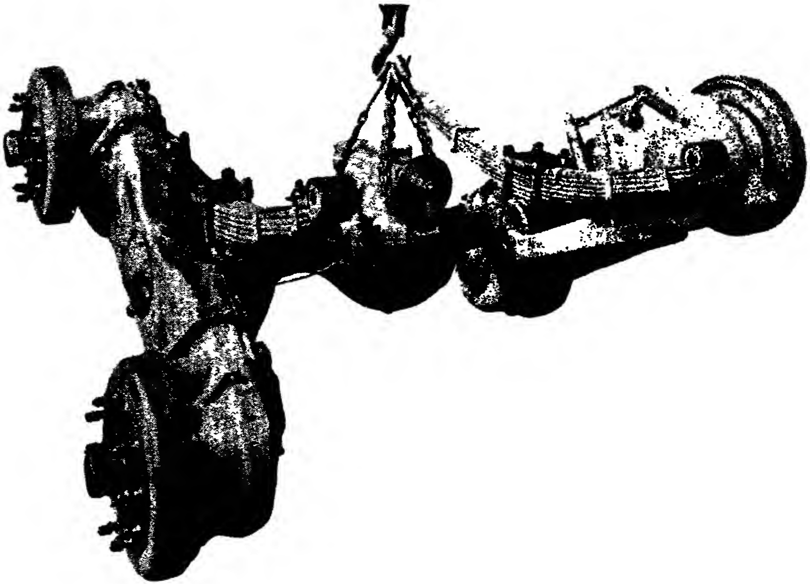
Trailing or Dead-axle Bogies. So far we have only considered the type of bogie in which both axles transmit a proportion of the driving torque. In certain circumstances only one axle performs this duty, whilst the other—usually the trailing one—is "dead." Obviously the success of this arrangement is dependent on the ratio of the maximum torque to the adhesive weight on the driving wheels; neither would it be suitable for the military type of vehicle. However, it has the advantages of simplicity and lower first cost, and there is some saving in weight.

Vehicles of this type fall into two classes—

(*a*) Those which are designed and built with a dead axle, and

(*b*) Conversion outfits.

The former comprise a few makes of heavy lorry intended for operation on main roads. Conversion units



(*Scammell Lorries, Ltd*)

FIG 26. REAR AXLE, SCAMMELL "PIONEER"

(*Scammell Lorries, Ltd.*)

FIG. 27. SCAMMELL "PIONEER" LORRY

are usually of a proprietary nature; they enable a standard type of four-wheel vehicle, such as the Chevrolet 30-cwt. van, to be given an increased carrying capacity without overloading the driving axle and wheels. The unit, therefore, comprises a dead axle with suitable

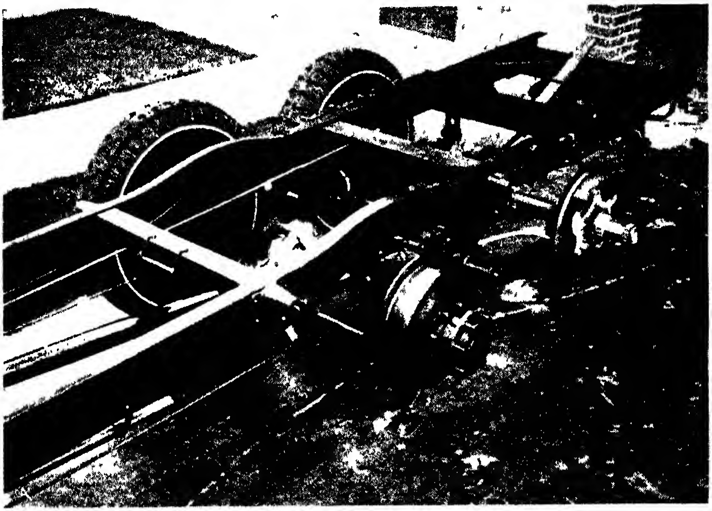
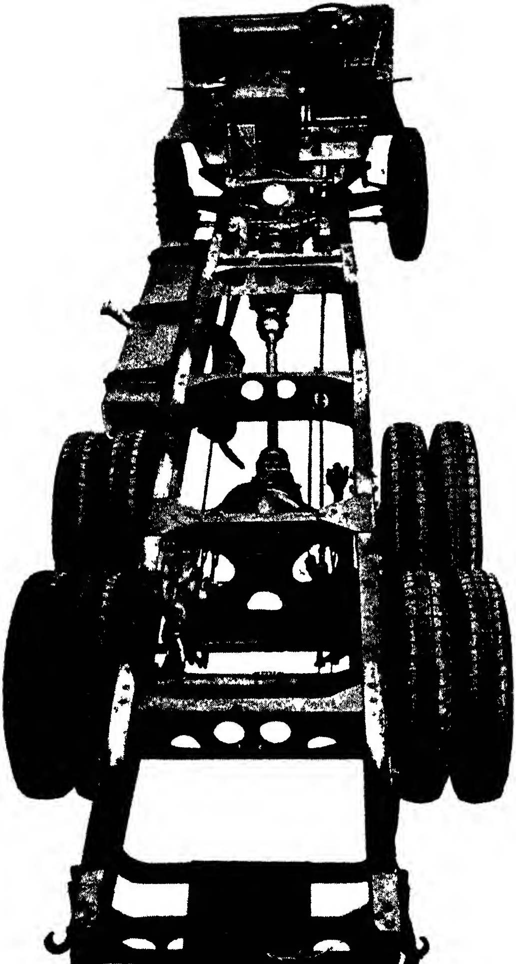


FIG. 28. FLEXION CONVERSION EQUIPMENT
(Flexion Products, Ltd.)

wheels and tyres, an equalizing system which can be attached to the existing rear springs, and an extension which can be bolted to the rear of the existing frame.

The Flexion Conversion Unit. This is shown in Fig. 28. The frame extension is bolted on just behind the arch over the driving axle, and carries on each side an equalizing beam. The latter is attached by a link at the front to the rear shackle of the spring above the driving axle; the rear end of the beam supports one end of the spring on the trailing axle. The parts are proportioned



(*A E C , ltd*)

FIG. 29. A.E.C. "MAMMOTH MAJOR" GOODS CHASSIS

so that 60 per cent of the total weight on the bogie is transmitted to the driving axle in order to ensure adequate adhesion. The equalizing beam is a malleable iron casting of substantial I-section, steadied laterally on the trunnion by hardened steel rubbing plates.

The A.E.C. "Mammoth Major." This is an example of class (*a*), and is marketed as a goods chassis of 12-ton carrying capacity. The leading axle (Fig. 29) is of the double reduction type, and is designed to transmit a maximum torque at the driving wheels of 19,000 lb. ft. The bogie suspension (Fig. 30) is similar to that adopted for the "Renown" chassis already described. In this case, however, the torque reaction is taken to the frame. A bracket (*as shown facing this page*), bolted to the axle casing, is pin-jointed to one end of a torque rod. The other end is attached to the frame by a spherical bearing. It is evident that in this case the torque reaction will not disturb the load on the driving axle, and satisfactory adhesion is obtained under all normal conditions.

Wheelbase (front axle to centre of bogie)	16 ft. 10½ in.
Track	6 ft. 3⅜ in.
Overall length	27 ft. 7 in.
Chassis weight	6·1 tons
Engine	6 cyl., 110 × 130 mm. 120 b.h.p.
Tyres	40 in. × 8 in. (front and twin rear) 13·5 in. × 20 in. (alternative single rear).

Brake Gear on Six-wheel Vehicles. The bogie usually carries a greater proportion of the total weight than the rear axle of a four-wheeler, although not twice as much. It is, therefore, usual to provide brakes on both axles of the bogie, whilst, not infrequently, front wheel brakes are omitted. The design of brake shoes and

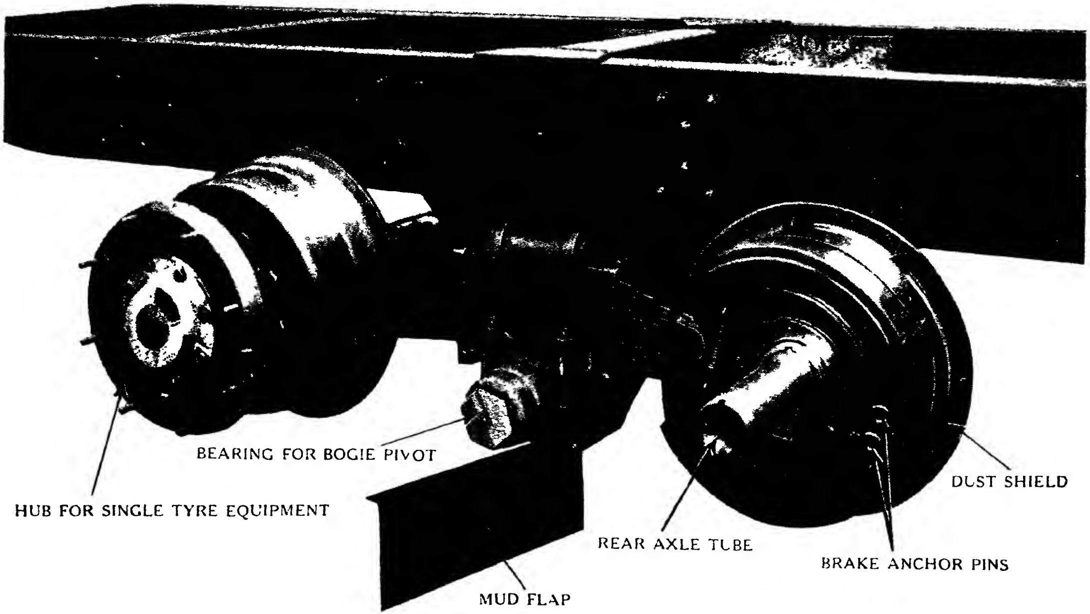


FIG. 30. "MAMMOTH MAJOR" SUSPENSION (A.E.C., L'd.)

drums is dealt with in another section, but the operating gear calls for special consideration.

The cam lever pull rods must be arranged so that the brakes are unaffected by axle movements: hence they usually converge on a point near the axis of the trunnion.

Unless a third differential is used it is not necessary to duplicate the controls to both axles. For instance, if the hand brake is applied to shoes in the leading axle the trailing wheels must likewise be retarded. Some manufacturers, therefore, couple the hand brake to one axle and the foot-brake to the other. In this case the final drive gearing and inter-axle cardan shaft must be capable of transmitting powerful braking torques.

It must not be overlooked that in designs where the driving torque sets up unequal loading of the axles, the braking torque will transfer the load in the opposite direction. In conjunction with a third differential this may result in a tendency for the wheels on one axle to skid, unless the brake leverages are suitably proportioned.

Anti-puncture Flaps. The tendency of a rotating wheel as it moves along the road is to throw up loose stones. In the case of the leading wheels of a bogie the stones are thrown on to the tyres of the adjacent trailing wheels, and may promote punctures. This can be obviated by a flap beneath the centre trunnion (Fig. 30); its position is dependent to some extent on the ratio of the wheel centres to their diameter, and must be determined by trial. Experience shows that the necessity for a flap is least when the wheels are as close together as possible.

SECTION XV

LUBRICATION

BY

MAJOR A. GARRARD, WH.EX.

SECTION XV

LUBRICATION

General Principles. The essential principles in accordance with which oil is utilized in the bearings of an engine have been dealt with in a separate section, while the sufficient but not excessive lubrication of the cylinder walls and pistons is discussed in the section entitled "Connecting Rods. Pistons, and Gudgeon Pins."

The methods by which oil is supplied to the bearings, cylinder walls and other parts, and the various considerations concerning its effectiveness as a lubricant and cooling medium, are dealt with in this section.

The primary consideration is always the crankshaft and big end bearings, since these suffer from a combination of heavy loading and speed which causes heating, but the various other parts must all have an adequate and continuous supply of oil. These parts include the cylinder walls, timing gear, gudgeon pins, pistons, valve mechanism, tappets, valve guides, and cams. A considerable excess over what is needed to maintain the essential thin film of oil must be supplied not only to ensure lubrication but for cooling.

Brief reference to lubrication methods of the earlier motor vehicles will be found instructive. Up to about 1909 many engines were provided with drip feed lubrication, each bearing or other special point being supplied with oil from a raised tank passing through a glass sight or drip feed indicator on the dashboard. The method was uncertain and the supply insufficient to have any cooling effect, so that the power that could be developed was very limited. Such systems, however, often utilized the splash principle in a crude manner. The surplus oil drained down into the bottom of the crankcase, which was shallow, so that the connecting rods dipped into the oil. When the level of the oil was too high the cylinder walls would be splashed too freely causing oiling up of the sparking plugs. When the sump was enlarged and troughs provided the oil was circulated by a pump which was above the oil level and trouble was liable to occur, particularly when starting up, from failure to draw oil up from the sump. In many modern systems the pump works in the oil or is drowned or submerged, and when it is arranged at a higher level the construction ensures that it is always automatically primed ready for starting.

All modern systems depend for their effectiveness upon the continual circulation through the bearings of a considerable quantity of oil. The pressure at which oil is delivered by the pump is far below the oil film pressure in the bearings. This point is dealt with in the section entitled "Bearings." Oil can only be forced to the low pressure or unloaded parts of the bearing or to the oilways. The rotating shaft drags in what it requires to maïntain the high pressure of the load-resisting film.

Oil surplus to the requirements of the bearings is splashed about and forms the oil mist which is essential for general lubrication in all engines. The maintenance of an adequate oil mist has only been rendered possible in modern engines by effective scraper rings in the pistons, which, while allowing proper lubrication of the cylinder walls, prevents oiling up of the plugs, and avoids excessive loss of oil through burning with the consequent blue smoke in the exhaust.

The various lubrication systems are classified broadly as follows, according to the method by which the oil reaches the bearing surfaces—

1. Forced feed in which oil is continuously forced

under a pressure of 25–100 lb. per sq. in. to the main crankshaft bearings, from which it passes through a drilled crankshaft to the big end bearings. This system is most widely used. Oil is often supplied under pressure to the camshaft bearings and to other parts.

2. Splash or trough systems in which oil is supplied to troughs arranged one under each cylinder, the surplus overflowing into the sump so that a constant level is maintained in each trough, the level being such that the dippers on the connecting rod caps collect a little oil and convey it direct to the crank pins, and also splash about a quantity which is sufficient but not excessive for other purposes. Oil flows down the cylinder walls and is collected in small cups or reservoirs above the main bearings to which it flows by gravity. Such pure splash systems are seldom used to-day on motor vehicles.

3. Combination systems in which oil is forced under pressure to the main bearings while the surplus is led to troughs under the dippers on the big end caps. The camshaft bearings may also be supplied under pressure.

4. Gravity systems in which the oil is raised from the sump and flows by gravity through pipes or passages to the various bearings and to troughs under the big ends. This has been used on certain Ford engines.

5. Dry sump systems which are similar to force feed systems with the exception that the bulk of the circulating oil is retained in a reservoir entirely separate from the engine instead of in the sump. The usual oil pump draws oil from this reservoir, and supplies it to the bearings and a separate pump drains the sump and returns the surplus oil to the reservoir.

In systems (1), (3), and (5) oil is also often forced to the camshaft chain or toothed gearing, and to the bearings and rockers of an overhead camshaft.

Examples of Lubrication Systems. Some examples of the different systems will now be considered in some detail. One striking characteristic of all modern systems, and more particularly engines designed for the more exacting commercial purposes, is the careful provision made for the definite feeding of oil to many of the working parts in addition to the bearings and big ends. The first system to be considered is that employed in the six-cylinder petrol engine manufactured by the Associated Equipment Company, and referred to in the section on "Bearings." Transverse and longitudinal vertical sections of this engine are shown in Figs. 1 and 3. An underneath view of a similar four-cylinder engine with the sump removed is also shown in Fig. 2.

The gear wheel type pump is driven very simply by toothed gearing from the front end of the crankshaft at 65 per cent of the engine speed. The suction side of the pump is supplied by a large diameter pipe from the rear end of the engine through a wire gauze strainer, which prevents foreign particles or large pieces of carbon from being led to the bearings. The strainer is carried on a plate bolted to the bottom of the sump, and particles of iron or steel are collected by a magnet secured to the plate. It is intended that the strainer and magnet shall both be removed from the sump and cleaned whenever the oil is changed.

The main delivery pipe from the pump runs the whole length of the engine on the opposite side to the suction pipe, and it delivers oil to each of the seven main crankshaft bearing caps through smaller pipes which are given one complete turn to impart flexibility and reduce the risk of breakage or strain when tightening up the unions. Copper always hardens when worked or bent, and is then likely to fracture as the result of vibration unless annealed in the usual way by heating to redness and plunging in water.

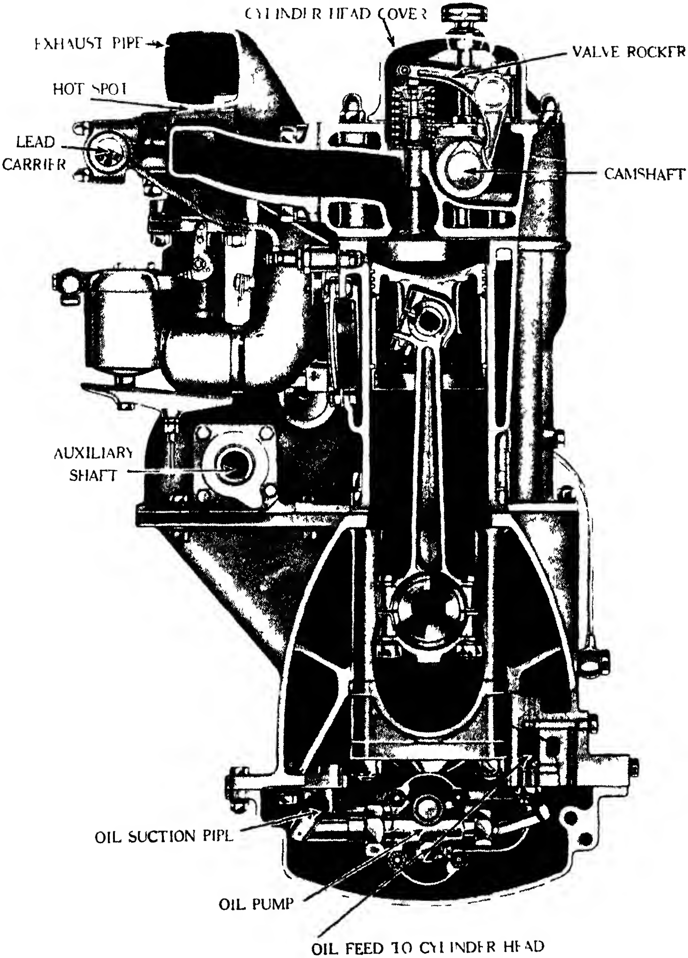


FIG 1. TRANSVERSE SECTION OF A E C ENGINE SHOWN IN FIG 3

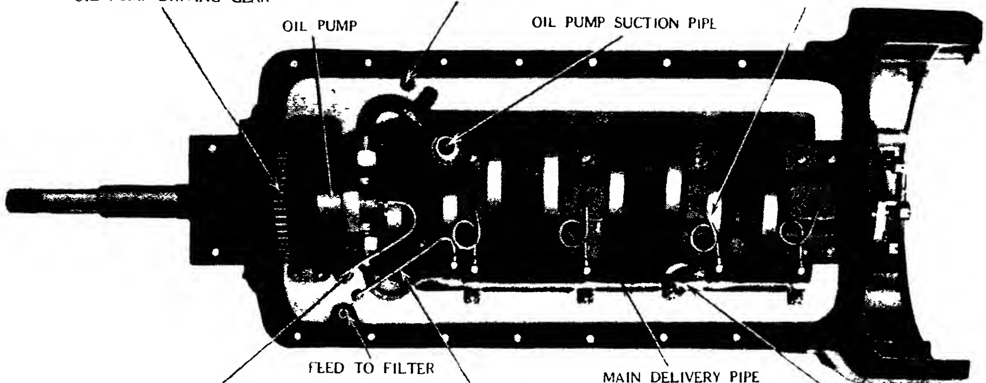


FIG. 2. UNDERNEATH VIEW WITH SUMP REMOVED OF A.E.C. FOUR-CYLINDER ENGINE

The further passage of the oil through the bearings and to the crank pins of this A.E.C. engine is described in the section on Bearings.

On the main delivery pipe is arranged a relief valve pressed on to its seat by a spring against the pressure of oil. The position of this valve is shown in Fig. 3, and its method of control in Fig. 4. The force which the spring exerts on the valve is varied by a rod which passes up through the top of the crankcase. Variation of the oil pressure in accordance with the load on the engine is effected by interconnecting the oil pressure regulator rod and the accelerator pedal in such a way that depression of the pedal increases the spring load on the valve, thus increasing the pressure in the main delivery pipe. An arm on the throttle lever (not shown) bears on the upper end of the rod. This variation of oil pressure in accordance, not only with the speed, but with the load on the engine, is a very desirable feature to secure adequate lubrication when the engine is working hard, while an excessive supply of oil when it is idling is avoided.

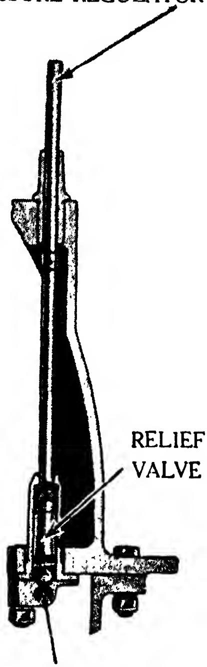


FIG. 4. A.E.C. OIL RELIEF VALVE

A small pipe, not shown, leads from the main oil pipe to a pressure gauge within sight of the driver. The pressures at which the engine, when hot, is intended to work are as follows—

When idling 5–12 lb. per sq. in.;

At full throttle 45–50 lb. per sq. in.

A screw adjustment between the throttle lever and the upper end of the regulator rod enables these pressures to be secured.

The cylinder walls, pistons, and gudgeon pins, are lubricated by splash due to the surplus oil thrown about by the cranks and forming an oil mist.

Provision is made for the regular supply of a relatively small quantity of oil to the valve and timing gear, and with this supply is associated a specially fine filter. This oil is bled from the oil pump by a special device. The fixed spindle for the driven, that is the lower, wheel of the pump is made hollow with a small radial hole which opens towards the delivery side. Between two of the teeth on the driven pump wheel is drilled a similar hole which is arranged to coincide with the first hole once every revolution, when a small quantity of oil is forced into the interior of the spindle. This oil passes through a small copper pipe to the filter, from which it is led up through a passage drilled in the crankshaft and timing case to the cylinder head.

The filter consists of a thick felt cylinder with a gauze outer cover carried in a cylindrical chamber completely partitioned off from the sump. The small supply of oil from the pump is led into this chamber and passes radially inwards through the felt to the centre of the cylinder, from which it is fed to the cylinder head. The whole of the oil is thus passed slowly through a very fine filter, and is cleaned very effectively. Such a fine filter could not be used in the main circulating system without retarding the flow unduly. To guard against interruption of the supply to the cylinder head, should the filter become choked owing to neglect, a by-pass

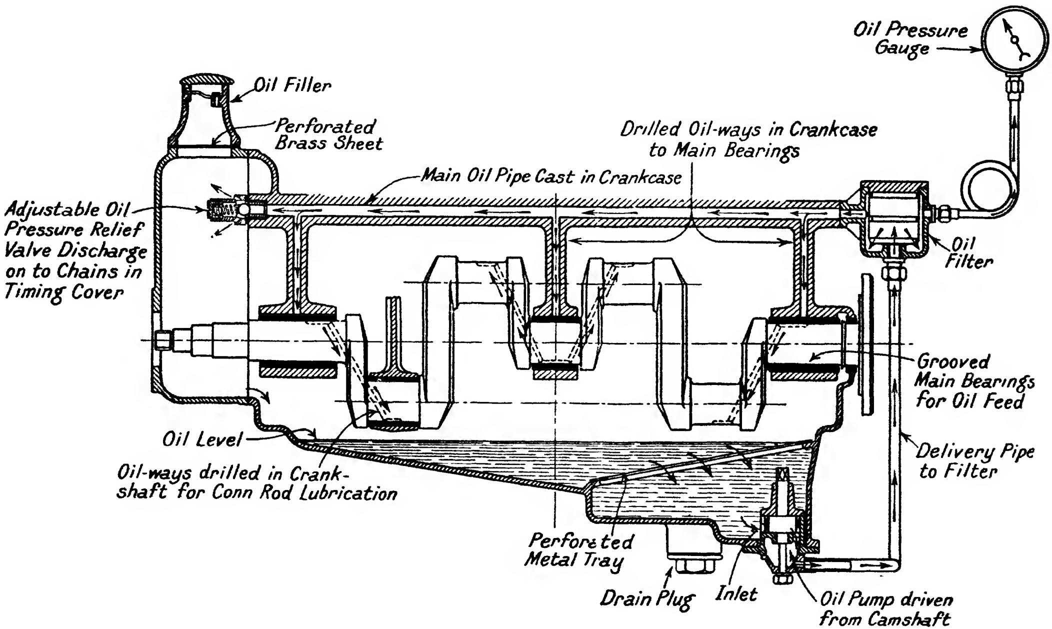


Fig. 5. Dorman Fully Forced System

valve allows the oil to pass direct to the cylinder head when the resistance of the filter becomes too great.

The oil from the filter is led upwards to the front camshaft bearing and through a special metering pin formed by the lower end of the valve gear cover stud at the front end of the engine. Part of the oil is fed over the toothed timing chain and part is fed to the camshaft trough. The automatic chain tensioner is lubricated from this part of the system. Oil flows along the camshaft trough the whole length of the engine, and the surplus runs down a vertical oil drain at the rear end into the crankcase. It is necessary to remember that the engine is well inclined so that the oil would flow rapidly along the trough were it not for the presence of a number of transverse baffles, one being provided slightly to the rear of each cam. A number of small troughs are thus formed into which each cam dips as it rotates.

A fully forced system applied to a four-cylinder side valve engine (95 × 140) made by Dorman & Co., Ltd., is shown diagrammatically in Fig. 5.

The sump is much deeper at the rear end, the gear type pump being arranged at the lowest point and driven in the usual manner by a vertical shaft. A perforated metal tray protects the suction side of the sump from foreign matter. An external pipe delivers oil under pressure to the filter which communicates on the one hand with the pressure gauge, and on the other with the main oil supply pipe, which is cast in the crankcase and supplies the three crankshaft bearings through drilled pipes leading downwards at an angle. The crank pins are supplied from the main bearings through diagonally drilled holes in the crankshaft. Pipes lead from the main oil pipe to the camshaft bearings. An adjustable relief valve of the ball type is arranged at the end of the main oil pipe, and the surplus oil is discharged

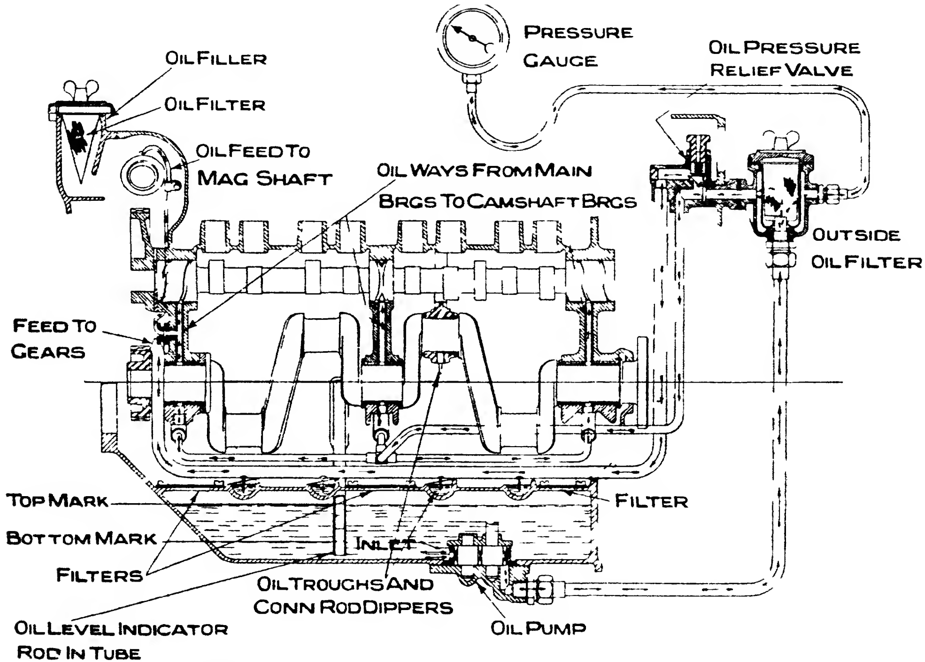


Fig. 6. Dorman Combined Trough and Pressure Lubrication System

on to the chains in the timing cover. A perforated brass sheet filters the oil supplied through the oil filler.

A carefully worked out design of combined splash and pressure feed is applied by Messrs. Dorman & Co. to a light four-cylinder (69 × 100) engine, the scheme being shown somewhat diagrammatically in Fig. 6.

The gear type pump is driven by a vertical shaft and skew gearing from the camshaft, and it is always completely submerged. An external delivery pipe conveys the oil under pressure to a large filter, placed very accessibly on the valve side of the engine. From the filter oil passes directly to the three main crankshaft bearings through a pipe inside the crankcase. The three bearings of the camshaft are also each supplied with oil from grooves passing round the divided crankshaft bearings. Shallow helical grooves on the camshaft journals work the oil along the bearings.

Adjacent to the outside oil filter is an adjustable pressure relief valve which is intended to maintain the pressure of the oil supplied to the main bearings at approximately 10 lb. per sq. in. when the engine is running. The output of the pump is such that a considerable surplus passes the relief valve and supplies each of the troughs under the big end bearings with sufficient to keep them full at all times. The end of the pipe finally discharges over the timing gears. The rearward side of each trough is higher than the forward side, so that the engine may be inclined and that the level of the oil in each trough may rise when the engine is more inclined than usual, as when climbing a hill. A slight increase in the supply at such times is always desirable.

The magneto shaft is supplied with a small quantity of oil from the connection between the front crankshaft and camshaft bearings. A conical gauze filter is arranged in the oil filler opening.

The Morris-Cowley engine, which in its main characteristics and general disposition has been little altered for a number of years, affords an example of a combined pressure and splash system which has been well tested.

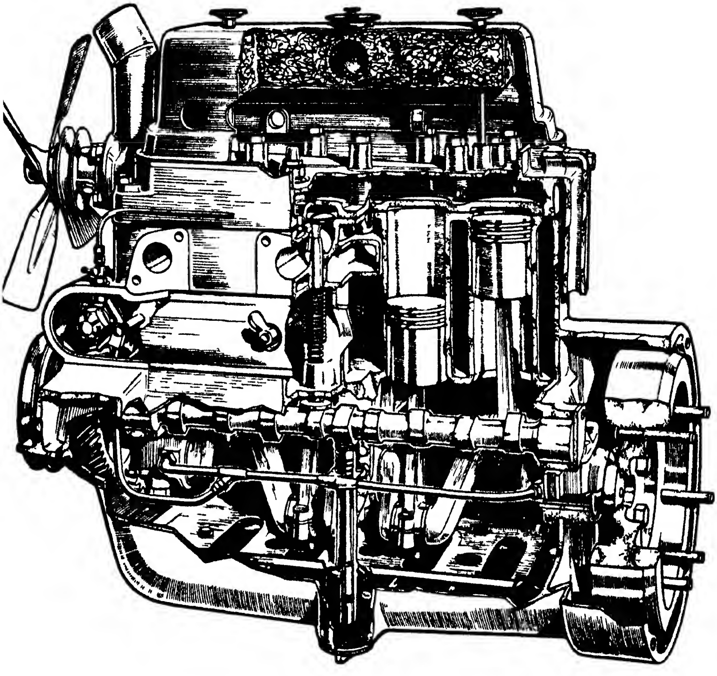


Fig. 7. Morris-Cowley Combined Trough and Pressure System

The plunger oil pump shown in the middle of the length of the engine in Fig. 7 is described in detail later in this section. The oil pump draws oil through a gauze filter, and delivers it direct to the middle crankshaft bearing, and also into pipes which lead to the two end bearings. The pressure varies considerably not only at different times in any given engine but in different engines. When warm the oil will give a reading of about 10 lb. per sq. in. at a speed of about 20 m.p.h., but the actual pressure is of little importance except as an indication that oil is actually flowing.

From the pipe leading to the forward engine bearing, a branch conveys oil to the pressure gauge and to a small jet which lubricates the timing gear. The surplus oil from the crankshaft bearings flows down to the pressed steel tray formed with troughs. The depth to which the dippers on the ends of the rods dip into the oil is determined carefully to avoid on the one hand insufficient splashing of oil, and on the other hand an excess beyond what the scraper rings are able to deal with. In a well designed connecting rod, the dipper is shaped as a slightly curved scoop up which oil runs through a hole in the cap direct on to the journal. No rules for dimensioning these various parts can be given, since the final refinements in any particular design are necessarily made as the result of careful testing and experiment. Various technical characteristics, including oil pressure, size of oil outlet to timing gears, viscosity of oil, depth and extent of oil grooves, crankshaft bearing clearances, and normal operating temperatures, are all interconnected.

In Daimler sleeve-valve engines, the question of cylinder lubrication is of even greater importance than in ordinary engines, since the sleeves as well as the pistons must be lubricated. The lubrication difficulty at starting has here been overcome by a temporary but automatic change of the system from force feed to splash.

The transverse vertical section through the engine, Fig. 8, and the underneath view of the engine, Fig. 9, with the sump removed, show the essentials of the system. The oil pump shown at about the centre of Fig. 8,

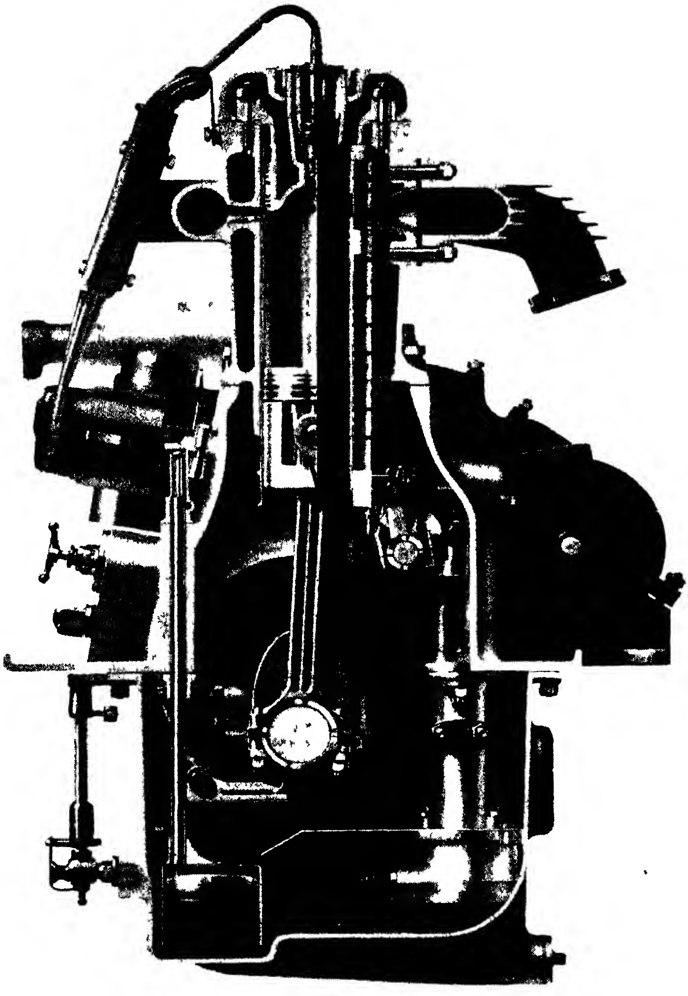


FIG. 8. DAIMLER SLEEVE-VALVE ENGINE WITH FULLY FORCED SYSTEM FOR NORMAL RUNNING AND SPLASH WHEN STARTING

surrounded by its strainer, delivers oil under pressure to the central longitudinal pipe, from which each of the seven main crankshaft bearings is supplied directly. Oil also passes through drilled channels to the big ends, the gudgeon pins and the bearings for the shaft operating the sleeve valve eccentrics. This shaft drives the vertical pump shaft through skew gearing.

When the oil pressure is high at starting, a special relief valve opens and allows oil to pass to a pipe running the length of the engine along one side and discharging into six small slotted troughs projecting about half-way across the engine, and having open ends. Dippers on the caps of the connecting rods enter the ends of the troughs and distribute oil generously over the sleeves, pistons, and gudgeon pins some time before the ordinary force system would provide any surplus from the bearings. As soon as the oil becomes warm and the pressure falls the relief valve closes and only the forced system is operative.

This method of priming establishes satisfactory lubrication even on the head rings of the sleeves, that is, the packing ring between the cylinder head and the inner sleeve, in half a minute from a cold start. The pistons and gudgeon pin are lubricated much sooner.

On the left of Fig. 8 will be seen the oil level indicator, consisting of a thin vertical rod supported by a float, and also the overflow tap.

The oil in this engine is cooled, so as to maintain it at practically a constant temperature, by circulating it through an oil cooling radiator consisting of six tubes fitted into the water cooling radiator. An auxiliary pump forces the whole of the oil through the oil cooling radiator in about a minute.

The lubrication system of the Buick eight-cylinder engine is of the fully forced type. A secondary pipe from the pump is connected to the pressure gauge and

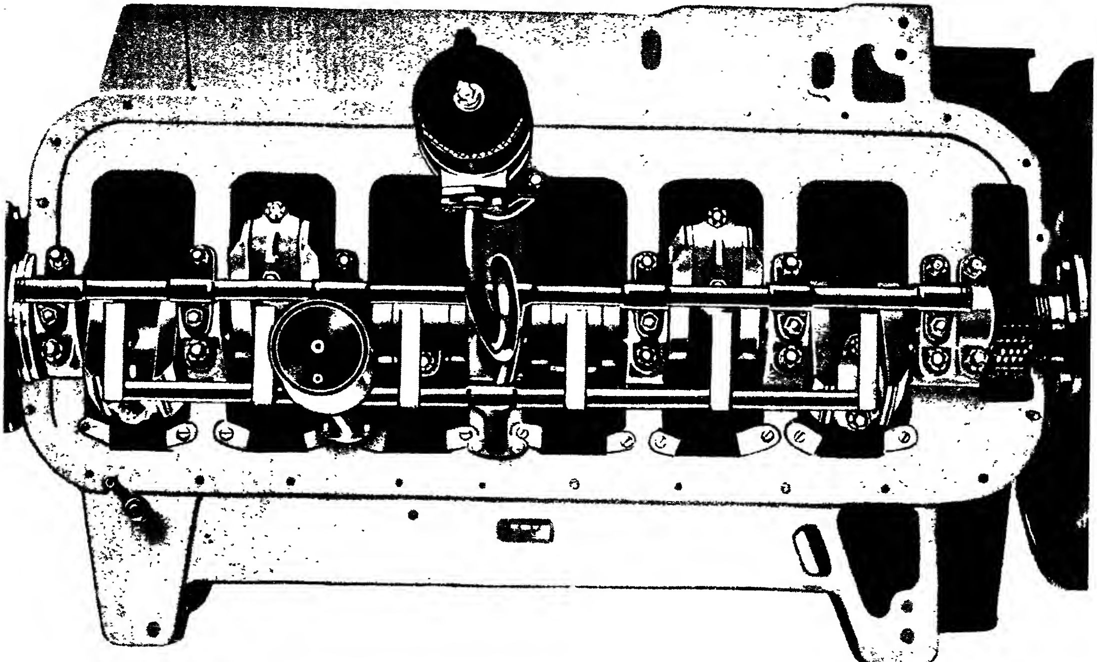


Fig. 9. Underneath View with Sump Removed of Daimler Sleeve-valve Engine

also delivers oil to a fine oil filter and thence to the hollow shaft for the valve rocker arms, so that only a part of the total oil in circulation is treated at a time by the filter. The surplus from the rocker arm shaft lubricates the timing gear and generator bearings.

When the filter is clean it will pass the whole contents of the sump in about five minutes at a car speed of about 25 m.p.h., but this rate of flow gradually decreases as the filter clogs.

The cylinder, pistons, and gudgeon pins are lubricated by oil forced through a small hole in the shoulder of each of the big ends, and registering once each revolution with the oil supply hole in the crankshaft. This is particularly valuable when starting from cold as mentioned below.

Of particular interest in the Buick engine from the lubricating point of view is the oil temperature regulator and the crankcase ventilator. The former, which will be described first, is arranged on the side of the engine, the detailed construction being shown in Fig. 10. A honeycomb arrangement of tubes is arranged in casings in such a way that all the cooled water from the outlet at the bottom of the radiator is pumped through the tubes, while oil direct from the oil pump is led round the outside of the tubes from which it passes to the main pressure supply pipe to the bearings.

