BALL AND ROLLER BEARINGS.

Although not coming directly under any one of the above headings, it might be worth while to say a word or two about ball and roller bearings. These had only comparatively recently come into vogue, especially in the cycle and motor car industries. The friction in their case belonged more to the rolling kind than the sliding. Goodman deduced the following laws from experiments carried out by him with Rudge ball bearings, using white "neutral" and pale American oils:—

1. The co-efficient of friction was nearly constant for all ordinary loads; hence, unlike lubricated cylindrical bearings, the frictional resistance varied directly as the load.

2. The friction was unaffected by change of temperature, and, therefore, by moderate changes in the viscosity of the lubricant.

Where steady heavy loads were to be carried, rollers were used instead of ball bearings. Due to their method of manufacture, they were very brittle and would not stand severe shocks. It was only with the advent of pneumatic tyres that these types of bearings were able to be used in cycles and auto-cars.

Solid lubricants, such as graphite and soapstone, were used in all kinds of bearings which were subject to very high pressures and very slow speeds. It was often found beneficial to mix powdered graphite with oils and greases used to lubricate the more heavily loaded bearings, since, besides offering very little resistance to motion itself under ordinary circumstances if a bearing happened to run dry, particles of it became jammed between the surfaces at the dry spots, and so prevented excessive heating. The graphite must be very pure, however, as the slightest trace of grit would counteract its usefulness. Graphite grease, besides being an excellent lubricant for gearboxes, also helped to deaden the noise of running when wear had taken place.

THE INFLUENCE OF LUBRICATION ON THE DESIGN OF BEARINGS.

Formerly, it did not worry designers whether their bearings could be lubricated or not; but, in these more enlightened days this important matter was not left out of consideration so much. Credit for this was largely due to Beauchamp Tower, whose experiments in this connection were quite classical. Till comparatively recently just enough oil was run into bearings to keep them from running hot. Whatever saving was effected in lubricants was more than lost in renewals and repairs.

POSITIONS FOR OIL-WAYS.

In practice the positions for oil-ways depended more on the individual taste of the erecting engineer than on any scientific principle. He often thought, when learning the trade at Eveleigh Workshops, that the cutting of a large groove in the crown of the axle-box brass would remove metal where it seemed to be most needed. This practice was, no doubt, the result of long experience, especially since, seeing that a large mass of metal was exposed to rushing air, there was plenty of opportunity for the box to keep cool.

In stationary machinery all the moving parts rested upon the bearings, and lubricants could be fed in through the holes and grooves in the cap, and so "perfectly" lubricate the bottom brass which took most of the pressure. But, in the case of axle bearings on trains and trams, the top brasses took the load, thus necessitating pad lubrication. In vertical engines the lubricant must be fed in from the sides of the brasses, as both of them alternately took the load.

Fig. 6 was intended to show the movement of the oil film in a bearing. It was drawn in on the "on" side, and expelled at the other three sides, as indicated by the arrows.

To quote Beauchamp Tower again, he found that, if the oil was distributed by means of a groove in the crown of the brass (see Fig. 7) the journal would not run cool when the load exceeded 100 lbs. per sq. inch. Being practically in the centre of pressure, it allowed the supporting oil film to escape, thus admitting of metal to metal contact just at the point where the oil ought to prevent wear. By releasing the weight for an instant, oil was allowed to flow down and lubricate the journal; but, as soon as the load was re-applied the oil filled the hole to its former level and the journal became dry. This arrangement clearly showed that it was more suitable for collecting and removing the oil than for lubricating purposes. Of course, in this experiment, the brass was very well fitted to the journal, and the latter revolved very steadily. This state of affairs was seldom realised in practice, for, besides indifferent workmanship, the vibration and side play helped to spread the lubricant over the surfaces. The same experimenter also tried the effect of cutting the groove parallel to the axle, but at the sides of the brass, as shown in Fig. 8, and found that here the bearing did not seize until the load reached 380 lbs. per sq. inch. The arrangement shown in the next figure (No. 9) did not allow the brass to take more than 178 lbs. per sq. inch. By using pad lubrication with any oil-ways he was able to apply loads of from 550 to 580 lbs. per sq. inch before breaking the oil film. When the lubrication was "perfect," i.e., when the lower side of the journal dipped into an oil bath, a bearing had been noticed to carry successfully as much as 620 lbs. per sq. inch.

