Causes of Failure in Ball Bearings.

By G. F. BARRETT.

NOTE.—The Paper was taken as read.

On the 3rd of May, 1900, Mr. P. L. Renouf read a paper before the Cycle Engineers' Institute—which Institute has now been incorporated with your Institution—on the subject of "Ball Bearings as applied to Cycles." This is now almost twelve years ago, and since then the motor car industry may be said to have come into existence.

Mr. Renouf in his paper, published in the Proceedings of the Cycle Engineers' Institute, Vol. II., page 35, went very fully into the historical part of the subject, alluding to the fact that ball bearings had been used—and apparently with success—in a race from Paris to Rouen in 1868, and he then traced the development of ball bearings in cycles through those of Messrs. Thomas Humber, William Hillman, Bown's Aeolus, and those designed by Mr. O. P. Clements for the Birmingham Small Arms Company.

The advent of the motor car, however, created the necessity for ball bearings suitable for carrying heavier loads than those which were met with in cycle construction, and great credit is due to the foresight of the Deutsche Waffen and Munitionsfabriken of Berlin who requested Professor Stribeck to carry out a series of experiments at the Technical Laboratory of Neubabelsberg, on the load carrying capacity of steel balls and ball bearings. These historical experiments—an extracted account of which appeared in *Engineering* on April 12th, 1901—paved the way for the modern commercial ball bearing, and the success of this is evidenced by the large number which are now being used every day, not only in the automobile industry, but in general engineering. This is, indeed, a case where the general engineer has benefited by the automobile industry, as it is largely the experience gained in motor cars which has led to the perfection of ball bearings, and made their application possible to nearly every form of machine met with in engineering and its many allied trades.

It must be evident that among users of ball bearings there are many who maintain that they are absolute perfection, that they have used them by hundreds and never had a single instance of trouble with them, but, on the other hand, there are some who maintain that they are unreliable and who will have nothing to do with them. It is hoped that these remarks will indicate the causes of trouble and point the way to their elimination.

The failure of ball bearings, apart from that due to direct overload, may be roughly attributed to five causes: faulty design, material, workmanship and mounting, and bad usage.

As regards the first of these, Professor Stribeck proved beyond doubt that in a ball bearing to carry journal loads, the ball should have only two points of contact; that is to say, one on the stationary and one on the revolving race, and that these two points of contact should be in a line at right angles to the axis of rotation of the bearing and parallel with the load.

Mr. Renouf, in his paper already alluded to, went into the theoretical points in connection with ball bearings, and described the so-called "spinning" action in connection with them. At the risk of going over what must be already familiar ground to most of you, it will be as well to describe this spinning action, as it plays such an important part in connection with the design of ball bearings, and, moreover, its ill effects are so much more pronounced in the larger bearings which are met with in automobile construction.

It seems natural to assume that, if ball races are so designed that the balls will carry the load across two diameters and touch at four points, two on the stationary and two on the revolving race, they would carry more load than if they had to do so across one diameter only. It is found, however, that a bearing designed on this four point principle very soon wears out, and this is almost entirely due to the so-called "spinning" action which takes place with this and similar designs.

Referring to Fig. 1, a ball is shown running in a "V"-shaped ball race, illustrated in cross-section, and the spinning action takes place at the points C, D, L and M. It will be clear that the ball has an instantaneous axis of rotation along the line AB; that is to say, if the ball be rolling in the direction towards us out of the diagram, the points e e will be travelling forwards and the points g g backwards, or away from us in relation to the line AB. If the observer's eye could be stationed at the point O, and move along at the same rate and in the same direction as the ball, and if he could observe the surface of the ball, it would appear to be revolving round the point D as indicated in Fig. 2. If he were to observe the points C, L or M in the same manner, the balls

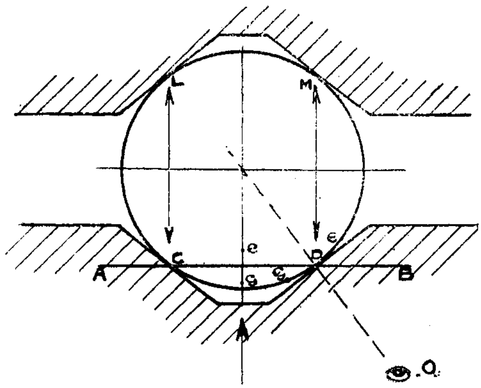

FIG. 1.

would appear to be revolving round these points in the same way. The bearing races, therefore, are not causing the balls to have a true rolling motion. Owing to the compressibility of the material, these points become surfaces, so that each point of contact becomes