When the engine is started from cold, the resistance to flow through the regulator due to the high viscosity of the cold oil results in such a pressure that a by-pass valve in the oil pump opens and allows oil to pass direct to the bearings without first passing through the regulator. To assist in the rapid heating of the oil at this time, the ordinary pressure relief valve discharges the excess oil back to the pump instead of into the sump.

When the temperature of the oil has been raised and its viscosity is sufficiently reduced, the by-pass valve closes, and all the oil passes through the regulator by means of which it is still further heated, since the cooling water, owing to the action of the radiator-shutters,

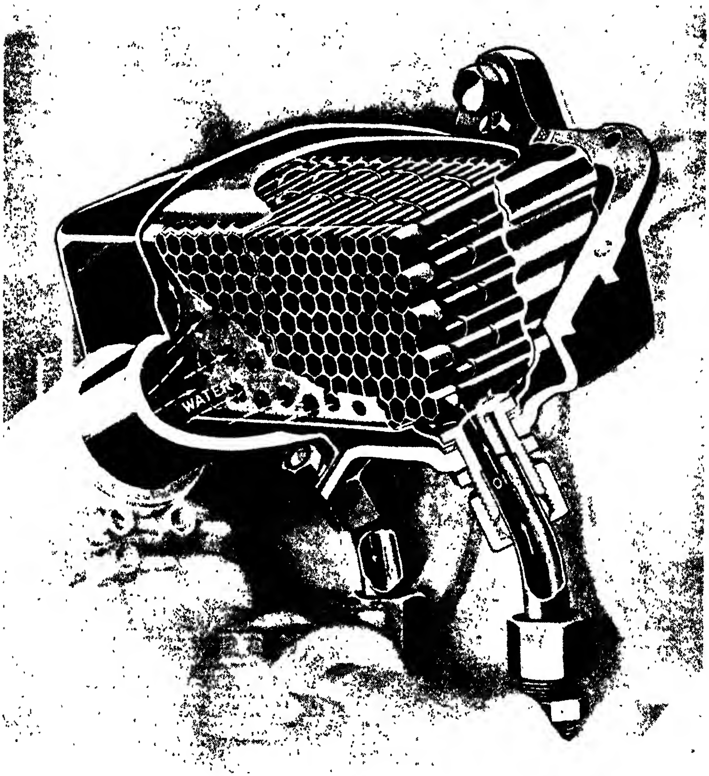


FIG. 10. BUICK OIL TEMPERATURE REGULATOR

which are controlled by a thermostat, warms up more quickly than the oil.

When the car is driven at high speed for any considerable time, the temperature regulator functions as an oil cooler, and checks that steady rise in temperature which may ultimately lead to failure of a big end bearing.

The primary purpose of the crankcase ventilator is the removal of petrol and water vapour, the latter being

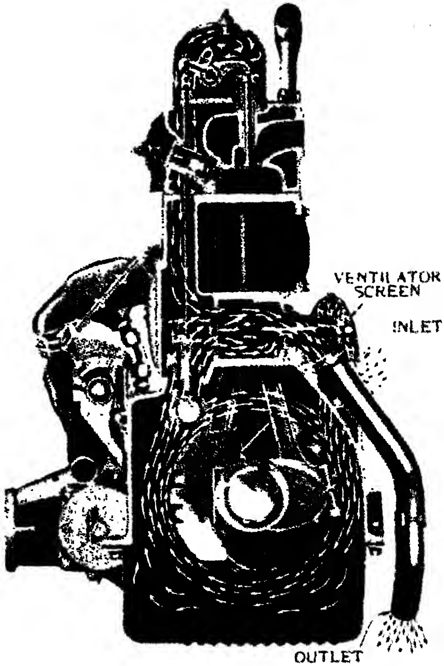


FIG. 11. BUICK CRANK CASE VENTILATOR

part of the products of combustion ; but the continuous circulation of air will also assist cooling of the oil and the big ends. The torsional oscillation damper at the front end of the engine induces movement of air upwards from the crankcase into the valve lifter compartment, as shown in Fig. 11. This compartment extends the whole length of the engine, and air passes from it between the fourth and fifth cylinders to the outlet which discharges below the undershield. Air enters the crankcase through a screened inlet at the rear of the engine. The valve rocker arm cover is also ventilated by air which enters at the rear of the cover and passes downwards into the valve lifter compartment.

Dry sump systems used in a limited number of commercial motor vehicles constitute a modification ot the

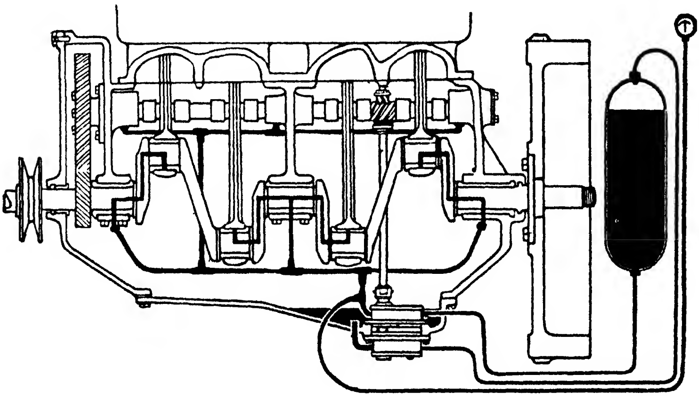


FIG. 12. DRY SUMP LUBRICATION
Diagram of system by Messrs C. C. Wakefield & Co., Ltd.

fully forced system. They differ, however, in that the circulating oil is not retained in the sump, but is stored in an entirely separate reservoir. A dry sump system is shown diagrammatically in Fig. 12. The pump is in two parts or two separate pumps are provided; in the case shown the two parts of the pump are driven in the usual way from the camshaft through skew gearing and a vertical shaft, the upper part of the pump drawing oil from the reservoir and supplying it under pressure to the crankshaft, camshaft, and big end bearings. The surplus oil which drains down to the bottom of the sump is removed by the lower part of the pump and returned to the reservoir. In such systems it is possible to keep in circulation a much larger body of oil than usual, and this oil keeps cooler and remains cleaner for a much longer period. This is a particular advantage in commercial vehicles in view of their heavy mileage.

Oil Pumps. The lubricating system in all engines depends upon the unfailing reliability of the oil pump. At one time the pump was usually arranged well above the level of the oil and its design was such that when the engine stopped the hot oil would drain out. On re-starting it was often necessary to prime the pump with oil before it would draw the oil from the sump up the suction pipe, and this difficulty increased greatly with wear. In modern engines the pump is often "drowned" or submerged in the oil, and when it is arranged at a higher level the design is such that it is rendered self-priming by a residue of oil.

Pumps as used on motor vehicles fall into three main types—

1. Gear pumps.
2. Plunger pumps.
3. Rotating vane, now rarely seen.

The gear type of pump is most widely used, and is very often arranged at the bottom of the sump, although it is carried quite independently by the upper part of the crankcase. A vertical shaft driven by skew gearing from the camshaft then drives one of the gears, the other rotating idly. When this type of pump is arranged with its axis horizontal and above the oil level, oil is retained, when the engine stops, in a well in which the lower gear works, or in a recess communicating with this well, so that the oil will prime the pump and thus fill up the gaps between the working parts.

A detailed description will now be given of a gear type pump as arranged with its axis horizontal on the side of the crankcase of a Standard engine. In Fig. 13 is shown a pictorial sectional view through the body of the pump, which is bolted to a seating on the near side of the engine. Fig. 14 shows a group of the components. Figs. 15 and 16 show essential components with certain

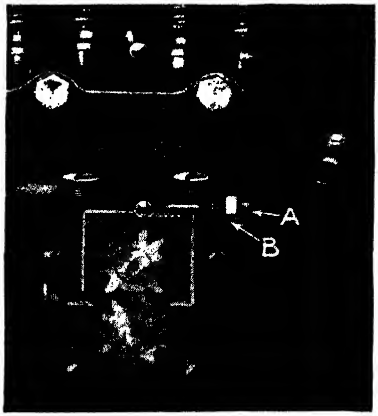


FIG. 13. STANDARD GEAR-WHEEL OIL PUMP

technical details, and Fig. 17 shows a diagrammatic view of the system.

The pump body, Fig. 15, is formed with a long bearing for the driving spindle projecting into the crankcase. The inner end of the spindle, Fig. 16, carries a skew gear which meshes with a complementary gear wheel integral with the camshaft. The suction hole in the pump body communicates with a vertical conduit in a projection cast on the inner wall of the crankcasing which forms part of the cylinder casting. This projection with its conduit extends down as far as the flange to which the sump is bolted, and a stand pipe or suction tube, Figs. 14 and 17, then extends the suction conduit down to

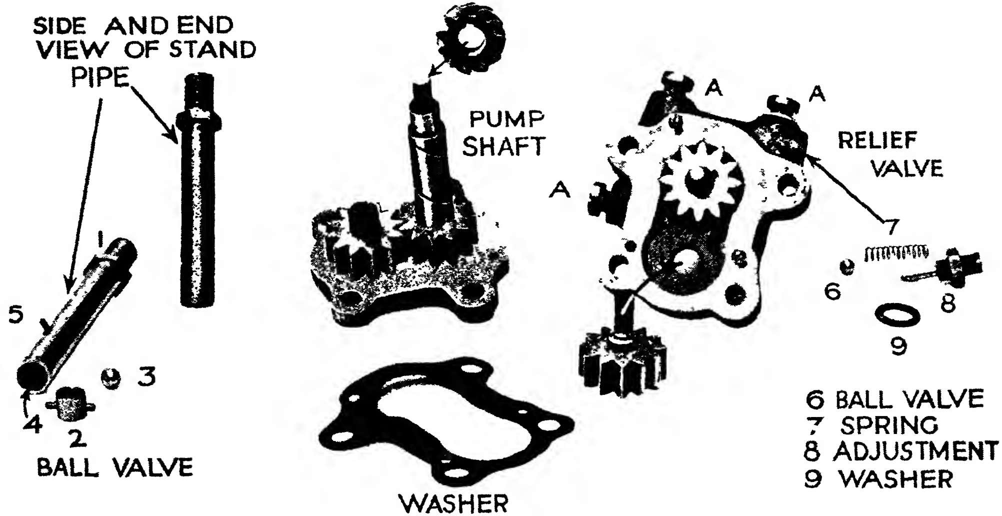


Fig 14 Standard Oil Pump Components

the bottom of the sump. The stand pipe is surrounded by a vertical filter, and is fitted at its lower end with a one-way ball valve which helps to retain oil in the tube when the engine is not working. The delivery hole conducts oil from the pressure side of the pump

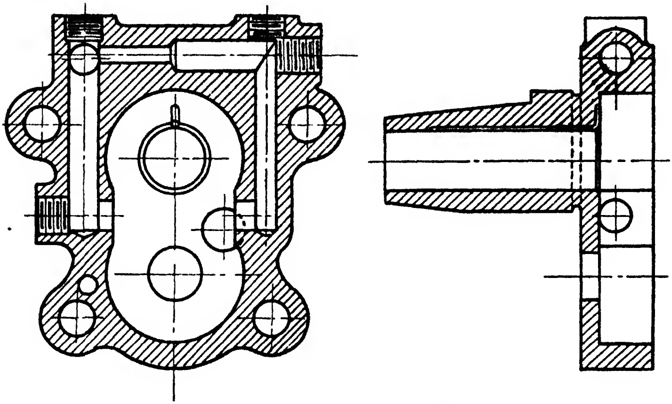


FIG. 15. STANDARD OIL PUMP BODY

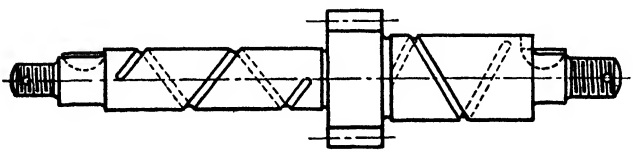


FIG. 16. STANDARD OIL PUMP DRIVING SPINDLE

into the interior of the crankcase, and thence to the bearings.

The outer side of the pump body is closed by a flat plate which is provided with a further bearing for the driving shaft, and a support for the spindle of the idler pump gear. This flat plate may form part of an ignition distributor or of the casing for the driving gear of a petrol pump.

In order that the pump may be effective as such and work with the minimum of liquid slip, it must be finished with considerable accuracy. The long bearing for the pump spindle on the inside of the crankcase is

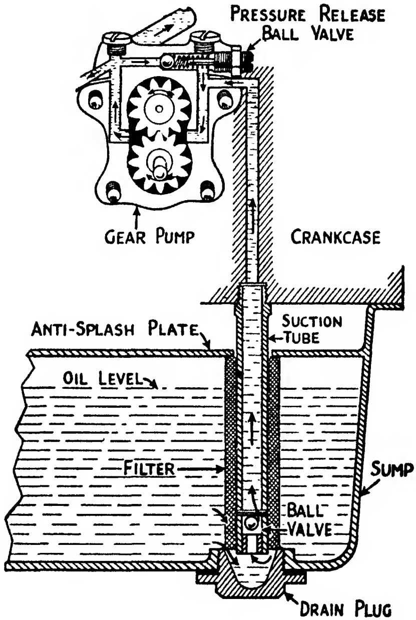


FIG. 17. DIAGRAMMATIC VIEW OF STANDARD OIL PUMP WITH FILTER

of $\frac{5}{8}$ in. diameter, the tolerances on the bearing being $\begin{smallmatrix}+ \cdot 0005 \\ - \cdot 0002\end{smallmatrix}$ while those on the spindle journal are $\begin{smallmatrix}- \cdot 0009 \\ - \cdot 0012\end{smallmatrix}$. The maximum possible diametrical clearance is thus seen to be $\cdot 0005 + \cdot 0012 = \cdot 0017$ in., while the minimum clearance is $\cdot 0007 - \cdot 0002 = \cdot 0005$ in.

The diameter of the recesses in the body in which the gears work is $1{\cdot}4 \begin{smallmatrix}+ \cdot 001 \\ - \cdot 0005\end{smallmatrix}$ in., while the diameter of the blanks from which the gears are cut is $1{\cdot}4\,{}^{-\cdot005}_{-\cdot006}$ in. From these figures it will be seen that the maximum diametrical clearance between the tips of the teeth and the casing is $\cdot001 + \cdot006 = \cdot007$ in., and the minimum clearance is $\cdot005 - \cdot0005 = \cdot0045$ in. If the spindle of the rotor and the exterior of the gear are concentric and the corresponding parts of the casing are also concentric, these clearances will be halved.

The next consideration is the fit of the gear sideways. The depth of the recess in the body of the pump is $\frac{9}{16}\,{}^{+\cdot001}_{-\cdot000}$ in., and the thickness of the gears is $\frac{9}{16}\,{}^{-\cdot0005}_{-\cdot0015}$ in. The sum of the clearances on the two sides may thus vary between the maximum limit $\cdot001 + \cdot0005 = \cdot0015$ and the minimum limit $\cdot0005$.

Finally, the backlash between the interengaging teeth is from ·003 to ·004 in.

These clearances at the three main points at which leakage may occur in conjunction with a relatively viscous liquid like oil allow of the attainment of a substantial pressure with a slip of about 15 per cent. That is to say, the actual output of the pump under normal working conditions is about 85 per cent of its calculated capacity.

The intermeshing teeth of the pump act as a barrier to the return of the oil from the delivery to the suction side, the oil being carried round in the spaces between the teeth and the casing, both wheels working simultaneously in this manner. In practice the pressure rises during a run until for a given speed and oil temperature a point is attained when it is kept constant by the release valve.

If the pump were working freely or discharging into a tank without any resistance, the volume of oil delivered would be equal to the volume of the spaces between the gears (for both wheels) multiplied by the speed of the pump, assuming no slip or leakage.

For the formula given below the writer is indebted to the Standard Motor Co.

Since the widths of the teeth and the gaps are the same on the pitch circle, the area of the space between the teeth is often taken to be half the area of the annulus formed by the outside diameter and the inside or root diameter of the gears, but it is, in fact, 10 per cent greater, since, owing to the curved shape of the teeth, more metal is removed to form the spaces than is left to form the teeth. Further, the total depth of the teeth should be taken as $\frac{2 \cdot 157}{P}$ instead of the more usual approximation $\frac{2}{P}$.

The volume of the tooth space thus becomes—

$$1 \cdot 1 f \times \pi \times D \times \frac{2 \cdot 157}{P} \text{ cub. in.}$$

where D = pitch diameter in inches.

f = face width of teeth in inches.

D = revolutions per minute.

P = diametral pitch of teeth.

For pumps having two gears of equal size the above formula becomes

$$1 \cdot 1 f \pi D \times \frac{2 \cdot 157}{P} \times \frac{N}{277 \cdot 3} \text{ gallons per minute.}$$

This reduces to the simple form

$$\frac{DfN}{37 \cdot 3P} \text{ gallons per minute.}$$

For the Standard pump mentioned, the free running delivery rate

$$= \frac{1{\cdot}2 \times {\cdot}563 \times 1000}{37{\cdot}3 \times 10}$$

$= 1{\cdot}81$ gallons per minute at a speed of 1,000 r.p.m. of the pump or 1,840 r.p.m. of the engine.

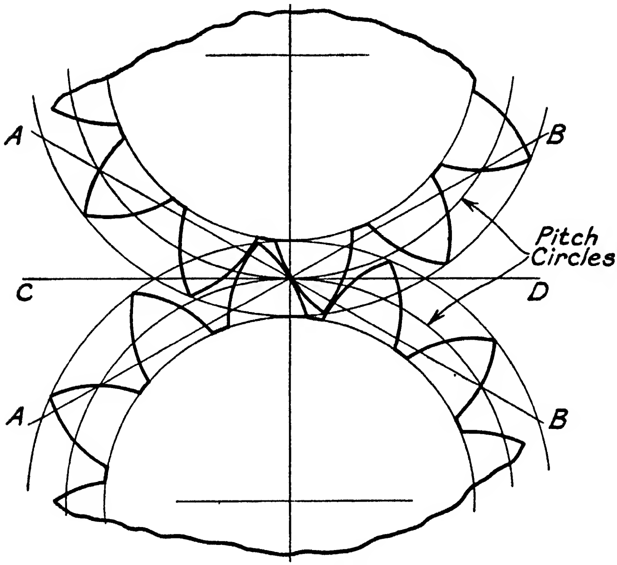


FIG. 18. STANDARD OIL PUMP TOOTH CONTOUR

In Fig. 18 is shown the pitch, tip and root circles for the pair of equal gear wheels used in the Standard pump. The diameter of each pitch circle is 1·2 in., and the pressure angle is 30°; that is, the point of contact of any co-acting pair of teeth, consisting of a driving tooth on one wheel and a driven tooth on the other, always lies on one of the lines AB which make an angle of 30° with the common tangent CD to the pitch circles. The tooth surfaces at any point of contact are at right angles to the line AB. There are 12 teeth, and as the pitch diameter is 1·2 in. the diametral pitch is

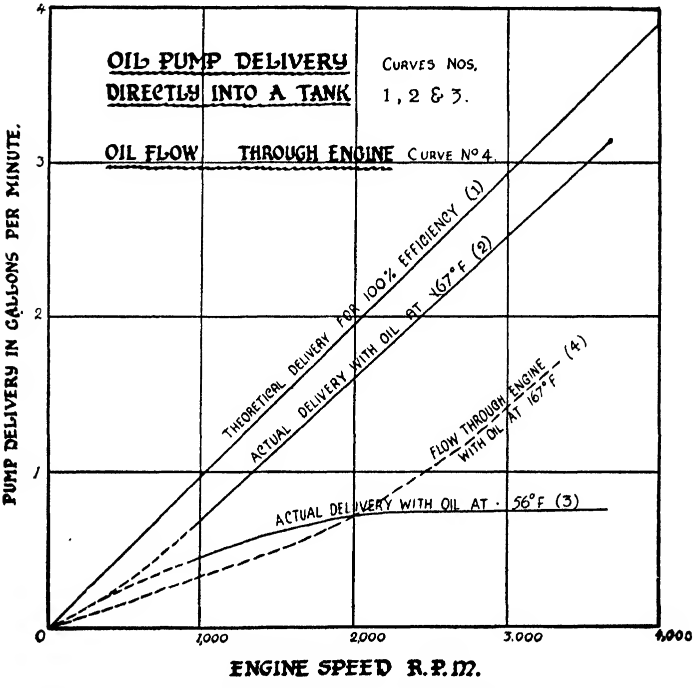


FIG. 19. OIL PUMP PERFORMANCE UNDER VARIOUS CONDITIONS

$\frac{12}{1{\cdot}2} = 10$. The base circles, not shown, for involute teeth touch the lines AB.

The performance of this pump under various conditions is shown in Fig. 19. Curve (1) shows the output at various engine speeds on the assumption that the pump is discharging freely into a tank without any slip or leakage. The actual delivery under identical conditions with the oil at a temperature of 167° F. is shown by curve (2). While curve No. (1) is a straight line through the origin of the co-ordinates, curve (2) is only straight above 1,000 r.p.m. engine speed, and is not in line with the origin. The efficiency of the pump under

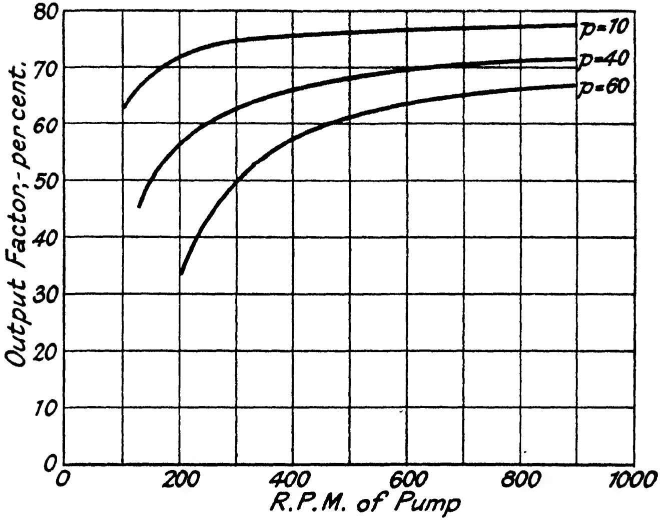


Fig. 20. Oil Pump Output

free delivery rises from about 70 per cent at 1,000 r.p.m. to about 86 per cent at 3,000 r.p.m.

Two other curves, (3) and (4), show the actual delivery to the bearings with cold oil at 56° F., and with hot oil at 147° F. In each case the pump is delivering against a pressure which can be regulated by the release valve, the normal working pressure being between 25 and 30 lb. per sq. in. Comparison of curves (2) and (4) shows that the capacity of the pump is about double the actual flow through the engine with hot oil at normal speeds, thus giving an ample margin, since approximately half the oil pumped is by-passed back to the pump.

Particulars of the output of another design of gear wheel pump are shown on Fig. 20. The oil was kept at a constant temperature of 140° F., and the output factor is shown for various speeds at three different pressures. The output factor is expressed as a percentage of the calculated discharge rate, assuming no leakage.

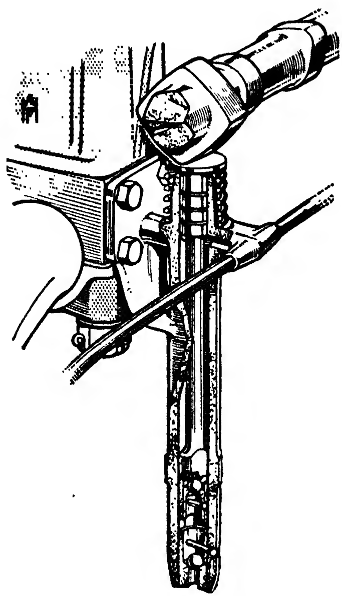


FIG. 21. MORRIS-COWLEY PLUNGER OIL PUMP

The Morris-Cowley plunger pump shown in Fig. 21 has been standardized on this car for a number of years, and is very powerful, efficient and reliable. The pump dips right down into the oil in the sump so that the inlet is always submerged, and the oil, when the engine is not running, rises up in the pump cylinder and covers the valves, so that the pump is always primed ready for starting. The plunger is reciprocated twice during each revolution of the crankshaft by an approximately square cam at the middle of the camshaft, this cam serving also as a camshaft stabilizer to prevent play due to backlash in the toothed timing gear. By locating the square cam suitably, relatively to the valve cams, it is possible to neutralize the tendency of the tappets to speed up the camshaft during their downward movement after passing over the peak of the cam. The square cam should then be compressing the spring on the pump so that the rotation of the camshaft is retarded.

Both of the enlarged ends of the pump plunger are a very close fit in the pump body. Leakage of the oil is rendered negligible by the retarding effect of a series of sharp ended grooves turned in the plunger. With small diameter pumps using oil such grooves are cheaper and simpler than piston rings, and are at least as effective while there is much less wear.

The lower suction ball valve opens upwards from a seat secured in the lower end of the pump cylinder, while the upper valve also opens upwards, but from a seat secured in the end of the plunger. The lift of the valves is limited in the case of the suction valve by a small cross pin, and in the case of the upper valve by securing the seating bush in a suitable position in the end of the plunger.

Too big an allowance for lift tends to slow up the rate of closure of the valve. This may occur as the result of wear of the cross pin which must then be renewed. In view of the high speed of reciprocation of the pump, the balls strike their seats and limit stops with considerable force, notwithstanding the retarding effect of the oil, and this accounts for wear of the pins.

With this arrangement of valves, the positive movement of the plunger on the down stroke forces the oil into the delivery pipe, the lower valve thus being closed while the upper valve is open, so that oil is in effect displaced from below to above the lower end of the plunger. The relatively weaker resilient up stroke forms the suction stroke. The pump discharges oil direct to the centre crankshaft bearing, to the side of which it is bolted, and also into copper pipes leading to the two end bearings.

In the one-way or non-return valves employed in most lubrication systems, hard steel balls, as made spherical within very close limits by ball bearing manufacturers, are used in conjunction with seats of relatively soft metal such as bronze. A good fit is more readily obtained with a narrow than with a wide seat.

The usual simple calculation for lift valves will show that L, the lift of the valve, need be only one-fourth of the diameter D of the opening in the seat to give a value for the circumferential area of opening equal to the area of the opening in the seat. This is represented by the equation

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

$$\text{from which } L = \frac{D}{4}$$

It is found, however, in practice, that the maximum rate of flow is obtained when the lift is rather more than this, for instance one-third to one-half of D. Too great an allowance for lift in the case of freely moving valves, such as those used in reciprocating oil pumps, slows up the rate of closing. The spring-controlled release valves seldom open to their fullest capacity. Details of the construction of a ball valve and seat are shown in Fig. 22.

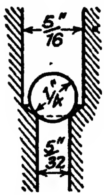


FIG. 22
BALL VALVE DETAILS

Oil Pressures and Starting Conditions. The oil pressure indicated on the gauge is primarily of importance as an indication that oil is passing to the bearings. The actual pressure, even in those engines which exhibit high pressure, for example 80 lb. per sq. in., is very small in comparison with the film pressure in the bearing, which may be of the order of 1,000 lb. per sq. in. It is, however, of importance that the usual pressure, whatever it may be, should be maintained to ensure that oil is being fed continuously to the low pressure region in each bearing.

When starting from cold, the viscous oil can only be circulated with difficulty, and the pressure rapidly builds up on the pressure side of the pump until it may show as much as 120 lb. per sq. in. on the gauge. The oil in the sump is too viscous to flow freely to the pump which is thus only able to discharge a frothy mixture resulting in a reduction of pressure. As soon as the oil in the sump warms up slightly and becomes thin enough to supply the pump more freely, the pressure rises again to, for example, 50 lb. per sq. in., but drops gradually to the normal running pressure of, say, 25 lb. per sq. in. at a speed of 25 m.p.h. This may, however, call for 20 minutes or even more of steady running. Oil pressure under various conditions is dealt with in connection with the description of the Standard gear type pump above.

The normal running pressure in different engines varies greatly. It may be as low as 10 and as high as 80 lb. per sq. in. Moreover, in some engines it may vary with the speed, and in others remains relatively constant. These differences are regulated by the relief valve, but depend primarily upon the oil system employed and upon details of design such as bearing clearances and dimensions of oilways.

It will be clear from the foregoing that the dangerous period occurs when the oil is cold, since the engine is then running practically without lubricant. This is a matter which is not generally appreciated by vehicle users, many of whom race their engines at starting instead of allowing them to warm up gradually. Improvements in the lubrication system, rather than an attempt to educate the public, form the best solution of this difficulty.

The cylinder walls and pistons are adversely affected more than the bearings, since the former are drained practically dry when the engine is stopped owing to their high temperature and to the retention of heat by the cooling water. Further, they receive no substantial amount of oil until it has warmed up sufficiently to pass to the bearings freely and provide a surplus.

This question is much more serious with forced than with splash systems. With the latter there is always a small residue of oil in the troughs, but with the former, the amount of oil thrown off is generally less than is distributed by dippers, and as mentioned above there is some delay when starting before any is thrown off at all.

The damage is due to rust formation and to friction of the rings on bone dry cylinders. When starting from cold, water and often sulphurous acid are produced by the combustion. Some of the complaints of excessive wear with aluminium pistons are due more to rusty cylinders than to any defect of the piston other than its greater sensitiveness to this trouble than cast iron.

The rich mixture used when starting which results in the deposition of petrol on the cylinder walls is not directly a cause of trouble. Sufficient liquid petrol reduces friction to some extent, but any petrol is dried up quickly by the heat of the first few strokes, long before the oil can be forced round, leaving the cylinder absolutely dry.

The problem then is to provide some oil spraying device which is operative immediately on starting, to counteract the delay in force feed systems before surplus oil is flung off from the big end bearings.

Some improvement in this direction may be effected by a small bleed hole in the shoulder of the big end of the connecting rod, which registers once every revolution with an oil supply hole in the crank pin. The bleed hole must not be connected with an oil groove in the bearing or the discharge of oil will be excessive. The best size of hole must be determined by experiment with any particular engine, but a hole of between $\frac{1}{32}$ and $\frac{3}{64}$ in. diameter is generally satisfactory. Oil is discharged from this hole at all times, and while it is sufficient to bring about an improvement when starting, it need not increase the crankcase oil mist beyond what modern piston scraper rings can deal with.

The Daimler system described above definitely provides a copious splash system at starting only, the system being fully forced at other times. The change over is entirely automatic, and depends upon the viscosity of the oil.

Oil Temperature Control. Thermodynamically, the most unsatisfactory feature of the internal combustion engine is the quantity of heat which must be dissipated into the air to ensure proper mechanical operation. The oil has to do its part in this connection. A considerable quantity of heat is generated by friction in the bearings, but in addition to this heat is flowing continuously by conduction from the crown of the piston towards the rings and skirt, and oil assists to carry away this heat at a sufficient rate.

When an engine is started up from cold the oil steadily heats up to a temperature which is such that the rate of radiation of heat equals the rate of cooling. A steady temperature may not be attained until the engine has been running 15 or 20 minutes, and the temperature will also tend to vary as the power developed varies.

The oil, which is continually splashed on to the cylinder walls and scraped off again, is thereby raised in temperature, since the skin temperature of the walls due to its intermittent exposure to the flaming gas is high. It is true that the temperature gradient, through the cylinder wall from the inside to the water jacket, is necessarily steep, but the temperature of the skin remains high enough to heat the oil, although not high enough to interfere with mechanical working.

Experiments made to ascertain the temperature of the metal on the cylinder walls and the temperature of the oil in contact therewith indicate that the oil in compression-ignition engines is about 30° F. higher than the water on the opposite side of the wall. The difference is somewhat less in petrol engines. This gives an oil temperature of not more than 250° F., which is well below the flash point but high enough to necessitate cooling before it will have sufficient body to form a properly resistant film in the bearings. The cylinder walls are only exposed intermittently, and the quantity of oil which is heated by them is small.

The temperature of the oil in the sump should not be allowed to exceed 140° F. if it is to retain its lubricating properties and if rapid oxidization is to be prevented. Temperatures as high as 180° F. do, however, occur in some engines during a prolonged run. Better results would be obtained in such engines if the temperature were kept down by, for instance, larger sump capacity, alteration of shape of sump to give larger cooling area or greater exposure to the current of air when running, or by the provision of a separate oil cooler or by crankcase ventilation. Special cooling means are used on a few engines.

Engine sumps are of varying shapes, and the few examples shown elsewhere in the section make that clear, the shape and area exposed to the atmosphere having considerable effect upon the cooling properties. Pressed steel and aluminium sumps are both used, the latter having in some cases external cooling fins cast integral therewith.

The quantity of oil carried in the sump varies considerably with different engines. As a rough guide the capacity in pints is generally between one-half and three-quarters of the treasury rating (R.A.C. formula $\frac{N.D^2}{2{\cdot}5}$ where N = number of cylinders and D = diameter in inches). The quantity of oil by itself is not of much significance. It is, however, one factor which affects the important matter of temperature. A sump of large capacity in relation to engine output will lower the temperature and require emptying and replenishing at longer intervals.

The lubricating oil receives heat from three main sources—

1. Bearing friction.
2. Piston friction.
3. Heat of combustion conducted through the pistons and cylinder walls.

Exact figures cannot be given, but some idea of the order of magnitude of the bearing frictional losses may be obtained in connection with the A.E.C. engine referred to previously. The seven main bearings and six big end bearings generate over 1,100 B.Th.U. per minute when the engine is running at 2,500 r.p.m.

The friction between the six pistons and their rings and the walls of the cylinders probably accounts for the generation of heat at a greater rate than the bearings.

There are further comparatively small frictional losses (all resulting in the heating of the oil) in connection with the driving of the oil pump, the friction of the cams and tappets (note that the energy expended in compressing the valve springs is recovered when they expand), the camshaft and rocker arm bearings, and the timing gear.

One result of the tendency in both private and commercial vehicle engines towards higher and more sustained speeds is increased attention to oil cooling. Frictional heat increases with speed, and the oil becomes hotter and thinner so that it is less able to resist the inertia forces which simultaneously increase as the square of the speed. Oil cooling reduces the risk of bearing failure. If the cooling is regulated in accordance with the speed, the oil can be kept under all ordinary running conditions at a sufficiently high temperature to give a low coefficient of friction without risk of a sudden or a sustained turn of speed raising the temperature to an unsafe figure. A lowered oil temperature allows a lighter oil to be used, and the rate of deterioration by oxidization and carbonization is reduced. The friction at starting is reduced and the oil is then distributed to the working surfaces more rapidly, while the necessity of using oils of different viscosity in summer and winter should be obviated.

A research carried out with the object of reducing big end bearing temperatures is described in *Engineering* (April 15th and 29th, 1932), the article being based upon a number of reports issued to members of the Research Association of British Motor and Allied Manufacturers.

In this research the obvious methods of reducing bearing temperatures by cooling the oil and increasing its rate of circulation were first investigated, and led to further work, which showed that the circulation of air through the crank case could be more effective.

If the assumption could be made that all the heat generated by friction in a bearing could be carried away by the oil, the problem of bearing temperatures would be a simple matter of calculation. But the experiments show clearly that this assumption is unjustified, and that a considerable proportion of the heat is imparted directly to the air. When the speed of rotation of the big ends is considered this would seem a reasonable expectation. For example, in an engine having a stroke of 4½ in. and running at 3,000 r.p.m. the linear speed of the crank pin is $\frac{4\frac{1}{2}}{12} \times \pi \times 3{,}000 = 3{,}536$ ft. per minute, or 40 m.p.h.

The experiments were conducted on a special machine which was essentially a dummy engine adapted to be

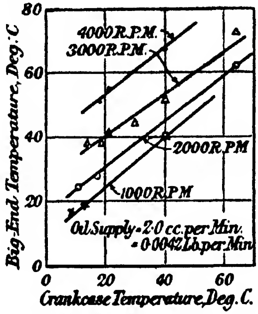


FIG. 23

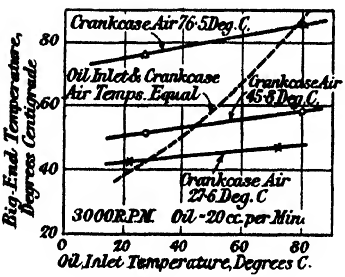


FIG. 24

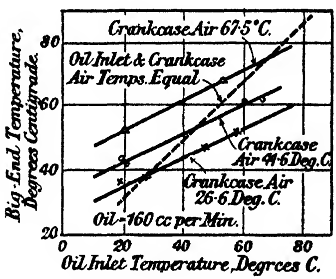


FIG. 25

driven at various speeds. Provision was made for varying and measuring the rate of flow of oil and its temperature, and for measuring the temperature of the big end bearings.

Fig. 23 shows the relation between crankcase temperature and big end temperature at different speeds when the rate of flow of oil is so low that its cooling effect is negligible. The big end temperature is clearly a linear function of the air temperature of such a character that a big end increase of 10° C. corresponds to an air temperature increase of about $7\frac{1}{2}$° C.

Figs. 24 and 25 show the influence on the big end temperature of variations in the rate of flow of oil and the oil inlet temperature. In each case the speed was kept at 3,000 r.p.m., but in Fig. 24 the oil was supplied at 20 c.c. per minute, and in Fig. 25 at 160 c.c. per minute. The small slope of the curves in Fig. 24 show how small was the effect of oil temperature on big end

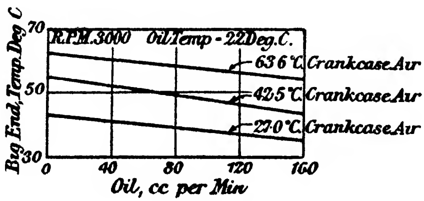


FIG. 26

temperature. A variation of 60° C. in oil inlet temperature corresponds to only a change of about 10° C. in bearing temperature. In Fig. 24 the rate of circulation of oil is eight times as great and the curves are somewhat steeper. In this case a variation of 60° C. in oil inlet temperature affects the big end temperatures by about 28° C. In both Figs. 24 and 25 it is obvious that changes in the air temperature of the crankcase have far more effect on the temperature of the big ends than variation in the oil temperature. For example, comparing the upper and lower curves in Fig. 24, a change of air temperature of $76{\cdot}5 - 27{\cdot}6 = 48{\cdot}9°$ C. affects the big end temperature by about 35° C. A greater effect on big end temperature would, of course, be obtained by varying both air temperatures and oil temperature, the result, if both were maintained at the same value, being shown by the dotted curves which have been drawn by cross plotting from the other curves.

Fig. 26 shows how very small is the influence of the rate of flow on the big end temperature, this having been referred to when comparing Figs. 24 and 25.

Fig. 27 shows how the speed affects the big end temperatures at different rates of flow, these curves having been derived from Fig. 26. When the flow is constant the temperature rises in a linear relation to the speed, but when the oil flow is made proportional to the feed

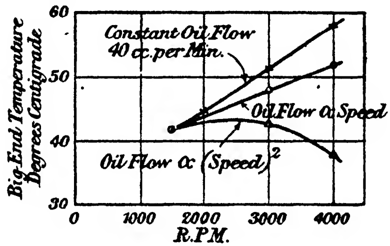


FIG. 27

the rise is less rapid. If the oil flow increases as the square of the speed a point is reached when the temperature falls as the speed increases.

Consequent on the above results a further series of experiments was carried out to ascertain the advantages and defects of the method of oil cooling by circulating air through the crankcase, a six-cylinder engine of 2·6 litres capacity being employed, and Castrol XL oil being used. It was not found practicable to measure big end temperatures, but the temperature of the oil in the sump and the air entering and leaving and in the crankcase were recorded. Two methods were employed to circulate the air. In one the crankcase outlet was connected to a blower, and in the other to the carburettor intake. Special means utilizing shields and centrifugal action were provided to avoid escape of oil particles from the crankcase.

Fig. 28 shows the influence upon oil sump temperature of variations in the quantity of air drawn through the crankcase by the blower, the speed of the engine being 1,500 r.p.m. full load. Under these conditions the engine consumed 196 lb. of air per hour. By reference to the curve it will be seen that if the air for combustion had been drawn through the crankcase, the oil temperature in the sump would have been reduced

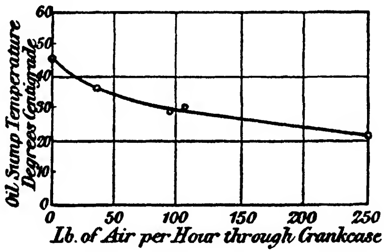


Fig. 28

from 45° C. to 25° C.—a reduction of 20° C. The temperature of the air in the crankcase was reduced simultaneously by 16° C. The combined effect on the temperature of the big ends of the lowering of both the oil and air temperatures can only be conjectured, but reference to Figs. 24 and 25 would suggest that a reduction in the neighbourhood of 18° C. might be anticipated.

The engine was then run at 1,500 r.p.m. and at 60 per cent full load for a number of hours both with and without air circulating through the crankcase; the effect upon oil consumption, petrol dilution, and acid formation was compared in the two cases.