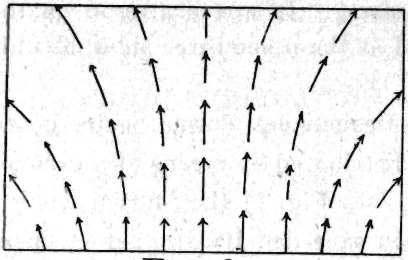

FIG: 6

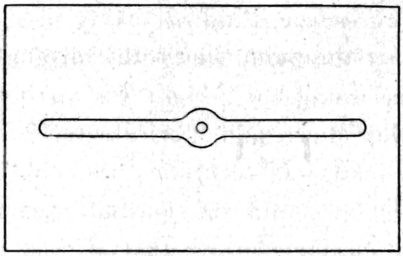

FIG: 7

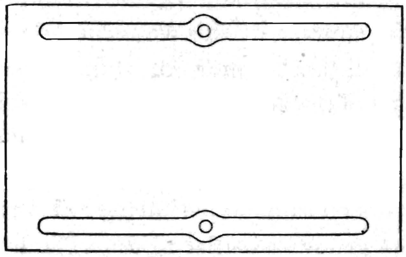

FIG: 8

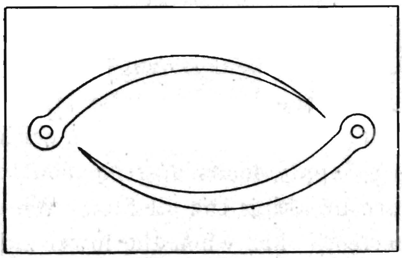

FIG: 9

Therefore, it was very desirable, in the design of bearings of all kinds, to aim at conditions of bath lubrication; and when this was not possible, forced lubrication should be adopted. This method was the only practicable one in most high speed machinery, e.g., steam turbines, Belliss and Morcam's, Sisson's, and other high speed reciprocating engines. Only small pressures need be applied, but it made all the difference to the life of the bearings. He had heard of gunmetal bearings, after they had been in constant use for two years under this system, preserving the tool marks just as they were when taken out of the lathe. "Splash" lubrication was also very successful in high speed engines, especially those worked by internal combustion.

HEATING OF LUBRICATING FILMS.

Bearings, even though well lubricated, generally increased in temperature after they had been running for some time. The amount of this heating depended largely on the efficiency of the lubrication. All oils decreased in viscosity with rise of temperature, especially the mineral; and, once the temperature reached that value which was able to destroy the viscosity of the lubricant necessary to maintain conditions of fluid friction, the bearing would rapidly run hot if not promptly attended to.

The number of British Thermal Units generated in a bearing in a minute was expressed by the formula:

$$\text{B.T.U.} = \frac{\text{Puv.}}{778}$$

where

P=load on bearing,

u=coefficient of friction,

and v=surface velocity.

Numerous experiments had shown that the greater the viscosity the heavier the load a bearing would carry, provided that the speed be not too high. The heat developed in the film must be conducted away through the metal of the shaft and bearing sufficiently rapidly to keep the temperature within the limits between which the lubricant was designed to work. The running hot of bearings was made use of sometimes, for instance, where tallow cups were employed, to melt the lubricant, so that it might flow to the bearing. Tallow cups were often used on lines of shafting.

ADMISSIBLE LOADS ON BEARINGS.

It was impossible at present to give a rule for finding the area of a bearing necessary to ensure cool running, as the conditions were so complex. The relative running positions of the journal and brass varied considerably with changing loads and viscosities. Fig. 10 showed that the relative positions with light loads and viscous lubricants did not alter much from those taken up when the shaft was at rest; whilst Fig. 11 would represent the conditions under heavy loads or with thin oils. Hence it was seen that increasing the load had the effect of bringing the surfaces nearer together on the "off" side, and widening their distance apart on the "on" side.

It was always found that, after some months' running upon a lubricated journal, a brass was worn so that its radius became greater than that of the journal. When in the running position, the new centre of the brass took up a position such as P (Fig. 11), by producing the line joining the two centres, P and O, as shown (the latter being the centre of the shaft), they could find where the thinnest part of the film was.

FIT OF BEARINGS.

If bearings had no end play circular grooves were liable to form on the rubbing surfaces. Therefore, wherever possible, provision should be made to prevent this undesirable effect from occurring. Last year when going over the Ultimo Power House with a party of students from this Section, he noticed that quite a large transverse movement was given to the armatures of the Willans and Robinson's dynamos to prevent this grooving action, although, in this case, it was done to preserve the commutators more than the bearings.

Another point to be noticed in bedding brasses was that they should never subtend an arc greater than 45 degrees on either side of the line along which the load acted, as, otherwise, it might lead to defective lubrication. When semi-circular, the metal should be filed away from the sides.