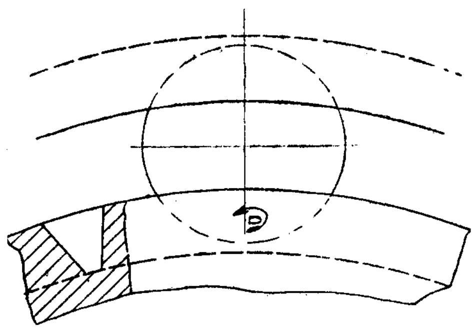

FIG. 2.

a step-bearing which is sliding or skidding round the surfaces of the ball race, and, being under pressure, the balls and races soon get destroyed. It is evident that the less the angle of the ball race is to the axis of rotation, the nearer the points of contact come to the top and bottom of the ball, so that when the races are parallel to the axis of rotation, this spinning action ceases entirely and a pure rolling action takes place, see Fig. 3. Another fault with this four point construction is that should the races not be mounted exactly opposite each other, or should side thrust come upon them, the load instead of being carried across LC and MD may be carried diagonally across LD, the ball rolling between these two surfaces and sliding and skidding at M and C.

The principle of two point construction was also discussed by Mr. Renouf, and he pointed out the great difficulty which arose when it was applied to a journal bearing, namely, the impossibility of making any adjustment which might be necessary owing to

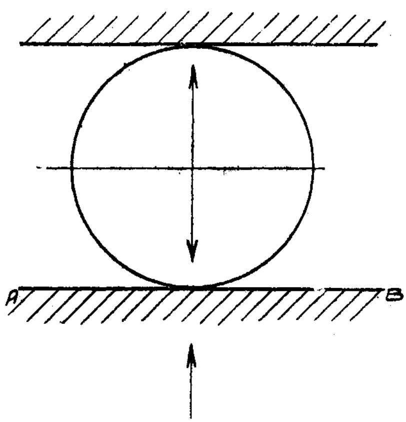

FIG. 3.

inaccuracies in manufacture, and also the impossibility of taking up any wear in the bearing. This difficulty was not at first easily overcome, and as a result bearings made on the cup-and-cone principle, such as were used in bicycles, but of larger dimensions, were largely used in the early motor cars.

Here it may be as well to point out the impossibility of making any adjustment for wear in radial bearings by means of lateral displacement of the races. Referring to Figs. 4 and 5: in the first case the shaft and inner race revolve in a stationary race and housing, so that any wear that takes place will be equal all round the revolving race, but will only occur on the stationary outer race where the load comes, namely, at the bottom. This outer ball race, therefore, will be larger across the vertical diameter CD, where the wear has taken place, than across the horizontal diameter AB, where there has been no wear. It is evident, therefore, that it is impossible to bring the outer race nearer the inner race owing to the absence of wear across AB, and it is therefore impossible to take up the wear across CD. The same remarks apply to Fig. 5, where the inner race is stationary, the wear taking place on this in the same way as on the outer race in Fig. 4. Again, the groove or track worn in the ball race will leave a sharp edge, and when

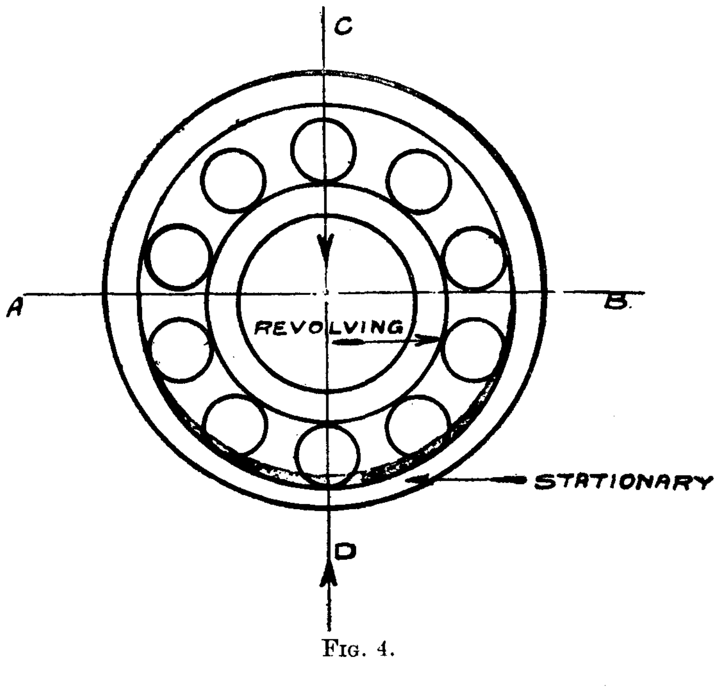

FIG. 4.

the lateral adjustment is attempted, the ball having to run on this edge, must quickly fail. This result must be found in all bearings which rely on adjustment by lateral displacement, including so-called adjustable roller bearings. It is, however, very doubtful if adjustment would be an advantage to the user even if it could be properly applied, as in inexperienced hands it would cause possible trouble due to too much or too little adjustment—a trouble that is often met with in connection with thrust bearings, where, of course, adjustment is possible.

