Forced Lubrication.

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(MEMBER).

THE theory and practice of lubrication are subjects of such importance, that it is not surprising to find a very extensive literature on them.

The mathematical theory has been well treated, and experimental figures of a varied nature are available to confirm or correct the theoretical treatment. While taking for granted the established results of those who have investigated the matter, it may be useful to any who wish to refer to the original communications to give a slight bibliography.

Beauchamp Tower.—Proc. I. M. E. 1883, 1885, 1888, 1891. Reports on Friction Experiments. These classic researches were made chiefly on pedestal and saddle bearings. The pressures were taken to high figures, but the speeds were not high. The investigations on the oil film are particularly interesting.

Osborne Reynolds.—Trans. Royal Society, 1886. This gives a full mathematical treatment of Tower's results, and may be taken as the foundation of the modern theory of lubrication.

Nicholson.—Trans. Manchester Association of Engineers, 1907—8. A useful paper and discussion. It co-ordinates the results of various investigators, among others—

Stribeck (Z. d. V. d. Ing. September 6th, 1902).
Heimann (Z. d. V. d. Ing. Vol. 49, p. 1161).
Sommerfeldt. (Z. f. Math. u. Physik, Vol. 50, pp. 97—155).

Ewing and Jenkin.—Phil. Trans. 1879. On starting from rest.

Lasche.—Traction and Transmission. January, 1903. Also published separately, "On the Design of Bearings for High Speeds." An interesting account of experiments for the A. E. G. Radiation from and cooling of bearings and variation of temperature from point to point are among its most interesting sections.

Morcom.—Proc. Inst. Mech. E. 1897. Deals particularly with double-acting engines using forced lubrication.

Goodman.—Pamphlet on "Friction and Lubrication of Cylindrical Journals." 1890.

Archbutt and Deeley.—Treatise on "Lubrication and Lubricants."

Thurston.—"Friction and Lost Work."

Michell.—Lubrication of Plane Surfaces (Z. f. Math. u. Physik, Vol. 52, p. 123).

A number of other papers might be quoted, but these give a considerable amount of information and fairly cover the ground.

The one outstanding conclusion is that the modern theory of lubrication is the theory of the oil film. The old ideal of a coefficient of friction gives place to the more suggestive theory that the resistance to motion is due to the shearing of a film of oil, which more or less completely prevents metallic contact and abrasion. The importance of the film is shown by considering that the resistance of a fully lubricated surface may be only one per cent. of a similarly loaded surface in which an oil film is not maintained. Resistance to shearing depends upon the viscosity of the lubricant, the thickness of the film, and the area of film in shear. The temperature of the film may alter its viscosity; the extent of the film may not be equal to the extent of the bearing; the thickness of the film may not be such as to entirely prevent abrasion, and the clearance in the bearing may be irregularly distributed and inaccurate, and similar disturbances may be created by bad alinement of the shaft or its springiness, so it is not possible to entirely solve the problem. As usual in engineering, theory may direct or explain practice, but experience must determine it. Certain positive conclusions, however, may be taken as established:—

(1) The resistance decreases with increasing thickness of the film.

(2) The resistance increases with the viscosity of the lubricant.

(3) The point of nearest approach is approximately 90 degrees from the line of load.

(4) The points of maximum and minimum oil film pressure are approximately at equal distances from the point of nearest approach.

(5) As the speed increases the points of maximum and minimum oil pressure get further and further apart, till at very great speeds they are in the line of load.

(6) As the speed increases the eccentricity of the oil film becomes less.

(7) The concentric position is the one of least resistance.

(8) Oil should be supplied at a point where the supply pressure is greater than the film pressure.

(9) The loading for a given speed must not exceed a certain limit at which the oil film is broken.

(10) This limit may be increased by lengthening the bearing, so increasing the cooling influence on the bearing.

(11) Oil grooves wrongly placed may destroy continuity of the film.

(12) A motion of pure rotation produces automatic maintenance of the film, provided the supply is adequate.

(13) The temperature varies throughout the bearing, the highest temperature being at the point where the film is thinnest.

Further, in the case of a reciprocating load we know that

(1) A reciprocating load irrespective of rotation produces automatic lubrication.

(2) Heavier mean loads can be supported if the direction of load is reversed, because the lubricant is more vigorously sucked in, and the retardation of surfaces approaching one another normally, increases very rapidly as the film becomes thinner.

Generally speaking, failure of lubrication is caused by rupture of the film due to :—

(a) Inadequate supply of lubricant.

(b) Reduction of the viscosity arising from excessive heating, either general or local.

(c) Badly placed oil grooves.

(d) Overloading.

(e) Grit.

(f) Impurities, such as water, reducing the film-forming quality of the oil.

Assuming that the bearings are not overloaded, that system of lubrication will be best, which best ensures that other causes of failure shall not occur.

The more one studies the question the more does forced lubrication appear best to meet the requirements. Its acknowledged superiority over other systems for high speed steam-engines suggests its adoption for the motor car engine. Various splash and gravity systems have been very carefully designed and worked out for car motors, and the success attained is great, so great that it is easy to argue that the success is good enough. But practice in the long run always pronounces in favour of the theoretically best, and it is this which explains the increasing favour with which pressure supply meets.

As a measure of its popularity, in Table A., p. 373, are given particulars of the lubrication systems of a number of well-known motor cars, of which no fewer than eighteen use oil under pressure.

The data which various manufacturers have kindly supplied will, it is hoped, serve not only to establish this point, but also to indicate accepted practice as regards the arrangement, etc.

There is considerable difficulty in obtaining full figures and particulars of up-to-date results and methods, such as oil consumptions, oil temperatures, wear over long periods, oil grooves, etc., and the table is necessarily long and incomplete. In fact information about forced lubrication, both descriptive and experimental, is rather scanty. Nearly all the experimental work has been done on bearings lubricated by other systems.

Some experiences with forced lubrication do not at once fit in with theory, and it may be of interest to describe a few observations and experiments.

One of the most obvious things to examine was the actual saving in friction, if any, which occurred with forced lubrication. A trial was made on a 120 b.h.p. engine at 450 revs. per minute. The engine was run unloaded with oil pressures of 30 lb. and 5 lb. per sq. in., and without pressure, the supply being maintained with a syringe. A large number of no-load cards were taken and the i.h.p. averaged out. The results of the trial showed :—

(1) That the engine was quieter the higher the pressure.

(2) The friction i.h.p. averaged 2·13, 2·41, 3·33, with 30, 5, 0 lb. per sq. in. pressure respectively.

There is some trouble in explaining this result, for one would expect the cooler oil and more complete oil film with the higher pressure to increase rather than decrease the resistance. Some light is thrown on the case by the quietening action, which means that the film thickness in reciprocating bearings was better maintained. In addition such a result may occur in a steadily loaded bearing, owing to the more copious supply preventing excessive local heating of the film, which would lead to rupture. This is borne out by the experience that an engine with forced lubrication takes longer to "run in" than one with splash or gravity supply, and further that its bearings take longer to take up a high polish.

An experiment on this point was made in the following way :—

A 4 in. shaft was run at 1,000 r.p.m. in two bearings 10 in. long, between these was a bearing $10\frac{1}{2}$ in. long, loaded by means of a spring and lever. The outer bearings were supplied by a gravity supply, the inner at varying pressures. The journal was driven by an electric motor. Curves 1 and 2, Fig. 1, give the horse power with loads of 186 lb. and 5,130 lb. respectively. The temperature was 168° F. in the one case and 200° F. in the other when run for some time at 25 lb. per sq. in. pressure oil supply.

A series of trials were taken with a load of 24 lb. per sq. in., applied by means of heavy discs on a bearing 10 in. long by 4 in. diameter. The bearing had a horizontal oil groove cut, and this was tried in four positions:—

(1) On the loaded side.
(2) Opposite the load.
(3) On the right-hand side at right angles to the load, with rotation as in Fig. 2.
(4) Opposite to (3) Fig. 2.

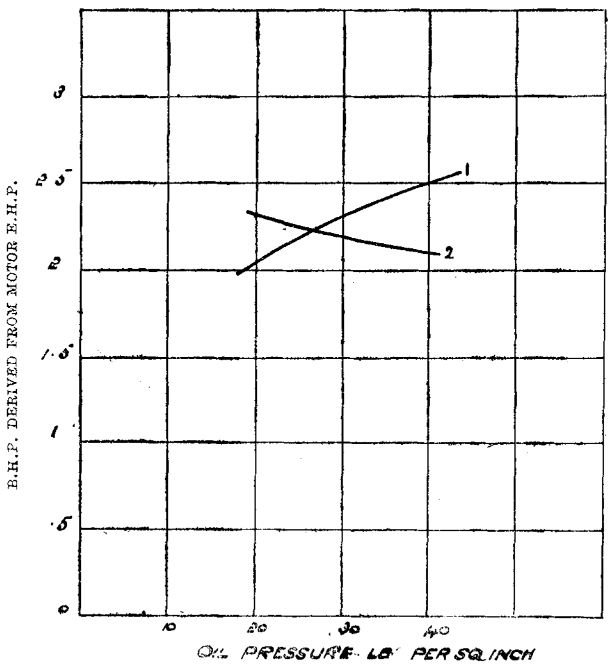


FIG. 1.

Diagrams were taken of the oil pressure at various points, a selection of such diagrams being given in Fig. 2: The curves may be compared with those composed by Dr. Nicholson from Sommerfeldt's figures. They differ noticeably from these curves owing to the forced feed, and the oil groove causes considerable modifications in their form. The pressures are not in equilibrium, because the pressures at other sections nearer the ends were out of balance in an opposite sense. In some cases high pressures were recorded nearer the ends than at the centre. Generally the diagrams show that forced lubrication is less likely to fail owing to faulty application than other systems.

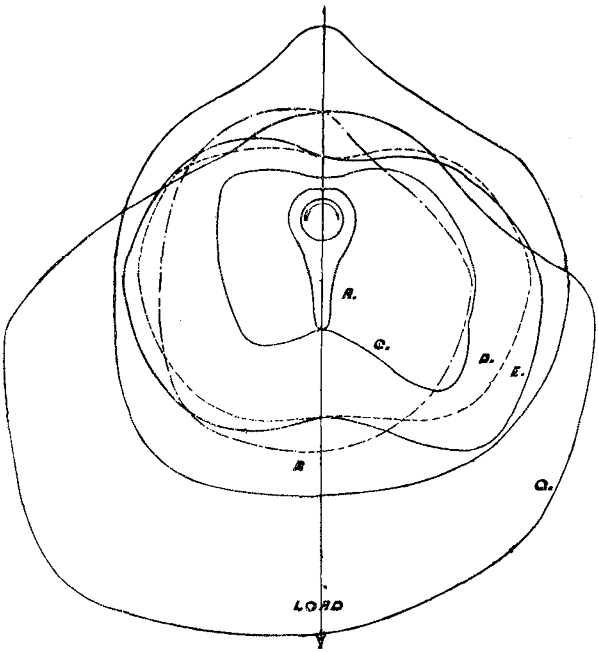


FIG. 2.

The index to the curves of Fig. 2 is as follows :—

A. 500 r.p.m. 20 lb. per sq. in. oil pressure.
Groove position (1) 24 lb. per sq. in.

B. 1,000 r.p.m. 20 lb. per sq. in. oil pressure.
Groove position (2) 24 lb. per sq. in.

C. 1,000 r.p.m. 20 lb. per sq. in. oil pressure.
Groove position (1) 24 lb. per sq. in.

D. 1,000 r.p.m. 40 lb. per sq. in. oil pressure.
Groove position (1) 24 lb. per sq. in.

E. 500 r.p.m. 40 lb. per sq. in. oil pressure.
Groove position (1) 24 lb. per sq. in.

F. 1,000 r.p.m. 40 lb. per sq. in. oil pressure.
Groove position (2) 24 lb. per sq. in.

G. 660 r.p.m. 20 lb. per sq. in. oil pressure.
Groove position (2) 126 lb. per sq. in.

Run at 1,000 r.p.m. and 40 lb. per sq. in. oil pressure, the final oil temperature reached was taken and found to be :—

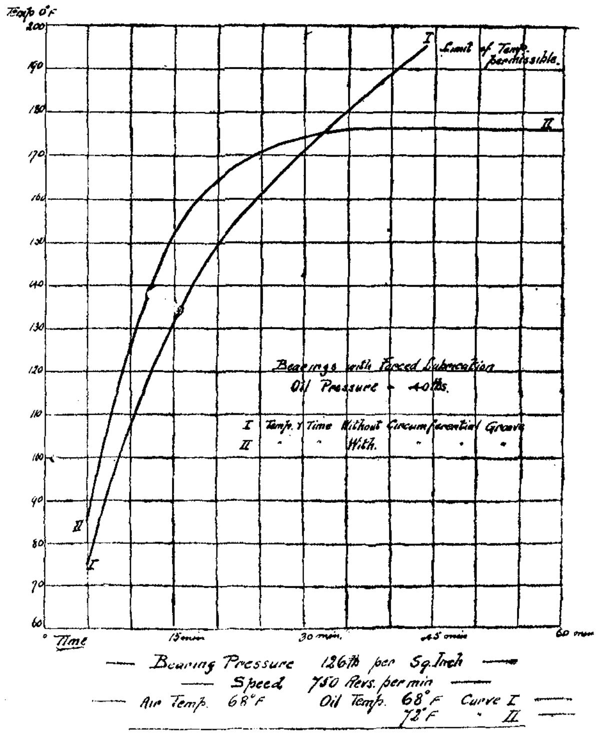


FIG. 3.