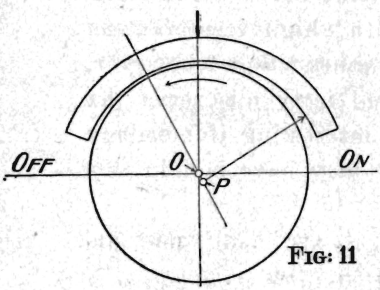

FIG: 11

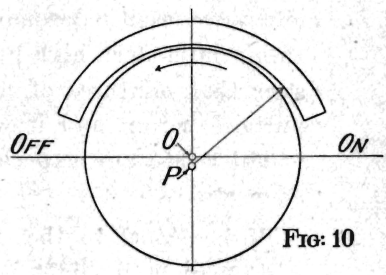

FIG: 10

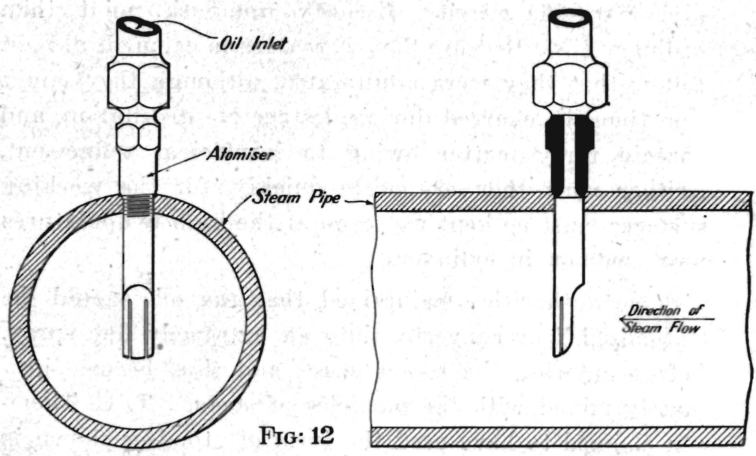

FIG: 12

No matter how well a bearing might be lubricated, it might, nevertheless, run hot if the workmanship was faulty. The designing draughtsman always assumed that the men at the machine and bench would faithfully follow his drawings. Burnishing the surfaces was frequently resorted to in order to decrease friction, and hence wear; and bearings that were exposed to dirt and dust must be enclosed in dust-proof casings. Grit very quickly cuts grooves in metal bearings. It seemed strange that the driving chains on motor lorries were not more protected.

INTERNAL LUBRICATION OF STEAM ENGINES.

This was not at all a settled question amongst engineers, but opinions were fairly unanimous that mineral oils alone should be used for internal lubrication, as fatty oils were easily decomposed at the high temperatures ruling in modern high pressure engines. Some, however, aver that mixtures of mineral and fatty oils serve the purpose better, and it would be interesting if members would relate any experience they may have had in this connection.

With regard to the properties which cylinder oils must possess, it might be stated that they should have a higher specific gravity, viscosity and flash point than ordinary oils. Because they were dark in colour it did not follow that they were adulterated, although they might sometimes be charred during destructive distillation, and contain tarry matter owing to insufficient refinement. Neither must they evaporate quickly, for the working surfaces must be kept wet even at the high temperatures now common in cylinders.

Some authorities maintained that the oil should be "atomised," or converted into an extremely fine spray, before entering the steam chest, and thus become intimately mixed with the particles of steam T. C. Thomsen claimed to have used the form of atomiser shown in

Fig. 12 with considerable success on hundreds of engines. From the figure it was to be seen that the oil was brought into the centre of the steam pipe where the velocity of the steam was greatest, and, therefore, had the maximum effect. Besides lubricating the parts more efficiently, he also found that this device was more economical than other systems of admitting oil. Of course, the atomiser could be fed equally well by sight-feed or mechanical lubricators.

When condensers were used and the condensed water fed back to the boilers, as in marine engines, no internal lubrication was allowed, owing to the injurious effect of the oil on the boiler plates. So far it had been found almost impossible to completely extract the oil from feed water of this kind. The concluding note of this paper would be devoted to the

LUBRICATION OF INTERNAL COMBUSTION ENGINES.

This did not differ very much from that of steam engines, but the normal temperature of explosion was much higher than that of superheated steam. Explosion temperatures ranged probably from 2000 to 3000 Fah., while superheated steam was never hotter than 600 or 700 Fah. Notwithstanding this intense heat, it was found that the working parts of the piston and cylinder had a temperature not much exceeding that of boiling water.