The practical results which Professor Stribeck demonstrated in the year 1901 directed serious attention to the commercial manufacture of a two point ball journal bearing which would conform with the results which he arrived at. Owing to the rapid advance in the design of machinery, it is now commercially possible to manufacture ball journal bearings of this class which require no adjustment from errors of manufacture, and which, if properly applied, are to all intents and purposes indestructible.

The journal bearings mostly met with are what are termed "single row ball journal bearings," and consist of a single row of

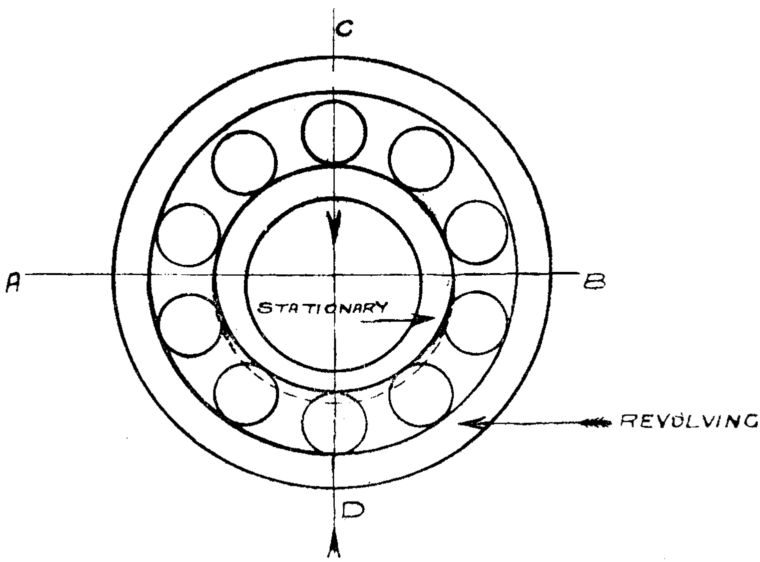

FIG. 5.

balls running round between two hardened steel races. There have been a great many different designs and methods of inserting the balls; the earliest method was to cut away a portion of one of the rings, through which cut-away portion the balls were inserted—this being afterwards filled up with a small piece of hardened steel held in position by a screw. This method did not prove satisfactory, as the small hardened piece broke away at the joint. Another method, which does away with this filling piece, is to cut transverse slots in both rings, which slots do not go right down to the bottom of the ball race, the balls being inserted by means of the elastic deformation of the rings. In order to do away with the transverse slots an ingenious method of assembling is to put in only half the maximum number of balls plus one—the centre ring being placed eccentric to the outer one, Fig. 6. The balls are then spread round between the two rings, when the centre one is brought concentric, Fig. 7. Owing to elastic deformation and heating it is possible to assemble half the total number of balls plus two. Another method is to cut away the sides of the ball race all round sufficiently to enable all the balls after they have been mounted in the inner race to be forced into the outer race when it has been expanded by temperature. There are also various modifications of these methods for doing the same thing.

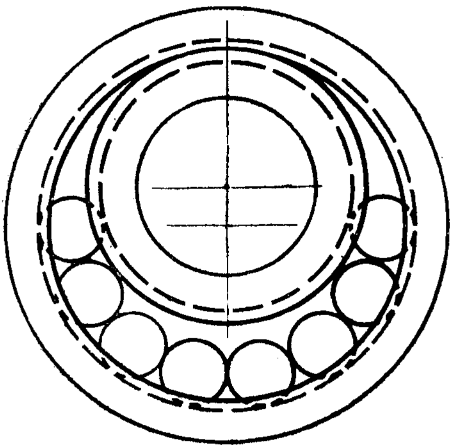

FIG. 6.

The curvature of the ball race naturally plays an important part in the life of a ball bearing. Owing to the fact that the path of the balls round the inner ring is convex and that round the outer ring concave, it follows that the track on the inner ring is the weaker of the two; and for this reason the curvature of the ball race in this ring is made to a smaller radius than the curvature of the outer race, so that the load carrying capacity of the two races shall be the same. The nearer the curvature of the ball races approaches that of the ball, the greater the load it will sustain, but the friction and wear, of course, go up in proportion. It is found in practice that a curvature about 5 to 10 per cent larger than the radius of the ball is the most suitable.