Oil groove in position (1) 74 degs. F. above air temperature.
,, ,, (2) 62 degs. F. ,, ,,
,, ,, (3) 60 degs. F. ,, ,,
,, ,, (4) 62 degs. F. ,, ,,

It would appear that position (3) Fig. 2 is the best position.

But where a horizontal groove is necessary (usually only on the flywheel end bearing) it is better to make a compromise, to allow for the conditions suitable for starting up; placing the groove on the side (3) from 15 degs. to 45 degs. up.

On a bearing with a reciprocating load a longitudinal groove is probably not necessary at all; if cut, the load should be assumed to act in the direction holding for the greatest proportion of the cycle.

A further series of trials were run at heavier loads with spring loading, which it is unnecessary to give in detail. The most important conclusion to be drawn was that the benefits of forced lubrication were best realized with a circumferential groove, which enabled heavy loads to be taken at high speeds. The quantity of oil passing is, of course, greater with this groove, and also the resistance.

As an example, the curves in Figs. 3 to 5 give temperature, e.h.p. and quantity of oil passing with 126 lb. per sq. in. loading,

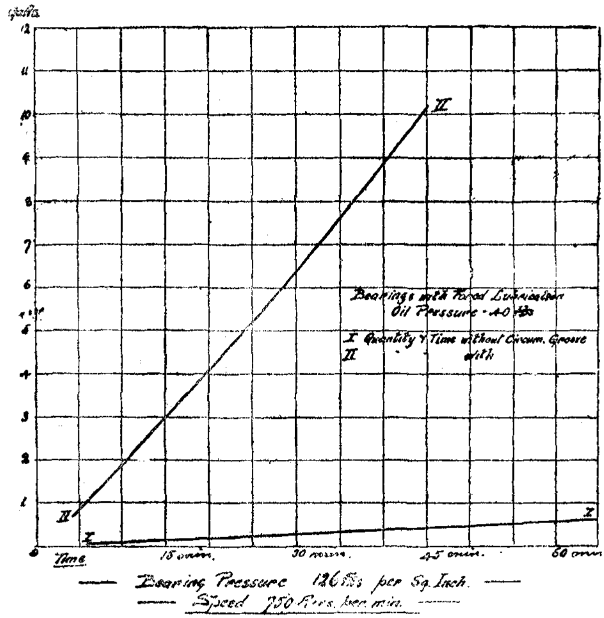


FIG. 4.

750 r.p.m. and 40 lb. per sq. in. oil pressure. The curves II. are with circumferential groove; the curves I. are with two horizontal grooves $2\frac{1}{2}$ in. apart on top of bearing, joined by a short circumferential arc. The oil inlet is at the top centre.

These trials have all been made with steady loads, and are consequently not of direct application to the case in point. Trials with reciprocating loads do not appear to have yet been made. Such experiments are difficult to make, but by no means impossible, and the results would be interesting. The author has made a few experiments, but, so far, of an inconclusive character. There is no doubt that reciprocation helps lubrication, and in particular forced lubrication is, under this condition, extremely efficacious. The quietening of an engine by raising the oil pressure is a direct proof of this.

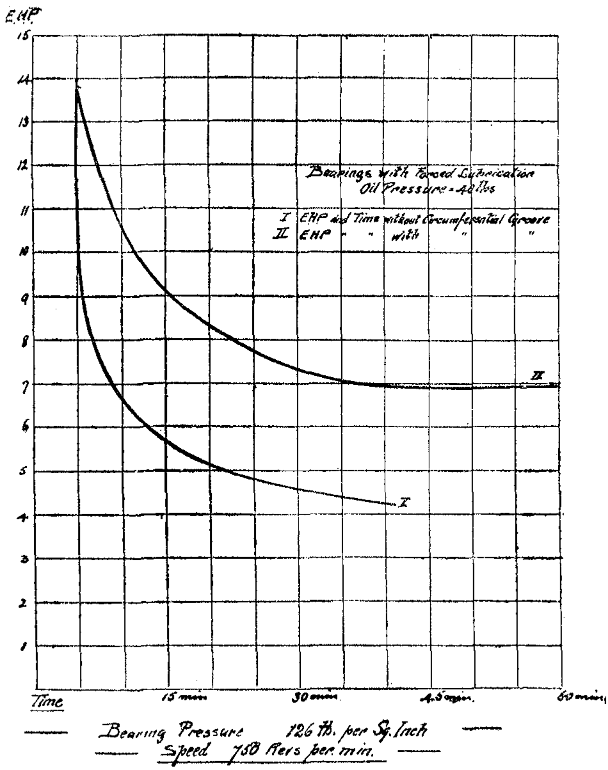


Fig. 5.

Possibly experiments on gelatine separating two cylinders would give valuable information, through the analogy pointed out by Osborne Reynolds and used by Michell in his sliding plate investigations.

The chief points in favour of forced lubrication brought out by the trials were, its more rapid adaptability to various conditions, its very positive maintenance of the all-important oil film, and the simplicity of the provisions necessary to ensure perfect lubrication. With splash and gravity systems elaborate oil grooves, troughs and

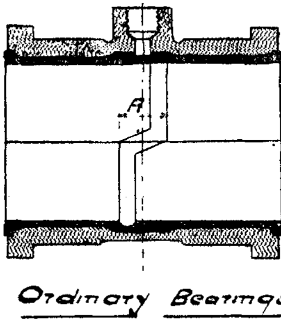


FIG. 6.

oil ways are required, and often they are cut without due regard to theoretical considerations. With forced lubrication very simple oil grooves are satisfactory. All that is necessary is to provide a

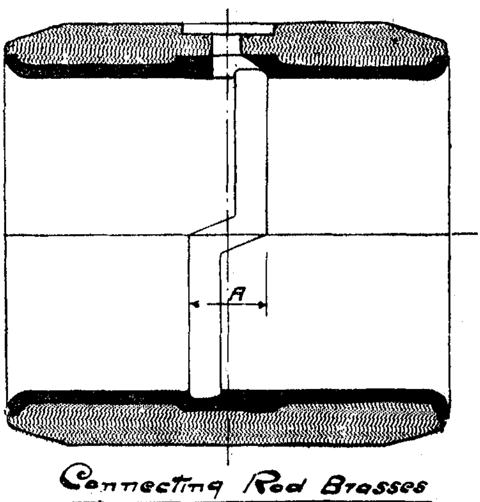


FIG. 7.

circumferential groove whereby a supply of oil is ensured at whatever point the minimum film pressure exists; the oil at this point will be forced right along the bearing, ensuring a perfect supply.

Where more circulation is required, one or more horizontal grooves may be cut in the bearing at suitable points, forming practically an oil pad, and also by increasing the circulation, having an important

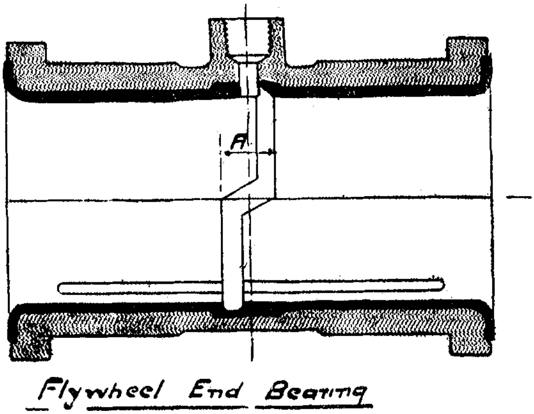


Fig. 8.

cooling effect on the bearing. Circumferential oil grooves should not form a continuous band, but should be staggered, thus preventing the formation of a ridge on the journal due to lapping action.

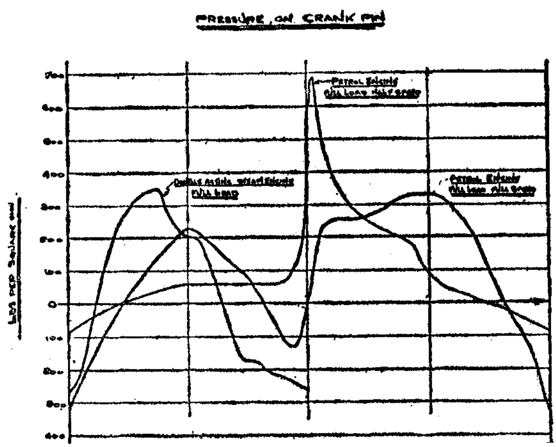


Fig. 9.

Figs. 6 to 8 give suitable designs of oil grooves for forced lubrication.

From various experiments it has been shown that bearings with forced lubrication will carry greater loads per square inch than others (this is largely due to the cooling effect of the copious supply and the certainty of the distribution), and it is interesting to consider the forces acting upon bearings in a motor car engine and those existing in a stationary engine of larger size. The load curves in Figs. 9 to 11 are fairly typical. The short period of reversal is noticeable in the petrol engine diagrams. The loads and rubbing speeds in neither case are very high, and are exceeded in many so-called slow speed engines, and considerably exceeded in steam turbines.

The extent to which forced lubrication is applied varies with different makers. Some apply it to main bearings only, and others

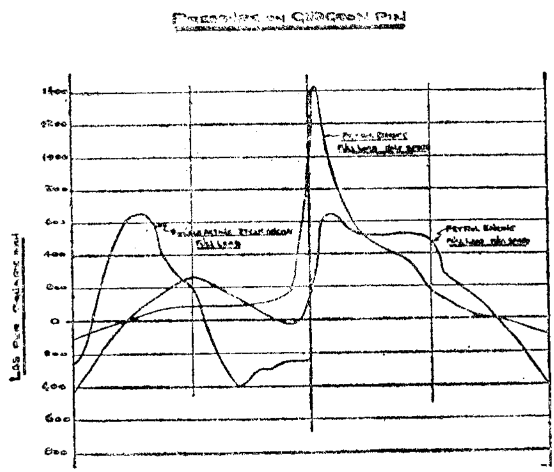


Fig. 10.

to main bearings and crank pins; others carry it to the gudgeon pin, and in some cases it is also carried outside to details of the transmission.

There seems little reason, if a pressure pump is included in the design, not to apply the pressure at any rate throughout the engine.

To consider the application to different parts of the engine in detail, the main bearings in a forced lubrication system are quite closed in, and provided a good oil filter exists can be kept very clean. In the splash or trough systems dirt may and often does collect in the open oil ways, with consequent trouble. The oiling of crankpins by forced lubrication is very certain and positive, whereas with splash lubrication great care is often required to ensure a correct supply. Further, in these and other bearings in which

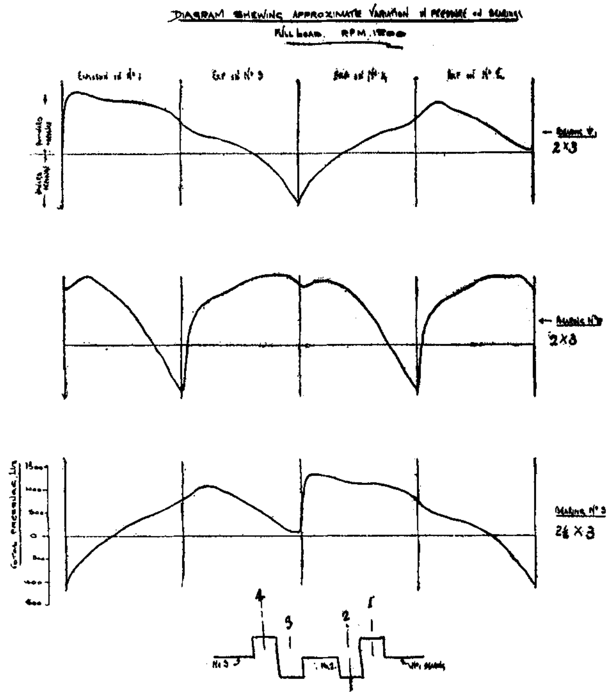


Fig. 11.

reversals take place, a thicker film is maintained by forced lubrication.

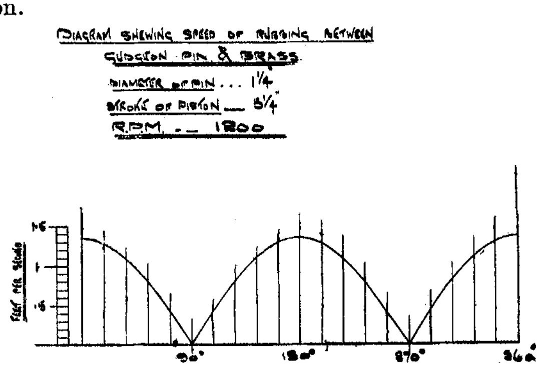


Fig. 12.