It will be seen from Fig. 29 that the oil consumption was increased by about 20 per cent. This would probably have been less if the big ends had been fed by pressure instead of splash. This increase of oil consumption is not, however, all loss. Upper cylinder and valve stem lubrication are improved and the oil is burnt, thus effecting a saving in petrol consumption. It was also found that when the engine was run at the same speed

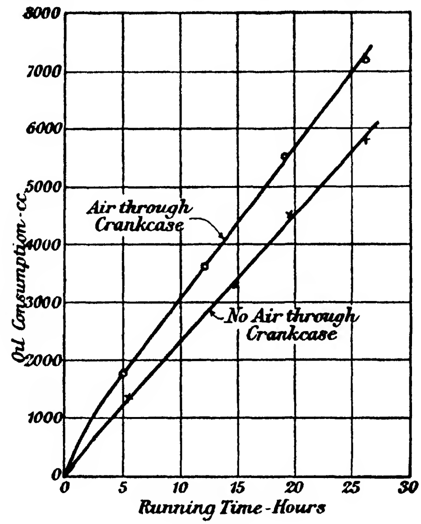


FIG. 29

but at full load, the maximum power developed was reduced by 1·9 per cent, and the fuel consumption was reduced by about 17 per cent. The slight reduction of maximum power is probably due to water vapour which reduces the proportion of oxygen in the air. The general effect of increased humidity is to make the engine run more smoothly, although with slightly less power. It is this effect which accounts for the common impression that an engine runs better after dark. The improved petrol consumption is probably due to the oil acting as fuel, but an improvement in this respect, comparable with that obtained during such a special test, can hardly be expected in practice. Crankcase ventilation has long been recognized as a method of reducing dilution of the oil by petrol. The Buick engine described above employs this method in conjunction with means for heating or cooling the oil according to requirements.

Fig. 30 shows how dilution progresses. Starting with clean oil, the percentage of petrol absorbed in the oil

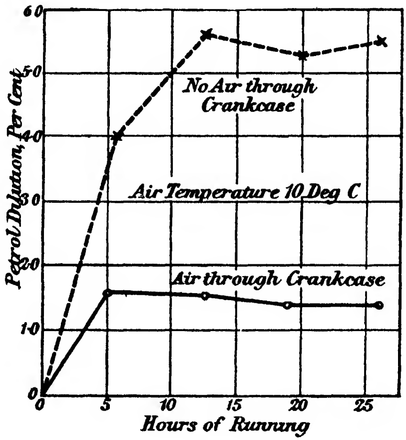


FIG. 30

steadily increases for about 5 hours when the crankcase is ventilated, and thereafter remains practically constant at about 1·5 per cent; whereas without ventilation the dilution attains 5·5 per cent. The tendency of a lubricant to acidity in use is avoided as far as possible by the oil manufacturers. The effect of crankcase ventilation in this respect is shown in Fig. 31 from which it will be seen that, starting with clean oil, the acidity was greater for some hours without ventilation, but that ultimately there was little difference.

Finally, Fig. 32 shows the result of a series of runs at different speeds and powers arranged to represent different practical operating conditions. During the first period of 60 minutes the engine developed only 9·4 b.h.p. at 1,080 r.p.m. During the second period 28·6 b.h.p. at 1,820 r.p.m., and so on, as shown at the upper part of the diagram. Air was drawn from the crankcase to the intake of the carburettor, so that the cooling effect of the air was roughly proportional to

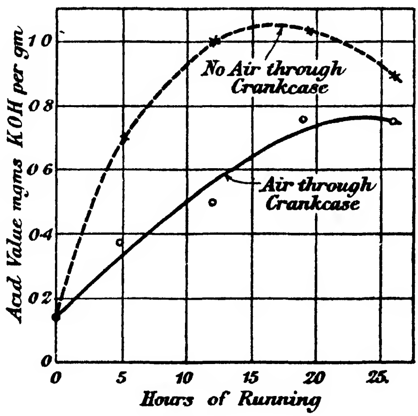


FIG. 31

the demand. With air circulation the oil sump temperature varies only a few degrees from 33° C. under all except starting conditions, while without air cooling the oil temperature rises substantially whenever the power output increases. The temperature of the air admitted to the crankcase was regulated in accordance with the temperature of the oil by a thermostat in the sump connected to a valve which admitted cold or heated air as required. This accounts for the rapid rise of oil temperature when starting, a very desirable characteristic; the low oil temperature, operating by means of the thermostat, then resulted in heating of the air supply to a temperature of more than 80° C. as shown by the peak on the air temperature curve.

The above series of experiments was undertaken, as mentioned above, with the object of reducing the

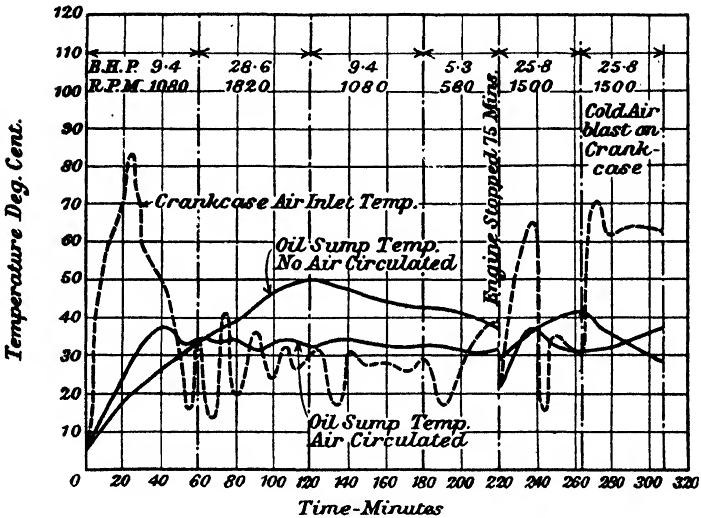


FIG. 32

liability to failure of big end bearings, the part of the engine most sensitive to any lubrication defect. The experiments suggested that an extension of the known methods of crankcase ventilation might well be made with the object of controlling the oil temperature within narrow limits, and that if the air drawn through the crankcase was led to the carburettor, the advantages would on the whole appear largely to outweigh the disadvantages. But apart from their particular object, the experiments afford the student a valuable insight into the conditions to which the oil in the engine is subjected, and its response to those conditions.

Oil Consumption. The enclosed self-contained lubricating system can only remain so if care is taken in the design and construction to prevent loss of oil. The crankshaft necessarily passes out of the crankcase at each end, and leakages of oil at these points is prevented by oil throwers, by collecting chambers, and by helical threads cut in the shafts. These threads tend to screw the oil back into the crankcase. Washers or gaskets, from oil sealing joints between the engine and the sump, side valve covers, overhead valve covers, and any other attachments. When the metal surfaces are rigid and properly machined, a jointing compound with or without washers of thick brown paper is sufficient, but when the faces are roughly finished or are not rigid a washer of some soft yielding substance, such as cork, must be used. Examples occur in light pressed steel sumps where the employment of a cork washer will avoid the necessity for very close spacing of the studs or bolts; also in side valve covers where it is desirable to employ one or two securing studs with finger tight nuts.

As a result of the attention given to the prevention of leakage the principal source of oil consumption is in connection with the lubrication of the cylinder walls and pistons. Lubrication of these parts is essential, but in the process a minute quantity of oil is lost on each stroke by passing upwards above the piston.

The consumption of oil in new engines varies greatly, being dependent not only upon size but upon design and perfection of workmanship. As a guide it may be stated that in the case of an engine of from 14 to 18 R.A.C. rating, a better result than 1,500 miles per gallon should not be expected. If the consumption is less than this the cylinder walls would not be adequately lubricated. The increase in the rate of oil consumption which takes place ultimately is due to wear of the cylinders, rings, or grooves.

In view of the necessity for a thin film of oil on the cylinder walls, and the fact that they are exposed on every fourth stroke to the heat of the explosion, the amount of oil lost per stroke is extremely small. A simple calculation determines the amount. In many cases the engine makes about 3,500 revolutions for every mile travelled on top gear. In a six-cylinder engine there are twelve strokes every revolution. If an oil consumption figure of 1,200 miles per gallon is assumed, the strokes per gallon of oil consumed amount to $1{,}200 \times 3{,}500 \times 12 = 50{,}400{,}000$.

A gallon of oil contains 4,545,000 cub. millimetres, which is a convenient unit in the circumstances. The amount of oil consumed on each stroke is then equal to

$$\frac{4{,}545{,}000}{50{,}400{,}000} = \cdot 09 \text{ cub. mm.}$$

Oil Filters. Foreign matter in the oil may increase the wear of an engine to such an extent that a great deal of attention is always given to the provision of adequate filters. The coarser particles are more readily separated than the fine particles, but the latter are more likely to do harm to the bearings. The filling orifice should always be fitted with a filter, for, although the oil from a can or drum is usually quite clean, there is practically certain to be grit or dirt in or near the filling orifice or on the neck of the can, particularly a spare oil can.

In the course of its circulation from the sump, through the bearings and other parts and back again, the oil is always passed through two or three filters which must provide sufficient area to avoid obstruction of the flow. The commonest filtering medium is fine wire gauze, but disks, felt and, magnets are also employed. As the foreign matter is heavy and tends to collect in the bottom of the sump, and as the oil is necessarily drawn from a low level, a filter is often provided on the suction side of the pump. In the Standard engine, the suction pipe draws oil from the bottom of a cylindrical gauze filter, as shown in Fig. 17. The filter in this case is carried by the drain plug, and this arrangement tends to secure proper cleaning of the filter with paraffin whenever the sump is drained and refilled.

In some engines the oil from the pump is passed through a filter before it is led to the bearings, while in others it passes direct to the bearings, and is then filtered before its return to the sump. In many force feed systems, the oil can only return to the crankcase by passing through a gauze filter forming the whole or part of a horizontal baffle arranged just above the surface of the oil. The whole surface of this baffle may be made of gauze or a sheet metal baffle may have gauze insertions.

In addition to the main filter, which deals with the whole supply in circulation, some of the oil may be by-passed through a separate filter, and in this case a finer filtering medium may be used. An example of this method is afforded by the A.E.C. engine, described previously, employing a felt by-pass filter.

Various arrangements of filters have been described above in connection with engine lubrication systems. Two further examples will now be described in detail. In the cylindrical gauze filter shown in Fig. 33 as used on some Crossley cars, the oil is led up through the central tube and passes outwards so that the foreign matter collects on the inside of the gauze. To remove the filter the nut *A* must be unscrewed, and the cup *B* and washer *C* removed. The spindle *D* is next unscrewed by the flats shown, and the filter complete with spindle can then be removed. The circular disk *E*, which is secured by the screws *F*, is then detached, when the gauze with the holder can be removed for washing in paraffin or petrol.

It is desirable that the area of the gauze should be as large as possible, so that the flow may be slow and a finer gauze therefore possible.

Oil filtering gauze woven from brass or copper wire

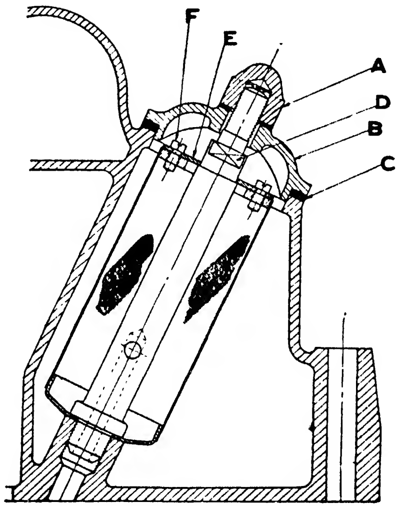


FIG. 33. CROSSLEY GAUZE FILTER

may be from 36 to 90 mesh, i.e. wires per inch. This is coarser than petrol gauze, which may be of 200 mesh, and will then prevent water from passing.

In the Auto-Klean Filter, Fig. 34, oil passes inwards through a series of fine slots in the wall of a cylindrical cartridge consisting of a large number of hollow disks fixed to a central spindle with a thin spacing washer between each disk. In each gap a small scraper is inserted to scrape away the foreign matter on the plates when they are rotated by the handle. The space between disks is ·005 in., which is equivalent to a wire gauze of 100 × 100 mesh. This is finer than is practicable with gauze owing to the rapidity with which fine gauze becomes choked. The spindle may be connected to the clutch pedal by a free-wheel, so that it is rotated through a small angle each time the clutch is operated. Alternatively a connection may be made to the brake pedal

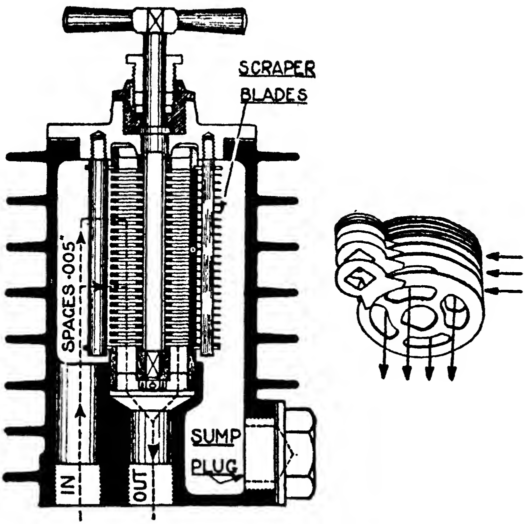


FIG. 34. AUTO-KLEAN ROTATABLE DISK FILTER

or accelerator. The strainer is thus kept clean automatically. The foreign matter which collects in the sump may be drawn off when the plug is removed.

Properties of Oil. The automobile engineer may set forth his requirements and the conditions which the lubricant for an engine should satisfy, but it is for the chemist who has specialized in the production of lubricating oil to meet those conditions or as many of them as he can.

It is not possible to gauge the quality or suitability of an oil by sight or touch. It may feel oily, and may appear to be suitably viscous, but only exact test can determine whether it has the necessary lubricating properties at the temperature at which it works in an engine.

A suitable lubricant must possess oiliness, a property which is difficult to define, but may be looked upon as molecular cohesion, so that it cannot readily be squeezed out from between the working surfaces, combined with a small resistance to shear so that it may allow the surfaces to slide freely over one another, giving a low coefficient of friction which varies with load, speed, and temperature. This subject is dealt with in the section entitled "Bearings."

The properties of oiliness and viscosity are interdependent. A lubricating oil must possess both to the proper degree. Treacle possesses viscosity without oiliness, and is clearly an impossible lubricant. Too high a viscosity results in loss of power due to unnecessary friction, while too low a value may cause failure of the bearing. The most suitable viscosity is the lowest which will stand up to the hardest running conditions.

The anti-frictional and cohesive properties are, however, not the only ones to be considered. All the oil in an engine is at times subject to violent agitation in the presence of air, and is moreover intermittently exposed to high temperature on the inside of the piston. Its tendency to carbonize, to oxidize, and to emulsify under the specified conditions must all be taken into account.

Oil suitable for lubrication may be obtained from mineral, animal, or vegetable sources, but for the particular requirements of the internal combustion engine animal oils have not been found suitable. Mineral oils are most widely used either alone or with the addition of suitable vegetable oil. The use of castor oil is, of course, well known in this connection. The mineral oils are obtained almost entirely from crude petroleum oil by distillation at certain temperatures after the lighter components of petrol and paraffin have been distilled off. A considerable amount of refining is, however, necessary before the oil is entirely suitable for use in a motor-car engine.

The two most important physical characteristics of an oil are the flash point and the viscosity. When an oil is heated up to the flash point temperature, the lighter constituents are volatilized at a relatively slow rate, sufficient only to cause a momentary flash if an open flame is played over or brought near to the surface of the oil. At this temperature the gases given off will not burn continuously, but when the oil is raised to a higher temperature, known as the burning point, the rate of volatilization is sufficient to maintain a continuous flame. The flash point is measured by the open or closed method, the two differing from one another in that in one case the flashing effect is obtained on the surface of the oil in a closed vessel, whereas in the other case it is measured in an open vessel. In the closed flash point test, the values obtained are slightly lower, owing to the protection of the oil from disturbing currents of air. Engine lubricating oils generally have a flash point in the neighbourhood of 400° F.

The viscosity of oils is measured by various methods and standards in different countries, but in this country the Redwood apparatus, shown in section in Fig. 35, is used. The oil to be tested is enclosed in the inner vessel, and is maintained at a definite temperature by an outer liquid both having supply and outflow pipes. The heating liquid may be uniformly distributed by a revolving agitator, and the temperature is indicated by a thermometer. The inner oil cylinder is also provided with a thermometer and an outlet is provided at the lower end ; this outlet is of a definite size, and is furnished with a valve consisting of a small metal sphere which may be raised clear of the outlet, when required, by

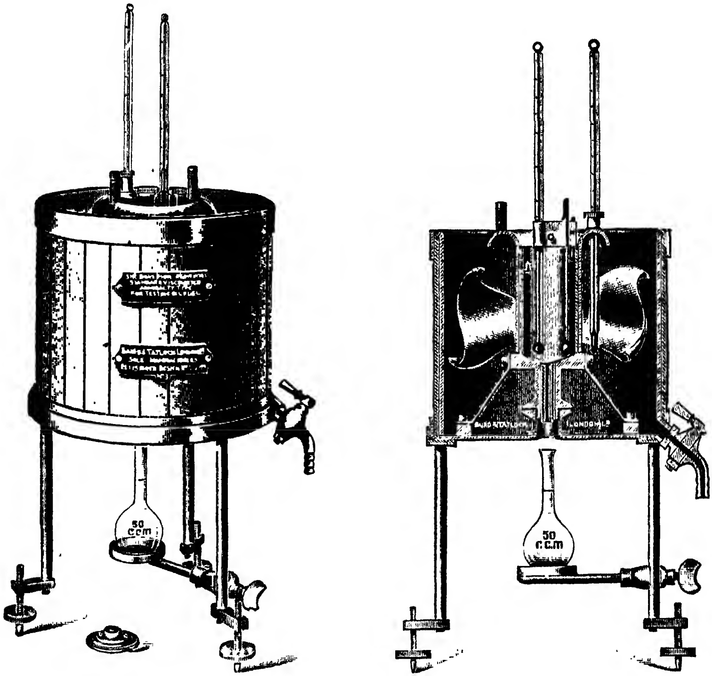


FIG. 35. REDWOOD ADMIRALTY-TYPE VISCOMETER
For determining the viscosity of fuel oil
(*By Griffin & Tatlock, Ltd.*)

means of a wire. The liquid in the bath is first brought to the required temperature and the oil is also heated to the same temperature and poured into the testing cylinder until it reaches a prescribed level, as indicated by the tip of the upturned point of a small arm shown projecting from the left-hand side of the oil cylinder. It is most important that the oil should be brought always to the same level so as to secure a constant head of liquid, and therefore uniform results. A flask

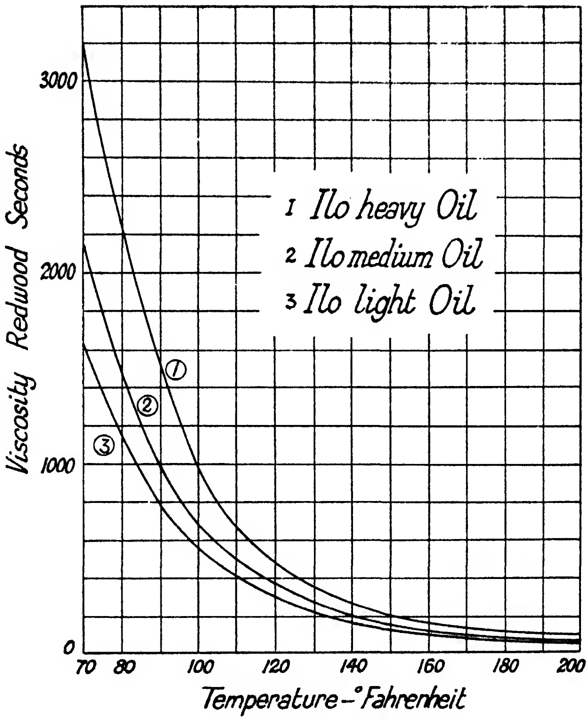


FIG. 36. RELATION BETWEEN VISCOSITY AND TEMPERATURE

with a narrow neck, holding 50 millilitres or cubic centimetres, is placed beneath the jet, and the ball valve is then raised, a stop watch being started at the same time. The time taken in filling the flask up to the 50 c.c. mark on the neck is then recorded, and expresses the viscosity of the oil in Redwood seconds. Care must be taken that the given temperature is maintained throughout the test.

The units of viscosity referred to above are, of course, of an arbitrary character, like the other units used in commercial testing apparatus in other countries, and suggestions have been made that viscosity should be expressed in absolute units. Such an absolute unit is the force required to move one side of a unit cube of oil at unit speed relatively to the opposite face of the cube, the two faces being, of course, separated by unit distance. It is thus a unit of the rate of shear of the oil.

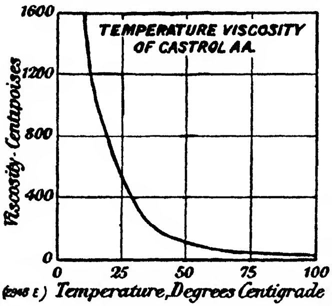


FIG. 37

If c.g.s. units are employed the result is expressed in dynes per square centimetre per second. The term poise has been given to this unit, and as it is too large for convenient use one hundredth part has been accepted as a unit and termed the centipoise.

It is important in view of the possible changes of temperature that the viscosity of lubricating oil should vary as little as possible with the temperatures. These variations are in all cases considerable, but they are in general less with mineral oils than with other oils. The several curves in Fig. 36 show how viscosity, as measured in Redwood seconds, varies with temperatures in the case of three grades of motor-vehicle engine oil. Fig. 37 shows a similar curve, but the viscosity is measured in centipoises.

SECTION XVI

BEARINGS

BY

MAJOR A. GARRARD, Wh.Ex.

SECTION XVI

BEARINGS

PLAIN BEARINGS

In various parts of a motor vehicle, shafts or spindles are mounted in bearings so as to rotate continuously or to oscillate through a limited angle.

Among the former are included the crankshaft and big-end divided bearings, the camshaft solid bearings, and the roller or ball bearings used for components such as the magneto, dynamo, starter-motor, or for various parts of the transmission, such as the gearbox, and axles.

The latter or oscillating class includes the steering swivel, steering gear bearings, and brake shafts, to which the terms "pivot" or "hinge" could perhaps be applied with greater accuracy. This class of bearing is, as a rule, not heavily loaded, and although it often wears much more than, for example, the crankshaft or camshaft bearings, this is due to exposure and to insufficient or irregular lubrication rather than to the amount of rubbing. Applications of such bearings or pivots are described in the sections of this work dealing with the parts of the vehicle where they are used.

The bearings to be dealt with in this section may be classed under the two headings: (1) plain or sliding contact bearings, (2) ball and roller bearings.

The engine crankshaft and big-end bearings, which work under the most exacting conditions of load and speed, form by far the most important application under this heading.

The bearing problem was an urgent one in connection with high-speed, heavy-duty steam engines more than fifty years ago, and the basic principles underlying the working of all such bearings, whether applied to steam engines or to motor vehicles or aircraft engines, were discovered and investigated at that time. For this we are chiefly indebted to the researches of Beauchamp Tower, who established certain general principles and

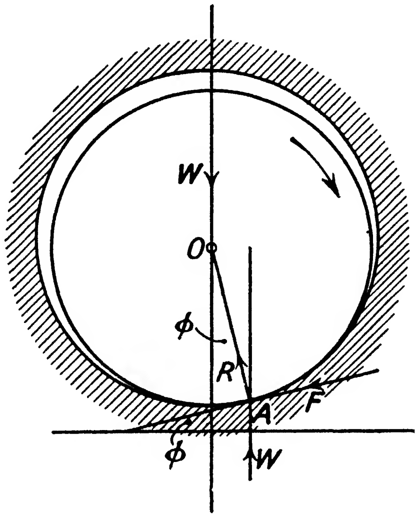


FIG 1. FRICTION OF SHAFT ROTATING IN BEARING

investigated the relations between friction, speed, loading, etc., although it must be recognized that comparatively recent experiments and developments in connection with automobile and aircraft engines have resulted in some modification of these laws, and in higher speeds and bearing pressures than were at that time considered possible.

Consider in detail a shaft rotating in a clockwise direction in a plain bearing, and subjected to a steady downward load. The shaft may be regarded as rolling up the bearing, in which it is assumed to be a very slack fit, until it begins to slip. The point, or rather line of contact, will then be along the radius OA, Fig. 1, making an angle ϕ with the vertical. The resultant force of friction F acts tangentially to the point A; that is, at right angles to the radius OA; and it opposes the rotation of the shaft. The shaft is then in effect just slipping at a steady speed down a plane slope inclined at an angle ϕ to the horizontal. This angle is called the limiting angle of dynamic or sliding friction, and it is such that

$$u = \tan \phi$$

where u is the coefficient of friction; u is also equal to $\frac{F}{R}$ where R is the pressure between the shaft and the bearing, and is exerted in the direction of the radius OA.

For all properly lubricated bearings, the value of u, and therefore the angle of friction ϕ, is so small that R is practically equal to W, the vertical load on the bearing, so that

$$u = \frac{F}{W}$$

As the shaft rotates it must continually overcome the retarding force of friction. The energy expended in this way is converted into heat, which is conducted away through the bearings to the adjacent metal parts, and is to a considerable extent carried away by the oil. Finally, this heat is, of course, dispersed into the atmosphere. During steady running a state of balance is reached when the heat per minute generated by friction is equal to that conducted away, and the temperature remains constant. In an internal combustion engine the problem is further complicated by the heating effect of the explosion on the lubricant, and indeed on the whole engine. The designers must see that the temperature of the bearings is never too high and that plenty of oil is supplied; otherwise the oil will become too thin, and will be forced out of the bearings, friction will increase, more heat will be generated, and the temperature will rise until the white metal liner or facing melts.

The energy expended per minute in overcoming the friction of a bearing is

$$E = \frac{\pi DFN}{12}$$

$$= \frac{\pi DuWN}{12} \text{ foot-pounds}$$

where D = diameter of bearing in inches
F = force of friction = uW
W = load on bearing
N = revolutions per minute.

The mechanical equivalent of heat being 778—

$$E = \frac{\pi DuWN}{12 \times 778} \text{ British Thermal Units of heat.}$$

In an actual example of a main crankshaft bearing in a large heavy vehicle engine, $D = 3$ in.: $W = 2{,}290$ (mean load): $N = 2{,}500$ r.p.m. A probable value of u under these conditions is ·015.

$$\text{Then } E = \frac{3{\cdot}1416 \times 3 \times {\cdot}015 \times 2290 \times 2500}{12 \times 778}$$

$$= 87 \text{ B.Th.U. per minute.}$$

As in this engine there are thirteen similar bearings, namely, seven crankshaft and six big-end bearings, it is evident that the heat generated by friction is by no means negligible.

Every properly lubricated shaft journal is separated from its bearing by a thin film of oil, so that no metallic contact takes place, and this accounts for the very small frictional resistance and the minute amount of wear. Such a thin film of oil or other liquid is capable of resisting very great pressures; that is, it is itself at a high pressure, and the thinner the film, the greater the pressure it is capable of resisting. The shaft, in fact, floats on a thin film of oil, the thickness of which varies according to loading and other conditions, but which is of the order of ·001 in. Such a thin film resists being forced out from between the surfaces very strongly, and it may with a loaded bearing be under a pressure of over 1,000 lb. per sq. in.

This remarkable effect is due to the property of liquids known as surface tension. The molecules of all liquids attract one another, but with comparatively small force. The attraction or cohesive effect is sufficient to prevent them from spreading, but not sufficient to hold them in a substantially rigid relation to one another as in the case of solid substances. The liquid molecules can be separated very easily and do not, in fact, cohere sufficiently to resist the force of gravity, although their mutual attraction is sufficient to prevent them from separating entirely, so that a liquid breaks up into drops; hence it is quite unlike a gas in which the molecules on the contrary do not possess the property of cohesion, and may even be regarded as exhibiting mutual repulsion.

Each molecule in the interior of the liquid attracts and is attracted by the adjacent molecules which surround it completely, but those molecules which are on the surface of the liquid are exposed on one side. The surplus force of attraction on this exposed side is then distributed in other directions, and more particularly in the direction of adjacent particles which are also on the surface of the liquid; as a result all the molecules which are exposed on the surface attract one another more strongly than those in the interior of the liquid, and we thus have the phenomenon of surface tension which gives the same effect as if the boundary or exposed surface consisted of a thin elastic membrane. The value or intensity of the surface tension varies considerably with different liquids, and also depends upon whether the surface of the liquid is in contact with air or with a solid or liquid surface.

Only when the two metal surfaces are properly smooth and fitted to one another, as the result of careful "running-in," is there an absence of metallic contact. In the early stages of the life of a bearing this smoothness of contour does not obtain. Each of the surfaces will have projecting from it small particles which must be rubbed down or broken away. These irregularities are of microscopic dimensions and, therefore, quite invisible to the naked eye, but until the bearing has been run-in, the coefficient of friction will not have reached its minimum value, and the best results will not have been obtained. Moreover, apart from these microscopic irregularities, the white metal yields or flows slightly until the journal and the bearing fit one another.

This peculiar property of a thin film of oil was discovered originally by Tower in the experiments referred to previously, in which a rotating shaft was engaged by an inverted bearing pressed down on to it by a known weight; the bearing was restrained against rotation by a long arm, and the force of friction could be measured with a considerable degree of accuracy. It was found necessary at one stage of the experiments to drill a hole through the bearing, and this hole was then lightly plugged. The experimenter discovered, however, to his surprise that the plug was very quickly forced out by the pressure of oil, and the same thing happened when a cork was driven tightly in. He found after further investigation that the film of oil in some parts of the bearing was subjected to a pressure of more than 600 lb. per sq. in., and that the pressures were greatest where the film of oil was thinnest, at or near the centre of loading. At points on either side of its maximum value the pressure diminished as the film of oil increased in thickness. Higher pressures were found to be possible without failure when the pressures were intermittent; for example, in the case of worm gearing, where the actual surfaces in contact are continually changing, a higher loading, for example, 2,000 lb. per sq. in., is permissible without risk of the film being broken down and metallic contact taking place. Not only does the surface tension of a thin film result in considerable resistance to the forcing out of the oil, but there is a powerful drawing-in effect, so that the oil tends to move inwards strongly towards the part where the film is narrowest. This effect of surface tension is familiar in other connections; for instance, the rise of a liquid in a fine tube or a wick. These form applications of the principle of surface tension to which the term capillarity is applied. Later experimenters have confirmed Tower's general deductions, and their researches have carried knowledge of the subject of lubrication considerably further forward.

A shaft loaded vertically and supported but not rotating in a bearing is shown in Fig. 2, the varying thickness of the supporting film of oil being much exaggerated. The pressure in the film at various points is shown by the polar diagram, and it will be seen to vary as some inverse function of the thickness of the film, the resultant R being vertical and equal to the load W.

When the same shaft is rotated in a clockwise direction, as shown in Fig. 3, oil is dragged round by the shaft and wedged into the gap to the right of the point D, where the film is thinnest, with the result that the pressure in this region is increased, as shown in dotted lines at b. The converse action takes place on the left of point D, where the pressure is reduced down to or even below that of the atmosphere, as shown at c. The direction OA of the resultant pressure on the bearing is

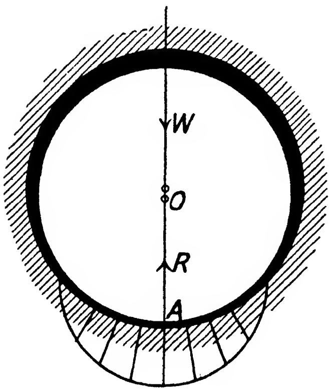


FIG. 2. VARYING THICKNESS OF OIL FILM IN NON-ROTATING BEARING

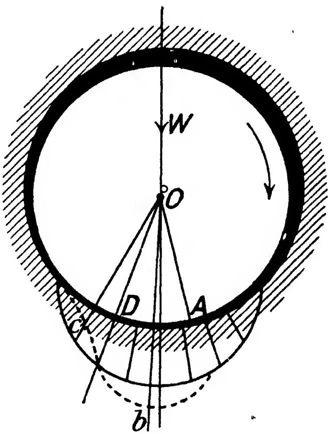


FIG. 3. OIL FILM IN ROTATING BEARING

moved towards the leading side, as in Fig. 1, although the point D, where the journal and bearing most nearly approach one another, is on the opposite side of the vertical.

There is some evidence to the effect that the oil circulates in the manner shown in Fig. 4. In practice this simple circulating system is, of course, modified by the fact that oil is continuously being fed to the bearing and discharged from it, although surface tension tends to prevent the expulsion of oil from the pressure region where it is relatively undisturbed.

The conditions in an internal combustion engine are very different from those existing in a bearing subject to a steady load, since the resultant force is continually changing both in direction and amount. These varying forces are greater than permissible steady forces.

The terms "thick film" and "thin film" lubrication have been used to distinguish between the conditions

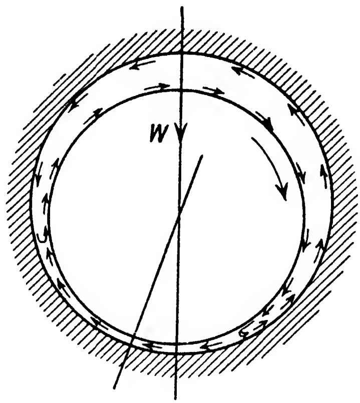


Fig. 4. How the Oil in a Vertically Loaded Bearing is Supposed to Flow

in which a film of oil always separates the sliding surfaces and those when lubrication is so intermittent or the lubricant is so thin that the surfaces are not definitely separated. For all the bearings in internal combustion engines thick film lubrication is essential.

Consideration of the foregoing principles will make it clear that no ordinary oil pump can possibly force oil between the working surfaces in or near the region where such pressures exist. Oil must, therefore, be fed to the bearing at some low pressure region where the working surfaces are slightly farther apart, and it will then be drawn in by surface tension and by the rotation of the journal to the high pressure region where the film is thin. It is thus an essential condition in all bearings that oil shall be introduced to low pressure regions, and shall not be allowed to escape freely from high pressure regions.

The shape and location of the oil ways is of great importance in this connection, and will be discussed in more detail later when bearing construction is considered. It may, however, be mentioned that oil ways in the high pressure regions should be avoided as far as possible. They represent so much loss of high pressure area, since the oil in the grooves can only be at a comparatively low pressure, and they also tend to break up the film. At parts of the bearing remote from the high pressure regions, the pressure of the oil may even diminish below atmospheric pressure, and this is likely to be the case in splash lubrication systems. Such systems are to-day much less favoured than forced feed systems in which the oil is forced to the bearing under a pressure of at least 25 lb. per sq. in., but even these pressures are small compared with the actual pressures in the thin part of the oil film, and forced feed systems should merely be regarded as rather more efficient methods of introducing the oil into the low pressure parts of the bearing.

The question of the most suitable clearance to allow room for the film of oil will be seen from the foregoing to be a matter of considerable importance. If the journal and its bearing were made exactly of the same diameter relative movement would be impossible; a difference of diameter of at least ·001 in. is necessary. The proper degree of tightness of the bearing of the big-end of a connecting rod is tested in the well-known way by ascertaining whether a rod will turn round under its own weight when moved over to an angle of about 30° to the vertical. This method of testing is used in repair shops, but in many factories the bearing and the journal of the shaft are finished each within very fine limits, so that the one will fit in the other and give a clearance which lies within definite limits. Two examples of modern practice are given below. They relate to the main crankshaft bearings and big-ends of the petrol engines fitted on bus and other chassis by the Associated Equipment Company.

Journal diameter $70 {}^{-\cdot 025}_{-\cdot 05}$ millimetres

Bearing diameter $70 {}^{-\cdot 05}_{+0}$ millimetres

Maximum clearance $= \cdot 05 + \cdot 02 = \cdot 07$ mm. $= \cdot 0028$ in.
Minimum clearance $= \cdot 025 + 0 = \cdot 025$ mm. $= \cdot 001$ in.

Big-end crank-pin diameter $58 {}^{-\cdot 01}_{-\cdot 03}$

Big-end bearing diameter $58 {}^{+\cdot 032}_{+\cdot 019}$

Maximum clearance $= \cdot 032 + \cdot 03 = \cdot 052$ mm. $= \cdot 0024$ in.
Minimum clearance $= \cdot 019 + \cdot 01 = \cdot 029$ mm. $= \cdot 0011$ in.

Reference has already been made to the frictional resistance to rotation causing the generation of heat which has to be conducted away largely by the oil. Tests have been carried out by The Hoyt Metal Company on various tin base white-metal alloys to determine, among other things, how the coefficient of friction depends upon speed and loading. A selection of the results is given below to indicate their general character.

Table I deals with tests on four different alloys at a constant speed of 2,760 ft. per minute, but with varying loads. The value of u, the coefficient of friction, is given in each case.

TABLE I

Alloy	Tin %	Load in lb. per sq. in. 140	280	420	560	700	840	980
A	89	·0694	·0354	·0244	·0183	·0146	·0138	·0115
B	87·15	·0379	·0320	·0246	·0188	·0149	·0138	—
C	79·36	·0604	·0320	·0219	·0177	·0140	·0115	·018
D	83·03	·0618	·0320	·0230	·0199	·0196	·0160	—
	Mean value of u	0·574	·0329	·0235	·0187	·0158	·0138	·0148

Examination of these figures shows that u is independent of the nature of the rubbing surfaces, the differences between the figures in any column being only usual experimental variations. There is, however, quite a definite decrease in the value of u as the load increases.

Table II gives values of u when the load is kept constant, at 560 lb. per sq. in., and the speed is varied.

TABLE II

Alloy	Speed in feet per minute 920	1,472	1,932	2,300	2,576	2,760
A	·0180	·0211	·0211	·0181	·0192	·0183
B	·0233	·0131	·0170	·0173	·0176	·0188
C	·0190	·0166	·0181	·0203	·0170	·0177
D	·0280	·0260	·0163	·0190	·0176	·0196

A similar series of values of u at various speeds is given in Table III, but in this case with double the loading, viz. 1,120 lb. per sq. in.

TABLE III

Alloy	Speed in feet per minute					
	920	1,472	1,932	2,300	2,576	2,760
A	·0100	·0140	·0105	·0110	·0016	·0112
B	·0083	·0093	·0094	·0107	·0110	failed
C	·0096	·0094	failed	·0104	·0102	·0120
D	·0098	·0107	·0108	·0108	failed	failed
Mean value of u	·0094	·0099	·0102	·0108	·0109	·0116

From the last two tables it is again clear that u is practically independent of the nature of the surfaces, but increases slightly as the speed increases.

The number of failures in the last table would suggest that the figure 1,120 lb. per sq. in. is too high for steady loading, although it may be permissible on motor-car engines, since the loads are intermittent and reversed, and an engine is rarely run at high speed on full throttle for prolonged periods.

In any bearing, the relation for safe and reliable running between p, the pressure in pounds per square inch of projected area, and v the linear rubbing speed in feet per minute is of an inverse character. One formula used in this connection is

$$p \,.\, v^n = K$$

where n has a value close to unity and K is a constant which varies considerably according to conditions such as the materials, surfaces, and methods of lubrication; also the margin of safety desired.

A number of tests on various tin base alloys were carried out by The Hoyt Metal Company to determine the maximum safe running values of p and v relatively to each other. The mean value of these tests showed a close approximation to the law—

$$p\,v = 1{\cdot}5 \times 10^{-6}$$

It must be realized that the values in the formula represent what are regarded as safe maximum values for steady running. In practice, a lower value for the constant may be adopted to give a larger margin for long life, while a higher value may be used if the engine only occasionally develops its maximum power.

For example, a commercial vehicle which runs, say, 50,000 or more miles in a year and may, while carrying a heavy load, work near its maximum power for long periods, might well be designed to give lower values, while a car which is so powered that the engine can, in view of road conditions, never develop full power for more than a few minutes may safely stand somewhat higher values.

The term load factor has been applied to the product $p \times v$.

In no branch of engineering are the working conditions of bearings more exacting than in the high speed internal combustion engine, whether used for aircraft, motor-cars, or for marine purposes, and whether of the petrol or compression-ignition type. The speeds of the former result in very heavy loads on the bearings, and the immense forces exerted during the working stroke in the latter type more than make up for its rather lower speeds. It is in applications to this class of engine that the composition of bearing metals has received most attention in recent years, and to-day there is fairly general agreement as to the best type.

Two main types of bearing metal are used, namely, tin base and lead base; the former is used exclusively for heavy duty bearings, and the latter, which is cheaper, is used only for comparatively light duty, and hence finds no place in the modern internal combustion engine.

The first tin bearing alloy was introduced by Isaac Babbitt about 1860, and his name is still used in this connection. The terms Babbitt metal, white metal, anti-friction metal, and white lining metal are synonymous, and are applied to both tin base and lead base alloys. The exact composition of the original Babbitt metal is

Fig. 5. Microphotograph of a Hoyt Bearing Metal Showing Antimony Cubes

not known, but it is generally thought to have been composed of between 83 and 89 per cent of tin, about two-thirds of the remainder being antimony, and one-third copper.