A cage is necessary to separate the balls, as, if no cage is provided, the slight difference in the speed at which the balls travel round the races, caused by the deflection of the balls as they go round under the loaded portion, bunches the balls together, and the friction caused by the points of contact of the balls with each other, running in opposite directions at a very high speed, produces such intense local friction that the steel is worn or burnt away so as to form grooves or flats round the balls. The pressure required to keep the balls apart, provided that the bearing is properly mounted and the load applied correctly, is very slight indeed —so slight as not to cause any appreciable wear whatever on the

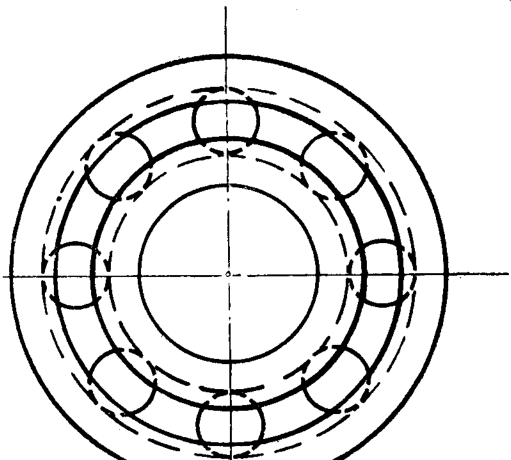

FIG. 7.

ball cages. If, however, loads come upon the races which tend to cant them out of true concentricity with each other, some of the balls will be driven up the sides of the track, and the alteration in the speed of these balls rolling round a larger diameter of track, will exert sufficient pressure to destroy any cage unless it is of substantial construction. A great deal of trouble in the past has been caused by the cages being too light and flimsy.

The comparative strength of the various designs of a journal bearing of a given size will depend—other things being equal—upon the number and size of the balls; the number of balls is regulated by the amount of space that it is considered desirable should be left for providing room for a strong and efficient cage and the size of the balls by the thickness that is required for the ball races. In bearings having unbroken rings—that is, without transverse slots—the number of balls is limited to that which can be got in by placing the rings eccentric. In bearings with a transverse slot for inserting the balls, the number is only limited by the cage, but great care has to be taken in producing these slots that they do not interfere with the continuity of the ball track and that the bearing rings are not overstrained in introducing the balls, or the balls themselves damaged. Providing that these points receive every attention in manufacture, the latter form of bearing is considerably the stronger.

Another form of bearing which has recently been brought into prominence is one having two rows of balls running in two grooves in the inner race, and in an outer race ground spherical from the centre of the bore. By this means two or three balls at the least must always be carrying the load as against one or two with the single row, but here, again, this is obtained at the sacrifice of making the outer ball race of so much greater curvature than the ball, that the load carrying capacity is not increased above the single row bearing.

As regards material and workmanship: it is hardly necessary to say that these should be the very finest possible. Only the most uniform and the very hardest material that it is possible to obtain should be used for the ball races, owing to the tendency that the metal has to flow away from beneath the ball under pressure. The ball races should be accurately ground and polished, as any roughness left from the emery wheels will make the bearings noisy.

A most important point is the accuracy of the steel balls. These can now be obtained commercially correct to standard size to within one ten thousandth part of an inch, and this is not, as with many articles, merely a figure of speech, but is absolutely true. It is this feature, as much as any, which has made ball bearings a success. The steel used for the balls should have a high elastic limit when hardened, as it is the elastic limit of the surface of the balls which is the limiting factor in their load carrying capacity. The cages should be made of a tough and uniform yellow metal, as steel or iron tend to lap down the balls.

Owing to the extreme accuracy which is now possible in the commercial manufacture of steel balls and ball races and the reliability of the material, failure due to faulty material or workmanship should be of rare occurrence.

When ball bearings were first introduced, the principles of mounting were not thoroughly understood, and a good many

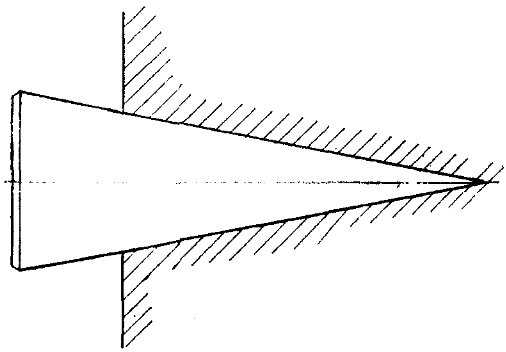

FIG. 8.

failures could be, and still are, very largely attributable to this cause. There are few appliances which behave so well when properly mounted, but which give trouble so quickly if one or two small details are not attended to.