For oiling the gudgeon pins, which are heavily loaded and have comparatively small reciprocating motion (see Fig. 12), much is to be said in favour of forced feed, though the fact that the

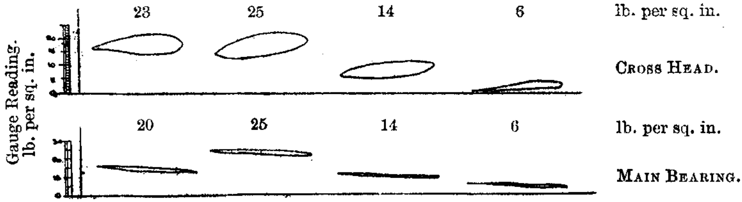
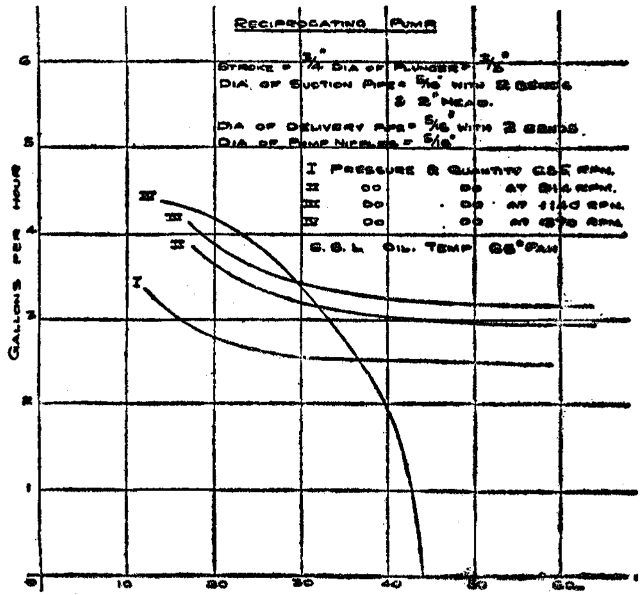


FIG. 13.

"change-over" occurs when the speed is high favours self-oiling. There is a tendency for oil to "cake" and often carbonize on the crank case side of the piston head, and the drip system usually



Pressure in lb. per sq. in.

FIG. 14.

arranged for with splash lubrication may be a source of trouble, carbonized oil and carbon dropping into the oil ways, and causing wear or clogging of the oil hole. Also, the oil is heated by the piston, and may be very hot and thin when it reaches its point of application.

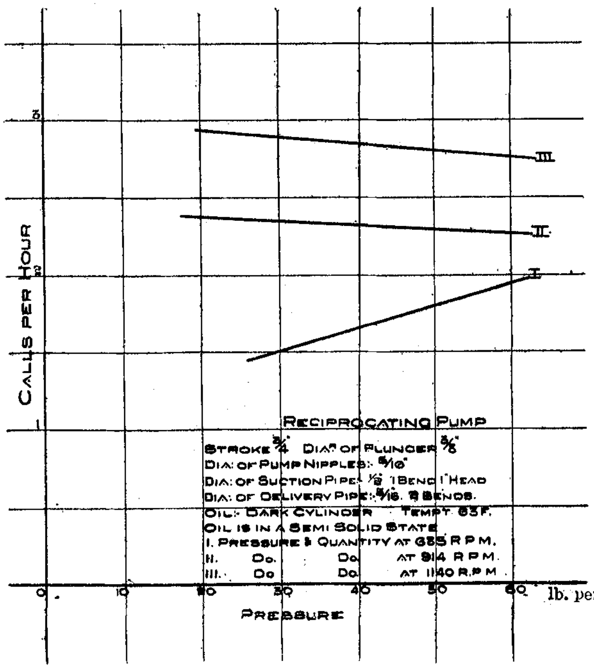

Calls per Hour
Reciprocating Pump
Stroke 3/4" Dia. of Plunger 3/8"
Dia. of Pump Nipples:- 5/16"
Dia. of Suction Pipe:- 1/2" 1 Bend 1" Head
Dia. of Delivery Pipe:- 5/16. 2 Bends.
Oil:- Dark Cylinder Tempt. 63 F.
Oil is in a Semi Solid State
I. Pressure & Quantity at 685 R P M.
II. Do. Do. at 914 R P M.
III. Do Do. at 1140 R.P M.
Pressure
lb. per sq. in.
I
II
III
0
10
20
30
40
50
60
1
2
3

Fig. 15.

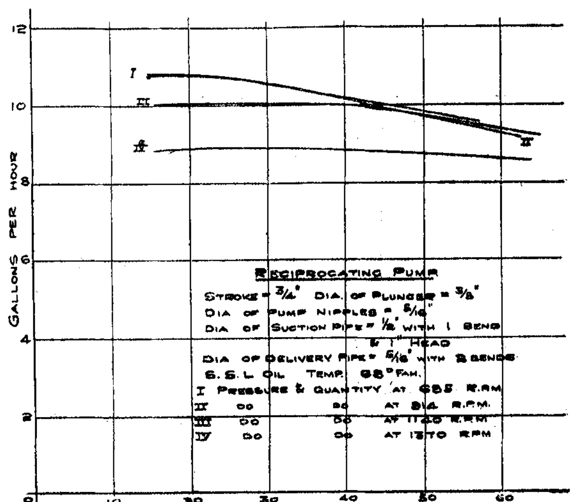
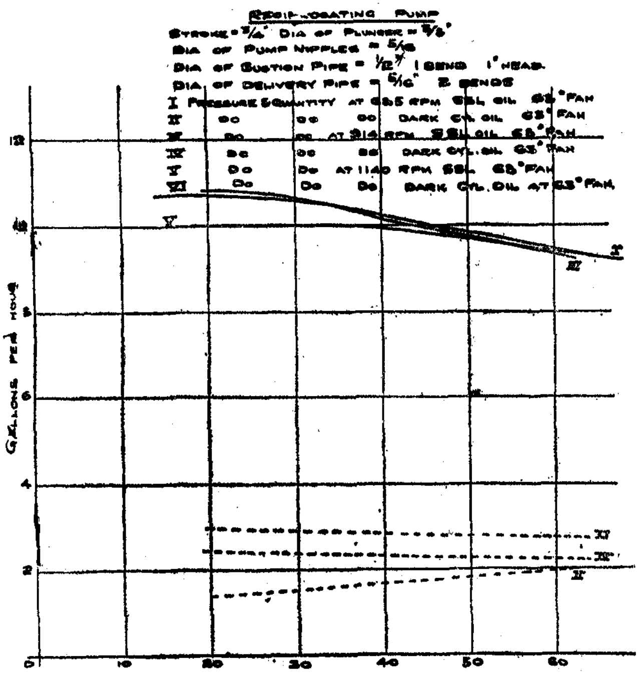

Gallons per Hour
Reciprocating Pump
Stroke = 3/4" Dia. of Plunger = 3/8"
Dia of Pump Nipples = 5/16"
Dia of Suction Pipe = 1/2" with 1 Bend & 1" Head
Dia of Delivery Pipe = 5/16" with 2 Bends
S. S. L Oil Temp. 68° Fah.
I Pressure & Quantity at 685 R.P.M
II Do Do at 914 R.P.M.
III Do Do at 1140 R.P.M
IV Do Do at 1370 R.P.M
Pressure in lb. per sq. in.
I
II
0
10
20
30
40
50
60
2
4
6
8
10
12

Fig. 16.

With forced lubrication supplied through a hollow rod or external pipe, a supply of clean pure oil is ensured. One of the great merits of forced lubrication is that it is easy to arrange baffles to prevent excessive oil being carried up the piston trunks, while with splash lubrication the difficulty of ensuring an adequate supply to the gudgeon pins often leads to excessive lubrication of the pistons, and subsequent trouble with cylinders, valves and ignition. A sheet of metal with slots for the rods placed just to clear the big ends has been found satisfactory in practice.



Pressure in lb. per sq. in.

FIG. 17.

The forced system requires means for preventing the efflux of oil from the main bearings. This is easily done by properly designed end plates, and baffles or thrower rings on the shaft.

Generally, in designing a forced lubrication system, the following points must be kept in mind:—

Have a pump of ample size, with good big suction and delivery pipes. There is a tendency to fit pumps and pipes which are too small to realise the benefits of forced lubrication. The effect of a choked suction is well shown in curve IV. (Fig. 14). From the curve of discharge given earlier it will be seen that the discharge from the clearance spaces is quite appreciable, and a pump or pipes which are too small may lead to the ends of the system being starved. A good rule is to make the discharge from the pump at full speed depend upon the total clearance. Thus, with a plunger pump, if the volume swept out per minute on the discharge stroke be V, and the sum of the peripheries of all bearings at each discharge point be P, then $V=8\times P$ is a good value.

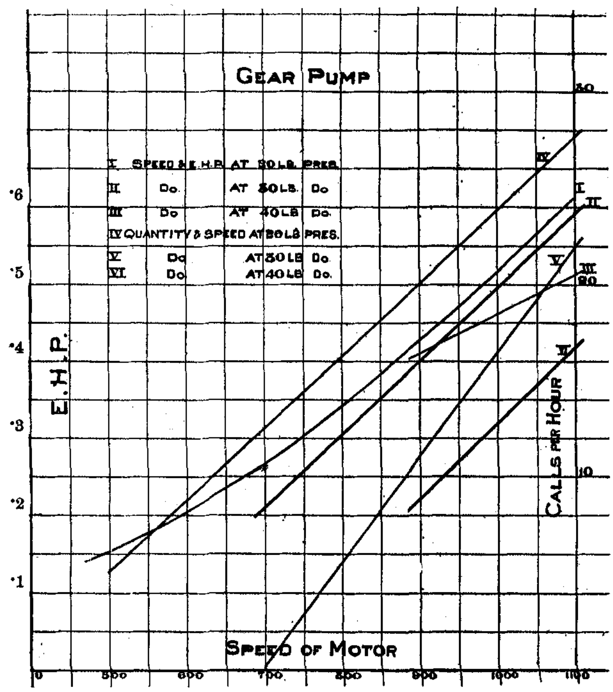


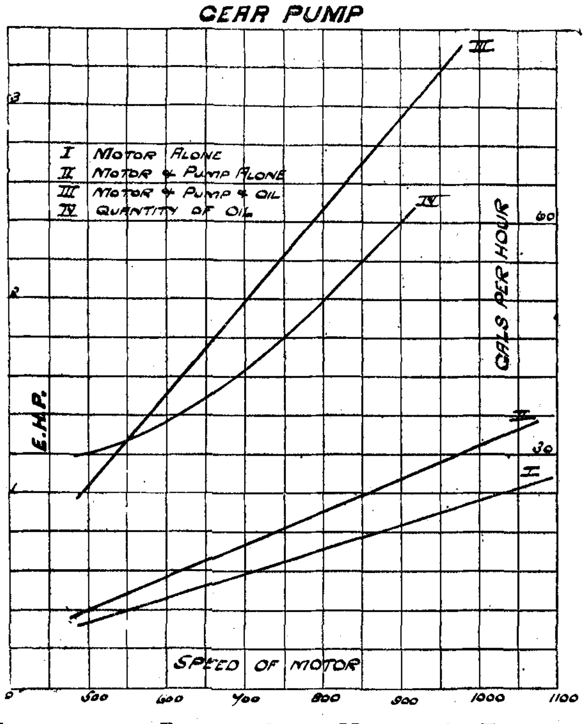
FIG. 18.

If B be the internal area of the pipe, for the main delivery pipe, take $B=P/480$; for the suction pipe, take $B=P/400$.

Another point which makes it necessary to have an ample pump is that, due to centrifugal and inertia effects on the oil in the moving parts, variations in pressure occur beyond those due to fluid friction and escape at clearances. To indicate the nature of such influences, a series of diagrams were taken with an ordinary indicator coupled direct to the main bearing, and through a flexible connection to the top end. The curves traced are given in Fig. 13.

Oil pumps of various types are used, the most common being :—

(1) Plunger pump.
(2) Gear pump.
(3) Vane pump.



Oil pressure zero. Pump speed 1·5 × Motor speed. Temp. 78° F.

FIG. 19.

The first is the most positive, and probably the most generally efficient. The second is good, and lends itself to simplicity of design. The third does not appear to be any good for high pressure. Centrifugal pumps are generally unsuitable, especially with thick oil. A series of trials run on different types of pumps are recorded in the curves of Figs. 14 to 19. It will be seen that the horse-power is quite small, so that the question of drive is an easy one, and there should be no difficulty in designing an absolutely reliable, well-protected, efficient drive for the pump. Some of the various drives used are mentioned in Table A.