The surface of a tin base bearing metal when magnified at least one hundred times is seen to consist of a mass of tin through which are distributed crystals of tin, antimony, and copper. The characteristics of the bearings, when magnified, depend upon the proportions of the several constituents and the manner of manufacture; for example, the differences due to rapid or slow cooling of the alloy. Figs. 5 and 6 show the microstructure of two Hoyt alloys; the first one contains sufficient antimony to show the well-known antimony cubes or crystals, and the second one contains less antimony and shows a fine lattice of copper-tin.

The principal component, tin, is a very soft, malleable metal which takes a high polish, but by itself is too soft to give the necessary support for the shaft. The addition of a small quantity of antimony produces

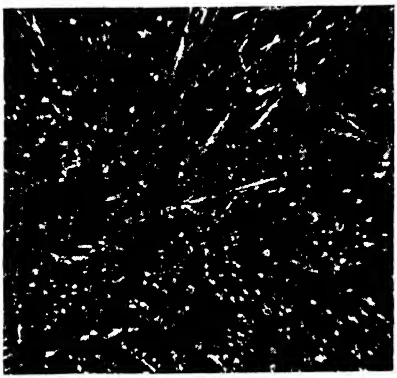


FIG. 6. MICROPHOTOGRAPH OF A HOYT BEARING METAL SHOWING A FINE COPPER-TIN STRUCTURE

a hard alloy of greater compressive strength, but these characteristics are accompanied by a certain degree of brittleness, and it is to overcome this characteristic and produce a tough alloy that a proportion of copper is introduced.

There is considerable difference of opinion why these white metal alloys offer resistance to wear, and at the same time support the load. A suggestion has been made that the hard antimony crystals are the most effective constituent from this point of view, and that the relatively soft mass of tin merely provides a background. There is, however, some doubt whether this is the case, since bearings, which after considerable running still show clearly the structure of the alloy, will generally be found to have worn too quickly. The effect in such cases would seem, in fact, to be akin to that of a grindstone or carborundum wheel, which consists of a number of very hard abrasive particles standing up from a softer base or bonding medium which wears away more quickly and always leaves them exposed and hence available for cutting. The theory has also been put forward that the minute spaces between the projections retain some lubricant which prevents scouring or other damage to the bearings when an engine is started up from cold, and before the oil has had time to circulate. It has been suggested that these interstices constitute a minute reservoir of oil when there is actual metallic contact. Such a residue of oil must, however, be present in such minute quantities that it is doubtful whether it can have any practical effect. Certainly under all ordinary running conditions the proper oil film must carry the load, and if it is broken seizure or melting of the bearing is inevitable. It is well known that the rate of wear when a cold engine is raced and before the oil has had time to warm up and circulate is much greater than at any other time.

The most dependable tin base alloys for internal combustion engine bearings contain approximately 85 per cent tin and are free from lead; the remainder consists of copper and antimony. Even slight variations from the standard formulae or variations in the purity of the components affect substantially the quality of the alloy. It must be realized, however, that there are various other grades which are cheaper and equally well suited for less exacting conditions.

It has sometimes been stated that the composition of the material in a bearing is relatively unimportant, and that the lubrication is of primary importance, since the film of oil carries the load. This is, however, by no means a complete statement of the case. It is true that the friction is largely independent of the material, but other factors, such as the wearing qualities, the resistance to spreading, cracking or breaking up of the bearing, are dependent upon the composition and properties of the alloy.

Bearing metal must have some plasticity to yield slightly in compression to compensate for minute irregularities, thus avoiding dangerous stresses and reducing overheating and wear. Plasticity is shown by the amount of compression before cracking takes place. Accurate machining and alignment reduce the necessity for much plasticity. A low rate of wear is desirable, but the white metal must nevertheless wear more than the shaft, since it can be renewed more cheaply. Tin base alloys have a lower fluidity than lead base alloys near the melting point, so that they must be poured 100° C. to 150° C. hotter, although the melting points are not very different. Pouring temperatures run from 400° C. to 500° C., depending upon wall thickness of the shell or cap and temperature of mould, all of which should be well pre-heated. Overheating of metal before pouring is to be avoided.

Tin and lead in the same alloy tend to harden each other, and are best used alone, the tin base alloys being superior mechanically. Table IV shows the results of tests and measurements made by Messrs. Hoyt on three of the tin base alloys referred to previously. The bearings were run at a speed of 2,760 ft. per minute, and the bearing pressure which produced failure was ascertained in each case.

The thickness of the white metal facing is usually from 1 to 3 mm., but may be rather more when cast direct into the big-end. In large bearings the shell may be undercut or dovetailed so that the facing is keyed

TABLE IV

	Tin per cent	Failure load. Lb./sq. in.	Shear stress. Tons/sq. in.	Compression yield point. Lb./sq. in.	Hardness Brinell
B	87·15	840	4·95	7,929	29
C	79·36	980	5·26	4,210	32
D	83·03	840	5·2	6,950	27·5

thereto, but this method is clearly impossible with the thin facings used in automobiles. Further, variations in thickness may cause uneven yielding and strain. Hence soldering is relied upon entirely. The steel must be well tinned beforehand, and must be heated before the white metal is poured to avoid unequal shrinkage, and in order that the solder may melt and bond the white metal and shell together. If the bonding is imperfect, oil will penetrate between the facing and shell, and as it is a poor conductor of heat the bearing will heat excessively. Solder and white metal will adhere better to a rough turned bronze or steel shell than to one which has been finely finished.

Good quality solder should always be used. The proportions of tin and lead may vary greatly; the common 50–50 is rather soft. The mixture which melts at the lowest possible temperature (180° C.) contains 63 per cent tin and 37 per cent lead. Whether the tin content is increased or decreased, the temperature of complete fluidity is increased.

The several types of half bearing may be broadly classified as follows—

1. Bronze liner with thin white-metal facing.
2. Steel liner with white metal facing.
3. Die-cast white metal liners.
4. White metal cast direct in the frame, bearing cap or connecting rod.

Of these several types, No. 4 is used more in connecting rods than in main crankshaft bearings.

No. 3 is much less used to-day than formerly, since the necessary thickness of white metal has, by itself, not sufficient toughness to resist the pressures and pounding effects of modern high-speed engines. For all petrol engines type No. 1 is generally regarded as the best practice, the thin white metal facing being well backed up by the bronze shell. It has, however, been found that even this construction of bearing is not adequate to resist the tremendous forces which occur on the explosion stroke in compression-ignition engines, and type No. 2 has been used for this purpose as well as in some petrol engines. As a general rule, to avoid risk of failure and to secure long life, all the bearings of an engine are made of the maximum practicable width and diameter. Engines differ so greatly in design that it is impossible to lay down any hard and fast dimensional rule. Design for a number of years has progressed by small steps, each one based more or less on empirical data obtained in earlier constructions.

In many small four-cylinder engines the crankshaft is carried in two main bearings only, and the rear one is generally made longer than the front one, partly because it has to support the weight of the flywheel in addition to other forces, and also in order that the wear should be negligible for very high mileages, since any substantial amount of taking up of the bearing might upset the alignment of the crankshaft, clutch, and gear shafts. This construction facilitates and indeed calls for the placing of the cylinders close together, so as to shorten the unsupported length of crankshaft between the bearings. The dimensions of the crankshaft including the webs must be sufficient to ensure stiffness quite apart from strength. The pressures per square inch on the bearings in any design can be calculated, knowing the maximum speed of the engine, the weights and dimensions of the moving parts, and the forces exerted on the working strokes. Such calculated forces must be well within the limits which have been found from previous practice to be satisfactory.

The loads on the two end bearings are, however, substantially less than in some other designs, since the centrifugal forces on the four cylinders balance one another; and the same effect occurs as regards the greater part of the reciprocating inertia forces. In some such engines the bearings are not divided, solid cylindrical shells faced with white metal being employed. If the bearings are of sufficient diameter and length the wear over long periods is negligible.

The exact opposite of this type of engine is the four-cylinder engine with a five-bearing crankshaft. The two end bearings may be fairly wide, but the intermediate bearings are necessarily comparatively narrow, if excessive overall length is to be avoided. The provision of a bearing between every crank makes the design of a rigid crankshaft a matter of some difficulty, and the necessity of providing sufficient material for the webs also limits the width of the big-end bearings. The dimensions decided upon for the bearings are necessarily the result of careful scheming in conjunction with the design of other parts such as the crankshaft.

In three-bearing crankshafts for four-cylinder engines there is much less difficulty in providing good bearing areas and a substantial crankshaft, but in this type of engine the middle bearing is greatly overloaded. Any three-bearing crankshaft, however rigidly constructed, is bound to yield sufficiently to load the middle bearing two or three times as heavily as the end bearings, resulting in greater wear.

Six-cylinder engines are to-day constructed either with seven or four bearings, although a few examples of three-bearing crankshafts are still in use. Similar considerations to those mentioned in connection with four-cylinder engines also apply. Further, in the seven-bearing type, the crankshaft is usually so elastic by

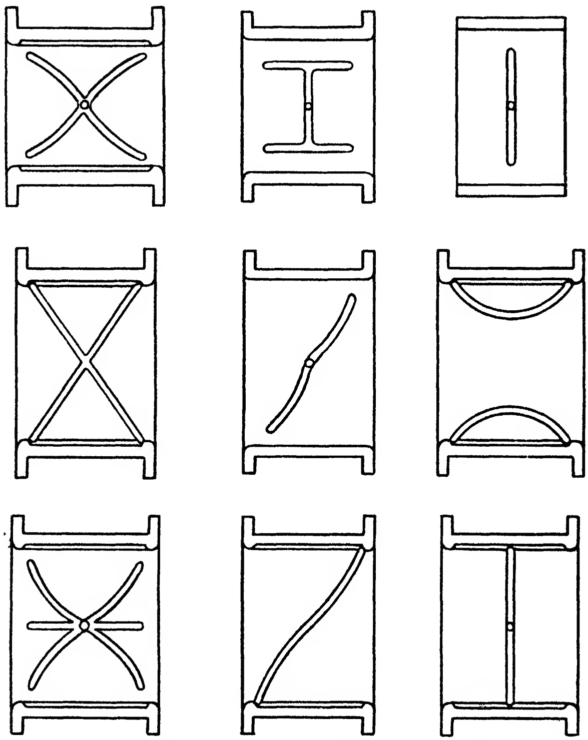


Fig. 7. Various Arrangements of Oil Ways

reason of its length and limited dimensions that torsional oscillation may be set up, and a frictional damper is usually provided. In engines having four-bearing crankshafts, there is, however, more room available for the bearings, and the crankshaft may be made stiffer, so that torsional oscillation dampers may be unnecessary.

In every engine at least one bearing must have flanges to prevent endwise movement of the crankshaft, although they are not intended to take any substantial end thrust.

Reference has already been made to the provision of oil ways in bearings, and to the necessity of avoiding any breaking up of the oil film by oil ways at the points where the pressure is greatest. It would seem, however, from a number of examples in practice that this point is not fully recognized. It is, of course, necessary that oil should have ready access to the bearings, but it should be realized that if oil is introduced at the parts where pressure is small, it will be drawn into the parts where it is required with a force which is far greater than can be exerted by any pump. A number of bearings with various arrangements of oil ways are illustrated in Fig. 7, but it must not be assumed that these all represent the best practice, since in many cases the grooves encroach seriously upon the high pressure area. The most common arrangement with crossed grooves has been widely used in steam and other practice for many years, although it is by no means the best for a high-speed engine. Sharp edges to grooves tend to break the oil film.

An example of modern commercial practice will now be considered in some detail. The petrol engine in question is manufactured by the Associated Equipment Company for buses and lorries, and has six cylinders of 110 and 130 mm. bore and stroke respectively. A sectional view is shown in the illustration facing page 766.

The forces on each bearing are due to—

1. Centrifugal forces exerted by the rotating masses which include the crank pins and a portion of the webs,

the big-ends, and part of the length of the connecting rods.

2. The inertia forces which are mostly vertical and are due to the reciprocating mass of the piston, the small end, and part of the length of the connecting rod. The angular movement of the connecting rods also accounts for lateral inertia forces.

3. The gas or explosion forces which are obtained from the usual indicator diagram. A brake mean-effective-pressure of 100 lb. per sq. in. is assumed in connection with the particulars given below.

The results of calculations under the above headings, when the engine is rotating at 2,500 r.p.m., are shown graphically in Figs. 8, 9, and 10, which relate respectively to the end bearings, the intermediate bearings, and the centre bearing. In each diagram a curve shows the amount and the direction of the radial forces on the bearing for every position of the crank pin, and it should be noted that the direction of the force does not always coincide with the centre line of the crank. The crankshaft is rotating in a clockwise direction, and the angle values on the curve show the direction of the force on the bearing when the appropriate crank is at the angle from the top dead centre mentioned. For example, in Fig. 8, the figure 60° indicates that when the end cranks are in the position CA (which is at 60° to the centre line) the magnitude and direction of the force on the bearing is represented by the line C 60°. The figures on the curve in Fig. 9 are given with reference to the position of one of the adjacent cranks, while those in Fig. 11 have reference to the two middle cranks.

Each curve in Figs. 8, 9, and 10, covers a complete cycle of operations, that is, rotation through two revolutions or 720°. The curve summarizes vectorially the three components specified above, but does not give each separately. The manner in which each curve is built up is, however, shown in Fig. 11, which relates

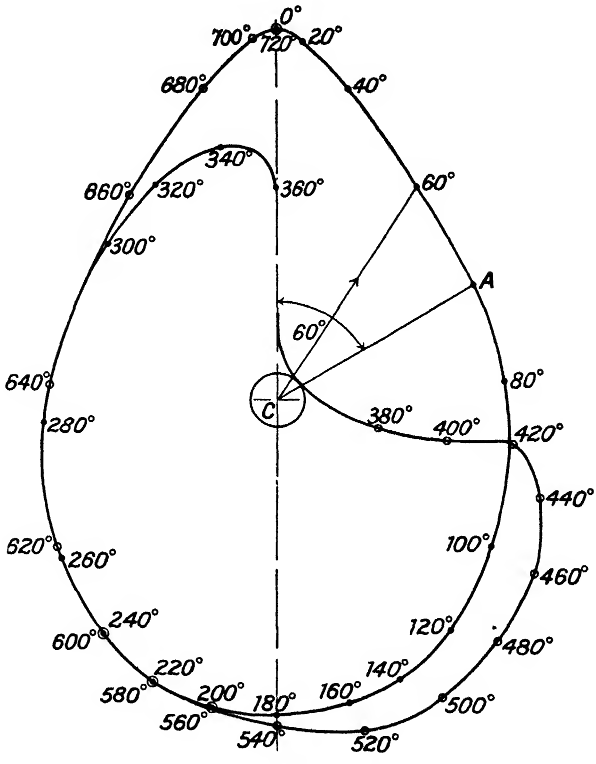


FIG. 8

Polar diagram showing forces on end crankshaft bearing of A.E.C. engine for every position of the end crank. Scale: 1,500 lb. per inch

to one of the end bearings. The circle (1) represents the centrifugal forces which are constant at any given speed, and act radially in the direction of the crank arm. The reciprocating (largely vertical) inertia forces are added vectorially to circle (1), and the result is indicated by curve (2). The obliquity of the connecting

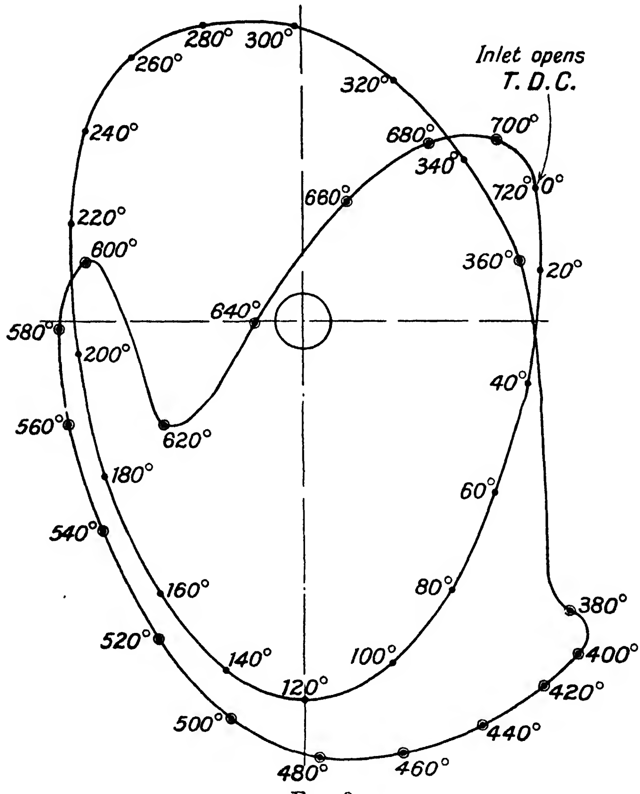


Fig. 9

Polar diagram showing forces on an intermediate bearing of A.E.C. engine for every position of an adjacent crank. Scale: 1,500 lb. per inch

rod and the consequent greater inertia forces at the top of the stroke account for the unsymmetrical form of curve (2). To curve (2) is added vectorially the downward thrusts due to the pressures on the compression and on the working stroke of the end cylinder

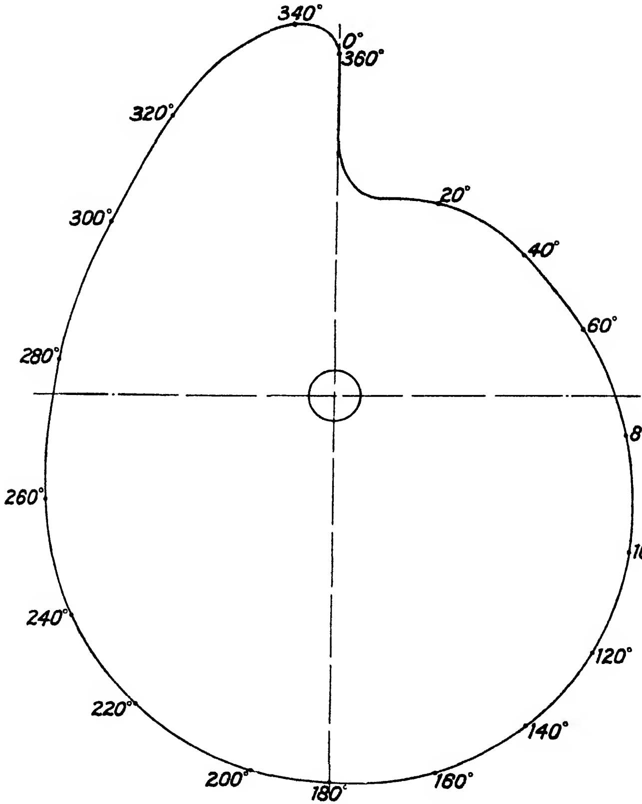


FIG. 10

Polar diagram showing forces on centre bearing of A.E.C. engine for every position of the adjacent cranks. Scale: 2,500 lb. per inch

which is the only one having any influence on the end bearings. These thrusts reduce or even reverse the centrifugal and inertia forces, and modify the previous curve as shown dotted at (3).

The curve of the centre bearing, Fig. 10, is simple in

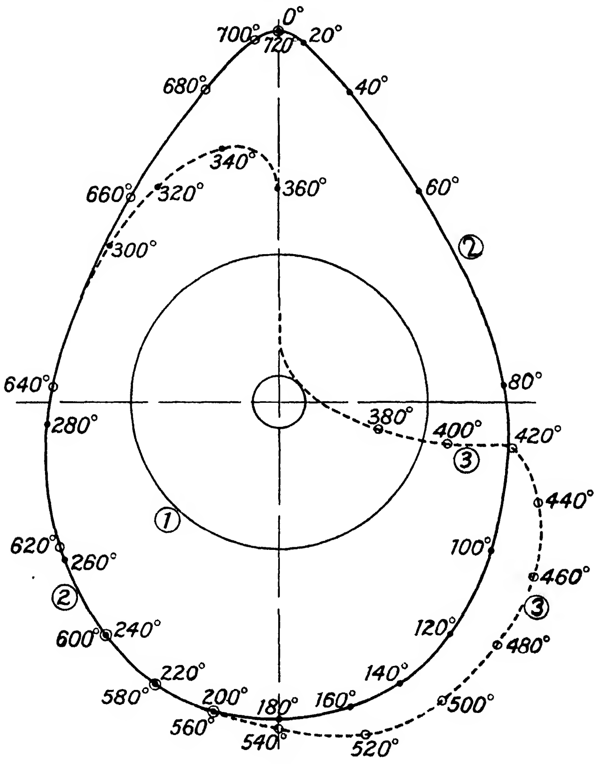


Fig. 11

Showing how the polar diagram is built up from the centrifugal, inertia, and explosion diagrams. Scale: 1,500 lb. per inch

form and is the same for both revolutions of the cycle. This is due to the fact that the two centre cranks, Nos. 3 and 4, coincide, although they fire 360° apart, so that one influences the bearing on one revolution, and the other influences it similarly on the next revolution.

The curve of any one of the four intermediate bearings is influenced by the cranks on each side which fire at intervals of 240° and 480°. The resultant inertia curve is thus unsymmetrical, and the modification of the inertia curve, due to the compression and explosion strokes of the two cylinders, increases the irregularity.

Examination of these curves shows not only the maximum values of the forces but indicates clearly the manner in which the direction of the force is continually changing. The maximum forces are approximately vertical, and the bearings are, therefore, quite properly divided horizontally, that is, in the direction of the minimum forces.

Certain figures derived from Figs. 8–10 are given in Table V, together with similar data relating to (*a*) the standard petrol engine as above, but modified by the

TABLE V

FRONT AND REAR BEARINGS

	Petrol Engine		Compression-ignition Engine	
	Without balance weights	With balance weights	Without balance weights	With balance weights
Maximum load (lb.)	2,940	1,950	5,300	5,000
Mean load (lb.)	2,230	1,390	3,750	3,400
Mean unit pressure (lb. per sq. in.)	290	180	469	425
Mean rubbing speed (feet per second)	30·6	30·6	36·4	36·4
Mean load factor (lb. per sq. in. per ft. sec.)	8,950	5,500	17,000	15,450

TABLE VI

INTERMEDIATE BEARINGS

	Petrol Engine		Compression-ignition Engine	
	Without balance weights	With balance weights	Without balance weights	With balance weights
Maximum load (lb.) . .	3,480	2,620	5,625	6,600
Mean load (lb.) . . .	2,290	1,430	3,940	3,200
Mean unit pressure (lb. per sq. in.) . .	655	408	885	718
Mean rubbing speed (feet per second) . .	30·6	30·6	36·4	36·4
Mean load factor (lb. per sq. in. per ft. sec.) .	20,000	12,500	32,200	26,200

TABLE VII

CENTRE BEARING

	Petrol Engine		Compression-ignition Engine	
	Without balance weights	Without balance weights	With balance weights	With balance weights
Maximum load (lb.) . .	5,220	3,060	8,900	5,550
Mean load (lb.) . . .	4,380	2,060	7,250	3,760
Mean unit pressure (lb. per sq. in.) . .	658	310	815	425
Mean rubbing speed . . (feet per second) . .	30·6	30·6	36·4	36·4
Mean load factor (lb. per sq. in. per ft. sec.) .	10,400	9,500	29,700	15,400

addition of balance weights on the crankshaft, which substantially reduce the forces on the bearings, and (*b*) a compression-ignition or heavy-oil engine of larger bore and stroke (115 × 142). The speed of the engine in each case is 2,500 r.p.m.

In the heavy-oil engine a much larger proportion of the total forces on the bearings is due to the explosion pressures. The relatively small reduction in loads and

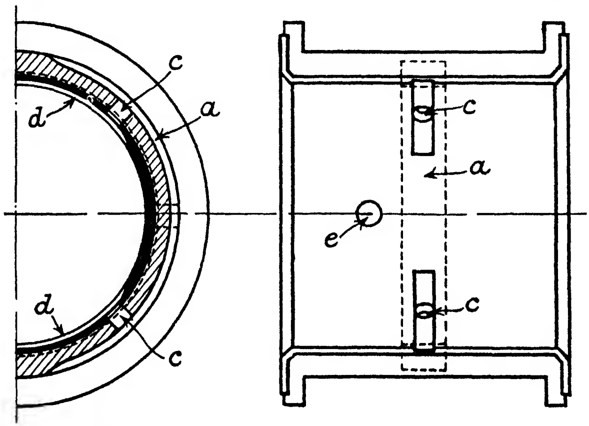


FIG. 12. A.E.C. FLANGED CRANKSHAFT BEARING (NO. 1)

bearing pressures resulting from the addition of balance weights to the oil engine crankshaft, in contrast with the considerable reduction in the case of the petrol engine is, however, largely due to the fact that the petrol engine was fully counterweighted and the oil engine only partially so.

The bearing shells with white metal faces used in the A.E.C. petrol engine to resist the forces set out in the foregoing tables are shown in some detail in Figs. 12–15.

Fig. 12 shows the forward bearing (No. 1) which is flanged to position the shaft endwise, all the other bearings being without flanges. Fig. 13 shows the rear bearing next to the flywheel (No. 7), Fig. 14 the heavily loaded centre bearing (No. 4), and Fig. 15 one of the

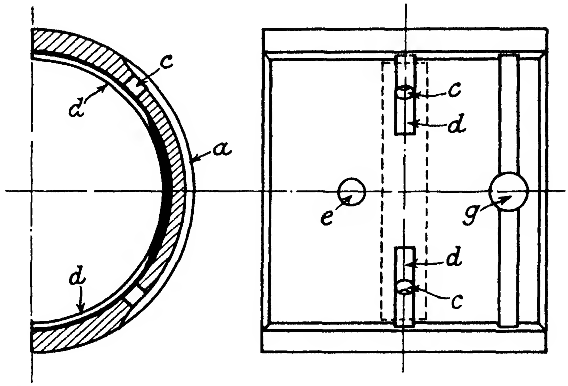


FIG. 13. A.E.C. REAR CRANKSHAFT BEARING (NO. 7)

four intermediate bearings (Nos. 2, 3, 5, and 6). In accordance with the invariable modern practice the caps

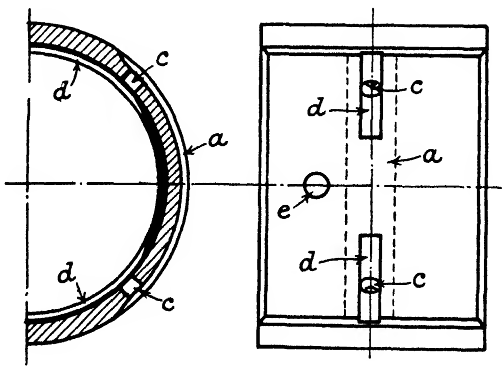


FIG. 14. A.E.C. CENTRE CRANKSHAFT BEARING (NO. 4)

are bolted up from underneath, the cap for the rear bearing being shown in Fig. 16, and that for the intermediate bearing in Fig. 17.

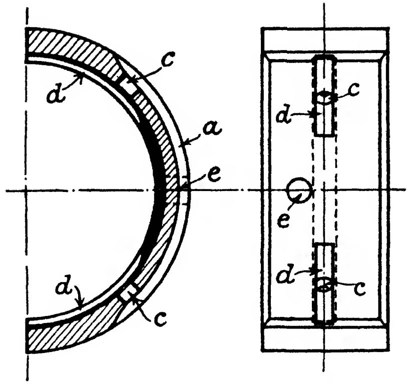


FIG. 15. A.E.C. INTERMEDIATE BEARINGS, NOS. 2, 3, 5, 6

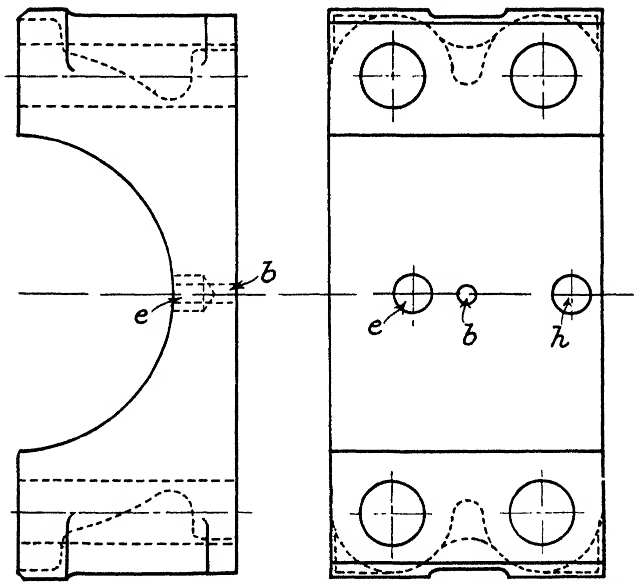


FIG. 16. BEARING CUP FOR REAR BEARING, (No. 7) OF A.E.C. ENGINE

Each bearing consists essentially of a bronze shell with a thin white metal facing which, in the case of No. 1 bearing, extends over the flanges. The top and bottom halves of each bearing are identical.

Considering first the provision for lubrication, each

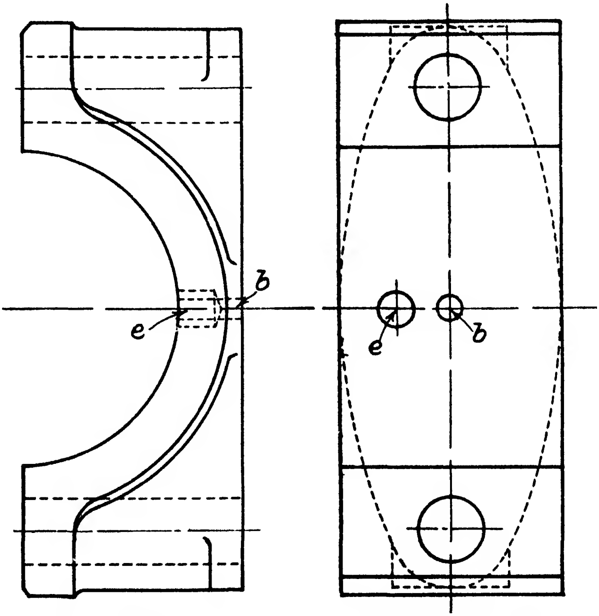


FIG 17 BEARING CUP FOR INTERMEDIATE BEARING OF A E C. ENGINE

bearing is turned with a part circumferential groove *a* on its outside to which oil is continually supplied under pressure from the oil passage *b* in the centre of the cap, the outlet from the oil pump in the sump being connected by a system of copper pipes to all the caps. Oil passes from the outer grooves through radial holes *c* to the inner grooves *d*, which extend to within a short distance of the middle of each half bearing. The central parts, where the oil pressures are highest, are thus quite unbroken, so that maximum resistance from the oil film is obtained, while the arrangement of the grooves following the direction of rotation leaves the oil films on each side unbroken. The grooves do not cut across the direction of rotation. They introduce the oil into the bearing, but leave the distribution in the bearing to the rotation of the shaft and to surface tension. The primary purpose of the grooves in the face of each bearing is to ensure an adequate supply of oil to the big-end bearings through the crankshaft. From each bearing one or two inclined holes drilled in the crankshaft conduct oil to adjacent crank pins. Each groove coincides lengthwise of the shaft with the entrance to an oil hole, and oil is thus forced under pressure to a big-end bearing during rotation through a certain angle sufficient to include a period when the crank pin in the neighbourhood of the oil outlet is not heavily loaded. The big-end bearing has no grooves so that the maximum resistance due to surface tension can be developed.

Each crank pin is drilled lengthwise, and the end of the hole closed. The diagonal oil holes discharge into the reservoir so formed, and a separate outlet is provided to the big-end bearing, the discharge being assisted by centrifugal force.

Each half of the bearing is held against rotation by a spigot *e*, the smaller diameter of which is screwed into the bearing.

At the rear end special provision is always made to prevent oil creeping along the shaft from the rear end of the bearing on to the flywheel. In this engine, in addition to the usual centrifugal thrower flange on the chankshaft, oil is collected in a circumferential groove near the rear edge of the bearing, and flows back to the sump through the holes *g*, *h* in the bearing and cap respectively.

In view of the magnitude of the forces it will be obvious that the bearing cap must be very rigid and not merely strong enough. The illustrations of the bearing caps show how this requirement is met.

Reference to the tabulated values of the forces on the bearings of the compression-ignition engines show that the maximum and mean bearing pressures are substantially higher than in the petrol engine. The design

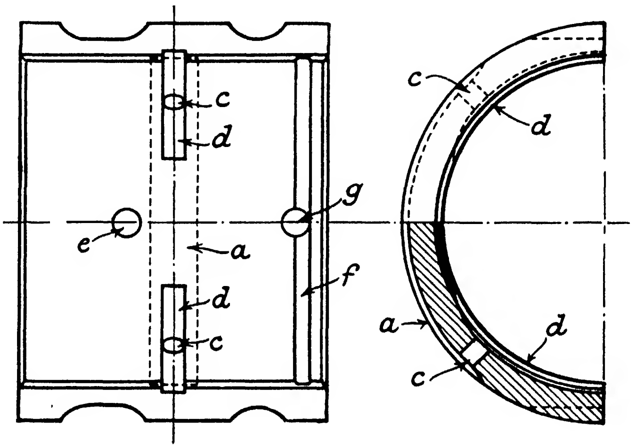


FIG. 18. REAR CRANKSHAFT BEARING (No. 7), WITH STEEL SHELL FOR A.E.C. COMPRESSION-IGNITION ENGINE

of the bearings has to be modified accordingly, and the rear half bearing together with its bearing cap are shown in Figs. 18 and 19. In this case the shell of the bearing is of steel, which is stiffer than bronze, and avoids the risk of the thin white metal facing breaking away. Also the bearing cap is of substantially larger dimensions and, therefore, more rigid. The caps in both petrol and oil engines are secured by bolts, extending right up through the cylinder casting to obtain greater rigidity and avoid distortion due to the heavy pressures. They

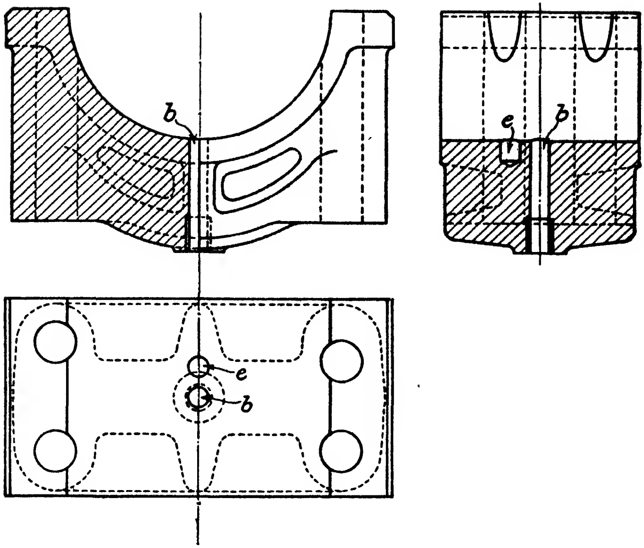


FIG. 19. HEAVY DESIGN OF REAR BEARING CUP FOR A.E.C. COMPRESSION-IGNITION ENGINE

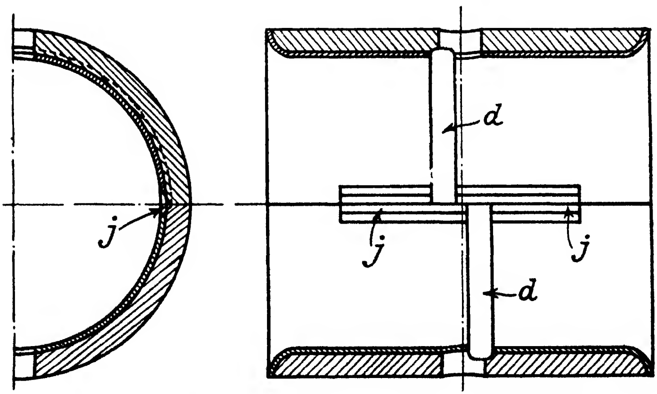


FIG. 20. PART OF DORMAN UNFLANGED BEARINGS, WITH WHITE METAL FACES

are only secured by nuts on studs screwed up into the main casing where through bolts cannot be used.

A pair of rear crankshaft bearing shells with white metal faces in contact with one another, as manufactured by Messrs. W. H. Dorman & Co., Ltd., is

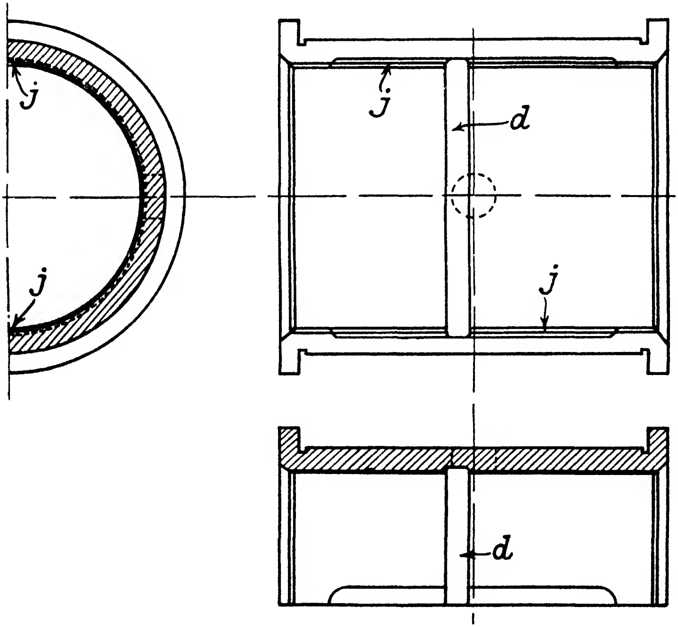


FIG. 21. DORMAN FLANGED HALF-BEARING

shown in Fig. 20. In this bearing the very common practice is followed of chamfering the meeting edges of the two half-bearings as shown at *j*; the two chamfers form in effect a transverse oil groove which connect with the circumferential oil grooves *d*, but must, however, not extend as far as the edge of the bearing to avoid loss of oil. These chamfers should not form a sharp edge with the bearing face.

Another design of Dorman crankshaft bearing with flanges is shown in Fig. 21, the corners behind the flanges being undercut.

The question of alignment with a number of bearings is of much importance. In an engine the bearing caps should be bolted up, and the opening in all the bearings bored in position. The half-bearings must then be bedded down properly into the caps and main castings with their faces flush. Very little bedding down will be necessary if the engaging surfaces have been finished within close limits, which must be arranged to avoid any possibility of jamming on the sides.

Provided that the white metal surfaces have been finished within close limits, and are connected with the outer surfaces of the shells, no further work may be necessary, but the linings may be left slightly full, and a long reamer will then ensure accuracy of diameter and correct alignment.

BALL AND ROLLER BEARINGS

Everybody is familiar with the great contrast between the resistance to movement when one surface slides over another, and when rollers or balls are interposed between them. Typical values for the coefficient of friction in a well designed ball bearing and in a properly lubricated and nicely run-in plain bearing are respectively ·0015 and ·015. The friction which does exist in ball and roller bearings is moreover of a totally different character. It is, in fact, not friction as the term is commonly understood, but is mainly the resistance due to deformation of the rolling surfaces, and is accompanied by a kind of hysteresis effect which depends upon speed.

When a ball or roller rolls over one of the races, the material of the race is forced up rather more in front (A) than behind (B), as shown in Fig. 22; and this happens however hard the material. The drawing, of course, greatly exaggerates the effect. The ball is at the same time slightly flattened, and the intensity of the crushing stresses may be very high. The actual areas of the hardened steel surfaces in contact are so small that the usual terms point and line contact for ball and roller bearings respectively are not inappropriate, although really incorrect. The shapes and dimensions of the contact areas and the crushing stresses depend upon the closeness with which the contour of the races fits the balls or rollers. The typical contact areas for different types of bearings shown in Fig. 23 may be circular, elliptical, or rectangular.

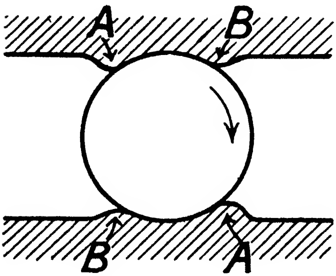


FIG. 22. EXAGGERATED DEFORMATION IN A BALL BEARING

In all types of ball or roller bearing it is essential that the balls or rollers should roll accurately without appreciable slip on any part of the race, and with the minimum of spin or rotation about the point of contact, so-called. The geometrical conditions for pure rolling are shown in connection with various types in Fig. 24, the generating lines for the rolling line of contact being shown dotted. It will be noticed that these generators form a tangent or a chord, according whether there are one or two lines of contact on each race. Also the generators are either parallel to one another or meet on the axis of the bearing. A certain amount of spin is inevitable with designs *B*, *C*, and *D*. For automobile purposes designs *A*, *B*, *C*, and *D* are seldom employed, and then only for the smallest and lightest applications.