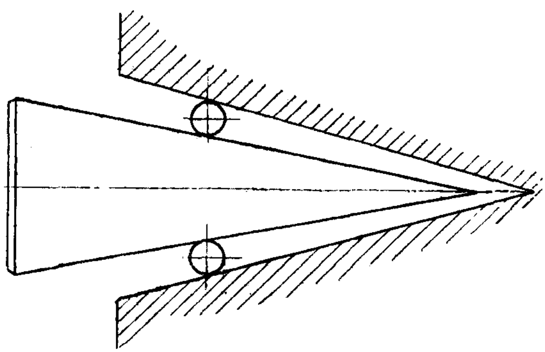

FIG. 9.

First and foremost is the question of side or axial thrust in connection with journal bearings. It is quite impossible to design a ball bearing with a single row of balls which will carry more than a small amount of both journal and side thrust, because of the enormous wedging action which tends to drive the balls up and into

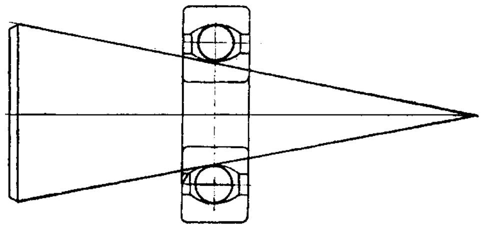

FIG. 10.

the races and burst them. If we take the ordinary wedge illustrated in Fig. 8 and remove from it as much friction as possible by

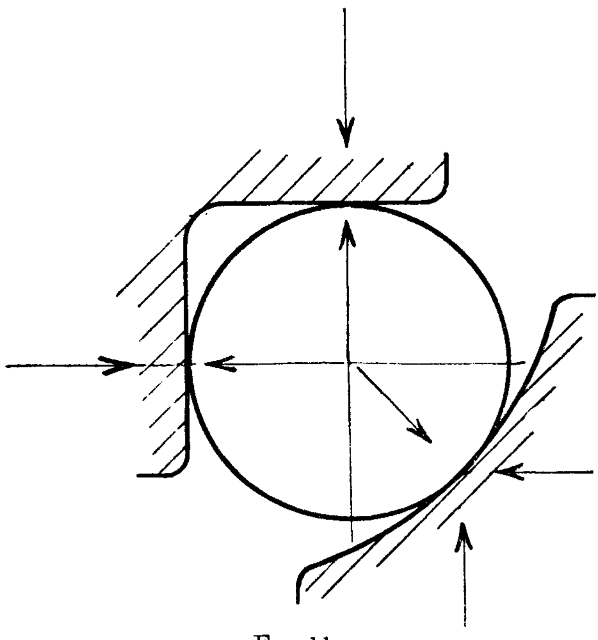

FIG. 11.

inserting a row of balls between it and the material into which it is being driven, as shown in Fig. 9, it will be precisely similar to the ball journal bearing shown in Fig. 10, but the friction of the wedge is again reduced by the fact that the balls are already in movement round the bearing and are therefore more ready to follow the movement of the wedge than if they were stationary. Owing to the fineness of the wedge the loads thus produced may readily become higher than the elastic limit of the surface of the ball or race can stand.

In this connection it is interesting to note that some manufacturers of this type of ball bearing advertise that their particular make of bearing will carry as much as 25 per cent of side thrust,

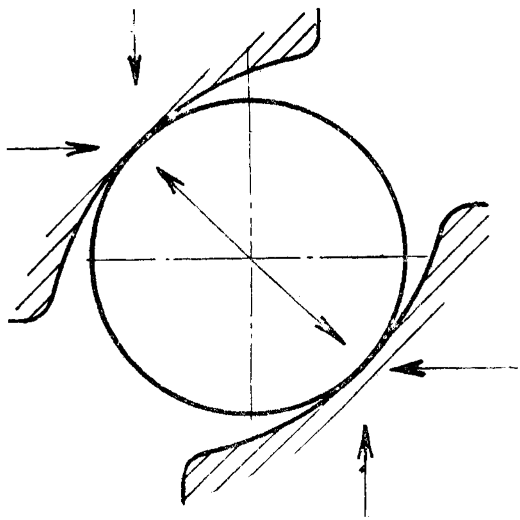

FIG. 12.

but when these bearings are supplied, the makers are particularly careful to print on the boxes which contain the bearings a warning to the effect that the bearing must not be clamped sideways, and that special care must be taken to see that the outer ball race has play sideways. It is, of course, obvious that if these instructions are carried out it is impossible for any side thrust to come upon the bearing. It is this wedging action which destroys the so-called cycle type of bearing, Figs. 11 and 12, and all others in which the load is not carried across the diameter of the ball parallel with the load and at right angles to the axis of rotation of the bearing.

Any other design but that shown in Fig. 13 must lead to trouble. This has been so abundantly proved in the earlier patterns of motor cars in which the cycle type of bearing was largely used with such bad results, that it is difficult to understand how this type can still be tolerated for anything but the light loads met with in cycle construction. It is, however, still met with under fancy names, and is claimed to carry both axial and radial loads.