The filter which must be fitted in the system should be efficient and accessible. It would be an advantage to place it in such a position that by lifting a cover it could be at once got at and periodically changed. The spare filter could then be cleaned ready for the next change over. The filter should have ample area, as the suction of the pump must be quite free. A good design of filter is shown in Fig. 20, but, of course, to suit special space requirements the design is subject to great modification. A point to be remembered which is often overlooked, is that perforated zinc and copper are not good neighbours in an oil well where water may be present. The oil pipes and oil ways should be ample in area, free from sharp bends and corners, and of adequate strength to

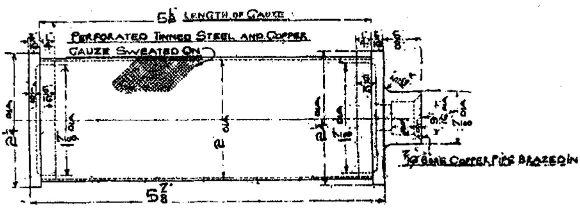


FIG. 20.

stand the highest pressures that may fall upon them. Steel pipes made to template are better than copper pipes, since the latter have been found to develop mysterious fractures. Where it is possible, hollow shafts and rods should be used to facilitate the distribution. All oil pipes should be carried in positions where they will not get in the way of overhauling, and where they will be protected from risk of damage. It is a good thing to bring the pipes up on the underside of the bearings in motor car engines, since they can be carried close to the cross-frames which usually support the bearings. In fact, the oil ways may actually be drilled in the crank base casting. The edges of oil holes in parts subject to stress should be carefully rounded, or they may become the origin of surface cracks leading to ultimate fracture.

In testing the pumps some trials (see Pump Trial Curves) were taken on heavy oils to show the effect of congelation in cold weather. Owing to the small quantity which can pass, a heavy pressure may come on the delivery pipe and gauge, if fitted. This is one reason for fitting strong pipes. A hand or automatic bye-pass may be fitted to relieve the pressure. A bye-pass valve alongside the gauge opening into the make-up oil tank was used with success in one instance. The best place for an automatic relief valve is on the delivery pipe close to the pump. Such valves are, however, a source of trouble, and should be avoided; as a rule, a gauge can be obtained to stand far greater pressure than will be put upon it in this way, and still be sensitive at the normal pressure.

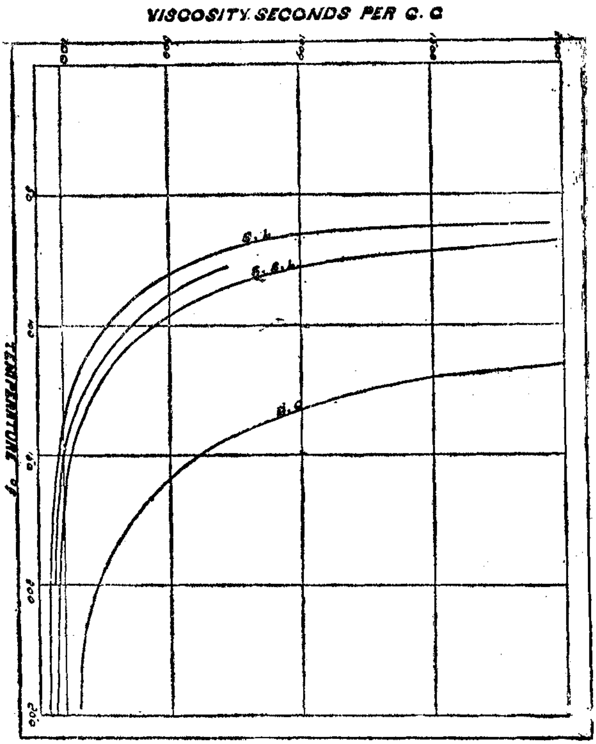


FIG. 21.

A very short time is required to warm up the oil, and the trouble from this source is probably less with forced lubrication than with any other system.

The chief wear in the bearings on a high-speed engine occurs at starting and stopping. A pump very quickly forces oil into a bearing, probably more quickly than the oil will get there in other systems. A non-return valve on the delivery pipe is possibly of value to ensure that the system shall remain full. The chief argument in favour of a pressure reservoir supplying oil is based on the need for oil at starting, and such a fitting for the pump to discharge into might be of value.

Any system in which a charge of oil is used for long periods may suffer from contamination of the oil. An experiment was made on a car fitted with forced lubrication. The sump was filled with a pure mineral oil called "SL," following the viscosity curve given in Fig. 21, and the car run for 3,000 miles in about four months on and off. The make-up was by a small scoop pump feeding the proper quantity into the crank case in the usual way. The oil after use was black in colour, and smelt of petrol. Tested for viscosity it gave the curve X (Fig. 21) on a Redwood viscometer. Its flash point, open test, was 214° F. against 435° F. On distillation at 300° F., $1\frac{1}{2}$ per cent. of petrol came over. After distillation the flash point was normal at 435° F. The oil was still dark in colour, but appeared to be a satisfactory lubricant. On burning the amount of ash was very little over normal.

On the subject of oil the information in Table A. is not very full. The general consensus of opinion is in favour of a high quality mineral oil of high flash point, well maintained viscosity, and low cold test. The viscosity curve of a satisfactory oil called "SSL" is given. (Fig. 21.) This oil has a flash point of 420° F., and is fluid at low temperatures. It is probably advisable to use a heavier oil in summer than in winter.

There are many satisfactory oils on the market, but also unfortunately many bad ones. Most makers will give sound advice as to the oil which suits their make of engine.

It would have been interesting to collect some figures of temperature normal to different engines after a long run, and it might be found that an oil cooler would be advantageous. This is fitted to steam-engines for hot climates, or long continuous running, with beneficial results, and is invariably a part of a steam turbine equipment.

The question of wear has not been considered, as the author has been unable, for obvious reasons, to collect reliable figures on this point. A car originally lubricated on the trough system has certainly run better, and shows less signs of wear, since forced lubrication was fitted; but one instance is not of much value. Many instances, however, of steam-engines with forced lubrication could be given in which the wear after long service has been infinitesimal.

An endeavour has been made to touch on the various points necessary to ensure good working of the system, and the conditions which have to be met. The author is of opinion that it is quite easy to meet all these conditions, and when they have been met it will be found that the forced lubrication system brings about a high mechanical efficiency, quiet running, and absence from wear.

TABLE A.

Name of Car	ADAMS.	ARIEL.	ARGYLL.	ALLDAYS.	AUSTIN.	ALBION.	B. S. A.
H.P.	16	31	15·8	10/12, 14/18, 20/25	15	16 and 24/30	15/20
Lubrication System	Forced feed, constant level troughs.	Forced.	Forced.	Forced.	Forced.	Albion patent.	Dip with constant level.
Type of Pump	Gear pump.	Gear.	Gear.	Rotary.	Vane.	Albion patent mechanical lubricator.	Gear.
Size of Pump	$1\frac{1}{16}$ in. by $\frac{7}{8}$ in. 16 pitch.	10 teeth, 8 pitch, $1\frac{1}{4}$ in. face.	—	—	—	Measured quantity forced to each bearing.	$\frac{1}{2}$ gallon p.m
Size of Suction and Delivery.	$\frac{3}{8}$ in. delivery. Other ducts drilled in crank case.	$\frac{9}{32}$	$\frac{5}{16}$ in. bore.	$\frac{3}{8}$ in.	$\frac{1}{4}$ in. bore.	—	$\frac{5}{16}$ in. bore.
Sight Feed or Pressure Gauge.	Positive forced tell-tale indicator on dash.	Gauge.	Gauge.	Gauge.	Gauge.	—	Sight feed.
Particulars of Filter	Gauze cylindrical.	Filtered thrice—(1) Pocket filter. (2) Barrel filter. (3) Surface filter.	Gauze filter above sump.	Gauze filter near bottom of crank case.	Inside reservoir.	Strainer at filling plug.	Gauze cup between pump and sump.
Relief Valve	Automatic.	Automatic.	Adjustable from dashboard.	Automatic.	Adjustable.	—	—
Working Pressure	2–3 lb. sq. in.	5 lb. sq. in.	10 lb. sq. in.	8 lbs. sq. in.	2–5 lbs. sq. in.	—	—
Speed of Engine	1,500	1,000	1,000	1,000	900	1,000	1,400
Speed of Pump	$\frac{1}{2}$ engine.	625 r.p.m.	500 r.p.m.	$\frac{1}{2}$ engine.	700 r.p.m.	6 strokes p.m.	$\frac{1}{2}$ engine.
Pump Drive	Spiral gears from cam shaft.	Spiral gears from cam shaft.	Cross shaft and worm gear from cam shaft.	Skew gears.	Spiral gear.	By ratchet from worm gear on engine.	By cam shaft.
Position of Pump	Bolted to side of sump.	Top of crank case.	Front left-hand corner of crank case.	Outside at bottom of crank case.	Central, outside reservoir.	In lubricator.	On cam shaft.
Oil Reservoir	Sump.	Sump.	Sump.	Sump.	Separate reservoir.	—	Sump.
Oil. Flash Point	Vacuum A.	Price's Motorine C.	320° F.–350° F.	Price's heavy engine.	213° F.	Vacuum A.	170° C.
Oil. Viscosity	—	—	$\frac{212° F.}{70° F.} = \frac{1}{25}$	—	367·6 at 15·5° C.	—	9·7 at 50° C. Water = 1.
Oil. Consumption	—	—	—	—	—	800 m.p.g. on 16. 600 m.p.g. on 24.	1,000 m.p.g.

Name of Car	CROSSLEY.	CHARRON.	CADILLAC.	DEASY.	DE DION.	DAIMLER.	F. I. A. T.
H.P.	15·6	15·6	30	14 and 18	9 to 35.	15, 22, 33, 38, and 57.	12/14
Lubrication System	Forced.	Gravity.	Splash.	Mechanical.	De Dion forced feed.	Semi-mechanical adjustable level troughs.	Forced.
Type of Pump	Gear.	Gear.	Double acting plunger.	Gear.	Gear.	Gear.	Centrifugal.
Size of Pump	Two $1\frac{1}{4}$ in. gears, 5/8 wide.	—	1 gall. p.m.	—	—	$1\frac{1}{2}$ gall. p.m.	—
Size of Suction and Delivery.	$\frac{5}{16}$ in.	$\frac{3}{8}$ in.	$\frac{1}{4}$ in.	—	—	3/8 delivery.	$\frac{1}{4}$ in.
Sight Feed or Pressure Gauge.	—	Sight feed.	Sight feed.	Sight feed 14. Pressure gauge 18.	Pressure Gauge.	—	Pressure Gauge.
Particulars of Filter	Gauze filter in sump.	In oil tank.	Oil does not pass twice through pump.	Gauze filter in sump.	Gauze covering sump.	Gauze in base chamber and drilled plate in pump.	Gauze filter in sump.
Relief Valve	Adjustable.	—	—	—	Automatic and adjustable.	—	Adjustable.
Working Pressure	10 lb. sq. in.	—	—	—	—	—	—
Speed of Engine r.p.m.	Up to 2,500	750–1,500	1,500	1,000	1,500	1,400	1,200
Speed of Pump	$\frac{1}{2}$ engine.	$\frac{1}{2}$ engine.	$\frac{1}{2}$ engine.	—	—	1,000	600
Pump Drive	Direct off cam shaft.	Direct off cam shaft.	Off magneto shaft.	Gear off cam shaft.	Parallel drive from cam shaft.	Spiral gears from eccentric shaft.	Keyed to cam shaft.
Position of Pump	Front corner, over $\frac{1}{2}$ time wheel.	End of cam shaft.	Near-side engine.	In sump.	Inside $\frac{1}{2}$ time gear cover.	In engine base chamber.	Rear end of cam shaft.
Oil Reservoir	Sump.	Oil tank.	Oil tank.	Sump.	Sump.	Sump.	Sump.
Oil. **Flash Point**	Price's H.	402° F.	Winter Vacuum A. Summer Vacuum B.	Vacuum A.	—	—	Vacuum B. Mobiloil.
Oil. **Viscosity**	Price's H.	980 at 60° F. 39 at 250° F.	Winter Vacuum A. Summer Vacuum B.	Vacuum A.	—	—	—
Oil. **Consumption**	500–800 m.p.g.	200–300 m.p.g.	750 m.p.g. A special trial gave 3,000 m.p.g.!	—	—	—	500 m.p.g.