It has not so far been found possible to build up a rational theory of operation for ball and roller bearings

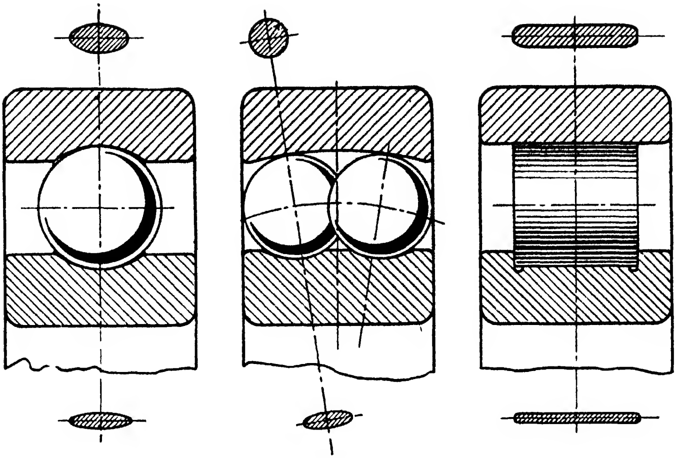


FIG. 23. TYPICAL PRESSURE AREAS

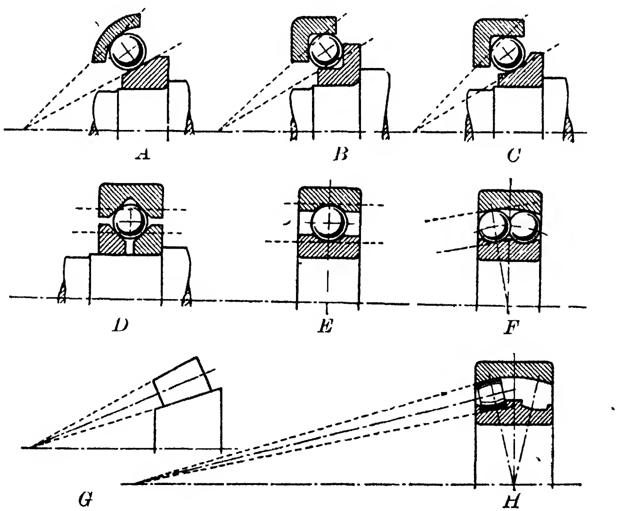


FIG. 24. GENERATING LINES

dependent upon contact areas or stresses in the material, the most useful formulae and data being of an empirical character and based upon endurance tests. But it may be useful to consider the manner in which a hard steel ball fails when loaded to the crushing point on a hard steel surface. When the ball is engaged by a plane or nearly plane surface, it is finally split more or less diametrically by the wedge-like action of a small cone, the base of which is formed by the contact circle, as shown in Fig. 25. If the contact surface is elliptical, fracture occurs across the minor axis. A simple failure of this kind due to excessive loading rarely occurs in practice. Failure of one of the races from flaking or other surface defect, or even surface cracking, is a more probable result.

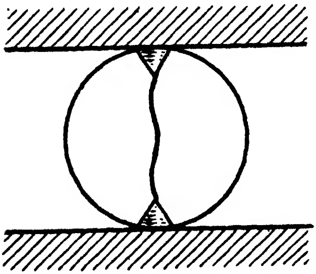


FIG. 25. HOW A BALL IS LIKELY TO FAIL WHEN LOADED EXCESSIVELY

The utmost care is taken by ball and roller bearing manufacturers in the selection of high grade steel practically free from even the minutest quantities of slag which renders the material porous or discontinuous. The loads are so concentrated that the steel must be as hard as possible. Both the balls and races are hardened by the usual methods of heating and plunging under carefully regulated conditions of temperature. They are then annealed only sufficiently to remove any internal stresses, but are not heated to such a high temperature that the hardness is appreciably reduced. For example, the Skefko Company harden their medium-sized ring in oil at a temperature of about 820° C., and temper them in oil at about 150° C. for one hour.

The smaller size balls, manufactured by the Skefko

Company are cold headed from wire, rough ground, annealed to move strain set up in the cold-pressing operation, heated up in rotary furnaces, hardened in water, and afterwards oil tempered. Subsequent operations consists of a series of grinding operations, firstly between rotating grinding wheels and stationary grooved cast iron discs, and subsequently by tumbling operations in which the balls are in contact with oil charged with a fine abrasive. They are then dried, and tumbled with small scraps of leather to impart a fine polish, after which they are subjected to ocular inspection and graded in steps of one-thousandth of a millimetre.

To obtain a high degree of hardness a high carbon steel with up to 1·2 per cent of carbon is used. For the smaller sizes of balls, say, up to $\frac{3}{8}$ in., a plain carbon steel may be used, but with this material the hardening does not go very deep. For larger balls and for the races the addition of chromium secures a sufficient depth of hardening. Also the chromium renders the steel much more resistant to fatigue which is in general the determining factor in most cases of bearing failure. Examples of the composition of three different kinds of steel, as used by the Skefko Ball Bearing Company, are given in Table VIII. The figures in each case indicate the limits or tolerances for each component, and are expressed in percentages.

TABLE VIII

Type of Steel	(1)	(2)	(3)
Carbon	·95 – 1·05	1·00 – 1·10	·95 – 1·05
Chromium	1·4 – 1·65	·40 – ·60	1·00 – 1·25
Molybdenum		·30 – ·40	·30 – ·40
Manganese	·15 – ·35	·25 – ·35	·25 – ·35
Silicon	·25 – ·35	·25 – ·35	·25 – ·35
Phosphorus (less than)	·025	·025	·025
Sulphur (less than)	·020	·020	·020

Phosphorus and sulphur are always undesirable impurities, the quantities of which must be kept to a minimum.

The engineer or designer who uses ball or roller bearings is concerned primarily to know what size and type he should obtain to carry certain loads at certain speeds under certain conditions. He cannot simply calculate stresses from given loads, allow a factor of safety, and hence determine the size, number of balls, etc. He requires a bearing which will under the specified conditions run indefinitely or at any rate for a number of years without causing trouble. For this he must rely upon the manufacturers who, working upon certain basic principles, have accumulated knowledge and experience which enable them to assess with a considerable degree of accuracy what each size and type of bearing is capable of doing. While each bearing must be large enough for its purpose, it must be no more than just large enough on account of cost, weight, and the room which it occupies in the machine.

The exact detailed design of the bearing is of primary importance in determining its load carrying capacity and endurance. The basic principles covering the main proportions of the several components have been investigated very fully by Professor Stribeck in Germany, and by Professor Goodman in this country. Professor Stribeck's researches were initiated in 1898, and the question of deformation in relation to loading was first considered by him. He found that a minute amount of permanent set occurred at the contact surfaces for loads which were quite small, and were, in fact, well below the clearly permissible working loads. The difficulties in the way of development of a rational theory based upon stress and strain rendered it necessary for him to proceed upon more or less empirical lines. It is practically impossible to determine exactly the point at which permanent set occurs. In the usual stress-strain diagrams applied to specimens with carefully prepared parallel lengths the yield point is generally fairly well defined by an abrupt departure from the straight-line which expresses the relation—

$$\frac{\text{stress}}{\text{strain}} = \text{constant}$$

But when a ball is compressed against a race no such definite change can be detected. It would seem that as soon as one particle yields, the area of surface in contact increases, and a ring of additional particles comes into action to resist further deformation, and so on as the pressure is further increased.

In the formulae developed by Stribeck and Goodman—

P = the maximum permissible working load in pounds.

d = the diameter of the ball in inches.

D = the diameter of the path of the centres of the balls in inches.

m = the number of balls in the bearing.

n = the revolutions per minute of the shaft.

x = a coefficient depending upon n.

k and c are constants.

The simplest basic formula suggested by Stribeck was as follows—

$$P = K\,d^2$$

This formula was further elaborated as follows—

$$P = 12{\cdot}8\,x\,m\,d^2$$

The value of x depends upon the speed, a number of corresponding values of x and n being given below—

n . .	1	300	500	1,000	2,000	3,000	4,000	5,000	10,000
x . .	48·4	24·2	10·9	15·62	12·1	9·68	8·58	7·7	5·06

The main results of the researches carried out by Professor Goodman were expressed in the formula—

$$P = \frac{k\,m\,d^3}{n\,D\,c\,d}$$

The constants k and c used in this formula have the following values—

Type of Bearing	k	c
Radial bearings—hollow races, best quality,	2,500,000	2,000
flat or ungrooved races	1,000,000	2,000
Thrust bearings—hollow races, best quality,	1,250,000	200
flat or ungrooved races	500,000	200

The two formulae do not agree exactly, but both show that the load-carrying capacity of the bearing is governed by the diameter and the number of balls and the speed of running.

In connection with this early work of Stribeck and Goodman, Messrs. Ransomes and Marles have published curves, Fig. 26, showing the load carrying capacity of one of their light type radial bearings at various speeds, the inside and outside diameters of which are respectively 50 and 90 mm., while the width is 20 mm. The safe working loads indicated by the curve for the *R* and *M* bearing are substantially above the loads at corresponding speeds suggested by the earlier investigations. This difference results chiefly from developments in (1) quality of steel, (2) heat treatment to secure hardness, resilience, and fatigue-resisting properties, and (3) detailed design of the parts to employ the maximum number of balls of the greatest possible diameter, and also to secure the best relation between the radius of the balls and the curvature of the races.

Ball and roller bearing manufacturers either publish

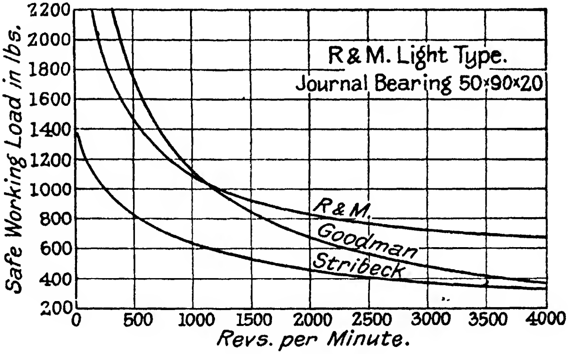


Fig. 26. Comparative Load Carrying Capacities

tables giving the steady load capacity of their bearings at various speeds, or will give the necessary information on request. Figures of this kind are based upon the result of much research and experiment, and are largely dependent upon the element of fatigue. They represent a scale of loading based upon a long if not an infinite life, the load varying as some inverse function of the speed.

The bearing manufacturers encourage inquiries from users as to the sizes and types suitable for any particular set of conditions, and the designer or other user should, therefore, when in any doubt submit his problem to them, and avail himself of the mass of information which they have necessarily accumulated. The industry is a progressive one; recent advances are considerable, and further progress may be anticipated, so that any published figures are always subject to revision, as further experience is obtained or as construction and methods of manufacture progress.

The table opposite relates to Ransome and Marles' ordinary medium radial or journal bearings with filling slots. A range of sizes suitable for automobile practice

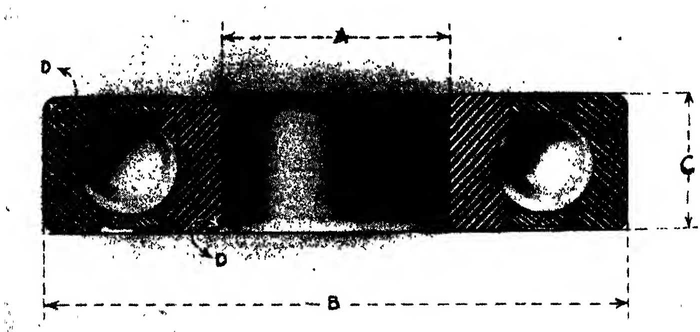


FIG. 27. LEADING DIMENSIONS OF RADIAL BALL BEARING

is included, but only as an illustration of the kind of information available, since very comprehensive tables of loads have been prepared for various other sizes and types. The leading dimensions as indicated in Fig. 27 are given for the various sizes in Table IX.

The rated figures apply to steady service without shocks, but where sudden variations of the load occur, amounting, perhaps, as in the case of front axle journals, to shocks, an additional margin of safety should be used. Hub bearings require a factor of 2·5 or 3 over the rated capacity; the calculated load being thus substantially less than that given in the tables. Gear-box bearings, on the other hand, may be greatly overloaded on the

TABLE IX

SAFE WORKING LOAD IN LB.

A	*B*	*C*	*D*	Approx. weight in lb.	Revolutions per Minute				
					600	1,000	1,500	2,000	4.000
$\frac{1}{2}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{16}$	0.25	515	430	365	320	250
$\frac{5}{8}$	$1\frac{13}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	0·31	570	490	435	310	280
$\frac{3}{4}$	2	$\frac{11}{16}$	$\frac{1}{16}$	0·41	685	565	490	430	335
$\frac{7}{8}$	$2\frac{1}{4}$	$\frac{11}{16}$	$\frac{1}{16}$	0·52	840	680	580	500	380
1	$2\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{32}$	0·69	1,000	850	735	660	510
$1\frac{1}{8}$	$2\frac{13}{16}$	$\frac{13}{16}$	$\frac{3}{32}$	0·93	1,300	1,095	950	860	630
$1\frac{1}{4}$	$3\frac{1}{8}$	$\frac{7}{8}$	$\frac{3}{32}$	1·22	1,710	1,425	1,230	1,100	780
$1\frac{3}{8}$	3	$\frac{7}{8}$	$\frac{3}{32}$	1·60	1.870	1,600	1,420	1,280	940
$1\frac{1}{2}$	$3\frac{3}{4}$	$\frac{15}{16}$	$\frac{3}{32}$	1·95	2.075	1,750	1,520	1,375	1,000
$1\frac{5}{8}$	4	$\frac{15}{16}$	$\frac{3}{32}$	2·22	2,320	2,000	1,725	1,530	1,100
$1\frac{3}{4}$	$4\frac{1}{4}$	$1\frac{1}{16}$	$\frac{3}{32}$	2·84	2,500	2,100	1,800	1,600	1,175
$1\frac{7}{8}$	$4\frac{1}{2}$	$1\frac{1}{16}$	$\frac{3}{32}$	3·22	2,700	2,230	1,920	1,700	1,260
2	$4\frac{1}{2}$	$1\frac{1}{16}$	$\frac{3}{32}$	3·10	3.060	2,520	2,100	1,850	1,350
$2\frac{1}{4}$	5	$1\frac{1}{4}$	$\frac{1}{8}$	4·40	3,800	3,140	2,600	2,250	1,600
$2\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{8}$	5·39	4,800	4,000	3,250	2,850	2,000
$2\frac{3}{4}$	$6\frac{1}{4}$	$1\frac{3}{8}$	$\frac{1}{8}$	7·83	5,700	4,450	3,500	3,000	2,200
3	7	$1\frac{9}{16}$	$\frac{5}{32}$	11·38	6,600	5,200	4,000	3,500	2.400
$3\frac{1}{4}$	$7\frac{1}{2}$	$1\frac{9}{16}$	$\frac{5}{32}$	13·13	7.900	6,300	4,800	4,000	3,100
$3\frac{3}{8}$	$7\frac{1}{2}$	$1\frac{9}{16}$	$\frac{5}{32}$	12·80	7,900	6,300	4,800	4,000	3,100
$3\frac{1}{2}$	$8\frac{1}{8}$	$1\frac{3}{4}$	$\frac{5}{32}$	17·22	9,400	7,400	5,700	4,900	3,600
$3\frac{3}{4}$	$8\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{32}$	18·52	9,400	7,400	5,700	4,900	3,600
4	$8\frac{1}{2}$	$1\frac{3}{4}$	$\frac{5}{32}$	19·09	9,400	7,400	5,700	4,900	3,600

The loads and speeds given are for general purposes only. In some cases the figure stated can be substantially increased. (R. & M.)

lower gears quite safely, since most of the running is done on top gear with something less than full engine torque. The bearings may be selected on the basis of top gear loading, and a factor of 2 or 2·5 over rated loads then adopted.

The various considerations which affect the bearing capacity will be further considered in connection with the detailed description of the many types in use.

Balls, rollers, and races are always ground and polished finely with the greatest care and with the smallest departure possible from the proper dimensions. It is, however, impossible to obtain exact accuracy, and very fine tolerances or limits are therefore decided upon and the components, when finally measured up, must fall within these limits. The limits decided upon as commercially possible and suitable by all manufacturers are in remarkably close agreement. They are commonly expressed as plus or minus decimal fractions above or below the stated dimension to indicate the extent to which the finished component may be large or small without, however, going beyond permissible limits.

Examples of tolerances allowed in ball and roller journal bearings, as manufactured by the Ransome and Marles Bearing Co., Ltd., are given in Table X.

Table X gives the manufacturer's tolerances for all sizes likely to be used on motor vehicles. It will be seen in most cases that the possible range of variation in any component amounts only to ·005 in. On other types of bearing similar tolerances are adopted.

The tolerances on these two dimensions which have been considered, namely, the outside and the inside diameters of the races, should, from the point of view of the user, be as small as possible to secure interchange ability. The manufacturer, on the other hand, would like them to be as large as possible so as to reduce the necessity for accurate and, therefore, costly workmanship, and to diminish the proportion of rejects. The limits now generally adopted effect a compromise between these two conflicting desires.

The above tolerances affect the user as well as the manufacturer, but certain other tolerances which call for special attention on the part of the manufacturer have also to be considered. The track or rolling surface in every race must not only be circular but must be

TABLE X

Inch Sizes

	Size	Tolerance
Bores	Up to and including 6 in.	·0002 in. to – ·0003 in.
	Above 6 in. and including 12 in.	·0002 in. to – ·0005 in.
Outside diameters	Under 2 in.	– ·0003 in. to – ·0008 in.
	2 in. and under 3 in.	– ·0005 in. to – ·0010 in.
	3 in. and under 5 in.	– ·0008 in. to – ·0013 in.
	5 in. and under 12 in.	·0013 in. to – ·0018 in.
Widths	All sizes	·0000 in. to – ·0020 in.

Metric Sizes

	Size	Tolerance
Bores	Up to and including 80 mm.	·005 mm. to – ·010 mm.
	Above 80 mm. and including 200 mm.	·005 mm. to – ·013 mm.
Outside diameters	Up to and including 75 mm.	·000 mm. to – ·013 mm.
	Above 75 mm. and including 150 mm.	·000 mm. to – ·020 mm.
	Above 150 mm. and including 420 mm.	·000 mm. to – ·025 mm.
Widths	All sizes	·000 mm. to – ·050 mm.

concentric with the other surfaces. Accuracy of this dimension within ·0001 in. is the kind of limit desired.

In order that each ball in a ball bearing shall take its proper share of the load, it is essential that it shall be spherical within such very fine limits that it may be described as a perfect sphere. In addition it is desirable that the diameter should not vary by more than ·00002 in.

The diameters of the rollers in any bearing should be very nearly constant—for example, within ·00005 in., while they should be perfectly cylindrical. The lengths of rollers should also vary very little—for example, within ·0002.

The manufacturers tolerances just mentioned are those adopted by the Ransome and Marles Bearing Co.

When the balls have been finally polished and their spherical characteristics verified they are graded according to their diameters, being arranged, for example, within three groups, so that the diameters of the balls in any one group will be within ·00002 of one another. The dimensions of the different groups may differ from one another by more than the tolerances mentioned, so that one bearing may be filled with slightly large balls and another with slightly small balls.

The races will, of course, have similar small variations from the specified dimensions, and in many of these variations, the races and the groups of balls are paired up and assembled by a process of selection or "selective assembly." For example, a slightly plus inner race will be paired up with a slightly minus set of balls to ensure the proper final fit of the bearing without slack or binding. This selective process is, however, so carried out that the bearings may have slightly different degrees of freedom or diametrical clearance. These grades are commonly indicated by lightly stamped marks on the sides of the races: 0, 00, and 000. The closest fit is marked 0, the medium fit 00, and the slackest 000. The three classes of fit may be designated by the letters *X*, *Y*, and *Z*, and are also known as one dot, two dot, and three dot fits. The diametrical clearance in the case of the 0 fit may be from ·0001 to ·0006, according to size and type. When ordering races, attention should be given to these markings which are important in view of conditions of working and of installation. For example, where a bearing is likely to become heated, a slack fit 000 is generally desirable, while in cases where a tight fit is important as in motor-car axles, the closest fit 0 should be specified. In most cases the medium fit 00 is suitable.

The question as to which fit of bearing is best is further associated with the "interference" fit of the races with the shaft or housing. One of the factors of considerable importance in the life of a ball or roller bearing is the choice of tolerances in all the interfitting parts and the section of bearings having the appropriate fit between the rolling elements and the races. In practically all applications of ball and roller bearings in motor vehicles, an attempt is made to secure such a fit that the races cannot creep on or in their shaft or housing, while there is only the slightest diametrical clearance when the assembly has been completed, allowance being made, as mentioned previously, for any unequal expansion due to the temperature of the shaft increasing to any substantial extent. It will be clear that the diametrical clearance will be reduced and possibly absorbed as the result of tight interference fits between the races and the shaft or housing.

All manufacturers publish appropriate figures for the fit of the inner race on the shaft. These figures must be such as to ensure that the inner ring cannot turn on the shaft under any working conditions. This condition requires that the shaft shall be finished so that it is definitely slightly larger than the bore of the race, and the interference fit thus obtained will prevent creep and, at the same time, reduce the diametrical clearance in the bearing. The same considerations apply to the fit of the outer race in its housing, but in races of the larger diameter the interference need not be quite as much. In short, the rotating race should be a light driving fit, and the non-rotating race a good push fit.

The statement is sometimes made that a slight creep should be permitted in the case of the non-rotating elements in order that the wear and stress shall not be concentrated on the same part of the track. If, however, such creep is definitely provided for, the friction resulting from the movement will wear the surface of the shaft or housing which are generally softer than the races. The movement will also cause a kind of surface flow of the metal resulting in slackness and increased wear and movement, the effect being rapidly cumulative and possibly resulting in early failure of the bearing. Apart from these considerations much greater accuracy with individual fitting and consequent expense is required to obtain the exact fit, and the bearings could hardly be interchangeable.

It is estimated that 60 to 70 per cent of the interference provided between the shaft and the inner race is transmitted through to the rolling surfaces, and thus increases the tightness of the bearing. The necessity for accurate gauging when fitting bearings will thus be obvious.

When selecting bearings for any particular purpose, the user finds available an almost bewildering range of types and sizes, but the field rapidly narrows on a careful examination of the problem. It is first necessary to investigate thoroughly the conditions under which the bearings are required to run, with particular reference to the speeds and loads, the latter including an investigation as to the direction of the load, and whether it is constant or various with speed, or in accordance with any other conditions. This question of loading and conditions is dependent upon the design of the various parts of the vehicle which are dealt with in detail under other sections.

This investigation will probably narrow the choice of a suitable type of bearing cons'derably, and consideration may then be given to questions of mounting, protecting, and lubrication, all of which are essential factors to ultimate success.

The ordinary radial type bearing, which has been so

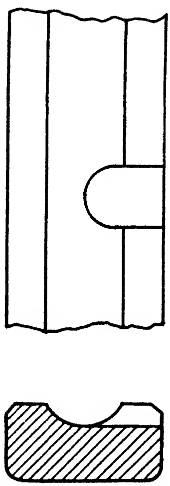


FIG. 28. OLD TYPE NOTCHED BEARING RACE

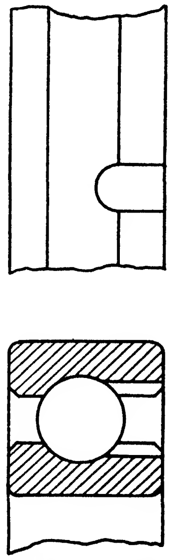


FIG. 29. MODERN NOTCHED BEARING RACES

widely used, was at one time supposed to be capable of dealing with radial loads only, but of recent years its capacity for dealing with substantial thrust loads. instead of, or in addition to, radial loads, has become recognized.

Two main types are manufactured, one with and the other without filling slots, known also as notched and notchless. The latter is also sometimes known as the deep groove type. The early radial bearings were provided with notches in both the inner and outer races, and these notches extended as far as the middle of the tracks in the races, as shown in Fig. 28. When these notches coincided, the last few balls could be readily inserted into the races, and a closing piece was sometimes introduced. The principal weakness in this construction was the interruption of the track by the tips of the notches, and filling pieces rarely fitted sufficiently

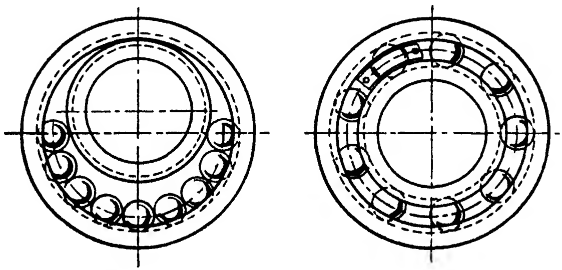


FIG. 30. METHOD OF ASSEMBLING NON-FILLING SLOT BEARING

well to surmount the difficulty. The next step resulted in the modern highly efficient notched type, the notches being carried only partly into the track, as shown in Fig. 29, so that an uninterrupted run is provided for the balls. A number of the balls can be put into position without using the notches, but the last ones can be forced in, the balls and more particularly the races being sufficiently elastic for this to be done. The utmost advantage is taken of the elastic properties of the components, so as to enable the filling notch to be as shallow as possible, and it is possible to exert considerable force without risk of damage. No filling piece is necessary. A large number of balls can be inserted, it being only necessary to provide sufficient gaps between them to accommodate the cage. Heavy radial loads may thus be carried.

In the notchless type, which is of more recent introduction, the balls are arranged when filling as close as possible together, as shown in Fig. 30, their centres extending not more than half-way round the circum-

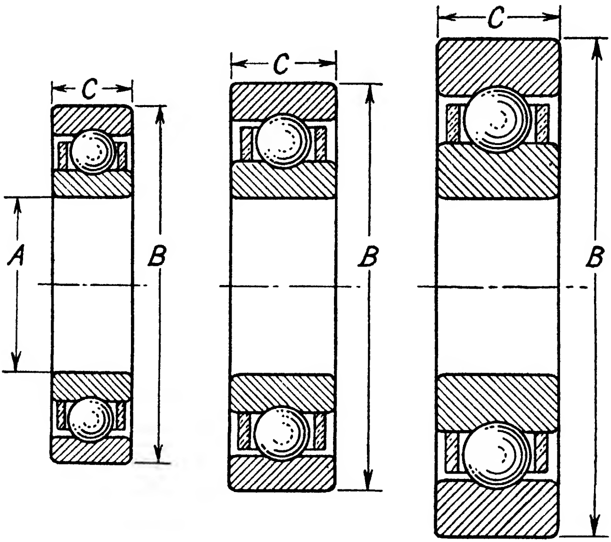


Fig. 31

Relative proportion of light, medium, and heavy ball bearings all fitting the same size of shaft

ference. It is this condition which limits the number of balls that may be inserted. With more balls the inner race cannot be placed in position. The balls are afterwards spread out and spaced properly by the cage. It will be obvious that fewer balls can be inserted than is possible in the notched type, the gaps between the balls being about equal to their diameter. Theoretically, a notched bearing will have a greater radial capacity, but in practice this is of little importance, mainly due to the fact that the notchless type can transmit heavy thrust loads in either direction. Further, experience has shown that the notched bearing is less suited for exceptionally high speeds.

Fig. 31 shows sectional views of three bearings, all

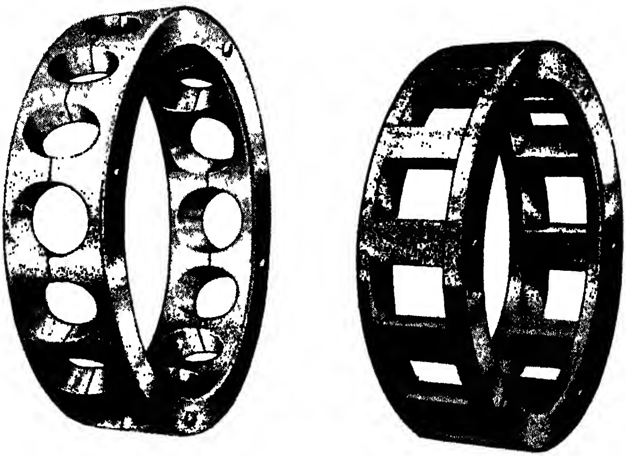


FIG. 32 R AND M. TWO-PART CAGES FOR RADIAL BALL BEARINGS AND PARALLEL ROLLER BEARINGS

having the same internal diameter and representing the light, medium, and heavy patterns made by Ransome and Marles. These are made in a wide range of sizes by steps of $\frac{1}{8}$ in. and $\frac{1}{4}$ in. from a shaft diameter of $\frac{1}{2}$ in. to $6\frac{1}{2}$ in. Table XI extracted from the published tables gives the recommended safe working loads in pounds at various speeds for bearings of each pattern fitted on shafts 44 mm. in diameter. The leading dimensions are also included.

TABLE XI

A	*B*	*C*	Approx. weight in lb.	Revolutions Per Minute				
				600	1,000	1,500	2,000	4,000
40	80	18	0·91	1,080	890	780	700	550
40	90	23	1·58	1,870	1,550	1,350	1,210	950
40	110	27	3·13	2,800	2,320	2,040	1,840	1,400

Many cages in this type of bearing, as well as for roller bearings, are made in two parts as shown in Fig. 32, the parts being riveted together in position. The material used is brass or white metal. If the cage is made a smooth running fit on the shoulders of the inner race, the balls do not have to locate it, and therefore run freely. This avoids the possibility of excessive rubbing between the balls and the cage which might result in what is called "smearing," the ball developing fine hard cracks and flaking unless the cage gives out first.

In some specially light duty bearings, the cage is dispensed with, but only for comparatively low speeds. A larger number of balls can be used, but there is bound to be a certain amount of grinding between them. Moreover, when two balls make contact, their meeting surfaces are rotating in opposite directions at a very high speed, and considerable heat is generated. The cage is, in fact, a necessary evil.

The built-up type of cage already referred to has proved reliable when carefully made, but numbers of separate parts are always regarded as introducing risk. In view of this, the one piece pressed steel cage has been developed by most manufacturers for all types of bearing, and this tendency has also been influenced by the cheapness, accuracy, and uniformity which can be secured by production methods utilizing pressed tools. The built-up construction is, however, often preferred for high-speed work and vibration, and is, moreover, appropriately used in the larger sizes where the numbers manufactured do not warrant the expense of pressed tools.

When it was recognized that the ordinary single row radial bearing was also capable of dealing with substantial thrust loads in addition to, or instead of, radial loads just as well as, and in certain cases better than, the self-aligning types, experiments were carried out to determine the relation between the two, and the results were expressed in the following formulae—

If W = rated capacity in lb
R = actual radial load
A = actual thrust load
N = speed in revolution per minute

Then for bearings without filling slots

$$W = R\,(\cdot 0004\,N + 0\cdot 8)\,A$$

and for bearings with filling slots

$$W = R\,(\cdot 0004\,N + 1\cdot 4)\,A$$

Ransome and Marles use for notched bearings a simpler formula giving similar results

$$A = \tfrac{1}{2}\,(W - R)$$

The value of W obtained from this firm's tables of safe working loads depends upon speed.

As an example, consider a medium radial bearing for a shaft 40 mm. diameter. The working load according to Table XI above is 1,210 lb. at 2,000 r.p.m. Suppose the actual radial load is 500 lb., then the safe thrust load $A = \frac{1}{2}\,(1210 - 500) = 355$ lb.

Modern notchless bearings subjected to steady thrust loads only can, however, carry from 75 to 100 per cent of their rated radial loads, provided that the maximum

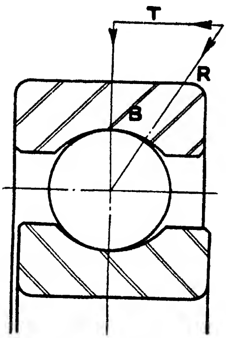


FIG. 33. R. AND M. BEARING WITH LARGE THRUST CAPACITY

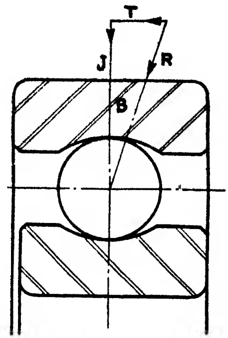


FIG. 34. R. and M. BEARING WITH SMALL THRUST CAPACITY PRODUCING SEVERE WEDGING EFFECT

diametrical freedom is allowed, while the sum of combined thrust and radial loads may equal the rated radial load. Intermittent loads or shocks will reduce the safe capacity, a factor of 2 being sufficient for most conditions, but in such cases the manufacturers might well be consulted.

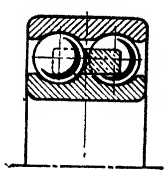


FIG. 35. DOUBLE ROW BALL BEARING

If a bearing is to carry substantial axial thrust, the tracks must be ground to a radius which is not much greater than that of the balls; for example, one-third. That difference is sufficient to avoid binding of the balls and to give a good angle of inclination when thrust loads are applied. This condition is shown in Fig. 33, where J represents the radial load, T the thrust load, and R the direction of the resultant thrust on the bearing. Comparing these conditions with the large radii tracks of the bearing, Fig. 34, the thrust capacity T in the latter is much reduced, as the angle B of the resultant force on the bearing is so small that a severe wedging effect is produced. However close fitting the ball and tracks

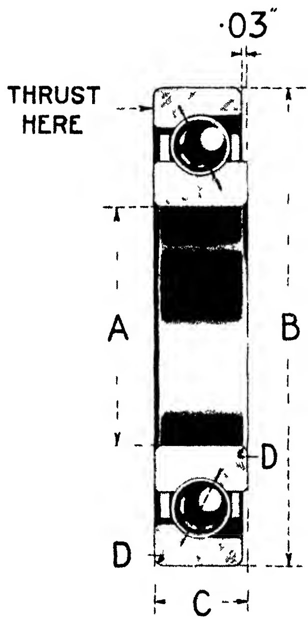


Fig. 36. R. and M Combined Radial and One-way Thrust Bearing

may be, it is clear that a thrust component always produces some wedging effect, and that slack must be provided to allow the angle B to attain a sufficient value. For this reason the three dot fitting is always best for any but the lightest thrust loads.

The inner race suffers in use more than the outer race owing to its shorter length measured circumferentially, failure being always due to fatigue if the bearing has been properly looked after. When the outer race forms the rotating member, as in front wheel hubs, this effect is even more pronounced, since all the wear occurs on the under side of the inner race.

Double row bearings, Fig. 35, have a capacity which is approximately 75 per cent greater than single row bearings of the same dimensions; or the capacity may be even more under some conditions.

When the thrust is in one direction only, the design

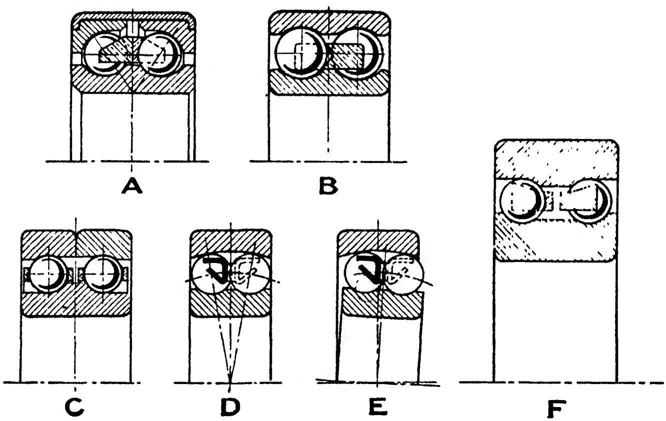


Fig. 37. SKF Self-aligning Double Row Ball Bearing

shown in Fig. 36 is highly effective, due to improved race contour, and also the fact that the outer race on one side of the track may be cut away to allow a larger number of balls to be inserted. The dimensions are such that the balls must be sprung into position when assembling. It may be used in rear axles of the bevel or worm type.

The self-aligning double row double thrust bearing illustrated in Fig. 37 was originally introduced by the Skefko Company. The outer race is spherical so that the bearing will work perfectly well even when the inner and outer races are substantially out of alignment. Each row of balls resists thrust in one direction only. The outer tracks do not fit the balls very well, but this is to a considerable extent counteracted by the fact

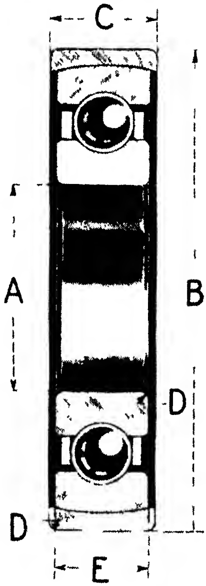


FIG. 38. R. AND M. WITH SELF-ALIGNING OUTER RING

mentioned previously, that the inner race always suffers more than the outer race due to its shorter circumferential length. Pressed steel or built-up cages may be used, and the balls are easily sprung into the recesses in the cage, the whole bearing being readily assembled. A large number of balls arranged close together may be used.

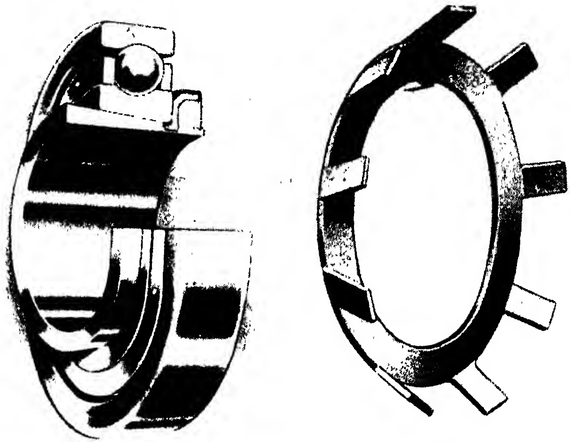


FIG. 39. R. AND M. ADAPTER BALL BEARING WITH LOCKING WASHER

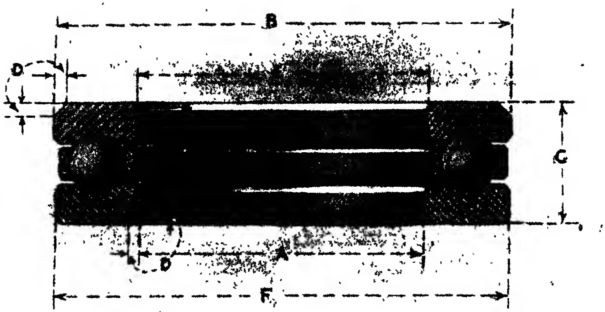


FIG. 40. R. AND M. DOUBLE THRUST BEARING

The ordinary radial type bearing may be made self-aligning by providing an outer ring having a spherical fit on the outer race. This is shown in Fig. 38.

To facilitate the fitting of the inner race on to a shaft in various types of bearing, adapters are provided as shown in Fig. 39, in connection with a Ransome and Marles single row radial bearing. The nut tightens the conical surfaces on the inner race and the adapter, and it is locked when set by turned over tongues on a light sheet metal locking washer.

The ordinary thrust bearing, Fig. 40, is often made

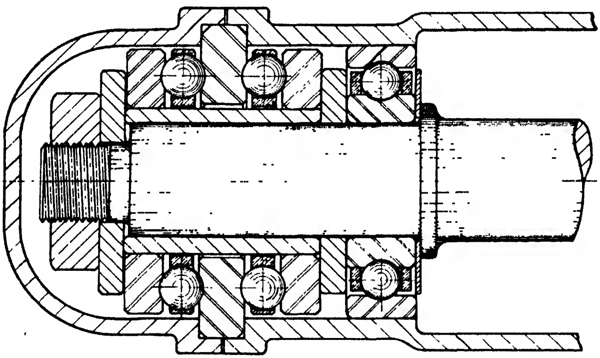


FIG. 41. R. and M. RADIAL AND DOUBLE THRUST BEARINGS

with the dimensions *A* and *F* of one race slightly less than the corresponding dimensions *E* and *B* of the other race. The smaller race may thus register on a shaft and the larger in a housing. One or both races may have spherical outer faces to secure alignment. Fig. 41 shows the application of double opposed thrust races on a worm shaft.

This type of thrust bearing is unsatisfactory at high speeds owing to centrifugal force which causes pressure against the outer part of the race, and leads to excessive spin of the balls. Such bearings wear better at speed when there is a substantial thrust load to keep them in position in the races. When two are used, the idle one will in general wear more than the loaded one for the reasons given. Careful adjustment is, therefore, necessary to avoid slack without introducing tightness. The pure thrust bearing has in many cases been superseded

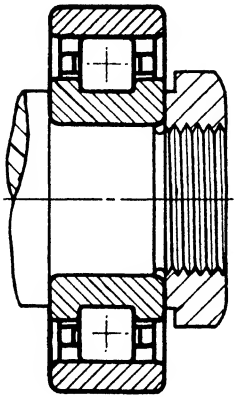


Fig. 43. Hoffman Simple Radial Roller Bearing

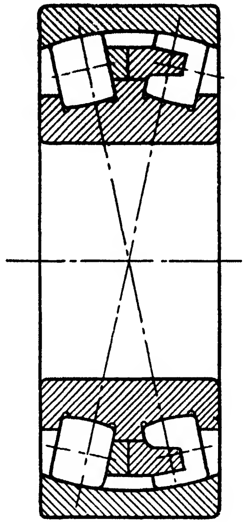


Fig. 44. Skefko Self-aligning Double Roller Bearing with Barrel Shaped Rollers to Fit Over Spherical Race

by other types adapted to deal with both radial and thrust loads.