For the front axles of moderate-sized touring cars the single

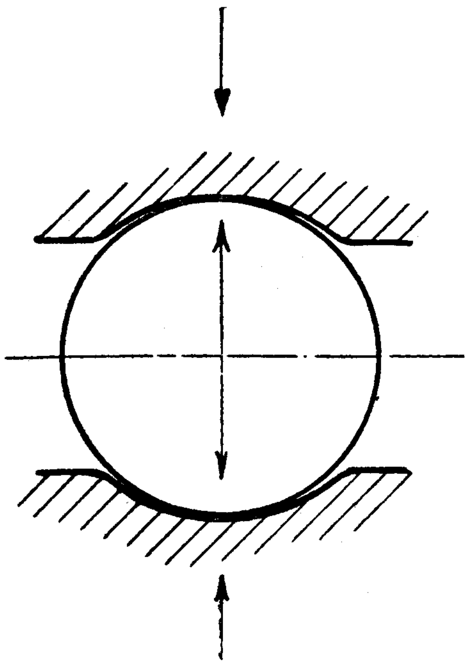

FIG. 13.

row type of journal bearing is found sufficient to control the shocks and jars which come upon it on the road, and also the side thrust when turning corners. On commercial vehicles, however, which have to carry heavier loads and run considerably greater mileage, and more particularly on taxi-cabs, which have, by the police regulations, to turn round in a very small radius, it has been found absolutely essential to provide thrust bearings in the front hubs where ball journal bearings are used, so that the latter have only to carry the pure journal load. This, of course, also applies to the heavier touring cars. It is not, of course, always essential that a ball thrust bearing shall be used to carry the side thrust when this is only slight, provided some other efficient means is employed, such as, for instance, on the lay shaft in a gear box, where hardened steel pins can be employed on the ends of the shaft.

Another cause of failure has been due to the mounting of the ball bearings in split housings in the same way as the brasses of a plain journal bearing. This often leads to the distortion of the outer ball race either by being nipped across the joint in the housing or at right angles to this. When this form of housing has to be employed, the bearings should be inserted in the bore after the cap has been firmly screwed down to see that it is a correct sliding fit. The same thing happens when the outer race is held in position by means of a set screw; the bearings invariably fail on the ball race just underneath the mark caused by the set screw, and at the opposite end of the diameter, showing that the race has been distorted and the balls nipped between the two races. If the inner race is forced on to a shaft which is out of round, the ball race will be distorted and cause trouble. Another cause of failure has been found to be due to the holding of the inner race on the shaft by means of a key—when this is driven up tight it has distorted the inner race and caused failure for the same reason. Both the inner and outer races, where they have to be held, should be a tight fit and clamped sideways against a flat shoulder truly bored or turned at right angles to the axis of rotation, and clear of the radius and burrs. The outer ball races are sometimes found distorted through being mounted in housings that are not bored truly round. It should not be forgotten that ball races are only hardened steel liners and therefore very easily deformed, and that, owing to the parts in contact being all of hardened steel, the least deformation will nip the balls. It is therefore extremely important that the bearings should be truly and firmly supported both on the shaft and in the housings.

Another trouble is the "creep" of journal bearing races. As this is not yet quite thoroughly understood by all, I have here a model which will assist to explain it. It is sometimes thought that the inner race slides round on the shaft upon which it is mounted owing to the friction of the balls driving it round. This is not the cause. It is due to the slight difference in diameter between the shaft and the bore of the inner race of the bearing, the revolving shaft rolling round inside the bore of the inner race as it rotates. If the diameter of the shaft is one thousandth of an inch smaller in diameter than the bore of the bearing, its circumference is approximately three-thousandths of an inch less, and in each revolution a point on its surface would travel three-thousandths of an inch less than a point on the bore of the bearing. If the shaft makes one thousand revolutions per minute, the surface of the bore will, therefore, theoretically "creep" forward three inches in each minute. Anything in the nature of a key or grub screw which is put in to prevent this is quickly worn away, as it is subjected to the full load on the shaft in either direction during each revolution, and is also subjected to a sliding of the keyway on the key owing to the slop of the bearing. It will be seen that this effect only takes place on the revolving member, that is to say, between the shaft and the inner ball race in gear boxes, etc. where the shaft rotates, and between the outer race and the hub-shell of, say, front axles, where the outer race rotates. The only way to prevent this "creep" is to make quite certain that the revolving races are a tight fit. There is no tendency whatever to "creep" between the outer race and its housing where these are both stationary, or between the shaft and the inner race where these are both stationary. This is because the load remains continually in the same direction. When the load itself revolves, as is more often met with in vertical shafts having an unbalanced rotating mass, then the outer race will tend to revolve in the opposite direction to the load.