Name of Car	LANCHESTER.	LANCIA.	MINERVA.	METALLURGIQUE.	MOTOBLOC.	N. E. C.	NAPIER.
H.P.	28	20 and 24	26	18	14/16	20, 30, 40.	45
Lubrication System	Forced.	Forced.	Combined splash and pressure.	Forced.	Gravity.	Forced for crank-shaft. Gravity for cylinders	Forced.
Type of Pump	Gear.	Gear.	Gear.	Gear.	(Alternative).	Plunger.	Gear.
Size of Pump	2–$\frac{3}{8}$ in. gear wheels, 2 in. long, 12 teeth in each.	—	$1\frac{1}{4}$ in. wheels.	—	$\frac{1}{10}$ litre.	—	80 gall. p.h. at 600 r.p.m.
Size of Suction and Delivery.	Suction, 4–$\frac{1}{4}$ in. holes. Delivery, 1–$\frac{5}{16}$ in., 2–$\frac{1}{8}$ in., and 1–$\frac{3}{16}$ in. bore.	Suction, $\frac{3}{8}$ in. Main delivry., $\frac{3}{8}$ in. M. B., $\frac{3}{16}$ in. Gudgeons, $\frac{1}{16}$ in.	$\frac{1}{4}$ in.	$\frac{3}{8}$ in.	1·5 and 2·0 cm.	—	—
Sight Feed or Pressure Gauge	Tell-tale.	Pressure gauge.	Pressure gauge.	Pressure gauge.	Sight feed.	Sight feed to cylinders,	Pressure gauge.
Particulars of Filter	Over bottom of pump.	In sump.	Grill full length of crank case.	Gauze filter in bottom of sump.	On dashboard.	Seven gauzes.	Gauze cylinder.
Relief Valve	—	Automatic.	—	Automatic and Adjustable	—	—	Automatic and Adjustable.
Working Pressure	40 lb. sq. in.	15/17 lb. sq. in.	—	2 lb. sq. in.	—	Slight.	10 lb. sq. in.
Speed of Engine r.m.p.	2,000	1,000/1,600	1,800	1,500	1,200/2,200	—	1,200
Speed of Pump	4,000	$\frac{1}{2}$ engine.	$\frac{1}{2}$ engine.	750	$\frac{1}{2}$ engine.	Engine speed.	600
Pump Drive	Gear wheels from crank shaft.	Off cam shaft.	Shaft connected to $\frac{1}{2}$ time shaft.	Skew gears from cam shaft.	Eccentric off cam shaft.	—	Skew gear off cam shaft.
Position of Pump	Bottom of crank chamber.	Rear end cam shaft.	Outside crank case lower half.	Back of engine bottom of sump.	Dashboard.	—	Back end alongside sump.
Oil Reservoir	Engine base.	Sump.	Sump.	Sump.	Tank.	Filter tank.	Sump.
Oil. Flash Point	—	520° F.	—	—	Vacuum A.	—	620° F.
Oil. Viscosity	—	75 secs. at 210° F.	—	—	Vacuum A.	—	—
Oil. Consumption	—	700 m.p.g.	250 m.p.g.	1,500 m.p.g.	150 to 200 m.p.g.	500 m.p.g.	—

Name of Car	PHŒNIX.	PANHARD.	RENAULT.	ROVER.	ROLLS-ROYCE.	SUNBEAM.	SINGER.
H.P.	9·9	18–30	14–20	15	40/50	12/16	16/20
Lubrication System	Splash and hand pressure pump.	Panhard patent.	Gravity.	Gravity.	Forced.	Forced.	Forced.
Type of Pump		Panhard patent.	Centrifugal.	Vane.	Gear.	Gear.	Gear.
Size of Pump		Very small.	—	1 quart p.m.	—	—	—
Size of Suction and Delivery.		$\frac{3}{8}$ in.	$\frac{1}{2}$ in.	$\frac{3}{16}$ in.	$\frac{1}{2}$ in.	$\frac{1}{4}$ in. outside.	$\frac{3}{8}$
Sight Feed or Pressure Gauge.		—	Sight feed.	Sight feed.	Pressure gauge.	Pressure gauge.	Pressure gauge.
Particulars of Filter		Gauze filter on return vent to oil tank.	In centre of radiator.	30 mesh gauze n sump.	Lowest point of sump.	Gauze in centre of sump.	Bucket shape in top of oil tank, easily removable.
Relief Valve		Automatic.	—	—	Automatic, can be set to required pressure.	Automatic.	—
Working Pressure		—	—	—	15 lb. sq. in.	—	5 lb. sq. in.
Speed of Engine r.p.m.		900	1,100	1,000	1,000	1,000	1,360
Speed of Pump		450	$\frac{1}{2}$ engine.	500	500	500	680
Pump Drive		Off end of exhaust cam shaft.	Off cam shaft.	Bevel gear.	Skew gears.	Direct from cam shaft.	Off exhaust cam shaft through skew gears.
Position of Pump		Between fly-wheel and crank case.	In crank chamber.	Below front end crank chamber.	Lower portion crank chamber.	End of cam shaft.	Back end bottom of base chamber.
Oil Reservoir		Oil tank.	Sump.	Sump.	Sump.	Sump.	Oil tank.
Oil. **Flash Point**	420° F.	—	—	—	500° F.	—	Vacuum A.
Oil. **Viscosity**	55 secs. at 180° F.	—	—	—	1,130 secs. at 120° F.	—	Vacuum A.
Oil. **Consumption**	1,000 m.p.g.	—	500 m.p.g.	600 m.p.g.	1,000 m.p.g.	—	700/800 m.p.g.

Name of Car	SIZAIRE.	SHEFFIELD-SIMPLEX.	TALBOT.	VAUXHALL.	VALVELESS.	WHITE.	WOLSELEY.
H.P.	12	45	12, 15, 20, 25, and 35.	20	22	20	Various.
Lubrication System	Drip.	Semi-mechanical.	Mechanical.	Forced.	Forced.	Forced.	Forced.
Type of Pump	—	Gear.	Plunger.	Plunger.	Vertical plunger and distributor diaphragm.	Plunger.	Gear.
Size of Pump	—	6 gall.p.min.	15 m/m dia.	$\frac{3}{4}$ in. diameter. $\frac{1}{2}$ in. stroke.	$\frac{5}{16}$ in. bore. $\frac{3}{16}$ in. stroke.	$\frac{1}{4}$ in. bore. $\frac{5}{8}$ in. stroke.	—
Size of Suction and Delivery.	—	$\frac{1}{4}$ in	10 m/m.	$\frac{1}{4}$ in. bore.	$\frac{1}{4}$ in.	$\frac{3}{16}$ in.	$\frac{3}{8}$ in.
Sight Feed or Pressure Gauge.	Sight feed on dash.	Pressure gauge.	Both.	Pressure gauge.	Pressure gauge.	Both.	Pressure gauge.
Particulars of Filter	—	Gauze in mouth of sump.	Detachable tubular gauze.	$3\frac{1}{2}$ in. dia. gauge surrounding suction valve.	Gauze in tank.	Wire gauze in top of sump.	Copper gauze in sump.
Relief Valve	—	—	—	Automatic and Adjustable.	None.	—	None.
Working Pressure	—	4 lb. sq. in.	—	10/15 lb. sq. in.	—	—	5–20 lb. sq. in., depending upon speed.
Speed of Engine r.p.m.	1,000	1,000	1,100	1,000	1,000	—	—
Speed of Pump	—	500	550 strokes.	500 strokes.	20 strokes.	—	Cam shaft speed.
Pump Drive	—	By spiral gears from cam shaft.	From cam shaft.	Eccentric on cam shaft.	By worm gear.	Off half-time shaft.	Spur gear off cam shaft.
Position of Pump	—	Low down on crank chamber.	In sump.	Rear end of cam shaft.	Front of engine.	In oil sump.	Side of engine.
Oil Reservoir	—	Sump.	Sump.	Sump.	Oil tank.	Sump.	Sump.
Oil. **Flash Point**	Price's heavy or Vacuum C.	250° F.	—	Price's Motorine "C."	470° F.	—	420° F.
Oil. **Viscosity**	—	—	—	Price's Motorine "C."	50 secs. at 210° F. (Sayboult M/C).	—	175 at 140° F.
Oil. **Consumption**	500 m.p.g.	200 m.p.g.	—	1,150 m.p. gall.	1,130 m.p.g.	—	600 to 1,400 m.p.g.

B B 2

THE DISCUSSION.

Colonel R. E. B. CROMPTON: Mr. Morcom's paper is the outcome of long experience in the lubrication of high speed engines. His firm was practically the first to introduce forced lubrication as an improvement on the crank-chamber splash system which it has largely superseded. The matter was very thoroughly thrashed out during the discussion which took place in this room thirteen years ago on a paper on the same subject then presented by Mr. A. Morcom. No doubt many of the improvements since introduced were suggested by that discussion. Most engineers are agreed that the staggered circumferential groove described in the paper is the best method of getting the oil to the position of highest pressure in a bearing.

I should now like to ask Mr. Morcom if he has been troubled with the oxidisation of steel surfaces in bearings by either acid in the oil or small quantities of water mechanically mixed with the oil? Also if he has taken any special precautions against water getting into the oil? I have already referred* to the very serious oxidising effect which shows itself on ball or roller bearings, in fact on any bright steel work exposed to the action of lubricants containing small quantities of water. It is not mechanical but oxidising action which has been the real cause of wear of such bearings, due to the removal of the oxidised surface by the mechanical action, and the re-oxidisation when the engine is standing. An effective remedy for this has been found in the use of special oils, chosen because they seem to stick closer to the polished steel surfaces and so prevent the oxidising matter obtaining access to them. Hitherto animal oils have been suggested for this purpose, but what we now want to find is a mineral oil sufficiently viscous and yet able to adhere to the metallic surfaces and so protect them from this oxidisation.

Again, we are looking for a mineral oil which combines with effective lubricating quality that degree of viscosity which will allow it to follow up to the suction side of the oil pump when everything is cool and when the engine is first started up. This difficulty appears to be a very real one with forced lubrication.

* See p. 231, *ante*.

The more viscous oils follow the pump plunger very badly during the first few strokes, so that just at the time at which we want the oil to be forced under pressure into the bearings, the pumps are frequently found not to give their full supply. I should like to ask Mr. Morcom how he proposes to cool his oil? Willans, in his early engines, used to cool the splash lubrication in his crank-chamber by passing the cold feed water through on its way to the feed heaters. Of course, this method is not available for the class of lubrication we are now considering.

I am in agreement with Mr. Morcom as to the necessity for making the whole pipe system of steel. Copper pipes are peculiarly liable to give way under vibration such as generally occurs in motor cars and in most high speed engines. It is particularly annoying that a set of bearings should be ruined by such a breakage which is not likely to be observed before serious damage is done.

Mr. G. H. Baillie: Mr. Morcom has very strongly advocated the forced system of lubrication as being the right one. No doubt it is an extremely good system, but I question whether it is any better than a splash system for engines doing the irregular work which motor car engines have to do. Certainly there are a great many engines running on splash systems, and I do not think it has ever been shown, or seldom suggested, that their bearings wear any more rapidly than those of engines with forced lubrication. My own opinion is that it is not of great importance whether splash or forced lubrication is adopted, but the important thing is to have the system, whichever it is, thoroughly well carried out, and it is in comparatively few cars that this is the case. Really the chief difficulty in a forced lubrication system is to get an effective pressure actually at the bearings, and this is due to the variations of temperature of the oil. Most cars I find show no pressure at all when their engines are really hot. Those fitted with an oil pressure gauge show a high pressure at starting which gradually diminishes, and when the oil pressure drops to practically nothing, I think the forced lubrication is extremely bad, because the oil ways through the crankshaft are small and tortuous and very liable indeed to get blocked up, much more so than with any splash system. In order always to get an effective pressure at the bearings, the only way would be to fit a relief valve quite close to the bearings, the result of which would be to subject the oil pump to undue pressure when the oil is cold. I see Mr. Morcom is strongly against the relief valve. Probably there is no need for it in engines running at constant speed, but in motor car engines where the pump has to deliver oil at any speed between 300 and 1,500 r.p.m., I consider it essential. I know that in one engine which had no relief valve the apron was always full of oil which squirted out of the glands on racing the engine. It was fortunate that it did squirt out or something would have burst.

Another difficulty is the wear of the bearings. With slack main bearings the oil pressure is much less on the big ends than on the main bearings. In order to get sufficient oil under all conditions at the big ends—for instance, under the worst condition when the main bearing is slack and the oil is cold—there must be too much oil at the big end when conditions are good, and then there is the difficulty that too much oil is thrown off into the cylinder. Baffle plates do a good deal, but not enough, because there has to be a slot, and the throw off is generally more or less in the direction of the slot. These difficulties do not exist to the same extent in stationary engines running at constant speed for many hours, and it does not therefore follow that the best system for them is also the best for motor car engines.

Another point I would like to mention is that Mr. Morcom's suggestion of forming oil-ways in the walls of the crank chamber has often been tried, but it is found that aluminium crank chambers are too porous to hold the oil.

Mr. A. E. TUCKER: I have had some experience in the mechanical testing of oils for purposes of engineering work, and I am alive to the extraordinary difficulty there is in getting reliable results. On page 352 Mr. Morcom states that "the one outstanding conclusion is that the modern theory of lubrication is the theory of the oil film. The old idea of a co-efficient of friction gives place to the more suggestive theory that the resistance to motion is due to the shearing of a film of oil which more or less completely prevents a metallic contact and abrasion." It seems to me that Mr. Morcom has introduced a dangerous word here—shearing a film of oil; by shearing, I take it, you imply a division into two parts, but surely in the action of a film of oil between two rubbing surfaces you get no such shearing, rather you get a streamline action which distributes itself over the whole mass of the intervening oil, and therefore it seems to me that the idea of a co-efficient of friction of the oil used still stands; we have to deal with a flow of liquids, and to recognise that this flow is a function of internal friction in the same way that the flow of solids is determined by their internal friction.