Many of the different types of ball bearing have their counterpart in roller bearings which may be fitted with cylindrical, conical, or barrel shaped rollers.

The simple radial type, Fig. 43, has parallel rollers, and is not adapted to resist end thrust, since the track in the outer race is a plain cylinder. The outer track as well as the inner track may, however, be grooved when resistance to end thrust as well as radial thrust is required. Some makers supply either ball or roller bearings of the same overall dimensions so that they are interchangeable. Comparing two such bearings of the same dimensions, substantially heavier safe loads might be expected in the roller bearings owing to the much

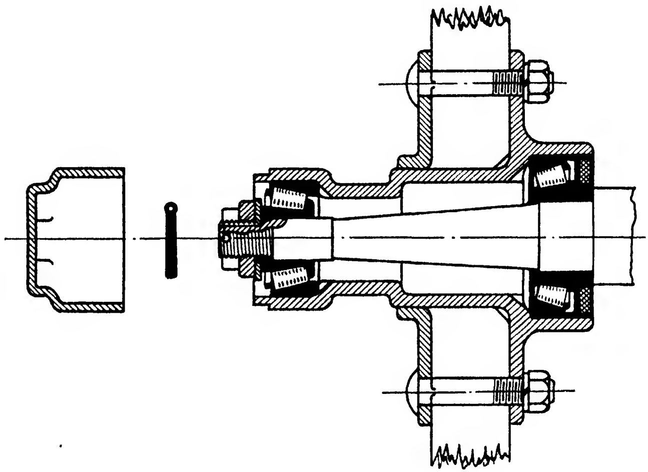


FIG. 45. TIMKEN OPPOSED CONICAL ROLLER BEARINGS

greater bearing surface resulting from line contact as opposed to point contact. It is, however, found that only from 50 per cent to 80 per cent more load can safely be carried. The main reason for this comparatively small increase is undoubtedly due to the difficulty of ensuring that each roller always bears along its whole length simultaneously.

The simple radial roller bearing may be made self-aligning by an outer ring spherically mounted, in which construction it is essential that the roller shall be central in the outer race to avoid uneven loading.

The Skefko self-aligning type of double roller bearing shown in Fig. 44 is adapted to take very heavy radial loads and thrust loads from either direction, and is particularly applicable for heavy commercial vehicle front and rear hubs. The rollers are barrel shaped to fit the spherical outer race.

The well-known conical roller bearings are nearly always used in pairs opposing one another as shown, for example, in connection with a front hub in Fig. 45. A section through a Timken bearing is shown in Fig. 46, and the several components in Fig. 42.

The generating lines of the rollers and cones meet on the axis of the bearing to ensure correct rolling, but the rollers are, in addition, aligned by engagement of the outer ends of the roller with a shoulder on the inner race. This avoids the necessity for alignment by the cage and reduces friction. This type of roller bearing will safely resist even higher thrust loads than radial loads, even where the inclination of the generating lines is small, but in designs in which the cones are steep the thrust loads may be twice the radial loads at corresponding speeds.

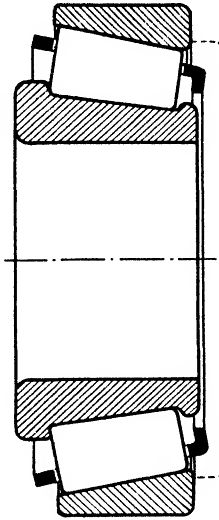


FIG. 46. TIMKEN TAPERED ROLLER BEARING

Lubrication. The suggestion has been made that balls and rollers will roll on their races equally well whether lubricated or not, and that lubricant is required for no other purpose than to prevent rusting of the working surfaces. This is, however, not the whole truth, for it is certain that oil or grease is essential to prevent early failure, and it is quite possible that the minute deformations which occur at the contact areas do account for some rubbing or friction, the effect of which is reduced by the presence of lubricant. In any case, lubrication is necessary to reduce cage friction, to preserve the finely polished surfaces from corrosion, and to protect the bearing from the intrusion of moisture and foreign matter.

Grease of quite thin consistency is usually recommended for all usual speeds under normal conditions. For high speeds, for example, over 6,000 r.p.m., oil is recommended, but this depends upon the size of the bearing. The larger the bearing the lower the speed at which the change from grease to oil should be effected; there can be no definite rule. The question of whether oil or grease should be used on motor vehicles is often settled by considerations apart from the bearings. For example, ball or roller bearings in the engine, gear-box, or the middle or back axle necessarily run in oil.

The carefully finished and polished surfaces of the balls and races, although so hard, may be easily damaged by chemical effects due to moisture, acids, or alkalis in the lubricant. Care must, therefore, be taken that only the best qualities are employed, since the slightest roughening of the smooth surfaces may increase rapidly for mechanical reasons. The consistency of grease is of some importance. It should somewhat resemble petroleum jelly, in being of such a nature as to adhere readily to every surface it touches. In this respect it is different from some of the cheap greases sold in bulk which may be so solid that a bearing will churn a dry channel in it.

SECTION XVII

CRANKSHAFTS AND FLYWHEELS

BY

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SECTION XVII

CRANKSHAFTS AND FLYWHEELS

CRANKSHAFTS

THE crankshaft is of first importance in ensuring the satisfactory running of an internal combustion engine. It may be regarded as the backbone of the engine, and its correct design and proportion determine the smoothness of its running as well as its durability. The designer must endeavour to obtain a shaft which is as nearly as possible rigid at all times, and which is in static and dynamic balance.

It must have bearing surfaces which will ensure reasonable working pressures and adequate lubrication, and it must be as hard as possible on the working surfaces to resist wear.

The actual strength of a crankshaft hardly enters into the problem, since if a shaft is heavy enough to be immune from deflection in any direction, and stiff enough to withstand vibration without momentary distortion, it will be found to have adequate strength to resist with safety any mechanical forces applied to it in ordinary operation.

The journals of a crankshaft are subject to torsional stress and to transverse bending, but this is, or should be, minimized by the bearings: if the shaft is weak in this respect, the bearings will soon give trouble from wear. For this reason the bearings should be as near as possible to the cranks. In this section we will consider crankshafts for one-, two-, four-, six-, and eight-cylinder engines, and many variations of each are possible. As a rule they are made of integral forgings, but occasionally they are built up for special designs, as, for example,

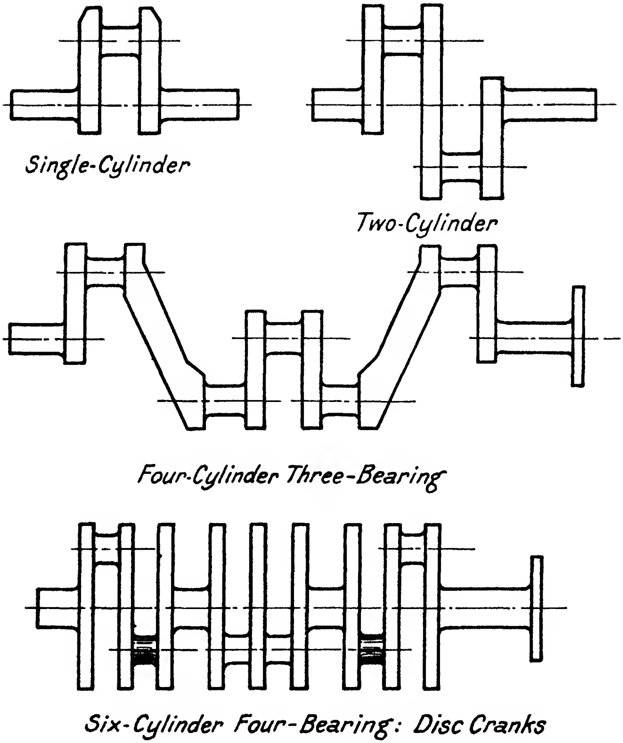

Fig. 1

when roller bearings are used, or when a central flywheel is fitted.

Types of Cranks. The most common types of crankshafts for one or more cranks are shown in Fig. 1.

It will be noticed that a four-cylinder crankshaft may have two, three, or five bearings; and for six cylinders, four or seven bearings, but it by no means follows that the multiplicity of bearings is advantageous. A small four-cylinder crankshaft, if designed with liberal proportions, will function perfectly with only two bearings, and it is very doubtful if there is any justification for going beyond three: five bearings mean excessive length (and weight) of engine, and there is probably greater chance of torsional vibration. The same remarks apply to six-cylinder cranks with four or seven bearings · perfect satisfaction can be obtained with four bearings.

Bearings, nowadays, are more generally made approximately "square," the length and diameter being equal. This is not always possible with six- or eight-cylinder engines, as the length would become excessive. It is not unusual to see that in an attempt to arrange a bearing between each crank (as with seven bearings in a six-cylinder crankshaft) the designer has reduced the width of the journals, until, allowing for the radius at the corners, there is little useful surface left, and the resulting absence of proper oil seal, renders lubrication difficult. In such a case, four properly-proportioned bearings would unquestionably give better results.

Strength of Shafts. It will be as well to remember here that the strength of a shaft depends on its moment of resistance to twisting, and if this is represented by M, we have for a solid shaft

$$M = \frac{\pi}{16} D^3 f$$

and for a hollow shaft

$$M = \frac{\pi}{16} \left(\frac{D^4 - d^4}{D} \right) f$$

Where D is the outside diameter of the shaft.
d is the inside diameter of the shaft if hollow.
f is the maximum *shearing* stress lb./sq. in.

The moment of resistance to twisting must be equal to the twisting moment in inch pounds multiplied by a factor of safety.

When the shaft is subject to transverse bending the bending moment $BM = fZ$, where f is the maximum *tensile* stress (lb./sq. in.) and Z is the modulus of the section.

For a solid shaft

$$Z = \frac{\pi}{32} D^3$$

and for a hollow shaft

$$Z = \frac{\pi}{32}\left(\frac{D^4 - d^4}{D}\right)$$

Hollow Shafts. Crankshafts are frequently made hollow to secure lightness, particularly in multiple cylinder engines, but attention must be paid to secure the best ratio of external to internal diameter. Except in special circumstances, it is not advisable to bore the shaft larger than half its outside diameter. Since, however, the weight decreases rapidly after that, so also does the moment of resistance, and the optimum value of the ratio d/D is about 0·5. As in any case, the crankshaft will have to be drilled for oil passages, the small extra expense of making it hollow is generally justified in any but the cheapest engines.

Journals and Crankpins. As all the bearings on a crankshaft may be considered to be running at high speeds, say, 8 ft. per second or over, in such a case the maximum pressure is of little account, as it is of momentary application only, and is not, therefore, applied for a long enough period to squeeze out the oil.

Load Factor. For a slow-running bearing, the maximum pressure should not exceed 800 to 1,000 lb. per sq. in. projected area, but for a high-speed engine, the load factor, i.e. the product of load multiplied by speed, becomes the limiting figure. The highest load factor which can safely be permitted depends upon—

1. The system of lubrication.
2. The viscosity and greasiness of the oil.
3. The facilities available for conducting heat away from the bearing.

With all these circumstances most favourable a load factor of 20,000 lb. ft. per second is permissible in any journal bearing with an alternating load, and this should always be checked when the design is under consideration.

If any distortion takes place, the load factor will be increased locally, and the pressure all concentrated on one point in the bearing; as the rubbing velocity is the same at all points, the load factor may become excessive at one point, causing rapid wear, or even seizing.

From the point of view of keeping down the load factor, it is evident that it is better to lengthen the journal rather than increase its diameter, as any increase in diameter will produce a corresponding increase in rubbing speed. From the point of view of rigidity and freedom from distortion, however, a long journal is very undesirable; the load cannot be evenly distributed, and the oil circulation is impaired. On the score of distortion alone, it is seldom possible to make any journal wider than one and a half times its diameter, and with forced lubrication it is desirable to make the length of the journal approximately equal to the diameter, or not more than one and a quarter times this.

Rate of Wear. The rate of wear may be taken as being directly proportional to the load factor, but the journal bearings will wear more rapidly than the crank pins, because in the latter the oil is flung out to them by centrifugal force, in addition to the pressure at which the oil is supplied.

We find, therefore, that with other things equal—

1. The bearing friction is nearly proportional to the load factor, and the smoother the surface the lower will be the friction.

2. The rate of wear is also proportional to the load factor.

3. The viscosity of the oil does not greatly affect matters. With high viscosity the friction is increased at first, but as this produces greater heat, the viscosity will fall and something approaching a balance is obtained.

Balancing Cranks. There are three conditions of balance which have to be satisfied in any engine crankshaft if it is to run without vibration.

(*a*) Torsional balance.
(*b*) Static balance.
(*c*) Dynamic balance.

Torsion of Crankshafts. It is easily shown that the distribution of torsional stress is greatly affected by the firing order of the various cylinders, if more than one.

Thus, in a single-cylinder engine of four cycles, let

WS = working stroke.
ES = exhaust stroke.
SS = suction stroke.
CS = compression stroke.

One Cylinder (Fig. 2)—

	180	degrees	$= WS$
	180	,,	$= ES$
	180	,,	$= SS$
	180	,,	$= CS$
Total	720	,,	1 impulse per two revolutions.

Here the impulses are 720° or two revolutions apart and the turning moment will be very uneven, and a relatively heavy flywheel will be required to smooth out the torque.

TWO CYLINDERS (Fig. 3)—

	1st Cylinder	2nd Cylinder
180° . .	*WS*	*SS*
180° . .	*ES*	*CS*
180° . .	*SS*	*WS*
180° . .	*CS*	*ES*

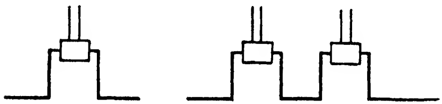

FIG. 2 FIG. 3

Here we have one impulse per revolution, i.e. the firing strokes are 360° apart, the turning moment has been greatly improved, but the reciprocating parts are entirely unbalanced.

If we place the cranks opposite to one another

TWO CYLINDERS (Fig. 4)—

	1st Cylinder	2nd Cylinder
180° . .	*WS*	*CS*
180° . .	*ES*	*WS*
180° . .	*SS*	*ES*
180° . .	*CS*	*CS*

Here we have two impulses in one revolution, and none in the next. So that while the reciprocating parts are much better balanced, the torque is badly distributed, and no amount of balance weight would correct it.

Two Cylinders Opposed (Fig. 5)—

	1st Cylinder	2nd Cylinder
180° . .	*WS*	*SS*
180° . .	*ES*	*CS*
180° . .	*SS*	*WS*
180° . .	*CS*	*ES*

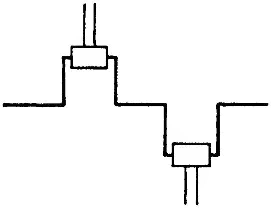

Fig. 4

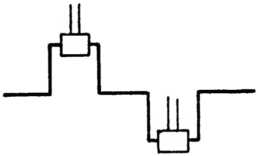

Fig. 5

In this case we have perfect reciprocating balance and even torque distribution, with one impulse per revolution.

Unfortunately an opposed cylinder engine presents many difficulties in design, e.g. gas distribution and difficulty in disposing of the engine in the chassis.

Four Cylinders (Fig. 6)—

	1st Cylinder	2nd Cylinder	3rd Cylinder	4th Cylinder
180° .	*WS*	*ES*	*CS*	*SS*
180° .	*ES*	*SS*	*WS*	*CS*
180° .	*SS*	*CS*	*ES*	*WS*
180° .	*CS*	*WS*	*SS*	*ES*

Firing order 1, 3, 4, 2.

FOUR CYLINDERS (Fig. 6)—

	1st Cylinder	2nd Cylinder	3rd Cylinder	4th Cylinder
180°	*WS*	*CS*	*ES*	*SS*
180°	*ES*	*WS*	*SS*	*CS*
180°	*SS*	*ES*	*CS*	*WS*
180°	*CS*	*SS*	*WS*	*ES*

Firing order 1, 2, 4, 3.

In both the above cases there are two impulses per revolution 180° apart.

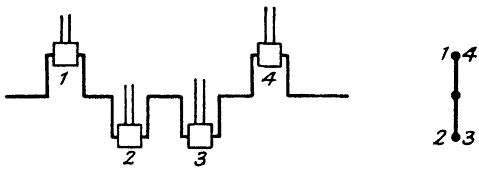

FIG 6

SIX CYLINDERS (Fig. 7)—

Firing order 1, 5, 3, 6, 2, 4.
Impulses 120° apart.
Three impulses per revolution.

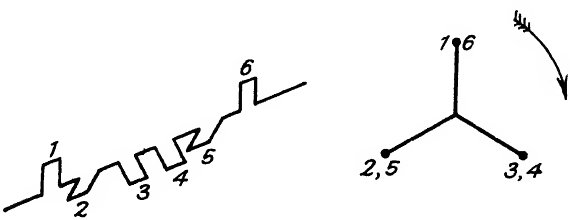

FIG. 7

In this example, as shown in Fig. 7, the two end cranks 1 and 6 are in the same plane, 2 and 5 are in another plane, and 3 and 4 in a third plane.

Six Cylinders (Fig. 8)—

Firing order 1, 5, 3, 2, 6, 4.

Again we have impulses 120° apart or 3 per revolution. With this order of firing, Nos. 1 and 2 cranks are in the same plane; Nos. 3 and 4 in another plane, and Nos. 5 and 6 in a third plane.

It is, of course, possible for the firing to take place in any desired order, but very different results are obtained by varying this. The worst possible order would be 6, 5, 4, 3, 2, 1, since each explosion following on the previous one, progressively from the rear end of the shaft, would add to the angular deflection or twisting

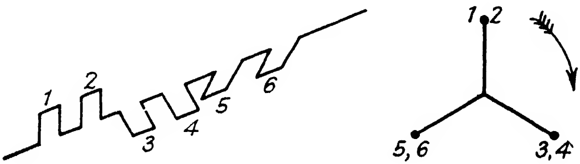

Fig. 8

of the shaft. After six explosions, this would be considerable, and when No. 6 again fired, the front end of the shaft would be released, and would spring back in the opposite direction, producing a violent oscillation or torsional vibration.

We may conclude, therefore, that succeeding explosions should take place at alternate ends of the shaft. It is, of course, impossible to make the torque vibrations symmetrical, since number 1 crank must always give a greater angular deflection than number 6, as the deflection increases as the distance of the working crank from the flywheel. This clearly shows that a firing order of 1 5 3 6 2 4 is better than 1 5 3 2 6 4, as with the former the explosions never occur in adjacent cranks, while in the latter they do, at least once. The effect of this is discussed more fully later.

Eight Cylinders.—Many orders of firing are possible in an eight-cylinder engine. In any case there will be four impulses per revolution following each other at intervals of 90°. The remarks just made on six-cylinder crankshafts clearly apply to eight cylinder cranks with added force on account of the greater length of the crankshaft, and greater liability to angular deflection. In multiple cylinder engines, however, the problems of gas distribution enter largely into the question of firing order, but the principle of distributing the explosions over opposite halves of the shaft must be followed as pointed out in the foregoing remarks on six-cylinder engines.

Position of Flywheels. It is a convention of design to place the flywheel at the rear of the engine, where it

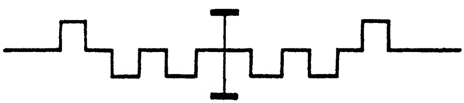

Fig. 9

can be combined with the clutch. It is easy, however, to see that this is not the ideal position from the point of view of torsional vibration. The flywheel acts as a reservoir of the energy developed in each cylinder, and from this point of view six flywheels would give the best results for a six-cylinder crank, with one flywheel adjacent to each crank. With a flywheel at the rear end, there is, as we have seen, a material difference in angular deflection due to numbers 1 and 6 cylinders. It follows that the ideal arrangement will be to place the flywheel in the *centre*, as in Fig. 9, so that the flywheel is never more than two cranks away from a firing cylinder. A friction damper, to be described later, should always be fitted to the front end of the shaft.

Static Balance. With the exception of single- and two-cylinder engines, where special balance weights must be provided, the ordinary arrangement of 4, 6, and 8 cylinder cranks is symmetrical about the principal axis, hence they all—assuming accurate workmanship —will be in static balance in any position. It is quite usual, however, at least in four-cylinder engines, to employ stampings for crankshafts, and only to machine

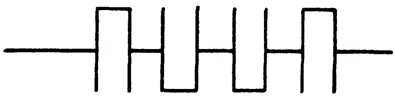

FIG. 10

the journals and crank pins, in which case some balancing will be required, and, generally, the simplest way is to prolong the crank webs as in Fig. 10, so that by rough grinding these as required, the shaft may be put in static balance.

Any crankshaft will be in static balance when its centre of gravity coincides with the principal axis, which is, of course, the case with any symmetrical arrangement.

Having regard to both static and dynamic balance it is always advisable to balance each crank with its own balance weights to minimize the inertia stresses, otherwise these have to pass through the journals with resulting complications.

It is now necessary for the reader to obtain a clear idea of the difference between Static and Dynamic Balance. Suppose a shaft has two unbalanced masses W_1 and W_2, Fig. 11, at radius r_1 and r_2 respectively. Let $U_1 = W_1 \times r_1$ and $U_2 = W_2 \times r_2$, then U_1 and U_2 are their equivalent at unit radius. Draw a triangle of forces, making the sides U_1 and U_2 equal to U_1 and U_2 on some convenient scale; then the closing line U is the equivalent in direction and magnitude of W_1 and W_2, and is their resultant force. It can, of course, be balanced by an equal mass X at the same radius as U but in the opposite direction. The system $X\ W_1\ W_2$ will then be in static balance, and if the masses act about a rigid shaft, it is not material whether or

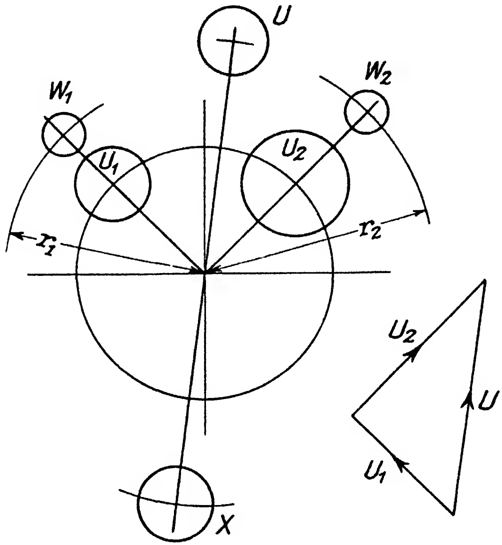

Fig. 11

not they are in the same plane, provided the shaft is freely suspended by centres on its axis. We will take a numerical example.

In Fig. 12 we have a weight $W_1 = 2$ lb. at a radius r_1 of 2·5 in., and a weight $W_2 = 3$ lb. at a radius of r_2 of 2 in.

$$U_1 = W_1 \times r_1 = 2 \times 2{\cdot}5 = 5$$
$$U_2 = W_2 \times r_2 = 3 \times 2 \quad = 6$$

We draw the two forces U_2 and U to a convenient scale of 5 and 6 parallel to W_2 and W_1: the closing line

U is found to have a length of 8 on the same scale, and we draw a line parallel to this which is the resultant of the forces W_1 and W_2; as it has a magnitude of 8 it can

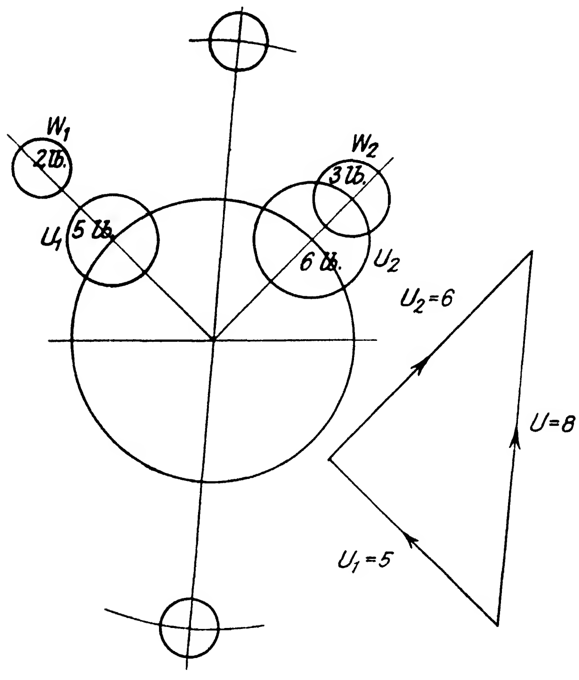

FIG. 12

be a mass of ? lb. acting at a radius of 4 in. or 4 lb. at a radius of 2 i 1., and it can be balanced by a similar mass × radius X in the same plane produced.

Dynamic Ba'ance. Next consider the same system rotating, and as ume first that W_1 and W_2 rotate in one plane: we see tl· n that they will be in dynamic balance if the centrifuga force of X is equal and opposite to the centrifugal force of the resultant of W_1 and W_2. We will take a numerical example.

Let $W_1 = 3$ lb., $W_2 = 4$ lb.
$r_1 = 3$ in., $r_2 = 4$ in.

find the mass and position of X to secure dynamic balance at 1,000 r.p.m.

Centrifugal force $CF = Wv^2/gr$ where $v =$ velocity in ft. per second; $g = 32{\cdot}2$; and $r =$ radius in feet.

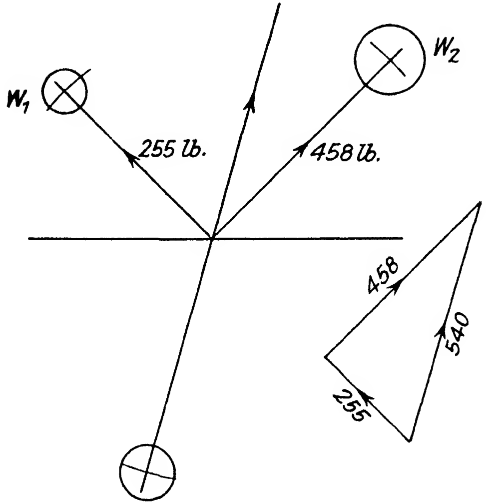

FIG. 13

For W_1, $CF = 3 \times v^2/32{\cdot}2 \times {\cdot}25 = {\cdot}372\, v^2$

here $v = \dfrac{3 \times 2 \times \pi}{12} \times \dfrac{1000}{60} = 26{\cdot}18$ f.p.s.

and $v^2 = 685$, so $CF = 685 \times {\cdot}372 = 255$ lb.

For W_2, $CF = 4 \times v^2/32{\cdot}2 \times {\cdot}33 = {\cdot}376\, v^2$

here $v = \dfrac{4 \times 2 \times \pi}{12} \times \dfrac{1000}{60} = 34{\cdot}9$ f.p.s.

and $v^2 = 1218$, so $CF = 1218 \times {\cdot}376 = 458$ lb.

We next draw the triangle of forces and find the resultant, Fig. 13, is 540 lb. to be balanced by a centrifugal force of equal magnitude in the opposite direction.

Assume the centre of gravity of the balance weight is at 4 in. radius, its velocity will be the same as that of W_2 and its velocity squared $= 1218$: then putting x for the weight,

$$CF = \frac{x \times 1218}{32{\cdot}2 \times {\cdot}33} = 114{\cdot}6\,x$$

and $\quad CF = 540$ so $114{\cdot}6x = 540$

whence $x = 4{\cdot}71$ lb.

Fig. 13 shows the system drawn to scale which is in dynamic balance at 1,000 r.p.m. all the weights being in one plane.

Where the masses to be balanced are *not* in the same plane we have another condition.

Assume they are 10 in. apart as in Fig. 14. Here when $W_1 = 255$ lb. and $W_2 = 458$ lb., both being centrifugal forces, we know the resultant is 540, so it follows that the position of X is such that

$$a : b \text{ as } 458 : 255$$

From this it is clear that different mass systems are needed for static and dynamic balance. It was at one time customary to balance shafts and flywheels by rolling them upon two horizontal straight edges, when they were considered satisfactory if they came to rest at any point; from the foregoing, however, it is seen that such a method, which only depends on static balance, is of no use whatever when the shaft is in motion, and centrifugal forces have to be dealt with.

One of the well-known balancing machines is the only thing which will show up the defects. Owing to the fact that in practice all engines of four cylinders and upwards have their cranks symmetrically disposed about the axis of the shaft, there are no cases where serious masses have to be balanced, though the writer came across a six-cylinder crankshaft machined all over

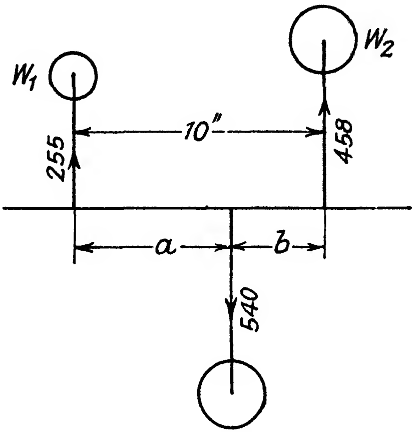

Fig. 14

where it was found to be 5 oz. out of rotational balance, presumably due to a chance accumulation of errors in one direction.

We may here mention the variation in torque resulting from varying the number of cylinders, as this has an important effect on the torsional stiffness of the shaft. If the torque variation is great, there must be a corresponding torsional vibration set up in the shaft, and for an engine of given power, it is clear that by increasing the number and decreasing the size of the cylinders, all the stresses are reduced, and the vibrations are smoothed out. The fluctuation of driving torque varies throughout all four strokes of the cycle, as shown in Fig. 15, for in addition to the varying pressure on the pistons during the working and other strokes, we have the effect of inertia pressure which is sometimes positive and sometimes negative. The algebraic sum of the pressure gives the energy available as power, and the energy in excess of the mean value has to be stored by the flywheel and given out again by it during those parts of the cycle when the energy is below the mean.

The diagram Fig. 15 is, of course, for a single-cylinder engine, and as we multiply the cylinders we obtain

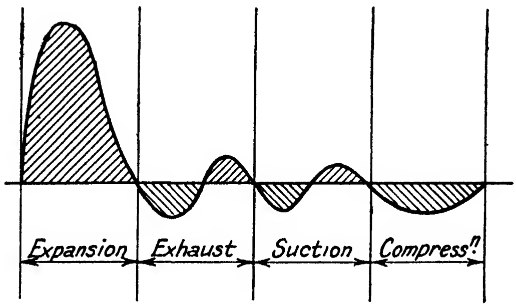

FIG. 15

similar diagrams overlapping each other, so that by adding together all the areas above the zero line, and deducting all those below it, we obtain the mean effective torque throughout the cycle of the revolutions.

In Fig. 31 (page 938) the diagram is drawn for four cylinders from which the effect of multiplying the cylinders is shown, though it does not illustrate the effect of cylinder dimensions. To obtain the *same power*, the size of the cylinders is reduced as the number is increased, and with smaller cylinders the variation of torque is through smaller limits, so that the maximum and minimum variation becomes much nearer to the mean in multi-cylinder engines.

It is sometimes stated that the ratio of excess energy to the mean energy per revolution has approximately the following values. If r is the ratio of the excess to mean energy (we will discuss this more fully later)—

Single-cylinder	engine		$r = 3{\cdot}7$ to $4{\cdot}5$
Four	,,	,,	$r = 0{\cdot}3$ to $0{\cdot}4$
Six	,,	,,	$r = 0{\cdot}19$
Eight	,,	,,	$r = 0{\cdot}17$

While these values must not be taken as empirical it is easily seen from them how the necessity of a fly-wheel diminishes as the number of cylinders is increased; in fact, for six and eight cylinders it is scarcely required at all. It will be seen from the foregoing that torsional vibration cannot be eliminated from any crankshaft, and whilst it is decreased with a larger number of small cylinders it is increased to some extent owing to the longer shaft required.

A crankshaft may be considered in the light of a torsional pendulum, and like any other pendulum, has an inherent frequency of oscillation; the number of complete oscillations (clockwise and counter-clockwise) is given by the formula—

$$N \times \frac{30}{\pi} \sqrt{\frac{J \times Et \times g}{L \times k^2 \times m}}$$

where J the polar moment of inertia of the shaft $=$ $\pi\, d^4/32$

$Et =$ the modulus of elasticity (shearing)
$L =$ the length of pendulum subject to torsion
$k =$ the least radius of gyration of revolving mass
$m =$ sum of revolving masses
$g =$ gravity acceleration.

It is, therefore, evident that if the maximum variations of torque occur at such time intervals as to synchronize with the natural vibration period of the shaft, very serious vibration may be set up, and if this persisted the shaft would break. There is not much danger from this as a rule, as the speed of any engine is never actually constant, but varies between a maximum and minimum. The phenomenon may, however, account for unexplained breakages in isolated cases.

The subject of shaft vibration under stresses is only amenable to abstruse mathematical treatment, and cannot be more than touched upon here, but one other point must be noticed. In Fig. 16 we have a diagrammatical arrangement of a torsional pendulum in which

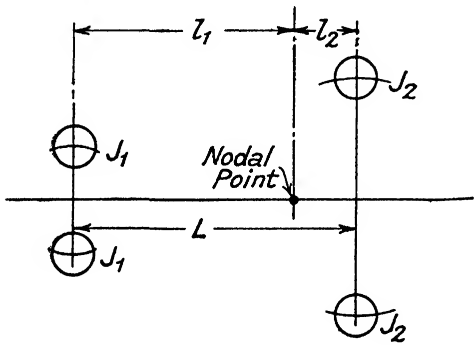

FIG 16

a fixed point is replaced by two masses swinging in opposite directions to the masses of the pendulum. Here the angular motions of the two ends will be directly proportional to the position of some point, called the nodal point; this position being such, that the angular motions of each section of the pendulum will be directly proportional to their distance from the nodal point. So that assuming the mass of one crank and its attachments to be concentrated at a point and to have a polar moment of inertia of J_1, and the flywheel to have a polar moment of inertia of J_2, then $J_1 \ l_1 = J_2 \ l_2$. From what we have seen it is clear that there will be more than one nodal point in a crankshaft, depending upon which cylinder is firing, which is a further complication. Enough has, however, been said to show that if the flywheel end of the shaft is moving with a constant angular velocity, due to the influence of the flywheel, etc., the front end will be in a constant state of change of velocity above and below that of the flywheel end, or in other words, in a constant state of torsional vibration.

The consideration of the position of the nodal point is sufficient argument in proof of the desirability of a *central*, rather than a rear end flywheel, already noticed.

With a flywheel at the rear, the nodal point will always be somewhere near the back end of the shaft, and it is theoretically the only point at which the angular velocity is constant; we see from this, that if quiet operation of the engine is sought, the drives for the camshaft and auxiliaries should be located as near as possible to the nodal point. This can only be brought about by placing the timing and other gears near the rear of the engine. It may be pointed out that the length of the shaft (within practicable limits) does not greatly affect its natural period of vibration as this only varies inversely as the square root of its length, as shown in the formula.

Harmonic Balance. The subject of crankshaft balance cannot be studied separately from that of the general balance of the engine as a whole, particularly in respect of the reciprocating parts which are attached to it, and we must remember that there are vibrations inherent in the engine itself which would still be manifest even if it were possible to assume each element of the engine to be absolutely rigid, while there are other vibrations due to the inevitable springing of those elements due to the elastic yielding of the materials.

If the problem had to be treated on a purely mathematical basis, it would be necessary to consider it from the physical basis of six degrees of freedom, namely, one of translation along each of its three co-ordinate axes, and one of rotation about any or all of these axes. The actual nature of the machine in question, however, narrows the problem very considerably, and a simple treatment can be pursued; for example, the reciprocating parts are confined to axes at right angles to the crank axis. We therefore know in what direction the unbalanced forces will act; torque variations will act about the crankshaft axis, and the reciprocating forces will produce rocking moments tending to swing the axis of the crankshaft about some point in its length.

There are two aspects from which this problem may be viewed. (*a*) We may consider the engine itself *fixed*, as if bolted to a perfectly rigid foundation, and (*b*) we may consider the engine freely suspended in space. It is clear that neither is quite correct for the engine of a motor vehicle. Commencing with the case of a single-cylinder engine, if the piston has a mass m, and the engine a mass M which is *fixed*, then the mass m will move with a strictly harmonic motion through a distance $s =$ the stroke. In this case at a speed of $R =$ r.p.m.; then with true harmonic motion of m, resulting from a connecting rod of infinite length, the forces in pounds set up, which have to be resisted by M (and the foundation bolts), are determined by the formula

$$F = 2\pi^2 msR^2/32{\cdot}2$$

with the result shown by Fig. 17.

When, however, we substitute connecting rods of normal proportions, we know that the piston has no longer a simple harmonic motion, and it is unsymmetrical. The accelerations are greater at the *in-centre* than they are at the *out-centre*, and the force diagram, still assuming the engine is fixed, will be somewhat as shown in Fig. 18. A periodic curve of this form, which is actually a sine curve, can be resolved by Fourier's Theorem into a number of simple harmonic components

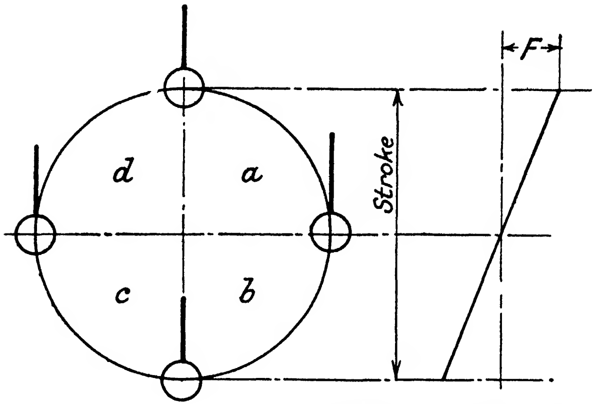

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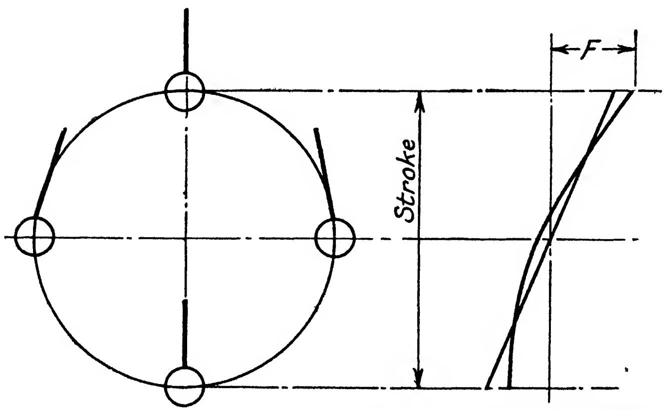
FIG. 17



(*Institution of Automobile Engineers*)
FIG. 18

which for our purpose may be represented by two, namely, a fundamental component shown by the sine curve Fig 19 at (*a*), and a secondary component (also a sine curve) of twice the frequency as shown at (*b*) By adding or subtracting the ordinates of the two curves, we obtain a third resultant curve (*c*) The second curve of twice the fundamental frequency is called the octave component, and expanding by Fourier's Series, it can be shown that there are other harmonic curves of four times and six times the fundamental frequency. As, however, the amplitude of these higher harmonic curves becomes progressively and

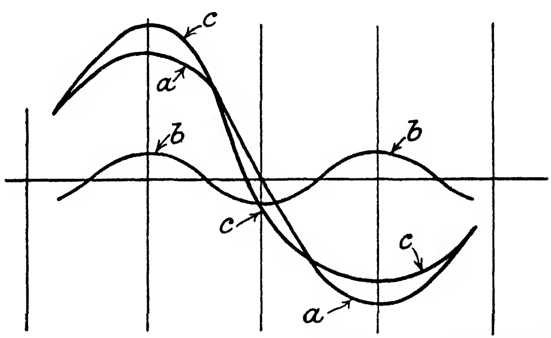


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FIG 19

rapidly less, to infinity, we can usually disregard them, though there are sometimes cases in which the third may be perceptible.

The following method of illustrating this effect is due to Dr. F. W. Lanchester, F.R.S. Three drums are provided having diameters in the relation of 4, 2, and 1. that is, the ratios of the frequencies to be examined, and strings threaded with a number of beads to correspond with the number of cylinders. The position of the beads must be such that when the string is wound *once* round the largest drum, the beads are equally spaced round the circumference. This drum we will refer to as the cycle drum, and the two smaller drums as the fundamental and octave drums respectively.