For the majority of bearings met with in motor car work the best fit to allow is to make the shaft half a thousandth of an inch larger in diameter than the bore of the inner race where this revolves, or to make the bore of the housing half a thousandth of an inch smaller in diameter than the outer race where this revolves; if a greater amount than this be allowed for a driving fit, the races expand or contract, nipping the balls and not making the bearing appreciably tighter on the shaft or in the housing. There is unfortunately a tendency to assume that, if a ball bearing is slack in the ball races to start with, it is carelessly made and it is rejected for one that is a tight fit. It would be better, in most instances, to look out for those bearings that are too tight to start with and to reject them. In commercial practice it cannot be expected that the internal diameter of a ball bearing can be more accurately finished than to within a total variation of half one thousandth of an inch, and it is also impossible to expect engineers to manufacture their parts to within a finer limit than this. If, therefore, it is decided to make the diameter of the shaft half a thousandth of an inch larger than the bore of the bearing, it will happen when the two extreme opposite limits are reached both on the shaft and in the bearing, that in one case with the smallest diameter of the shaft and the largest diameter of bearing, the dimensions will be exactly the same, which will be too slack a fit, while in the other case the largest diameter of the shaft may be a full thousandth of an inch larger than the smallest bore of the

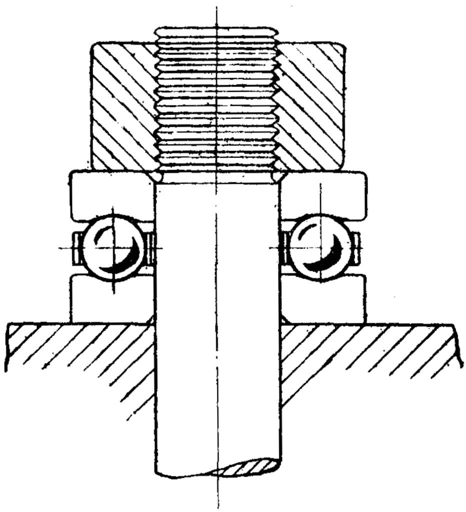

FIG. 14.

bearing, which will be too tight and tend to expand the race and nip the balls. In actual practice, however, the bearings and shafts are found to be well within the limits of dimension, but it would appear as though occasionally there must be a certain amount of fitting done, that is to say, when a bearing is too tight or too slack a fit, another must be tried. In repair work, if a race has started to "creep" on the shaft and worn it down ever so slightly, it is necessary to have a bearing of a special bore, or to have the diameter of the shaft enlarged.

The "creep" of thrust bearings is caused by the two races not being parallel with each other, or not being kept in rolling contact. A burr in a radius or on a shoulder is sufficient to throw the race out of square, and the load then all comes on one ball at a time. This load travels round the stationary race with the revolving race, and owing to the give in the washer it is driven round on its seating. This can always be prevented by mounting both washers perfectly parallel and concentric, when there is absolutely no tendency to "creep," and no pin or key is required to hold them. Mounting the stationary washer in a spherical seat does not get over this, as the race will still "creep" round if the revolving race is not square with the axis of rotation. For this reason mounting thrust races against nuts screwed on the shaft as shown

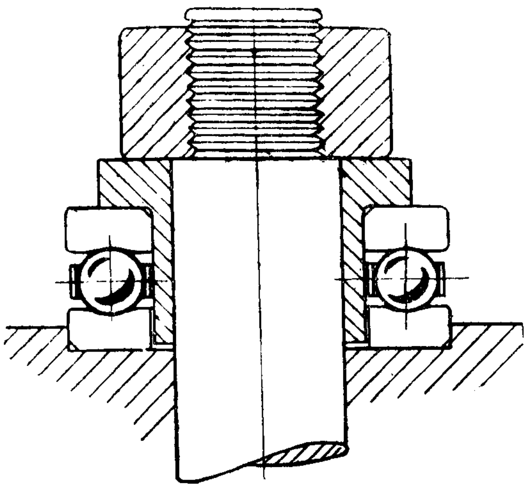

FIG. 15.

in Fig. 14 is not good practice, as, unless the greatest care is taken in machining the threads, the face of the nut is seldom at right angles to the shaft. When this cannot be done a sleeve sufficiently long to ensure its shoulder being kept at right angles to its axis of rotation should be used as shown in Fig. 15. When the balls are not kept in rolling contact the cage and balls, if on a horizontal shaft, will fall eccentric to the races and a similar effect will take place.