Mr. Morcom has given some curves which I am quite unable to follow. He says that they have been obtained by determining the pressures, as I understand it, at various points of the peripheries. It seems to me that in dealing with the practical question of motor cars and high class engineering components generally, when there are two working surfaces so concentric as are the journals and shafts in motor car work, the determination of the variation of pressure at different points of the journal concerned is practically impossible, and I shall be extremely obliged to Mr. Morcom if he will explain and amplify what he has stated, and give us some idea as to how these curves have been obtained. With respect to the point he raised—and it is a point of considerable practical importance, especially, I take it, in his own engines where some minor parts have been made of zinc and other parts of copper, and both have been at work in the same chamber—I think the deterioration and destruction of the zinc might have been anticipated. Colonel Crompton alluded to the formation of oily acids in the crank chambers, etc., but I may point out that with water alone you would get an electrolytic action with copper and zinc—a couple being formed, and that where you have water and oil churning up in the crank case and still more so in the Morcom engine, where the heat is considerable hydrolysis or a decomposition of the oil occurs with resulting acids, together with a curious saponifying action due to the incessant churning up of the water and the oil with some amount of metallic bases. Under such conditions you have here a disproof of the common saying that water and oil will not mix. They mix most perfectly. The mechanical action may be accompanied by a chemical action upon the oils used, and consequently upon the metals of the gear, so that deterioration owing to the production of such acid may result with serious effects. The amount of metallic oxide required to saponify some animal and vegetable oils is very small. I have seen instances where the originally thin oil had been thickened seriously in the crank case of an engine on account of this saponifying action; this thickening would, of course, greatly reduce the efficiency of the oil. The remedy is obviously to keep all surfaces as free from dirt, etc., as possible, and to use only oils of approved merit.

Mr. C. R. Garrard: I will refer first of all to Mr. Baillie's remarks as to the comparative wear on the bearings when splash or forced lubrication is used. This opens up and includes the question of materials as well as lubrication, but throughout the paper this evening there is scarcely any reference to the question of the materials used in the bearings. It is well known to most of us who are practical men, that a very little change in the composition of the material produces very remarkable results in the wear or otherwise of a bearing. I had under notice some phosphor bronze bearings carrying shafts of steel with a small percentage of vanadium, and the results were disastrous in a short time. The same vanadium steel was then tried with a white metal lining to the bearings, the bearings are running now, and they have never been taken down since 1904. Again, the same phosphor bronze with other crankshafts gave satisfactory results, though not quite so good as the white metal. If the film of oil theory holds good, the nature of the materials used in the construction of the bearings should not affect the result.

Another phenomenon we have remarked is that one of the two materials should be porous. If we case-harden a shaft, it becomes very dense indeed, but we should never put such a shaft in a solid ring where it would be subjected to an abnormal pressure with the ring strong enough to resist it. It would be all right if the pressure could not exceed a pre-determined amount. In that case such a bearing is admissible.

Mr. Morcom touched upon the introduction of a cross hole drilled in the shaft. We have been very reluctant indeed to adopt that practice when we are tied down by the size of the shaft, on account of the danger in drilling and the risk of breakage. Cycle engineers in the early days were in the same fix. They were afraid to drill an air hole in the tube because the tube broke through the air hole.

A very simple kind of relief valve or an equivalent to it can be arranged for by having a very small hole in the plunger pump, so that if the pressure becomes abnormal, the oil can surge back again until it tends to go the right way, and the normal conditions are restored.

The high speed of the engine and even of the cam shaft is somewhat against the plunger pump, and that is the reason why most of us have adopted the gear pump for the oil. While on the subject of gear pumps, I would like to ask if Mr. Morcom can explain the object of the little groove round the teeth of the Albany pump. From practical observation and experience I find that it is a very good arrangement indeed. With the little groove the pump can suck from a greater depth than it can without it even when quite empty. When charged with oil it seems to form a lock for the liquid, and to answer very well indeed.

The cooling of the oil is very generally practicable for motor boats where the engine runs under a heavier load than on a car. A car gets relief for short periods of time, but a boat runs for hours without relief, and it is very usual to cool the oil for marine work. The question of the use of castor oil has not been touched on. It has often been used with very good results at high temperatures; for very high speed bearings which would not run cool paraffin has been mixed with the oil, and the result has been satisfactory.

There is another thing I would like to draw attention to. It is a very dangerous practice, although it cannot always be avoided, to have several oil leads in parallel from one main lead. At some time or other greater resistance may result in one lead than in the others, and the oil will go the easiest way, leaving one bearing perhaps unlubricated. I recommend a tell-tale in one pipe, and no division should be made in the one pipe from its tell-tale so as to make sure the oil is going the right way. Turning to the splash system, something akin to a fair amount of pressure has been obtained with this form. In trough lubrication I have used a connecting rod designed with the cap at the bottom end milled out to such a shape that it formed a cup to scoop the oil up with. At 1,800 revolutions per minute this attained a considerable lineal speed, and the oil by its inertia knocked the air out of the way, owing to the form given to the scoop.

It would be well to direct attention to the lubrication of engines designed for aviation. In these, of course, the designers attempt to make the crank shafts as light as possible, and there have been fractures through the lubricating holes in such cases.

Mr. L. A. Legros: Perhaps it is possible to throw more light on the trouble occurring with nickel steel shafts to which Mr. Garrard has just alluded. In using crankshafts of this material I have only found one particular kind of phosphor bronze suitable for bearings; this is cast under special conditions for ensuring cooling during casting, with the result that the density is increased some five or six per cent. This bronze (known as Eatonia) does not adhere to, and act as a lap on, the journals. The same trouble did not occur with vanadium steel, which ran satisfactorily in bearings of ordinary phosphor bronze.

Mr. Morcom has drawn particular attention to the necessity for putting a large radius at the end of the lubrication cross holes in crankshafts; the risk of fractures originating in drilled holes in shafts has been recognized for some time past, particularly on railways and in electric supply stations. It is not necessary for a hole to be drilled deeply into a shaft for this risk to be incurred; merely pointing a drill in will do it. An

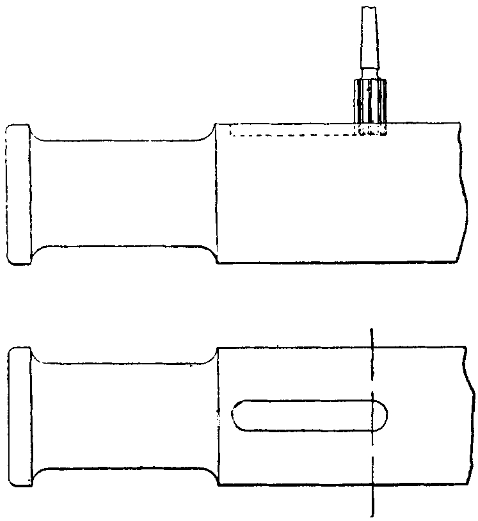


FIG. 22.

axle of a suburban train running out of London broke, and was examined by me immediately after the fracture; it had broken through a depression formed by a drill-end for receiving the end of a setscrew which kept the pulley for driving the electric light dynamo in place. The same method for retaining chain wheels in place has produced a similar result in tramcar axles, and was obviated by clamping the chain wheels on the axles. On the electric railways and tramways of London similar trouble has been experienced. If the keyseats for the wheels are cut with an end-mill, as shown in Fig. 22, fracture would in many cases take place through the diameter of the circular end of the keyseat. The risk can be avoided by milling the keyseat with an ordinary mill as shown in Fig. 23.

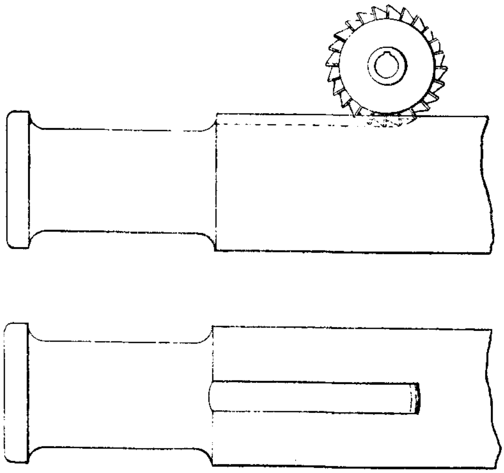


FIG. 23.

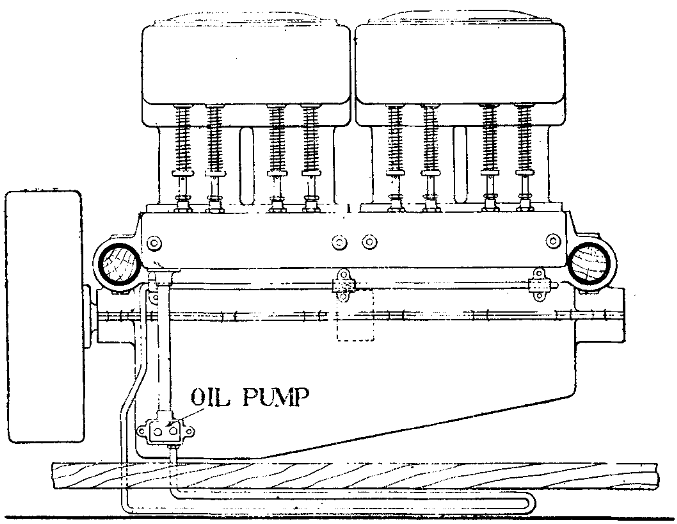
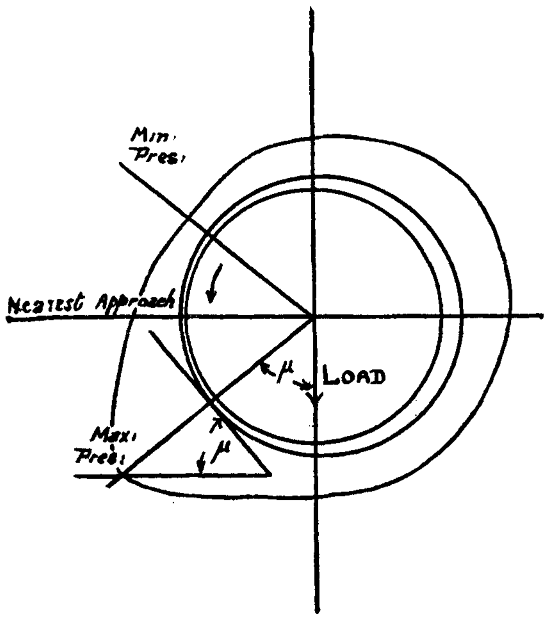


FIG. 24.

A good strainer for filtering the oil before entering the oil pump in forced lubrication systems, is a most essential detail; without an efficient strainer stoppage of an oil passage may occur by accumulation of small pieces of dirt. It is also necessary that the strainer should have a very large area so that it may not become blocked and thus cause failure of the circulation. The sump suggested by Mr. Morcom is an excellent idea.

Cooling the oil has been effected on motor boats by passing the oil from the pump through a short length of pipe, alongside



Curve representing oil film pressur in plain journal bearing

FIG. 25.

the keel, outside the shell of the boat. Such an arrangement is shown in Fig. 24, and has proved to be quite efficient in practice.

Mr. L. H. HOUNSFIELD: From Beauchamp Tower's experiments I had been led to believe that the point of maximum pressure was at the angle of friction with the line of load (μ in the sketch, Fig. 25), and I had imagined the journal climbing up the side of the bush until it reached this critical angle, which I also thought was coincident with the point of nearest approach.

From Mr. Morcom's paper I gather that these ideas are quite wrong, and that the line of maximum pressure may be situated anywhere between the line of load and that of nearest approach at right angles to the load.

I should like to know whether Beauchamp Tower's experiments were inaccurate, or whether I have misinterpreted them.

Referring to the sketch, the vertical components of the pressures acting on the top of the shaft must equal those underneath, minus the pressure due to the load; now, as this last is very great in petrol engines, the pressure of the oil film the whole way round the journal is very high, if the sketch is in anything like correct proportion. How, then, can oil, forced at only 40 lb. per sq. in. pressure, find its way into the bearing, which of course it does?

In the list of so-called forced lubrication systems at the end of the paper I notice 16 out of the 34 are specified as "forced." May I ask whether Mr. Morcom considers they are forced systems in the sense that he uses the word himself?

Mr. MORCOM: I have the maker's word for it.

Mr. HOUNSFIELD: Some of them I know are not forced, because the oil is forced into a trough and thence the oil reaches the bearings merely by splash. In regard to Fig. 13 it is stated that the ordinary indicator was used for obtaining these curves. May I ask Mr. Morcom if he means the steam engine indicator?

Mr. MORCOM: Yes.

Mr. HOUNSFIELD: I should have thought the ordinary steam engine indicator had too large a capacity to give a representative diagram. I was very much interested in Mr. Morcom's statements regarding rounding off the edges of the oil holes in shafts. I wish some further information might be given upon this point.