If a string with four beads be wrapped round the cycle drum, the beads will be found equally spaced in quadrants round the drum. Next let the string be wrapped round the fundamental drum: it will take two

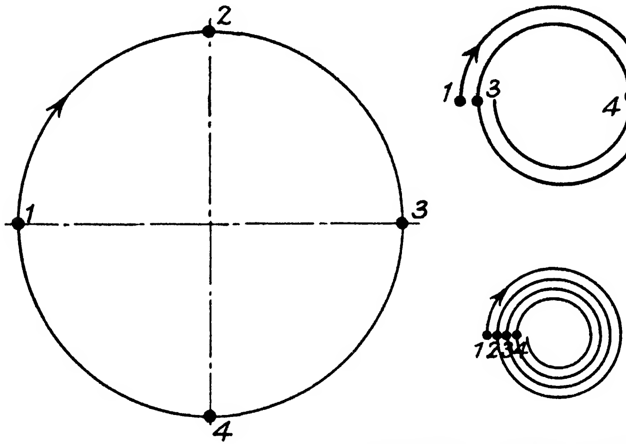

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FIG 20

turns and the beads will appear in pairs, one pair on one side and another pair diametrically opposite to them, Fig. 20.

Passing the string round the octave drum we find all four beads together at one point, thus demonstrating that in the four-cylinder engine, the secondary or octave vibration is totally unbalanced, the four cylinders synchronizing.

We then take a string with six beads, and wrapping this round the cycle drum, we find the beads are arranged at the corners of a hexagon. On the fundamental drum, the beads appear in pairs at three points, the corners of an equilateral triangle; lastly, wrapping the string round the octave drum, the beads are again disposed in pairs at the corners of an equilateral triangle, proving that the six-cylinder engine is truly

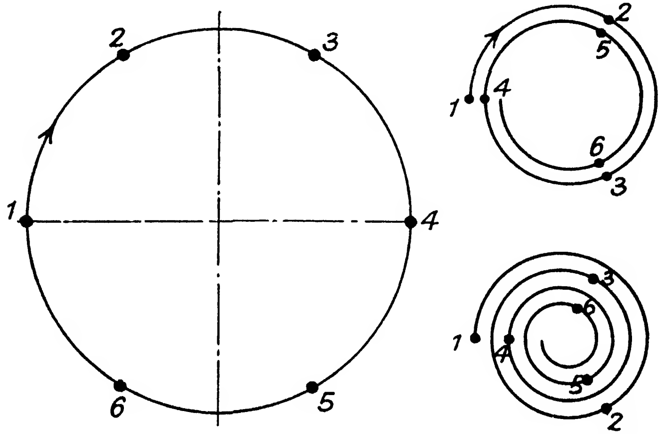

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FIG. 21

balanced in both fundamental and octave vibration, Fig. 21.

We will now take the case of two cylinders with cranks at 180° which is one-fourth of the cycle: Fig. 22 shows the string on the cycle drum. On the fundamental drum the beads appear opposite and on the octave drum they coincide, showing that the fundamental vibrations are balanced, and the octaves unbalanced.

Lastly, we have the case of the two-cylinder opposed engine, and as in this case the fundamental vibrations neutralize each other, having positive and negative values, we use a black and white bead, which on the cycle drum are spaced opposite each other; we then see that on the fundamental drum the black and white beads coincide, showing that the vibrations are

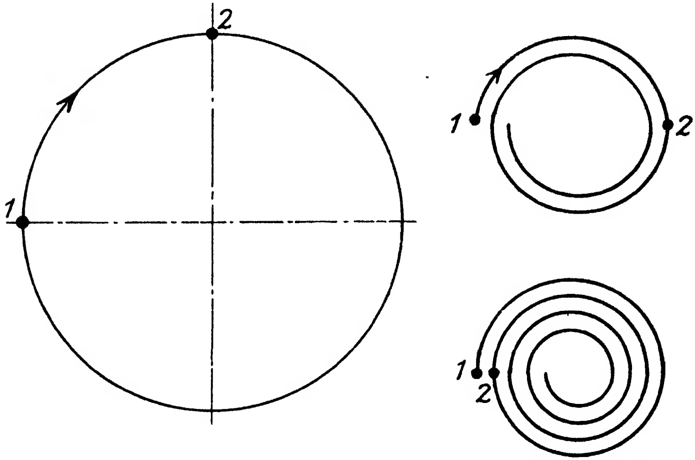

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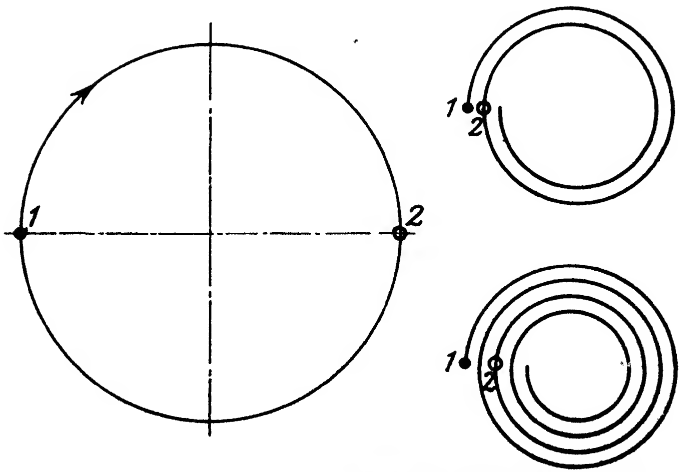
Fig. 22



(Institution of Automobile Engineers)

Fig. 23

neutralized, and we find the same result on the octave drum, which proves that the engine is truly balanced in both respects, Fig. 23.

From what we have seen so far, it is clear that a single-cylinder engine has two positions in each revolution when its kinetic energy is zero, and two positions where it is at a maximum, and, disregarding the connecting rod angularity, these conditions occur at the quadrants. After passing the top dead centre, the crankshaft is doing work on the piston; during the next quadrant, the piston is giving back its energy to the crankshaft, and the same conditions occur during the other half of the revolution.

Apart altogether from any changes in gas pressure on the piston, there is, therefore, an alternation of torque from positive to negative twice per revolution, due to inertia stresses alone. Clearly this torque has its equal and opposite reaction on the frame of the engine, in which, therefore, a periodic vibration is set up. In addition to this we have, of course, the compression pressure taking place every other stroke (in a four-cycle engine) and the ensuing expansion plus explosion pressure. In Fig. 24 we have a zero line AB representing one complete cycle which is divided into eight quadrants. We commence at c with a negative inertia stress which becomes zero on passing the first quadrant; rises to a positive maximum at the end of the second quadrant, and so through four strokes to the end of the cycle, as shown by the sine curve cd.

Then we have a negative suction pressure during the first two quadrants; a compression pressure increasing during the third and fourth quadrants, at the end of which ignition takes place; during the fifth and sixth quadrants expansion is in progress, until the exhaust opens; and finally we have the exhaust against some positive back pressure during the seventh and eighth quadrants. By adding and subtracting the ordinates of these superimposed curves we obtain their algebraic sum, which is the resultant torque variation curve for one cylinder throughout one cycle of two revolutions, shown in the figure by the dotted line.

We must not lose sight of the fact that this curve of gas pressures is only to a comparatively slight extent

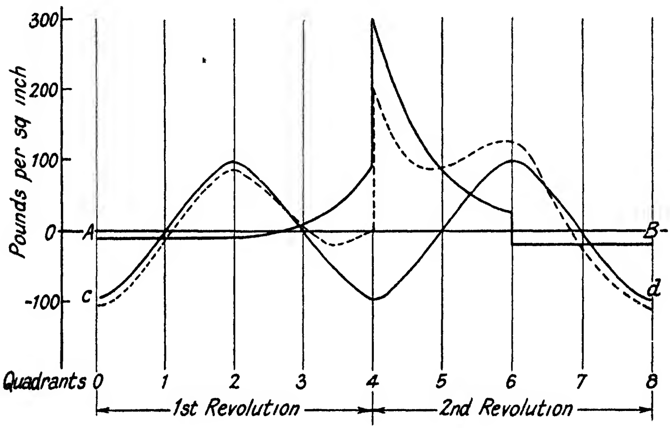


FIG 24

modified by the speed of the engine, whilst the inertia forces vary as the *square* of the speed. The curves as drawn in the figure would be for an engine running at high speeds.

In a single-cylinder engine, the crankshaft may be balanced by adding suitable counterweights to the crankwebs, such, that the mass of the crank as concentrated in the crank pin is balanced by the mass of the counterpoise on the opposite side; for estimating this weight, the weight of the big end of the connecting rod (as weighed in a scale when the gudgeon pin is supported) is added to that of the crank and crank pin, so that the weight of the small end only is added to that of the piston. There still remain unbalanced variations of a considerable magnitude, due to the mass of the reciprocating parts.

Thus, in the case of a motor-cycle engine with 85 mm. bore and 88 mm. stroke when running at 3,000 r.p.m., the forces due to harmonic motion of the pistons are of the order of 800 pounds, and those due to the secondary vibration of about 180 pounds; thus making a total of 980 pounds unbalanced vibrational forces. These may, of course, be partially counteracted by increasing the mass of the balance weights, but it is easily seen that such a recourse can at best be a very rough approximation to what is required, and the unbalanced forces have to be absorbed in the mass of the engine and those parts of the chassis to which it is attached.

If we assume it possible, so to arrange the balance weights that all the reciprocating parts are balanced as though they were rotating masses centred at the crank pin, *the resultant unbalanced forces would occur at right angles to the direction of the piston motion.* As the vehicle springs are approximately unyielding in this direction, some advantage may be taken of this circumstance. If, as is usual, *half* the reciprocating weight is balanced, the effect will be that the unbalanced resultant is the equivalent of a mass equal to the added balance weight rotating at the crank pin radius in the direction opposite to that of the engine; the maximum force will be at the same time halved.

In the case of a two-cylinder engine with cylinders side-by-side, and cranks *coincident*, the conditions are the same as in a single-cylinder engine, but if the cranks are at 180° the fundamental forces are balanced and in this case they will produce a rocking effect at right angles to the axis of the crankshaft (Fig. 25) about its central plane, depending on the distance apart of the cylinders.

In either of these cases, the octave vibrations as we have already seen are totally unbalanced. It follows that in a two-cylinder engine we have to choose between (*a*) the effect of the unbalanced reciprocating forces, when the cranks are coincident and the resulting firing strokes separated; or (*b*) balanced reciprocating forces when the cranks are 180° apart and two explosions 180° apart followed by 360° of idleness. From this difficulty, the two-cylinder engine with cylinders side-by-side has become practically obsolete, and where this number of cylinders is desired they are usually arranged *opposite* when the vibration difficulties disappear.

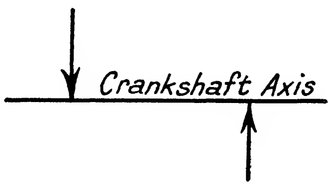


FIG. 25

If we refer again to Fig. 25 another and very important circumstance becomes apparent; it was noticed that with two cylinders at 180° the reciprocating forces are only balanced in one plane; that is to say when the shaft is viewed from one end. When we look at the shaft from its side, we see that the forces produce a rocking couple which may become very disturbing. Let us, then, consider the case of a four-cylinder engine. Here we have, Fig. 6, cranks numbers 1 and 4 coinciding, and also cranks 2 and 3; it is clear that the fundamental forces are balanced and, therefore, neutralized as they are arranged about a plane of symmetry located at the centre of the shaft in respect to its length.

In the case of a normal six-cylinder engine, whose crank is shown in Fig. 7, we have the following cranks coinciding: numbers 1 and 6, numbers 2 and 5, and numbers 3 and 4, and each of the three groups is placed at the corners of an equilateral triangle. It is evident that an engine possessing symmetry of this kind is entirely free from any rocking moment, because whatever couple may be set up at any instant in one half of the engine, is exactly balanced by an equal and opposite couple in the other half. This has been referred to by Dr. Lanchester as "looking-glass symmetry," since the whole of the engine on one side of the plane of symmetry may be considered as a reflection of the other half on the plane of symmetry considered as a mirror. Where the distribution of the cylinders is thus symmetrical there is no rocking moment.

Torque Reaction. The ideal engine, as regards freedom from torque vibration, is, of course, the steam turbine, in which the general torque is produced by the combined effect of many thousands of turbine blades arranged around the circumference. Of course, such a degree of smoothness can never be obtained in any reciprocating engine with a very limited number of cylinders, and, as already noticed, the torque variations require to be absorbed in the flywheel, a function which it performs very effectively. We must not, however, overlook the fact that the *torque reaction* falls to be taken up by the mass of the engine as a whole, to which vibrations due to torque are constantly being imparted.

If, for example, we take the case of a single-cylinder engine, the maximum torque occurs (Fig. 15) once in every two revolutions, and if we imagine the speed of the flywheel due to its inertia to be approximately constant, we must assume that during the explosion stroke there is an excess of energy put into the flywheel to carry it round at the same speed during the remainder of the cycle. Actually the flywheel speed is *not* constant and slight fluctuations in velocity are constantly taking place. When an impulse takes place, there is a backward impulse imparted to the frame of the engine which nothing can reduce, so that were some recording dynamometer fitted beyond the flywheel, this might show no very appreciable variation of torque; yet could some similar recording apparatus be applied to the engine, it would be apparent that it was constantly subjected to a series of kicks, which, when finally absorbed by the chassis of a motor vehicle, might become a source of very unpleasant vibration.

Various devices have been proposed to overcome this disability, the most successful of which are the two geared and oppositely rotating flywheels, used in the original Lanchester and the Lucas "Valveless" engines; but the additional complication proved too much for their ingenuity in both cases, and such methods have now been abandoned.

We know that the momentum imparted to a mass by the application of any force is proportional to the duration of the force; it follows that the slower an engine runs, and the greater the consequent duration of the torque impulse, the greater will be the vibration; hence the faster the engine runs, the less apparent this becomes, and provided we do not invade the region of speeds where the higher vibration periods appear, there is usually some optimum speed at which the engine, in the motorists' phrase, "goes to sleep," the frequency of the torque recoil impulses having become too high for perception by the senses.

Four-cylinder Engine Octave Vibration. We have seen that the principal cause of defective balance in a four-cylinder engine is due to the fact that the octave components of the motion of the four pistons are synchronized, hence the forces which bring about octave vibrations are four times as great in a four-cylinder engine as they are in a single cylinder (the cylinders being in each case of the same size).

In balancing a crankshaft in practice, it must be pointed out that a crankshaft which is in static balance may or may not be in dynamic balance, but if it is in dynamic balance, it will necessarily be in static balance.

While it is clearly advantageous to have all the moving parts appertaining to each cylinder of equal weight, meticulous accuracy in this respect is not necessary when it is remembered that the unbalanced forces in quite a small cylinder may be of the order of half a ton, so that a quarter of an ounce or less in the weight of the reciprocating parts is of comparatively insignificant importance.

We have now to consider the question of vibration due to lack of rigidity in the components themselves. Such vibrations may become very troublesome if they tend to occur in synchronization with any of the forces we have already examined; thus a natural vibration period of the crankshaft may fall in tune with the fundamental period of the piston, or its octave, or even one of the higher harmonic periods; indeed, it is generally only when the latter happens that the higher periods can be distinguished.

Sometimes the firing point may be the cause, from which we see that synchronous vibration may occur at several different speeds of the engine. A further complication arises if the vibration coincides with some natural period on the chassis itself, and we may frequently distinguish these periods at definite engine speeds—it is unnecessary to analyse these here.

Crankshaft Torsion. We have seen that the varying crankshaft torsion is a most troublesome factor to deal with, and in the early days of the six-cylinder engine the difficulties arising from this source nearly proved the undoing of this type. It is not sufficient alone to add stiffness to the shaft, and the trouble is increased by a large stroke-bore ratio, and by increasing this ratio the crankshaft designed for maximum stiffness becomes unduly heavy, and the larger bearings are an additional source of friction. We saw in Fig. 16 that the crankshaft may be considered as a torsional pendulum, and if flywheels of equal moments of inertia were fixed to the ends it would have a definite period of vibration.

The fact that in a six-cylinder engine we have one flywheel only, and a system of cranks instead of another, modifies the condition without fundamentally altering it, and whereas if there were two flywheels, it would be possible for them to vibrate in opposition without communicating any reaction to the engine frame, the connecting rods and cylinders provide a source whereby the vibration is communicated to the engine as a whole, and so the crankshaft vibration may be felt in the chassis.

The chief cause of torsional crankshaft vibration is the reversal of torque which, as shown in Fig. 24, occurs twice per revolution in each cylinder. It follows from this, that if the engine speed is one half that of the torsional period of the crankshaft, the maximum and minimum point of the torque reaction would synchronize with the periodic vibration of the shaft.

In a six-cylinder engine, where the torsional vibrations would occur three times per revolution, the synchronous vibration would occur when one revolution of the engine corresponds to three complete periods of the crankshaft considered as a torsional pendulum. Hence when torsional vibration is present in a six-cylinder engine, we may find two marked periods having a speed relation of 2 to 3; the lower when the piston pressure is the cause of synchronization, and the higher when it is due to piston inertia.

Of course, the cylinder farthest from the flywheel acting at the extreme length of the shaft, where the vibration amplitude is greatest, is the prime cause of vibration, but the other pistons operate in a similar though diminishing manner, so that the problem becomes very complex, but in any case the periodicity due to the first cylinder is the greatest. It will further be seen that there is a torsional variation due to the octave vibration, so that if the period of explosion vibration can be located at 1,000 r.p.m., there will be another period at 1,500 r.p.m. due to piston inertia and another of a lesser extent at 750 r.p.m.

Enough has been said to show that the whole question of crankshaft vibration is a very complicated one, and being an inherent property of an elastic structure, cannot be *wholly* eliminated by any precautions in designing, and in spite of everything remains as something to be reckoned with. Some device, therefore, for dealing with the trouble *after* it occurs is indicated, and such a device, known from its inventor as the "Lanchester balancer," is the only remedy which can be applied.

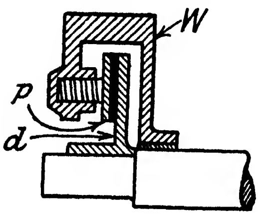


FIG. 26

It is shown diagrammatically in Fig. 26 where *W* is a flywheel capable of rotating freely on the outer or front end of the crankshaft; beyond it, is a friction disc *d*, keyed or otherwise secured to the shaft and rotating with it. A friction plate *p* is provided in the flywheel *W*, which is pressed against the disc *d* by means of a series of helical springs. So long, therefore, as there is no sudden change in the velocity of the crankshaft, the flywheel rotates with it at the same speed. When, however, changes occur suddenly in the crankshaft speed, as when tortional vibration takes place, the inertia of the flywheel acting through the friction surfaces acts as a drag, and in consequence this sudden movement of the shaft is restrained; as a result the objectionable vibrations are damped out and to a great extent rendered imperceptible. A common arrangement is to build up a small multidisc friction clutch, as shown in Fig. 27, and by adjusting the pressure of the springs, the apparatus may be adapted to any ordinary conditions of crankshaft vibration.

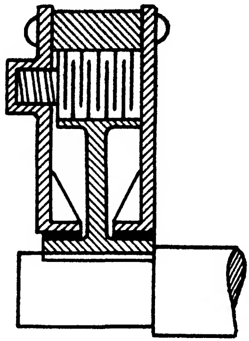


FIG. 27

It may be remarked here that in a six-cylinder crankshaft undamped with this device, there may exist an amplitude of vibration amounting to several degrees due to the various causes noted.

Crankshaft Stresses. It will be necessary now to consider further the question of crankshaft stresses from the quantitative point of view. We have already seen that it is necessary to combine two groups of stresses, (*a*) those brought about by the inertia of the reciprocating parts, and (*b*) those arising from the varying pressure of the gases within the cylinder. We have further seen that the former vary with the square of the running speed, while the latter are substantially constant for any speed (this statement being necessarily only approximate).

Dealing first with the inertia stresses, we may for the sake of clearness remind ourselves that the force of inertia is the variable accelerating or decelerating force on the reciprocating parts, and during one stroke, from top to bottom of the cylinder, the acceleration passes through a cycle. While the piston is being *accelerated* from zero to maximum velocity, which takes place during the first half of the stroke (while the crank is passing through 90°), the force is negative—it is a restraining one, and the piston is absorbing energy, let us say, from the flywheel, etc. Then, while the piston is being decelerated from maximum velocity to zero, which happens during the second half of the stroke (while the crank passes through approximately another 90°), the force is positive—it is a helping one; the piston is giving out energy, let us say, to the flywheel.

This cycle is repeated during the stroke from the bottom to the top of the cylinder, so that at a little after half stroke coming down, and a little before it going up, the piston velocity reaches a maximum, and its acceleration is zero; at the top dead centre, the piston velocity is zero, and acceleration at a maximum, while at the bottom dead centre velocity is again zero, but here the acceleration is somewhat less than at top dead centre due to the obliquity of the connecting rod.

Using the following symbols: $N =$ r.p.m.; $v =$ instantaneous velocity of piston f.p.s.; $f =$ acceleration of piston f.p.s.p.s.; $R =$ the crank radius in feet; $\theta =$ the crank angle from top dead centre in degrees; $c =$ length of connecting rod in feet; and $n =$ ratio of connecting rod to crank radius $= c/R$, then,

$$v = \frac{2\pi NR}{60}\left(\sin\theta + \frac{\sin 2\theta}{2\sqrt{n^2 - \sin^2\theta}}\right)$$

$$f = \frac{4\pi^2 N^2 R}{3600}\left(\cos\theta + \frac{n^2\cos 2\theta + \sin^4\theta}{\sqrt{(n^2 - \sin^2\theta)^3}}\right)$$

Here we may neglect those powers of θ which are higher than 1 and simplify the equation into

$$f = \frac{4\pi^2 N^2 R}{3600}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

from this the inertia can be calculated for any position of the crank.

For practical purposes, however, such accuracy is not needed, as we are, in general, only concerned with the maximum inertia forces, which as we have seen only occur at the top and bottom of the piston strokes. We may also remember that we are about to compare and combine the inertia diagram with the gas pressure, or indicator diagram, and this will commonly be drawn on some scale representing pounds pressure per square inch of piston area. It is, therefore, of advantage to express the inertia in the same terms, namely, pounds pressure per square inch of piston, so that the two quantities may be more readily combined. It is only necessary to express the weight of the parts in terms of the piston area; that is to say, if the reciprocating parts weigh 3·5 pounds and the piston has an area of 7 square inches, the weight of reciprocating parts per square inch of piston area will be $3{\cdot}5/7 = 0{\cdot}5$. We then have a very simple formula, and using the following symbols—

p = inertia pressure lb./sq. in. of piston area

w = total weight of reciprocating parts in lb./sq. in. of piston area

n = r.p.m.

s = stroke in inches

$\frac{r}{l}$ = ratio of crank radius to connecting rod length

then at top of stroke $p_t = 0{\cdot}0000142\ wn^2s \left(1 + \frac{r}{l}\right)$

and at bottom of stroke $p_b = 0{\cdot}0000142\ wn^2s \left(1 - \frac{r}{l}\right)$

Let us take a numerical example. A piston weighs with its gudgeon pin and rings 0·6 lb., it has a diameter of 3 in. (approximately), giving an area of 7 sq. in.; The weight of the upper portion of the connecting rod (weighed with the big end supported) is 0·45 lb. The total reciprocating weight is $0{\cdot}6 + 0{\cdot}45 = 1{\cdot}05$ lb., hence $w = \frac{1 \cdot 05}{7} = 0 \cdot 15$ lb./sq. in. of piston area. We will make $s = 5$ in. and the ratio of crank to connecting rod $\frac{r}{l} = \frac{1}{4} = 0 \cdot 25$ and $n = 1500$. then at the top of the stroke

$$p_t = 0 \cdot 0000142 \times 0 \cdot 15 \times 1500^2 \times 5\,(1 + 0 \cdot 25)$$
$$= 0 \cdot 0000142 \times 0 \cdot 15 \times 2{,}250{,}000 \times 5 \times 1 \cdot 25$$

whence $p_t = 30$ lb./sq. in. of piston area.

At the bottom of the stroke,

$$p_b = 0 \cdot 0000142 \times 0 \cdot 15 \times 2{,}250{,}000 \times 5 \times (1 - 0 \cdot 25)$$

whence $p_t = 18$ lb./sq. in. of piston area.

We can immediately convert this into pounds, by multiplying by 7, when we have

$p_t = 30 \times 7 = 210$ lb. inertia at top centre
$p_b = 18 \times 7 = 126$,, ,, bottom centre

Now let us see what these forces become at an engine speed of 3,500 r.p.m.

The square of 3500 = 12,250,000, so at top stroke

$$p_t = 0 \cdot 0000142 \times 0 \cdot 15 \times 12{,}250{,}000 \times 5 \times 1 \cdot 25$$
$$= 163 \text{ lb. per sq. in.}$$

while at the bottom dead centre

$$p_b = 0 \cdot 0000142 \times 0 \cdot 15 \times 12{,}250{,}000 \times 5 \times 0 \cdot 75$$
$$= 97 \cdot 8 \text{ lb. per sq. in.}$$

and again multiplying these by 7, we get the total forces, viz.

$p_t = 163 \times 7 = 1141$ pounds at top centre
$p_b = 97 \cdot 8 \times 7 = 684 \cdot 6$ pounds at bottom centre.

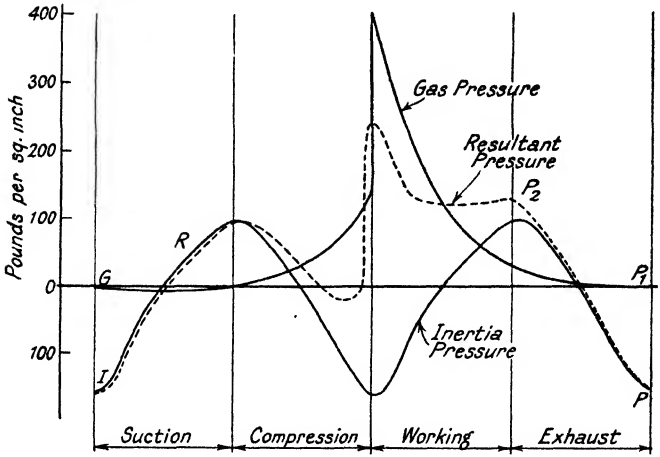

400
300
200
100
0
100
Pounds per sq. inch
Gas Pressure
Resultant Pressure
P_2
R
G
P_1
Inertia Pressure
I
P
Suction
Compression
Working
Exhaust

FIG. 28

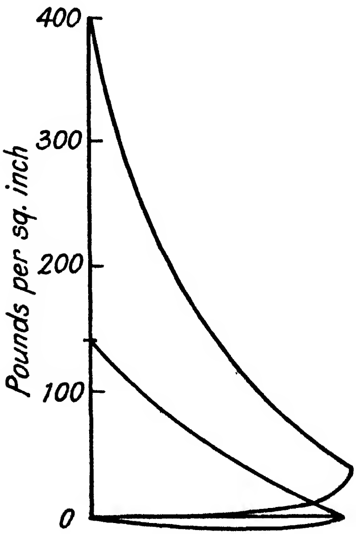

400
300
200
100
0
Pounds per sq. inch

FIG. 29

In Fig. 28 the curve IP shows the variation in inertia pressure for one complete cycle of four strokes. We have to combine this with the indicator diagram for the same period, and this we will assume to be as shown in Fig. 29. We first dissect the curves of the four cycles, and rearrange them more conveniently as shown in Fig. 28 by the line GP_1; we then make the algebraic sum of the ordinates about the zero line and the result of this addition (or in some cases subtraction) is shown by the line RP_2—the resultant pressure. It is a fortuitous circumstance that the maximum gas pressure coincides with one period of negative inertia, so that to some extent the amplitude of pressure variation is diminished. We will next draw this line RP_2 as a polar diagram showing more clearly what takes place during the two revolutions of the crankshaft. In this case the pressure ordinates are drawn as radii, and for clearness, the curve of the first revolution is drawn as a full line, and that of the second revolution dotted. The zero pressure line is shown as a full circle. This diagram is, of course, for a single-cylinder engine only (Fig. 30).

Next suppose we have to consider a two-cylinder engine as shown in Fig. 3 *ante*. There will be one explosion per revolution, so that we may superimpose a tracing of the diagram with the point P either coinciding with the point P of the first cylinder (but in phase with point X) if the cylinders are arranged as in Fig. 3, or we may place it at a suitable point in the circumference to coincide with the firing stroke of the second cylinder according to the firing sequence.

We will next examine a four-cylinder engine firing in the conventional order of 1 3 4 2. For the sake of clearness we will draw the diagram as in Fig. 31 with the four curves superimposed; we then see that the effect is to smooth out the irregularities which arise from the inertia stresses, so that in no case is there at any time

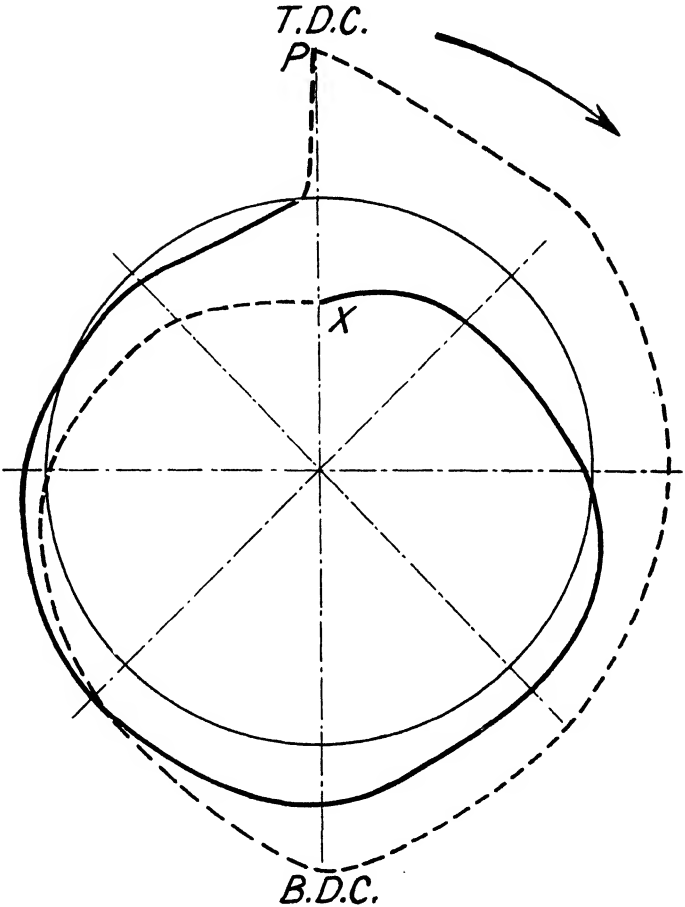


Fig. 30

Dotted Line = Working and Exhaust
Full Line = Suction and Compression

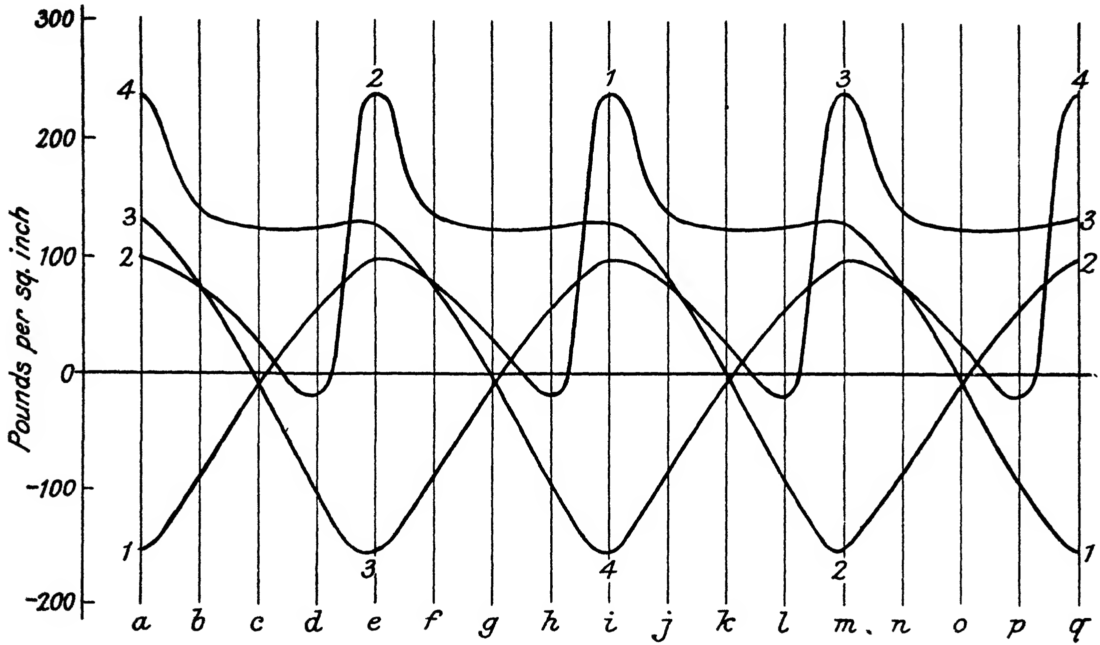

Pounds per sq. inch
300
200
100
0
-100
-200
4
3
2
1
2
3
1
4
3
2
4
3
2
1
a
b
c
d
e
f
g
h
i
j
k
l
m
n
o
p
q

FIG. 31

any negative force acting in opposition to the flywheel. If we take the algebraic sum of the ordinates at the points *a* . . . . *q* we find them to be as in Table I

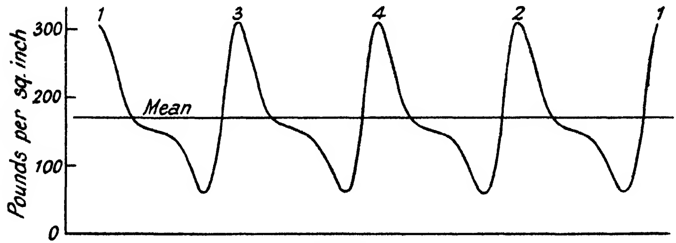


Fig. 32

below, and by drawing a diagram of these we obtain the final result shown in Fig. 32.

If we take the mean of the turning moments in this table we find it to be 179 (approx.) and the maximum variation from this is 310–179 = 131 lb.

TABLE I

Cylrs.	1	2	3	4	Total
a. .	− 160	+ 100	+ 130	+ 240	+ 310
b. .	− 90	+ 75	+ 75	+ 140	+ 200
c. .	− 10	+ 30	+ 0	+ 125	+ 145
d. .	+ 50	− 20	− 95	+ 125	+ 60
e. .					+ 310
f. .					+ 200
g. .					+ 145
h. .					+ 60
i. .					+ 310
j. .					+ 200
k. .					+ 145
l. .					+ 60
m. .					+ 310
n. .					+ 200
o. .					+ 145
p. .					+ 60
q. .					
				Mean	179·75 (approx.

We must remember that we have expressed our inertia and also our gas pressures in terms of pounds per square inch, and to obtain the actual crankshaft moments the figures in the last column of the table must all be multiplied by 7 sq. in. which we have taken as the area of our piston. This, of course, in no way affects the shape of the curves, but only their scale.

There is no need to draw another diagram for a six-cylinder engine, because the construction is similar except that as the impulses follow at 120° instead of 180° there will be three in each revolution instead of two, and six instead of four in the complete cycle. The student should construct a six-cylinder diagram of turning moments as an exercise.

Again, we may remember that the inertia pressure of 163 pounds was that occurring at a speed of 3,500 r.p.m., and that at a speed of 1,500 r.p.m. the inertia pressure at the top of the stroke was 30 lb. per sq. in. It is evident that at such a slow speed the inertia variation would be only about one-fifth of those shown in the figures, and consequently the variations of the gas pressure (which would remain the same) would be much less affected by the variation of inertia; from this we see that the torque moment curves would be very considerably modified in value, though their general characteristics would remain the same.

In making calculations, therefore, for a new design of engine, it is well to calculate the inertia stresses for three different engine speeds, of which one should be the highest we expect to attain; an inspection of the resulting torque moment diagrams for the full number of cylinders will afford a very fair idea of the roughness or smoothness of the running of the engine, and the consequent comfortable sensation or otherwise it will produce in the chassis.

In view of this discussion of the vibrations set up in the crankshaft we are better able to appreciate the advantage of fitting a central flywheel to minimize the disturbing effect of the whip in a long shaft, where one cylinder is at a considerable distance from the flywheel if this is placed at the rear end.

We cannot, of course, overlook the resulting disadvantages of a central flywheel which are—

1. The increased length (and therefore weight) of the engine case.

2. With side valves, and a camshaft in the crankcase, the shafts will be too close together to permit of a flywheel of the necessary diameter. (N.B. This does not apply in the case of an engine fitted with an overhead camshaft.)

3. The increased cost, as will be seen, of the flywheel, or alternatively the increased cost of a built-up crankshaft to permit of the mounting of a solid flywheel between two of the cranks.

We may now summarize what we have learnt as follows—

1. All crankshafts must be in static and dynamic balance.

2. Inertia stresses are positive and negative, and change in each quadrant of one revolution, namely, negative in the first quadrant starting from top dead centre; positive in the second quadrant; negative in the third; and positive again in the fourth as we approach the top dead centre.

3. Inertia stresses are unequal, and are substantially less at the bottom centre (positive) than they are at the top (negative).

4. The curve of alternation from negative to positive and *vice versa* is a sine curve, and as such can be resolved (by Fourier's Theorem) into a fundamental vibration, and an octave component of twice the frequency of the fundamental vibrations, with other harmonic vibrations

of four times or six times its frequency, and these harmonics occur with increasing frequency and decreasing amplitude to infinity.

5. As a general rule, in practice, only the second harmonic (the octave) is of importance though the higher frequencies—at least the fourth—may occasionally be perceived if they happen to synchronize with some other vibration.

6. These vibrations are inherent in any design, and so can never be eliminated.

7. As shown by Lanchester's model, in a four-cylinder engine, the fundamental vibrations are balanced, and the octave vibrations are unbalanced.

8. In a six-cylinder engine, both fundamental and octave vibrations are balanced.

9. In a two-cylinder engine with cranks together both fundamental and octave vibrations are unbalanced.

10. In a two-cylinder engine with cranks 180° apart the fundamental vibrations are balanced and the octave vibrations are unbalanced.

11. In a two-cylinder engine with cranks 180° apart and the cylinders opposed, both the fundamental and octave vibrations are balanced.

12. To avoid rocking couples, the right and left halves of the crankshaft must have "looking-glass" symmetry about the central plane.

13. The whipping of a crankshaft due to the inherent vibrations of torque can *only* be eliminated by means of some such device as the Lanchester friction damper.

Design of Crankshafts. We will now pass on to consider the design and construction of the crankshaft. There is a considerable variation in practice with regard to the material used for crankshafts, and this we will discuss more fully later: for the present it is enough to say that, at any rate for engines for normal use, quite a moderate class of steel is perfectly suitable. We must, of course, except the case of engines for aircraft purposes which do not in any case come within our province in this section, but by the time we have met all the requirements of torsional rigidity, freedom from whipping stiffness and rigidity of crank pins and journals, and the provision of sufficient bearing surface in a very limited length, we shall find that the dimensions of the shaft are such as to eliminate all risk of failure except from fatigue.

The two essential requisites are surface hardness and resistance to fatigue, and if the design is adequate from the view points we have enumerated above, it will certainly be found that the stresses we can calculate in the shaft come out quite absurdly low. We must not forget that the rigidity of any shaft depends upon the modulus of elasticity; now this is substantially the same for all steels, irrespective of their tensile strength, and since rigidity is the controlling factor in the design, any steel we may employ will give the same result in this respect. We may say at once that a straight carbon steel containing from ·35 to ·40 per cent of carbon will meet all the requirements of the case (excepting, as already mentioned, the case of aircraft engines).

There are advantages, too, from employing an ordinary carbon steel for car engines among which we may mention—

1. It is relatively cheap and easy to obtain.

2. Any engineering factory is familiar with the methods of forging it.

3. The necessary heat treatment is simple, and does not involve risks from slight errors in this respect.

4. It can be obtained in very uniform quality. From this we see that it is generally more dependable than the very high tensile steels which are very sensitive to slight variations in treatment, which, if wrongly carried out, may result in conditions which are highly dangerous.

Fracture of a carbon steel shaft, carefully designed, is a rare occurrence, and when it does take place is generally due to fatigue resulting from periodic vibration from torsional stresses. No elaborate alloy steel is proof against this. We have already seen that it can be avoided in two ways: (*a*) by fitting a vibration damper, and (*b*) by increasing the shaft diameter so that its natural period of vibration is raised above that of the inherent torsional vibrations, and so is not likely to synchronize with them.

There is another cause of failure which should be apparent to any student of machine design, and that is the lack of provision of a liberal radius wherever the diameter changes. Too often this is omitted on one pretext or another by even experienced designers who should know better—a frequent instance being in the case of the wedge-shaped projection known as an oil flinger, and as one of these is invariably required immediately behind the rear bearing (between it and the flywheel, where the nodal point, already described, is usually situated, and where the stresses from torsional vibration are greatest) it may be a common source of failure.

The student must be determined never to make this mistake in any circumstances.

It may be remarked that it is no very great task to design a four-cylinder crankshaft which is free from noticeable torsional vibration, though in the case of six-cylinder shafts, which, as we have seen, are more prone to this on account of their greater length, it is not so easy; and, in fact, it is best to assume that a vibration damper will always be necessary in sixes and eights.