Mr. MORCOM, in his reply upon the discussion, said: Colonel Crompton asked how to prevent the forced system of lubrication from introducing oxidising matter. This trouble, if it existed, would be equally present in any system in which oil is used over and over again. Colonel Crompton, and also Mr. Tucker, pointed out that where you have water present, as in a steam engine, oil, in the presence of water and air, undergoes in the churning up process some form of "hydrolysis" which produces saponifiable acids, and it very much puzzles chemists how to explain it. A paper was recently read on this subject of the saponification of oils in closed crank cases, and several chemists who spoke upon the subject denied that it was possible, and no one explained exactly what the action was that took place. In one instance at a central station where they presumed this was causing trouble, they went to the expense of putting a considerable amount of alkali into the crank case. The result was an even worse evil, as a very bad emulsion was formed, like brown frog-spawn in appearance, filling the crank case and overflowing through the doors when they were opened for examination. On a petrol engine, analysis showed very little water and therefore no emulsion, and though we do get CO_2 present in the gases of combustion, it is unlikely to do much harm as there is so little water. The tests I have made have shown the oil in petrol engines to be neutral even after long use. The action in the case of ball bearings would probably be worse owing to the intimate contact of the bearing surfaces.

Colonel Crompton asked how to introduce an oil cooler. On motor boats, as Mr. Legros says, the problem is easy, but on a motor car I should not like to prescribe the best method. I believe that in one of the old type Wolseley cars a cooler was put in the middle of the radiator. In order to cool the oil it is necessary to have an arrangement to divide it up very finely. Several forms of coolers have been tried; the best results were obtained with an apparatus called a concentric condenser, which consists of a large number of corrugated cylinders fitted one inside the other, jointed alternately for water and oil. Using that apparatus, which occupied little room, you could get a big cooling capacity. Probably a cooler built on similar lines to the usual radiator would be efficacious. Air cooling might be tried in the sump by making the oil reservoir like a radiator.

Mr. Baillie questioned whether forced lubrication was better than splash. Of course, as I said, great success has been obtained with splash systems, but I think it difficult to argue that the success is good enough. One of the strongest arguments in favour of forced lubrication is that high speed steam engines with splash lubrication would not run double-acting, and elaborations had to be made in order to prevent reversal of thrust, but as soon as forced lubrication was introduced it became possible to run a double-acting engine quietly, which without pressure fed lubrication ran noisily.

In a petrol engine the reversals are not so extreme as with steam. If anyone with a sufficiently sensitive ear would listen to the running of two engines, one with pressure lubrication and one without, or to the same engine run first with pressure and then with the pressure dropped right down to a small amount, I believe he would be able to detect, especially when running slowly, that it was quieter with pressure than without, and, further, I believe there would be found to be slightly less wear in the long run in the case of the engine run with the higher pressure than with the lower.

Another thing which points to the fact that forced lubrication is better than splash, is that at certain central stations, the engineers in charge of engines not fitted with forced lubrication want to convert the existing systems into forced lubrication.

Mr. Baillie has quoted several instances of trouble with forced lubrication. All I can say is that the systems of forced lubrication he quoted must have been extremely badly designed, and in no way prove that a really well-designed forced system is not better than splash. In favour of a relief valve he instanced a little pump squirting oil all over the place, so constituting a form of relief valve, but a leaky pump ought not to be put outside.

I do not think that a great deal more oil should be thrown up on to the pistons with forced lubrication than with a splash system.

If you have examined the running of an enclosed engine through an inspection door, you will have seen that a good deal of oil flies about whichever system may be in use. In any case it is by no means an insuperable problem to stop an excess of oil getting to the cylinders: in fact on some forced systems it has even been found necessary to provide special holes to lubricate the pistons.

Mr. Tucker objected to my phrase "shearing of an oil film." It is in common use when discussing viscous fluids. It is quite reasonable to regard the shearing of metals as a problem of flow. He asked me what the meaning of the film pressure was, and I cannot do better than refer him to Professor Nicholson's paper, where a good statement is given on the subject, and also to the papers by Osborne Reynolds, Sommerfeldt and Michell.

To take up a further criticism, of course it was obvious, as soon as one found it out, that the zinc plates and copper would give trouble. Such cases generally arise from want of thought, perhaps on the part of a draughtsman. He sees that a plate is necessary, and he finds a perforated zinc plate on the stores book, so he puts it under the copper gauze, and it is not until it is worn out that it is recognised as having been the cause of trouble. But these little matters are all the better for being pointed out.

Mr. Garrard says I do not refer to the materials of which the journals and bearing surfaces are constructed. Of course, that would be entering on a very large subject. Thurston, in his book on the subject, devotes a great number of pages to the consideration of this point. In the papers by Lasche some very interesting figures are given on different materials, with photographs of bearings which have seized badly when overloaded. According to theory there should be no metallic contact at all, and therefore no need to worry about what the surfaces are made of, but in practice a certain amount of abrasion is bound to take place. There will be a certain amount of grit in the lubricant which will find its way into the bearings, causing wear, and hence blackening of the oil. This blackening means that there are minute particles of metallic substances detached from the bearings, and those again cause wear. If there is perfect lubrication and pure oil, it is not important to consider the metallic substances of which the bearings are composed, but the causes that make it important to choose the right surfaces to rub together are imperfect lubrication at starting, and the presence of impurities. These make it necessary to be very particular in choosing how best to lubricate, and of what material to make the bearing journals.

To Mr. Hounsfield, I would recommend a careful perusal of the papers by Sommerfeldt, Nicholson and Michell, especially as he has shown his point of nearest approach on the wrong side. I think he has only read a part of Tower's report, as this very point was one of his most interesting conclusions.

The President: At another time some observations which I was prepared to make may be worth presenting to you. They deal with experiments which I have conducted on the pressure to which bearing surfaces are subjected.

I should like now to ask Mr. Veitch Wilson, who is present, and who is one of the greatest authorities upon lubrication, if he will contribute some remarks to our printed proceedings, upon some of the chemical points which have been raised to-night in connection with lubrication.

It only remains to ask you to pass a hearty vote of thanks to Mr. Morcom. I feel that we have listened to a man who has worked at the subject for a great number of years, and has freely given us the results of his investigation in a most attractive form.

COMMUNICATED.

Mr. F. L. Martineau wrote: Mr. R. K. Morcom, in his paper on "Forced Lubrication," puts down certain positive conclusions which may be taken as established. These conclusions taken together and interpreted by the light of the rest of the paper would seem to indicate that contrary to the practice of the author and his recommendations, the lubricant should be forced into a bearing, not at the point of the least pressure in the oil film, but at the point of nearest approach, and that the pressure under which the oil should be supplied should be greater than the maximum pressure obtained in the oil film.

Under these circumstances it would seem that the trials made with pressures up to 30 or 40 lbs. per square inch are entirely inadequate. By utilising higher pressures, as suggested, the varying thickness of the oil film is, in a large measure, corrected, and by causing the shaft to assume the concentric position the least resistance would be obtained.

The "Ilgner" system of electric mine-winding machinery has shown that this is so, for the fly-wheel used in many instances in this system is of such weight that it is impossible to start it rotating, or to maintain it in rotation, even with a 500 h.p. motor, unless the spindle is first lifted in the bearing by the lubricant being pumped into it from underneath, so that the resistance is reduced to a very minute amount. Schlick, in his gyroscope, utilises practically the same conditions.

This increase of pressure, according to the author's testimony, would tend to decrease noise, and also by increasing the minimum thickness of oil film would prevent any abrasive action taking place in a bearing, and as a consequence, there would be no metal worn away to act as a cutting material on the shaft or bearing.

In talking about Fig. 14, on page 364, the author remarked that after a certain speed the pump sucked in air and not oil. If the suction pipe was below the oil surface, this would seem to me impossible. What would really be taking place would be cavitation.

It would be interesting if the author would give the curves relating to the velocity of oil in both suction and delivery pipes of the oil pump, which would settle the minimum dimensions in a very much better manner than the method he has suggested.

Mr. R. W. A. Brewer wrote: Although it may be taken as an axiom that, "the success of a flying machine depends upon the efficiency of the engine lubrication," and as the most important aspect of the lubrication problem at the present time, when light engines are considered, is perhaps, that of the aeroplane engine, I regret to find that no mention is made of this branch of the subject in Mr. Morcom's paper. For some time past I have had considerable experience with the running of air cooled aero engines, and the Gnome engine used by Mr. Grahame White on his flight to Lichfield was entirely under my supervision.

The Gnome is an example of an engine of which the one weak point is the lubrication system, and whereas in some other engines, doubtful design has been rendered commercially possible by the perfection of the lubrication system, the success of the Gnome is due to design, as there is no lubrication system, and the waste of oil is enormous.

Air cooled engines, in fact, generally appear to lack the refinements of a good lubrication system. Take the Anzani for instance—the only oil feed is one to the crank chamber, and owing to the drilled cylinder walls, the supply of lubricant must be excessive since the bulk of the oil is blown out of the exhaust holes. The two engines mentioned are lubricated by splash in the most elementary form, but, of course, the two designs are not to be compared—the only wonder is that the Anzani engine with plain bearings throughout runs at all.

When we find two engines which will work in spite of such faults, it seems somewhat strange that some form of positive feed lubrication is not adopted.

Of aero engines perhaps the E. N. V. has received the most careful study in connection with its lubrication, but the refinement of lubricating the cylinder wall through the gudgeon pin is perhaps carrying matters to too fine a point.

Cylinder wall lubrication of aero engines, however, is of the greatest importance, particularly with air cooled engines, and the power developed is enormously influenced—I speak from actual measurement—by the efficient lubrication of these parts.

Now, as to the oil; the general practice which I follow myself is to use castor oil, but whereas it is usual to mix 25 per cent. of petrol with the oil, I use the pure medicinal castor oil alone.

As to the quantity of oil used by the Gnome engine there is a wide difference of opinion; the makers state that the consumption is 2 litres per hour, others set the lubricators (two in number) to give one drop per 14 engine revolutions. It is my experience that the consumption of pure castor oil is about 7 litres per hour, that is hearly half as much as that of the fuel. It is no doubt possible to greatly reduce this consumption, but with an engine of the cost and refinement of the Gnome, no risks should be run through lack of lubricant.

Castor oil leaves rather a sticky black deposit in the combustion chambers and on the liners of the exhaust valves; it is therefore advisable to clean these out thoroughly after a few hours' run, but as the cylinder covers containing the exhaust valves are very easily removable, no great inconvenience is caused.

I have not discovered any other disadvantages in the use of pure castor oil, though it is somewhat expensive in the ordinary way, and inconveniently viscous for treatment with a hand pump.

I recently designed a cooling system for use in Ceylon for cooling oil circulated continuously from a tank through the engine and back again; in this system all the oil before returning to the engine passes through cooling tubes situated below the machine, and thence through a strainer to the pump. The oil reservoir, of brass, is well exposed so as to help to keep the temperature of the oil down.

My experience in gas engine practice led me to the conclusion that a lubricant which had been used for the pistons and had been exposed to high temperatures, was unsuitable for returning for further piston lubrication owing to its loss of "nature," although it would suffice for the lubrication of bearings. From this it appears to me that oil which is continually used on a circulating system should be regularly supplemented with fresh oil, and that the surplus of old oil should be allowed to drain away.

Mr. J. Veitch Wilson wrote: The points raised in Mr. Morcom's paper are primarily, the relative merits of "forced" lubrication under pressure, as exemplified in the Belliss and Morcom engine, and of "splash" lubrication as used in the Willans and Robinson engine; secondly, the effect of these two methods of lubrication upon the lubricants, and incidentally the nature of the lubricants required to meet the conditions which are found to exist.

While these two systems differ in detail, they agree in adopting the principle, found also in ring, chain, and some other modern systems, of the continuous and repeated use of the same supply of oil in contradistinction to the older systems in which the oil was supplied by syphons or other lubricators, and was lost after once working its way through the bearings.

I had hoped that Mr. Morcom might have prefaced his paper by some little historical account of various forms of continuous use of the same oil, but as he did not do so, I presume that he does not regard it as germane to his subject, or perhaps he thought it would be too elementary for his audience.

The continuous use and re-use of the same oil did not, however, originate with either the "force" or the "splash" system, but is, I think, much older than either, as I remember that I was called upon some 25 or 30 years ago for advice regarding the oil to be used on some large engines in Lancashire, the crank shaft bearings of which—probably to secure a more copious supply of oil, and to avoid waste—had been fitted with a small force pump which raised the oil from a receiver below the crank shaft and delivered it to the lubricators on the top of the bearings. The engines were either by Musgrave's or by Hick, Hargreaves & Co.

I found accidentally a still earlier reference to "forced" lubrication in searching through a volume of patent abstracts for information on some other subject, when I came on particulars of a patent (No. 6204, of 1831) by Samuel Hall, for an engine to be driven by steam, gas or other medium. His third claim is:—"In a method of lubricating the engines which consists in repeatedly injecting by a small force pump of the same oil or other lubricating matter either into the working cylinder, or preferably into a pipe from the boiler or generator between the throttle valves and the valve or cock to the working cylinder. After passing through the cylinder and lubricating the piston and rod, the lubricating matter is ejected with the gas or other fluid into a vertical pipe whose upper end is open to the atmosphere, and whose lower end terminates in a vessel of water. The oil falls into and floats on the water, whence it passes by a pipe back to the forcing pump." Here we seem to have the germ of forced lubrication.

The amount of information contained in Mr. Morcom's paper indicates that an immense amount of work must have been undertaken in its preparation both in the shape of actual experiments and in considering and marshalling the results of them. Unfortunately, Mr. Morcom has not given us many particulars of his experiments, but has contented himself by postulating his deductions in axiomatic fashion. We have therefore few data by which to judge of the reasonableness of his deductions except by reference to the published reports of other investigators. For my part, I am constrained to fall back on Beauchamp Tower's reports, at the reading of some of which I was privileged to be present, and I supplement these by various experiments in which I have myself taken part.

In the light of this information I submit the following remarks regarding Mr. Morcom's conclusions. Beauchamp Tower's experiments (1883 report) showed that the co-efficient of friction was reduced—

(a) At constant speed as the load increased.
(b) At constant load as the speed increased.
(c) At constant speed as the temperature increased.
(d) At constant temperature as the body of the lubricant was reduced.

In the course of a series of experiments, extending over several months, which I personally undertook upwards of thirty years ago on an Ingram and Stapfer oil tester, driven by a slack belt, I found that the machine took longer to attain full speed and developed a higher temperature as the oils used increased in body.

In the course of a subsequent series of experiments which I undertook conjointly with Mr. T. J. Pullin, of Burton-on-Trent, the results which I had previously obtained were confirmed, and we found, on testing each oil at its normal working temperature, that the viscosity of all had been reduced to nearly one common value.

We observed also, and it is now generally recognised, that the thicker the oil, the more rapidly it loses body as the temperature rises. From this we deduced that the proper oil for any bearing is the thinnest which, when exposed to the temperature necessarily generated under normal conditions, will provide the thinnest film which will keep the surfaces apart, and that the use of a thicker oil throws upon the bearing the necessity for the development of greater heat to reduce the oil to the proper consistency, and every additional degree of temperature requires so much extra power to produce it.

These results may be taken as generally confirming and, perhaps in some part, explaining Mr. Morcom's conclusions. They also lead me to regard with favour the system of forced lubrication, inasmuch as by providing a constant supply of oil where it is required (barring accidents), it dispenses to a large extent with the necessity for the use of thick oil, that will adhere to the bearing merely by reason of its thickness, and by permitting the use of thinner oil, obviates the difficulties which may arise from any considerable change in the body of the oil while in use.

Mr. Morcom gives various causes for failure of lubrication, among which he includes, "impurities such as water," but the long and satisfactory use of a water and oil bath in the Willans engines shows that water, *per se,* is not objectionable, and I believe that, despite all precautions, water is not unknown even in the Belliss and other engines with similar systems of forced lubrication. Water troubles in such engines depend mainly upon the oil used. With hard water nearly all fatty oils are liable to saponify to a greater or less degree, and although pure hydrocarbon or mineral oils are not generally affected in this way, there are some mineral oils which display an unaccountable affinity for water, with which they remain in close combination for long periods, forming a magma which may interfere with the proper distribution of the oil.

Mr. Morcom, on page 354, calls attention to two facts, namely:—

1. That the engine was quieter the higher the pressure.
2. That the friction i.h.p. averaged 2·13, 2·41, 3·33 with 30, 5 and 0 lb. per sq. in. pressure respectively,

and he finds some trouble in explaining what appears to him to be an anomaly. It seems to me that the engine ran more quietly on the higher pressure on account of the better cushion of oil, and No. 2 seems to be in accordance with Beauchamp Tower's experience that resistance diminished with increased load.

The figures 3, 4 and 5 are interesting as confirming some of the arguments given above. In each of these the curve I. represents the test without a circumferential groove, *i.e.*, with restricted supply of oil, while curve II. represents the test with circumferential groove and an ample supply of oil. The quantities of oil passing are shown in Fig. 4.

In Fig. 3 it is shown that the temperature of the limited oil supply, curve I., continued to rise until the maximum permissible was attained, while the ample supply, curve II., reached constant temperature apparently about 177 deg. F.

Conversely, we learn from Fig. 5 that curve II., representing the unlimited supply, found its constant e.h.p. at 7, while the curve I., representing the restricted supply, attained a higher temperature and reduced the e.h.p. to a little over 4, probably because the oil had been further reduced in body by the higher temperature.

This raises the question whether it is desirable or advisable to adopt artificial cooling for lubricating oil used in continuous circulation. My own experiments have shown that when, in the course of a test, fresh cold oil was applied to the bearing, the temperature fell and the resistance temporarily increased till the bearing again attained its normal working temperature. Therefore, except under circumstances in which the bearing, by conduction as in the case of a turbine, or by radiation in the case of other engines, is likely to be raised above the temperature of friction, I would suggest that it is not advisable to cool the oil, as by doing so we raise its viscosity and increase its resistance on the bearing. On the contrary, I recommend the use of a copious supply of good oil of moderate body applied by a powerful pump in good working order.

In conclusion, let me recommend airmen and others who use castor oil to remember and to profit by the experience of Willans and Robinson, who, after many years' use of that oil in the crank chambers of their engines, had to abandon it on account of its injurious action on the bearings and the interior of the crank case, the liquid contents of which I have in several instances seen as red as blood from the action of the fatty acids on the metal with which they came into contact. Castor oil must be used with great care upon phosphor bronze or white metal bearings.

Mr. Morcom wrote: In further reply to speakers at the discussion and to some written communications, I take this opportunity of dealing with a few points which still appear to call for notice.

Firstly, to deal with Colonel Crompton's remarks. There is no doubt that one of the reasons which makes it so necessary to have a very open suction to the pump, is the difficulty of making a very viscous oil follow the pump plunger well, and it is this same variation in the viscosity of the oil which gives one of its chief advantages to the plunger type of pump, which seems to pick up better with a thick oil than the other types.

Having now had an opportunity of reading Mr. Baillie's remarks, my opinion that he has in mind some very badly designed system for basing his criticisms upon is emphasized. In a forced system when the oil is cold there is considerable resistance in the various pipes, and plenty of pump capacity is necessary in order to get the viscous oil to the proper point of application when starting up. It is quite an easy matter to make your pump drive, pump and connections ample to withstand the extra work required by this condition, especially as the engine is usually not running very fast. The capacity of the pump should be made to supply ample oil when the engine is hot, to keep the lubrication system working efficiently; the oil pressure should certainly never drop to practically nothing. On the Lanchester car, even using a thin oil in hot weather, after a long run there is sufficient pressure to perform the process of lubrication, and the same can be said of steam engines running, say, in the south of India.

Then Mr. Baillie says forced lubrication is extremely bad because the small oil ways through the crankshaft are very liable to get blocked up. This is surely once more entirely a matter of good or bad design, and oil ways of the forced lubrication system can be made even less liable to trouble from this source than on the various splash systems.

The difficulty of getting adequate pressure throughout the system is also largely an imaginary one. If main bearings became so slack as to cause risk of breakdown of the system, it would be quite time to readjust them. As a matter of fact, with really good lubrication it will be found that a great deal of running is necessary before main bearings show any considerable wear at all.

With regard to the fear of oil getting into the cylinders, this again is liable to be exaggerated, and it seems to me that in the more or less haphazard splash systems there is more tendency to get oil into the cylinders than in the carefully directed forced system. In several cars using forced lubrication, the use of baffles has been found unnecessary, and certainly my experience has been that motor cars with forced lubrication give very little trouble from oil carbonisation in the combustion space.

Mr. Tucker asks how the oil pressures at various points of a journal were obtained. They were obtained in exactly the same way as Beauchamp Tower obtained his figures, that is, by means of pressure gauges opening into the bearing at various points, through small holes drilled straight through the bearing.

With regard to Mr. Garrard's remarks about the plunger pump, small plunger pumps can be run at quite considerable speeds provided the suction is made very free. Pumps of the type used on high speed steam engines, namely, what is known as the valveless oscillating type, have been run satisfactorily at as high speeds as 1,000 revs., but any difficulty on this score can be met by arranging a suitable drive, and it will be seen from the table that a great many different drives for the oil pumps are used, at speeds quite different from the engine or even the crank shaft speed. I take it that the groove on the teeth of the Albany pump is really in the nature of a form of packing, and Mr. Garrard seems to agree with me here.

The use of all animal and vegetable oils in any engine which runs for some time on a given charge of oil, has been found by experience on steam engines to be fraught with considerable danger, and the author's firm decry the use of anything but pure mineral oils. I cannot agree with Mr. Garrard that it is a dangerous practice to have several oil leads in parallel from one main lead. So long as the leads are ample in size, and the pump and pipes are all adapted to the maximum demand, the oil is bound to find its way to all the points of egress, that is to say, to each clearance space. If a system were designed in which the escape through less than all the clearance spaces was such as to take the whole output of the pump, trouble might occur as he suggests, but the design would be wrong. Of course, the centrifugal and inertia effects are utilised in various lubrication systems in a way which does provide pressure at the point of application, and to one who believes in forced lubrication, any such attempts are in the right direction, but do not seem to offer the same advantages of certainty and simplicity as the true forced system.

I am afraid I do not quite follow Mr. Martineau's deduction from my précis of the theoretical conclusions. The development of the film pressure throughout a bearing is practically an automatic process. What the designer has to ensure is an ample supply of oil to replenish this film as it gets destroyed due to efflux at the clearances. To do this it is necessary to have an ample supply at the point of least pressure, where it could be picked up by the revolving journal and carried round to establish a film.

The lifting of the journal in the Ilgner system by means of very high oil pressures is rendered necessary by the fact that, until a certain critical speed is reached, no oil film is established by rotation, and very great friction results. There is no doubt that high oil pressures do have certain advantages, but it is found that so long as the pressure is great enough to supply sufficient oil to maintain a film and carry away developed heat, it is unnecessary to go higher.

With regard to my remarks about the pump sucking air and not oil after a certain speed, what I meant was that, when this speed is reached the air, which is always in suspension in the oil due to the vacuum caused by the drag of the suction, expanded until it filled the whole of the pump's capacity, and therefore the action of the pump ceased. Probably this is what Mr. Martineau meant by cavitation.

To give the curves which Mr. Martineau asked for would undoubtedly be carrying the design back to first principles, but the empirical rule given is one which has been found serviceable in practice, and may be corrected by the experience of manufacturers. The form of the expression is at any rate simple and based on reason.

Mr. Brewer writes chiefly on the aviation engine, which was not dealt with as a special branch in the paper. His remarks on the lubrication of various types of light engines are most interesting, and I quite agree that insufficient attention seems to be paid to this important point in these engines. Probably at the present stage, however, they are so constantly tuned up and adjusted, that methods such as the use of castor oil, etc., and inefficient oil circulation are not of so much importance as they will be when the number of fliers is greatly extended, and the ordinary requirements of every-day engineering reliability have to be met.

Replying to Mr. Hounsfield's remarks about the indicator diagrams taken, the author does not maintain that these will be of extreme accuracy (indicator diagrams very rarely are), but they will do as their name applies—indicate. Mr. Legros in his remarks about the rounding off of edges on slots, etc., has saved me the trouble of supplying Mr. Hounsfield with some of the information he asks for.

Mr. Veitch Wilson's remarks are interesting, and I thank him for supplying some of the history which had not been included in the scope of the paper.

I do not think that the forcing of oil into a cylinder can be considered as a true case of forced lubrication, as it is not done with the same intention, and the subsequent history of the charge of oil is different.

I explained at the time of reading the paper that most of the experiments on which a good many of the deductions given were made, were made not really in connection with this paper, but for the purposes of design of bearings, pumps, etc., of steam engines and turbines, and it therefore seemed better before this Institution to give merely the results. I quite agree with Mr. Wilson's comments on Beauchamp Tower's experiments, which with modifications based on later researches agree with my own views.

With heavy loads and high temperatures, I cannot agree that water must not be regarded as a troublesome impurity. When the oil film is so small that the drops of water contained in an emulsion are squeezed out to an appreciable area, they have no separating power at all, and tend to rupture of the true oil film and subsequent inefficient lubrication. I always regarded the use of emulsion in the Willans engine as rather an over-bold piece of engineering, although undoubtedly considerable success was attained by this method. The fact that soapy water is used in a good many cases for the purpose of lubrication also is an instance where perhaps Mr. Wilson's ideas apply.

Mr. Wilson, in his comments on page 354, slightly misunderstands some of the figures. The pressures of 30, 5 and 0 lb. per sq. in. respectively refer to pressures in the oil pipe, and not pressures due to the load. I am rather inclined to agree with Mr. Wilson that oil cooling may only be necessitated when unsuitable oils and inadequate pumps are used. On the other hand, the amount of heat which has to be carried away from a bearing is considerable, and the final temperatures reached on a continuous run might become too high to give satisfaction with any known oil. In such cases a cooler is an absolute necessity, as for instance in turbine bearings, which are practically always either water jacketed or supplied with oil chilled in a cooler. I am glad to note that Mr. Wilson agrees with me in decrying the use of castor oil.