With regard to wear—as the speeds of engines have been increased, so has the difficulty from this cause. Working pressures also are much greater than formerly and the resulting load factor tends to become high. The only remedy for this is to make the crankshaft surfaces as hard as possible, and the bearing material as soft as is consistent with resistance to very high pressures without squeezing out. Grit in the lubricant cannot wholly be eliminated, and the softer the bearing metal, the more readily will it swallow up such particles as wash in between the surfaces.

At one time, the only way of obtaining a hard bearing surface (apart from the use of high alloy steels) was to resort to case-hardening—a very risky business with a solid crankshaft, owing to danger of distortion of the finished shaft during the necessary heat treatment. Almost the only way by which case-hardening could be accomplished was by using a built-up crankshaft, case-hardening the pins, and shrinking the whole into the crank webs, and then giving a finish grinding process to the entire shaft. At the present time, however, it is possible by nitrogen treatment to secure an exceedingly hard surface at comparatively low temperatures of heat treatment, so that no appreciable distortion is brought about, and the "core" of the steel is in no way adversely affected or made brittle. There is no doubt that surface-hardening of the crank pins and journals is always desirable and the extra cost is perfectly justified.

We will take first the simple case of a shaft for a single-cylinder engine.

In spite of what has been said we cannot altogether disregard calculations of stresses in crankshafts, if for no other reason, they are necessary to provide some starting point for the designer. The simplest way will be to make use of some approximate formula as a guide, and see where we shall arrive. We will first consider the stiffness of a plain shaft, subjected to a simple transverse bending load, Fig. 33. We will take an actual example.

If reference is made to Fig. 32 it is seen that the maximum pressure on the crank pin, taking inertia pressure into account, is, in the example we took, about 300 lb. per sq. in. We will assume a piston diameter of approximately 85·75 mm., which will give an area of 9 sq. in. The maximum pressure on the crank pin will, therefore, be $300 \times 9 = 2{,}700$ lb.; we will further assume the bearing centres to be 10 in. apart, and as we must provide for some deflection, we will allow, say, 0·003 or three-thousandths of an inch.

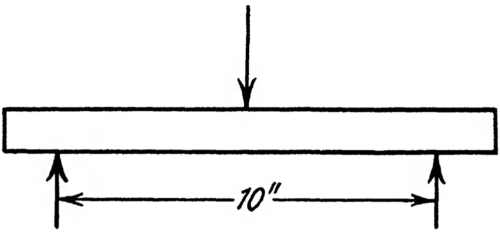


FIG. 33

The deflection is found from the formula

$$d = WL^3/48\,E.I.$$

where W = load at the centre = 2,700 lb.

L = length in inches = 10 in., so L^3 = 1,000

E = modulus of elasticity = 30,000,000

I = moment of inertia of the section = $\frac{\pi}{64} D^4$

for a circular section of diameter = D.

Substituting these values

$$d = \frac{2700 \times 1000}{48 \times 30{,}000{,}000 \times \frac{\pi}{64} \times D^4}$$

$$\frac{\pi}{64} = 0{\cdot}049$$

$$\text{so } d = \frac{{\cdot}0383}{D^4} \text{ . but } d = {\cdot}003$$

$$\therefore \frac{{\cdot}0383}{D^4} = {\cdot}003 \text{ or } {\cdot}0383 = {\cdot}003\, D^4$$

$$\text{so } D^4 = \frac{{\cdot}0383}{{\cdot}003} = 12{\cdot}77$$

$$D = \sqrt[4]{12{\cdot}77} = 1{\cdot}887$$

this being the diameter of shaft in the circumstances to give a theoretical deflection of $\frac{3}{1000}$ of an inch.

Experiments have been made to ascertain how nearly an actual crankshaft conforms to a straight bar in respect of stiffness, or ability to resist transverse bending stresses, and it has been found that a crankshaft is from ·25 to ·5 as effective as a straight shaft; that is to say, from ·5 in the case of a four-throw crank to ·25 in the case of a six-throw crank.

For purposes of rough computation we may take an all round figure of ·33 from which we understand that a straight shaft of given diameter will resist three times the transverse bending stress as will a normal crankshaft to give the same deflection.

Applying this to the example we have just used. we must assume a central load of $2700 \times 3 = 8100$ lb. Then proceeding as before

$$d = \frac{8100 \times 1000}{48 \times 30{,}000{,}000 \times {\cdot}049 \times D^4} = \frac{{\cdot}1145}{D^4}.$$

and assuming the same deflection allowable, viz. 0·003 in., we have

$${\cdot}003 = \frac{{\cdot}1145}{D^4} \text{ so } D = \sqrt[4]{38{\cdot}2}$$

whence $D = 2{\cdot}4279$

say 2·43 the required diameter for a normal crankshaft to give $\frac{3}{1000}$ in. deflection.

Now we assumed that the cylinder had a diameter of 3·45 in. and we see that the crankshaft is approximately two-thirds of the bore of the cylinder. We may take this as an approximate rule for a single-cylinder engine, and shall not be very far out. So we may write the rule thus,

Single-cylinder engine crankshaft, if D = cylinder bore and

$$d = \text{diameter of journals}$$

$$d = 0{\cdot}666\,D.$$

We have next to determine the length of the crank pin. On this, as the pressure is alternating, we may allow 1,000 pounds per sq. in. of projected area, and we know the piston pressure is 2,700 lb.: let l = the length of the bearing, then $d \times l \times 1000 = 2{,}700$, let d = 2 in. and substituting

$$2 \times l \times 1000 = 2700 \text{ or } l = \frac{2700}{2000} = 1{\cdot}35 \text{ in. only.}$$

We have seen that it is generally inadvisable to make the pin shorter than the diameter; consequently, on a crank pin 2 in. × 2 in. = 4 sq. in. of projected area we have a specific pressure of

$$\frac{2700}{4} = 675 \text{ lb. per sq. in.}$$

which will be a very moderate figure.

We will next see what the load factor on the journal bearings will be: the load factor is the product of pressure and rubbing velocity. We know the maximum pressure is 2,700 lb. and this is distributed over two bearings, so that they each carry 1,350 lb. at the worst condition. The speed in feet per second at, say, 1,200 r.p.m. will be—

$$v = \frac{\pi d}{12} \times \frac{\text{r.p.m.}}{60} = \frac{\pi \times 2}{12} \times \frac{1200}{60}$$

$$v = 10{\cdot}5 \text{ ft. per sec.}$$

$$10{\cdot}5 \times 1350 = 14{,}175$$

We know that this should not exceed 20,000 lb. ft., and we have assumed a somewhat hypothetical case, as a single-cylinder engine would be unlikely to carry so high a working pressure as 300 lb. per sq. in.

We will next see what the load factor on the journals will be. We will again assume a speed of 1,200 r.p.m. Then the pressure on *one* journal is 1,350 as before, and the diameter of the journal is 2·43, so the load factor will be

$$L.F. = 1350 \times \frac{\pi \times 2{\cdot}43}{12} \times \frac{1200}{60} = 17{,}150$$

It will be seen that we have taken the load as that due to piston pressure for the sake of clarity, but actually this is not correct, since the proper working of any bearing depends upon its remaining *cool*, which in its turn depends on the *average* amount of heat put into it; in other words, upon the *average* pressure. Now this is not the explosion pressure (which is of only momentary duration) but the *mean indicated pressure*—a much lower figure. In fact, this is not likely to be much above 140 lb., instead of 300 which we took from the maximum pressure: our load factor figures will be reduced in proportion with satisfactory results. Thus the crank pin load factor will only be

$$14175 \times \frac{140}{300} = 6100 \text{ lb. ft.}$$

and the journal figure becomes

$$17150 \times \frac{140}{300} = 8000 \text{ lb. ft.},$$

again a very moderate figure.

We have now obtained the main dimensions of the shaft, but we have not dealt with the crank webs. Suppose the stroke to be 5 in.; as the journals are 2·43 in. diameter, allowing for an adequate radius (not less than $\frac{3}{16}$ of an inch) and a flat surface to take side thrust, there must be a circular facing not less than $3\frac{1}{4}$ in. diameter, so the webs can be $3\frac{1}{2}$ in. wide. Again we find the necessities of stiffness far predominating over requirements of strength.

To withstand the turning moment a very small thickness would be necessary, but to prevent any tendency for the crank to "spread" when the explosion takes place, a thickness of at least $1\frac{1}{8}$ in. would be needed to furnish also a rigid support for the crank pin. There is no need to make any calculations as to strength of webs, their width is determined by the diameter of the journal and the surrounding fillet with the flat rubbing surface, and this, as we have seen, gives a very wide web: roughly speaking the thickness of the web may be made approximately one-third of its width, which is 1·33 the diameter of the journal, and to save weight it is usual, and quite permissible, to bevel off the outer sides of the webs, as shown in Fig. 34.

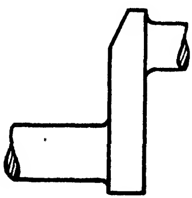


Fig. 34

For a single-cylinder engine we know that balance weights will be required. If possible, these should be integral with the webs, to avoid any possibility of danger from their flying off. If on the score of expense this is not deemed possible they may be made independently and bolted on, but in such a case there should be a registering fillet to take the centrifugal force, the bolt being then freed from shearing stresses, as shown in Fig. 35. An alternative method being to prolong the webs somewhat and bolt a slotted balance weight

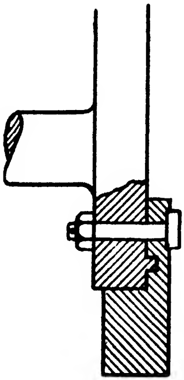


FIG. 35

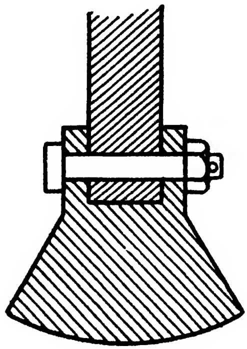


FIG. 36

through the web, Fig. 36; the bolt is then in double shear, and there is little, if any, risk of trouble.

Four-cylinder Shafts. These may have two or three bearings, only very large engines affording any neces-

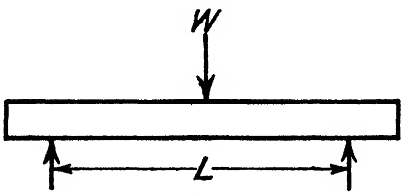


FIG. 37

sity for five. With a two-bearing shaft the diameter must be relatively great to withstand the bending stress, and here again we can pursue the method already followed for a single-cylinder shaft. The calculation of the bending moment is somewhat different, as it is unsymmetrical. For example, in a centrally loaded beam the bending moment is $\frac{WL}{4}$, Fig. 37. With an unsymmetrical load it is

$$\frac{W (L_1 \times L_2)}{L}, \text{ Fig. 38,}$$

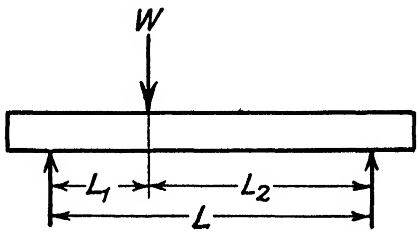


FIG. 38

and in this case the deflection of a straight shaft will be, if d = the deflection of a centrally loaded beam, and d_1 that of an unsymmetrically loaded one, then

$$d_1 = d\left(1 - \left[\frac{L_2 - L_1}{L} \right] \right)$$

We can then proceed to work out the maximum diameter, using as before three times the actual load on account of the shaft having only one-third the stiffness of a solid bar.

A two-bearing crankshaft will be of the form shown in Fig. 39, and will not be in balance unless there is an extra weight between the middle cranks to balance the two outer webs. This, however, is easily arranged by making the centre portion of large diameter, as shown in the figure, which will have the further advantage of making the central portion stiffer, though at the same time increasing the centrifugal force where it is least desirable. A better plan, therefore, is to prolong the

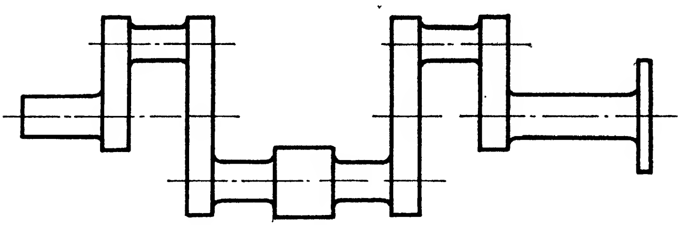


FIG. 39

two outer webs to form their own balance weights, as shown in Fig. 40. Here there is no increased unbalanced centrifugal force, and there will be less vibration.

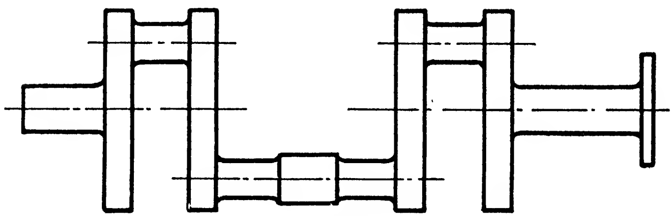


FIG. 40

Three-bearing Shafts. When the central bearing is fitted, as it should be in all but the smallest engines, the case is easier to deal with. The shaft, Fig. 6, is seen to be balanced at the outset, there is the same weight of metal on each side of the principal axis, so balance weights are not primarily necessary. As already stated, the writer is in favour of individual balancing, but opinions on this differ, and it may be well to point out that while they relieve the stress in the journal bearings, they do undoubtedly tend to increase torsional vibration, and they may, therefore, be regarded with suspicion.

Six-cylinder Cranks. All the foregoing is applicable to shafts for six-cylinder engines. As we saw in the earlier part of this section, a six-cylinder engine is fundamentally better balanced than a four, and consequently has less tendency to torsional vibration. As already mentioned, it may have seven or four bearings, and no hard and fast rule can be laid down as to which. Popular prejudice is largely responsible for supposing seven bearings to be better than four (largely influenced by the semi-technical Press), and this must not be lost sight of, but generally speaking it is easier to design a rigid six-cylinder shaft with four bearings, and the whole engine becomes simpler.

In any six-cylinder crank, however, a difficulty arises owing to the fact that the cranks are not arranged on one diameter. In other words, looked at from the end, a four-cylinder crankshaft is flat, while a six-cylinder shaft is triangular. This presents difficulties of manufacture. We will take the case of the crank shown in Fig. 7 with firing order 1 5 3 6 2 4, and for the sake of argument assume seven bearings, then the forging may be made in one plane and subsequently be twisted between each crank to bring the various cranks into their proper relative positions at 120° apart.

The disadvantage of this is that the twisting is not beneficial to the material, and though it is often done, it is clearly better (though more expensive) to forge the entire shaft from the solid, which, if properly done, actually results in an improvement in the condition of the steel. Such a forging is undoubtedly the best. As an alternative the shaft may be turned from the solid, and, in this case, the cranks may be formed as discs, resulting in a very stiff shaft. This is particularly suitable for one having four bearings, and the extra machining cost is justified. The question of weight must not, however, be overlooked, and, as will be seen, a great deal of latitude is afforded to the design in obtaining the best arrangement to suit a combination of circumstances.

Eight-cylinder Crankshafts. The straight eight-engine presents many problems, and none more involved than that of the crankshaft, since there are so many possible arrangements of firing order, and these, in turn,

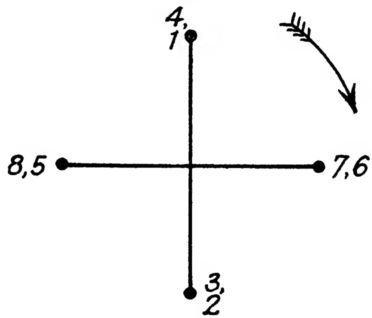


FIG. 41

are bound up with the difficult question of gas distribution. We will consider the matter step by step.

In the first place the engine as a whole may be considered as two four-cylinder engines coupled together: as a four-cylinder engine has two impulses per revolution, spaced 180° apart, so an eight-cylinder engine has four impulses per revolution, spaced 90° apart. To consider the simplest case we may assume the cranks of the front half of the engine to be arranged as a normal four-cylinder crankshaft, and the rear half to be another, but placed with its cranks 90° behind those of the first. The position of the eight cranks will be as in Fig. 41: the firing order of the front half will be 1, 3, 4, 2, and that of the rear half 5, 7, 8, 6; we can better illustrate it thus—

Front half . .	1		3		4		2	
Rear half . .		5		7		8		6

and as the cranks are interspaced we can assume these two lines of figures pushed together so that they read 1 5 3 7 4 8 2 6, which we may call Series 1, Order 1.

We can have other firing orders with the same arrangement of cranks, in fact, a series of eight, namely—

Order 1.	Series 1.	1 5 3 7 4 8 2 6
	,, 2.	1 5 3 6 4 8 2 7
	,, 3.	1 5 2 7 4 8 3 6
	,, 4.	1 5 2 6 4 8 3 7
	,, 5.	1 8 3 7 4 5 2 6
	,, 6.	1 8 3 6 4 5 2 7
	,, 7.	1 8 2 7 4 5 3 6
	,, 8.	1 8 2 6 4 5 3 7

See Fig. 42.

Now if we look at the diagram of this crankshaft seen in perspective, we find that it does not comply with the first condition for a multi-cylinder shaft; it is not in a condition of looking-glass symmetry, and as a consequence there would be a serious rocking couple in the longitudinal plane. We will, therefore, discard this arrangement. It is easily seen that we can arrange eight cranks in a large number of different ways, but we need not attempt to describe all these here, but inertia balance, for all these crank arrangements, is the same; the primary and secondary forces are in equilibrium, but the fourth harmonic is unbalanced. This may in any case be neglected as the force is very small. We have, therefore, to chose the arrangement which is least subject to the influence of rocking couples; we know that these can only be eliminated when the resultant of the inertia forces of all the reciprocating parts moving *downward* at any instant, is equal to the resultant of all the inertia forces of the reciprocating parts moving *upward*; also that these two resultants must be in opposite directions, and in line with each other, and the condition of looking-glass symmetry implies that the crank pins equally spaced from opposite ends must be in line with each other, namely, numbers 1 and 8, numbers 2 and 7, and so on.

With the arrangement shown in Fig. 42, each half of the engine is an ordinary four-cylinder, (the two halves being arranged at 90°, or one quarter of a revolution apart). Now in this, as it corresponds to one half the period of the secondary unbalanced force, we shall have the unbalanced forces at any instant acting in opposite directions, with, therefore, a maximum rocking couple.

We take another case in common use which we will call Order 2. Here the firing orders may be—

Order 2.	Series 1.	1 3 4 2 8 6 5 7
	,, 2.	1 3 4 7 8 6 5 2
	,, 3.	1 3 6 2 8 6 4 7
	,, 4.	1 3 5 7 8 6 4 2
	,, 5.	1 6 4 2 8 3 5 7
	,, 6.	1 6 4 7 8 3 5 2
	,, 7.	1 6 5 2 8 3 4 7
	,, 8.	1 6 5 7 8 3 4 2

See Fig. 43.

And, lastly, we take a third example—

Order 3.	Series 1.	1 3 2 4 8 6 7 5
	,, 2.	1 3 7 4 8 6 2 5
	,, 3.	1 3 2 5 8 6 7 4
	,, 4.	1 3 7 5 8 6 2 4
	,, 5.	1 6 2 4 8 3 7 5
	,, 6.	1 6 7 4 8 3 2 5
	,, 7.	1 6 2 5 8 3 7 4
	,, 8.	1 6 7 5 8 3 2 4

See Fig. 44.

In both Fig. 43 and Fig. 44 we have primary and secondary balance without any rocking couples, and, if there is any preference, it may perhaps be given to Fig. 44 which is easier to manufacture from the point

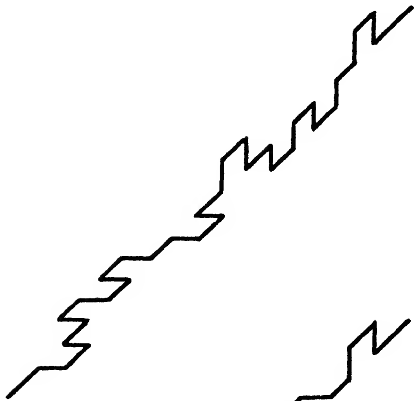


Fig. 42

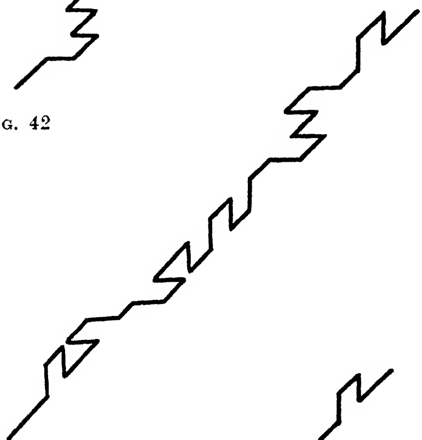


Fig. 43

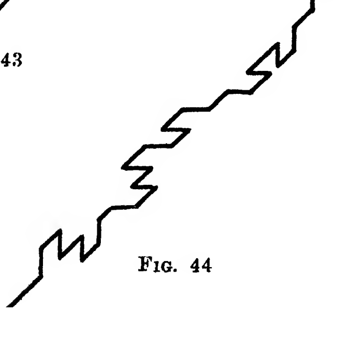


Fig. 44

of view of the forging. Fig. 44 has the further advantage when we take into consideration the load in the journal bearings due to centrifugal force. In the second example (Fig. 43) the shaft has all the adjacent cranks at right angles, and the centrifugal force of each pair is unbalanced in consequence.

With the third example (Fig. 44) each pair has the cranks *opposite* so that the centrifugal forces are balanced, and in consequence the bearings are less heavily loaded.

Rubbing Velocity. We have already discussed the question of load factor—the product of velocity × pressure—but velocity alone must receive attention and should not exceed 30 ft. per sec. with ordinary bearings, and that figure is somewhat high: if floating bearings are employed, the speed may go much higher as the actual rubbing velocity is halved, so that the figure of 30 ft. per sec. may be increased by 50 per cent to, say, 45 ft. per sec.

Flywheel Mounting. At one time flywheels were almost always mounted on a cone, the crankshaft being tapered at the rear end to fit into the flywheel. Theoretically the practice is correct from the point of view of stresses in the shaft, but the method is objectionable for several reasons; it is not by any means easy to fit the flywheel truly in its place, and it is apt to wobble, besides being very difficult to remove if this becomes necessary.

The modern method is to forge a flange on the end of the shaft and bolt the flywheel to it, but the flange must be carefully proportioned and stiff enough to withstand any disturbance due to the action of the clutch. A spiggot will be necessary to register the wheel to the shaft, and this should be formed on the *face* of the flange and should not be the edge of the flange itself. An illustration is given in Fig. 45. The bolts then can pull the surfaces together without fear of distortion, and a larger number of *small* bolts is preferable to a smaller number of large ones owing to the better distribution of stress afforded.

It is as well to remark here that the flywheel must, under no circumstances, be so mounted that it cannot easily be removed if necessary: for example, the nuts of the coupling bolts must be accessible when the engine is fully assembled, and the bolt heads must also be either accessible or held against turning, so that the flywheel may be taken off without removing the crankshaft.

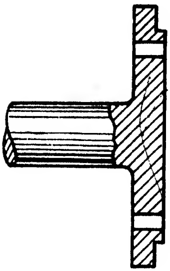


FIG. 45

We may also call attention to the point that keyways in any crankshaft are an abomination; timing wheels, for example, must be mounted on splines cut solid on the shaft. The reason for this is that although there may seem to be plenty of metal in the shaft to enable a woodruff key to be fitted, there is always the danger that the incessant torsional vibration will start a crack leading from the keyway, and it is impossible to say where this may extend to.

Materials for Crankshafts. We have already seen that quite simple steels are perfectly satisfactory for any ordinary crankshaft, but it will be as well to examine some of the simpler alloy steels which may be used, although their extra cost must not be overlooked. We will take three examples.

No. 1. MEDIUM CARBON STEEL

Chemical analysis.

Carbon	from 0·35 to 0·45 per cent
Silicon	0·30 per cent maximum
Manganese	from 0·40 to 0·85 per cent
Sulphur	0·06 per cent maximum
Phosphorus	0·06 ,, ,, ,,

Mechanical properties. As normalized see below.

Ultimate tensile strength .	35 tons/sq. in. minimum
Yield ratio	55 per cent minimum
Elongation	20 ,, ,, ,,
Reduction of area . .	40 ,, ,, ,,
Izod.	20 ft. pounds minimum

Forgings free from laps, cracks, twists, seams, and damaged ends.

Normalizing. Forgings to be normalized by heating to a temperature of 860° C., and after attaining a uniform temperature throughout they are to soak at that temperature for 15 minutes, and then be allowed to cool freely in still air.

Further Heat Treatment. Heat to 860° C. and quench in oil; reheat to 650° C. and allow to cool in still air.

Brinell hardness 217 to 196.

No. 2. 1 PER CENT NICKEL CHROME STEEL

Chemical analysis.

Carbon	from	0·30 to 0·40 per cent
Manganese	from	0·40 to 0·80 ,,
Silicon		0·3 per cent maximum
Sulphur		0·05 ,, ,, ,,
Phosphorus . . .		0·05 ,, ,, ,,
Nickel	from	0·9 to 1·1 per cent
Chromium		0·25 per cent maximum

Mechanical properties. After heat treatment as below.

Ultimate tensile strength .	40 tons/sq. in. minimum
Yield ratio	55 per cent ,,
Elongation	20 ,, ,, ,,
Reduction of area . .	45 ,, ,, ,,
Izod.	30 ft. lb. ,,

Heat Treatment. Normalizing. Heat to 850° C., soak for 15 minutes and cool in still air.

Further Heat Treatment. Heat to 850° C., soak for 15 minutes and quench in oil. Reheat to 600° C., soak for 15 minutes and quench in oil.

Brinell hardness 210 to 200.

No. 3. 3 PER CENT NICKEL STEEL

Chemical analysis.

Carbon	from 0·25 to 0·35 per cent
Silicon	0·3 per cent maximum
Manganese	from 0·35 to 0·75 per cent
Sulphur	0·045 per cent maximum
Phosphorus	0·045 ,, ,, ,,
Nickel	from 2·75 to 3·5 per cent

Mechanical properties. After heat treatment.

Ultimate tensile strength .	45 tons/sq in. minimum
Yield ratio	60 per cent ,,
Elongation	22 ,, ,, ,,
Reduction of area . .	50 ,, ,, ,,
Izod	40 ft lb. ,,

Normalizing. Heat to 850° C., soak for 15 min. and cool in still air.

Further Heat Treatment. Heat to 850° C., quench in oil, reheat to 550° C. and quench in oil.

Brinell hardness 225 to 215.

The above specifications explain themselves. They are all representative of good practice. The choice is largely a question of cost.

When the medium carbon steel is used it cannot have the wearing surfaces hardened by nitriting. This, giving exceedingly good wearing properties, provides an entirely satisfactory shaft at a relatively low cost.

FLYWHEELS

Function of Flywheel. We have now to consider the subject of flywheels. As everyone knows the function of a flywheel is to act as a reservoir of energy: that is to say, to store up and restore the energy furnished at the end of the crankshaft. We have seen that there is a perpetual fluctuation of this energy during the cycle of each cylinder. When there is only one cylinder the fluctuation extends over a cycle of two complete revolutions or 720°.

With two cylinders, as a rule, the fluctuation covers a period of one revolution or 360°.

With four cylinders it covers a period of half a revolution or 180°.

With six cylinders a period of 120°

With eight cylinders a period of 90°.

It is clear, therefore, that the fewer cylinders we have, the longer is the period available for an alteration in the speed of the engine, and in consequence the wider will be the variation in its speed. As we increase the speed we add energy to the flywheel, and as we decrease the speed the energy is given up.

Energy of Flywheels. Radius of gyration. In any revolving body, *the centre of gyration* is a point in the body, or system of bodies, at a certain distance from the axis of motion, in which the whole of the matter in revolution (as an equivalent condition) may be conceived to be concentrated.

The distance of the centre of gyration from the axis of motion is called *the radius of gyration.*

The moment of inertia is equal to the product of the square of the radius of gyration by the mass or weight of the body.

In what follows we will use the symbol I for the moment of inertia, and k for the radius of gyration. A flywheel may be analysed into a disc and a rim, see Fig. 46. And for a disc the moment of inertia is—

$$I = \frac{\pi}{64} D^4 = 0{\cdot}0491\, D^4$$

for a hollow cylinder

$$I = \frac{\pi}{64} (D^4 - d^4) = 0{\cdot}0491\ (D^4 - d^4)$$

The radius of gyration is—

$$\text{for a disc } k = \frac{D}{2} \times \sqrt{0{\cdot}5} = \frac{D}{2} \times 0{\cdot}7071$$

and for a hollow cylinder

$$k = \sqrt{\frac{D^2 + d^2}{8}}$$

and since the moment of inertia $= W \times k^2$, in which W varies as the square of the diameter we see that the flywheel effect must necessarily vary as the fourth power of the diameter, so a very small change in this will have a very considerable effect.

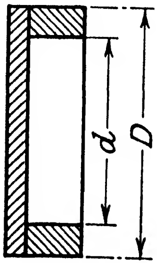


Fig. 46

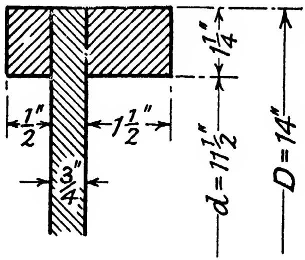


Fig. 47

It follows from this that if

W_r = weight of cylindrical portion in pounds

W_d = ,, ,, disc ,, ,,

k_r = radius of gyration of cylinder

k_d = ,, ,, ,, ,, disc

dE = energy stored or restored during a change of angular velocity from v_1 to v_2 ft. per sec.

$$\text{then } dE = (W_r k_r + W_d k_d) \frac{v_2^2 - v_1^2}{2g} = \text{ft. lb.}$$

Let us take an actual example, and assume a flywheel having the dimensions shown in Fig. 47.

The wheel is of steel weighing 0·284 lb. per cub. in., and is rotating at a mean speed of 2,000 r.p.m. The speed varies from 1,980 to 2,020 r.p.m.

for the disc $k_d = \dfrac{D}{2}\,0{\cdot}7071 = \dfrac{14}{2} \times {\cdot}7071 = 4{\cdot}95$

for the rim $k_r = \sqrt{\dfrac{D^2 + d^2}{8}} = \sqrt{\dfrac{14^2 + 11{\cdot}5^2}{8}} = 6{\cdot}4$

The weight of the disc is if t = its thickness

$$W_d = \frac{\pi}{4} D^2 \times t \times 0{\cdot}284 = \frac{\pi}{4} \times 14^2 \times {\cdot}75 \times 0{\cdot}284$$
$$= 32{\cdot}75 \text{ lb.}$$

and that of the rim is

$$W_r = \frac{\pi}{4}(D^2 - d^2) \times t \times {\cdot}284 = \frac{\pi}{4}(14^2 - 11{\cdot}5^2) \times 2$$
$$\times {\cdot}284 = 28{\cdot}4 \text{ lb.}$$

for the velocity we will take that of the mean radius of gyration, viz. $\dfrac{4{\cdot}95 + 6{\cdot}4}{2} = 5{\cdot}67$ in.

then its maximum velocity v_2 will be

$$\frac{2 \times 5{\cdot}67 \times \pi}{12} \times \frac{2020}{60} = 100 \text{ f.p.s.}$$

and the minimum velocity v_1 will be

$$\frac{2 \times 5{\cdot}67 \times \pi}{12} \times \frac{1980}{60} = 98 \text{ f.p.s.}$$

and substituting these values we have

$$dE = \left[(28{\cdot}4 \times 6{\cdot}4) + (32{\cdot}75 \times 4{\cdot}95) \right] \frac{100^2 - 98^2}{64{\cdot}4}$$

$$dE = \frac{344 \times 396}{64{\cdot}4} = 2115 \text{ ft. pounds.}$$

It is easily realized that the flywheel effect is very considerable on quite a moderate-sized flywheel. This is frequently made use of in emergencies: for example, a car, stalled on a hill too steep to climb, may be held on the hand brake while the engine is speeded up with the clutch disengaged, and by letting the clutch engage quickly the stored energy in the flywheel is made use of to assist the power of the engine.

A heavy flywheel has, however, its disadvantages, as its great moment of inertia is a hindrance to acceleration. Of recent years the fluid flywheel (which is necessarily of considerable weight) has come into use, and with this, acceleration to some extent is sacrificed to the other advantages the device possesses.

In the case of a multicylinder engine the cranks, with the attached masses of the big ends of the connecting rods, together form a very considerable rotating mass which has a large flywheel effect; as a consequence the necessity for a heavy flywheel disappears, and, as a rule, it becomes no more than a disc to which the single-plate clutch is attached and the combination meets all requirements particularly in view of the modern high speeds obtaining.

Materials for Flywheels. Cast iron, which was formerly used, is quite inadmissible owing to danger of bursting. The old rule for steam engine flywheels was, that up to peripheral speeds of 60 miles per hour (88 ft. per sec.) it was reliable; as the example just given shows, however, this speed is easily exceeded, and cast iron must, therefore, be ruled out.

Cast steel is not much better, as it is very apt to be spongy, and we are left with a steel forging as the only reliable material to use for the purpose. As the flywheel is generally entirely without any sort of boss, being attached to the crankshaft by a flange, it can be cheaply made by turning it out of a mild steel boiler plate.

For special purposes, e.g. for stationary engines, a heavy flywheel may be needed, and in that case the rim must be designed with regard to the tension in the material, and the following formulae are required.

Tension in Flywheel Rims.

Let s = stress in pounds per sq. in.

w = weight of a cubic inch of the material

d = *mean* diameter of rim in inches

n = r.p.m.

v = velocity of rim in ft. per sec.

$g = 32{\cdot}2$

$$v = \frac{\pi dn}{12 \times 60} = \frac{\pi dn}{720} = 0{\cdot}00436\, dn$$

$$s = \frac{12w}{g} v^2 = \frac{12}{32{\cdot}2} wv^2$$

$$w = 0{\cdot}26 \text{ for cast iron}$$

$$w = 0{\cdot}28 \text{ for steel}$$

then for cast iron

$$s = \frac{12 \times 0{\cdot}26}{32{\cdot}2} \times v^2 = 0{\cdot}097\, (0{\cdot}00436\, dn)^2$$

and for steel

$$s = \frac{12 \times 0{\cdot}28}{32{\cdot}2} \times v^2 = 0{\cdot}1043\, (0{\cdot}00436\, dn)^2$$

whence approximately

$$s = \frac{v^2}{10} \text{ for cast iron or steel.}$$

As in the case of the crankshaft, we may say that—providing mild steel (forged) is used for the flywheel and this has any usual type of clutch attached to it—no calculations for a flywheel are necessary; and a simple disc of adequate thickness to support the clutch without any danger of springing is all that is necessary in any multicylinder engine.

SUMMARY

The following summary will be useful to students and designers. Stiffness is the chief object to be attained, and the shorter the shaft is the better; hence in a four cylinder engine, three bearings are generally better than five; in a six cylinder four bearings are preferable to seven; and in an eight cylinder five are better than nine. The shaft will be stiffer, the bearings better proportioned, and the whole engine generally lighter and cheaper; this applies to all small cylindered engines, with very large cylinders the case may be different.

The approximate proportions to aim at in commencing a design are—

1. The journal diameter may be $\frac{2}{3}$ the cylinder bore; the length from 1 to $1\frac{1}{2}$ times the diameter.

2. Width of webs from 1·3 to 1·4 times the diameter of the journal, and thickness of webs one-third of their width.

3. Bearing pressures are determined by inertia pressure, plus gas pressure, plus centrifugal force of all rotating parts. Centrifugal forces in main journals *may* be balanced, those in crankpins are *unbalanced*.

4. Maximum rubbing speed of bearings should not exceed 30 f.p.s with ordinary bearings; above this floating bushes are indicated.

5. Load Factor (pressure × velocity) should not exceed 20,000 pounds feet with solid bearings; for this calculation the pressure should be the *mean* inertia, plus gas pressure, plus the maximum centrifugal force, all taken at the maximum engine speed.

FORMULAE USED

Moment of Resistance of Shafts.

Solid shafts $M = \frac{\pi}{16} D^3 f$

hollow shafts $M = \frac{\pi}{16}\left(\frac{D^4 - d^4}{D}\right)f$

where D = outside diameter
d = inside diameter
f = maximum *shearing* stress lb. sq. in—

Section Modulus.

Solid shaft $Z = \frac{\pi}{32} D^3$, hollow shaft $Z = \frac{\pi}{32}\left(\frac{D^4 - d^4}{D}\right)$

(Recommended size of bore $d/D = 0{\cdot}5$)

Centrifugal Force.

C.F. $= Wv^2/gr$
C.F. $= 0{\cdot}00017\ Wn^2s$ (stroke in feet)
C.F. $= 0{\cdot}0000142\ Wn^2s$ (stroke in in.)

where W = revolving mass lb.
v = velocity ft. per sec.
r = radius in feet
$g = 32{\cdot}2$
n = r.p.m.
s = stroke in in. or ft.

Velocity (instantaneous) *of Piston.*

$$v = \frac{2\pi NR}{60}\left(\sin\theta + \frac{\sin 2\theta}{2\sqrt{n^2 - \sin^2\theta}}\right)$$

where N = r.p.m. of crankshaft
v = instantaneous velocity of piston f.p.s.
f = acceleration of piston f.p.s.p.s.
R = radius of crank in feet

θ = crank angle from T.D.C. degrees
c = length of con. rod ft.

n = ratio of con. rod to crank radius $\frac{c}{R}$

Acceleration of Piston.

$$f = \frac{4\pi^2 N^2 R}{3600}\left(\cos\theta + \frac{n^2 \cos 2\theta + \sin^4\theta}{\sqrt{(n^2 - \sin^2\theta)^3}}\right)$$

$$\text{or } f = \frac{4\pi^2 N^2 R}{3600}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

Inertia of Piston at Ends of Stroke.

$$\text{At top of stroke, } p_t = 0{\cdot}0000142\, wn^2 s\left(1 + \frac{r}{l}\right)$$

$$\text{at bottom of stroke, } p_b = 0{\cdot}0000142\, wn^2 s\left(1 - \frac{r}{l}\right)$$

where p = inertia pressure lb./sq. in. of piston area
w = weight of reciprocating parts lb./sq. in. of piston area
n = r.p.m.
s = stroke in in.
$\frac{r}{l}$ = ratio of crank radius to con. rod length.

Deflection of Solid Shaft.

$d = WL^3/48$ E.I. for symmetrical loading, and

$$d_1 = d\left(1 - \left[\frac{L_2 - L_1}{L}\right]\right) \text{ for unsymmetrical loading.}$$

Diameter of Solid Crankshaft.

$d = {\cdot}666D$ (approx.)
where D = cylinder bore.

Load Factor.

$$pv = p \times \frac{\pi d}{12} \times \frac{\text{r.p.m.}}{60} = p \times 0{\cdot}00436d \times \text{r.p.m.}$$

Flywheels

Moment of Inertia, for disc.

$$I = \frac{\pi}{64} D^4 = 0{\cdot}0491\ D^4$$

for hollow cylinder $I = \frac{\pi}{64}(D^4 - d^4) = 0{\cdot}0491\ (D^4 - d^4)$

Radius of Gyration.

For disc $k = \frac{D}{2} \times \sqrt{0{\cdot}5} = \frac{D}{2} \times 0{\cdot}7071$

for hollow cylinder $k = \sqrt{\frac{D^2 + d^2}{8}}$

Energy Stored or Restored by Flywheel $= dE$

$$dE = (W_r k_r + W_d k_d) \frac{v_2^2 - v_1^2}{2g} = \text{ft. lb./sec.}$$

where W_r = weight of cylindrical portion lb.
W_d = ,, ,, disc lb.
k_r = radius of gyration of cylindrical portion
k_d = ,, ,, ,, ,, disc
v_2 = lower velocity of wheel
v_2 = higher ,, ,, ,,
g = 32·2

Tension in Flywheel Rims.

when s = stress in lb./sq. in.
w = weight of a cubic inch of material
d = *mean* diameter of rim in inches

$n =$ r.p.m.
$v =$ velocity of rim ft./sec.
$g = 32{\cdot}2$

$$v = \frac{\pi dn}{12 \times 60} = \frac{\pi dn}{720} = 0{\cdot}00436dn$$

$$s = \frac{12w}{g} v^2 = \frac{12}{32{\cdot}2} wv^2$$

$w = 0{\cdot}26$ for cast iron, $0{\cdot}28$ for steel

$$\text{for cast iron } s = \frac{12 \times 0{\cdot}26}{32{\cdot}2} \times v^2 = 0{\cdot}097\ (0{\cdot}00436dn)^2$$

$$\text{for steel} \quad s = \frac{12 \times 0{\cdot}28}{32{\cdot}2} \times v^2 = 0{\cdot}1043\ (0{\cdot}00436dn)^2$$

whence approximately $s = \dfrac{v^2}{10}$ for cast iron or steel.

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