Coming now to some common forms of ball bearing mounting on back axle gears, Fig. 16 shows a design where the differential housing is supported on two journal bearings and outside these a single thrust washer is shown at each end, the whole lot being held endways between two fixed abutments or flanges on the back axle casing, the adjustment being made by means of packing pieces against the thrust washers. The inner races of these journal bearings should not only be a tight fit on this differential sleeve, but they should be firmly clamped endways by means of a nut. The

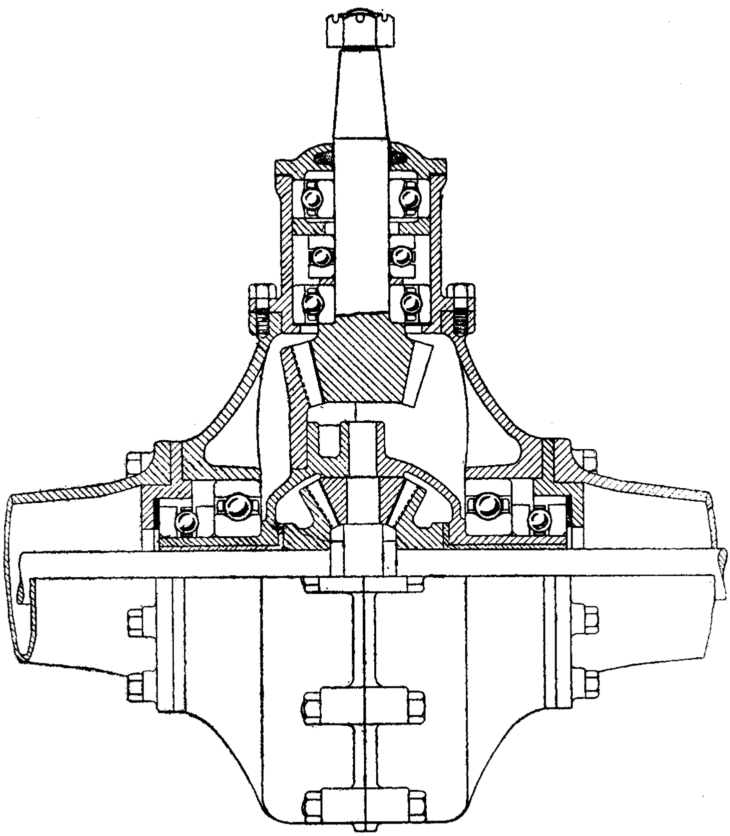

FIG. 16.

only clamping that they get in this particular design is through the ball thrust washer, and as this should be adjusted without pressure there is no clamping effect whatever on the journal bearings except when the thrust comes on. By this method it is quite impossible to observe accurately the adjustment of the thrust washer. The frictional resistance of ball bearings is still so slight when they are too tightly adjusted that when a back axle is mounted as shown it would be very difficult to detect a permanent overload of a ton on these bearings.

There are here two models: one showing a double thrust bearing carefully adjusted, and another one with a load of a ton, and although, as here mounted, the difference in adjustment can be detected, yet they will both spin round, and it will easily be understood that when these are hidden, and to the friction of the bearings is added the friction of the gears and the inertia of the moving parts, the difference will be so slight as to be practically unnoticeable. These bearings then may have a permanent load of a ton, to which has to be added the working load that they are designed to carry; this will probably overload them above their carrying capacity.

It is of course important that the large bevel wheel on the differential should be exactly in correct pitch with the small bevel pinion and be accurately maintained there. It is very difficult to ensure this being accurately done by means of these packing pieces. If the thrust bearings are permanently overloaded and the races brittle away, chatter of the bevel wheel will start and the gear will be noisy and wear away quickly. The bearings on the bevel pinion consist of a journal bearing threaded up against the pinion, a single thrust washer, a distance piece and an outer journal bearing, all held in position by means of an end cap which abuts against the outer ball race of the outer journal bearing. Neither of the inner journal races are clamped upon the shaft, and it is a difficult matter to thread the first bearing such a long distance over the shaft, and make it a tight fit. "Creep" of these inner races is soon bound to take place. A single thrust bearing only is employed. It is true that theoretically the thrust is only in one direction, but owing to chatter and spring of the parts it is just as important to prevent the bevel wheels getting too far in mesh as it is to prevent them getting too far out. The distance between the shoulder on the pinion and the end face of the housing which controls the distance of the pinion in gear with the bevel is filled up with as many as seven different pieces, thus making accurate fool-proof mounting impossible. The only means of holding the pinion from getting too far in mesh with the bevel is through the inner race of the outer journal bearing, so that this ball race has to take any side thrust which may come upon it in this direction, and should "creep" take place on this bearing, there is nothing whatever to hold the pinion from getting too far in mesh.

Another common form of mounting is shown in Fig. 17. Here one thrust bearing only is shown on the differential housing, the location of this housing in the other direction being controlled

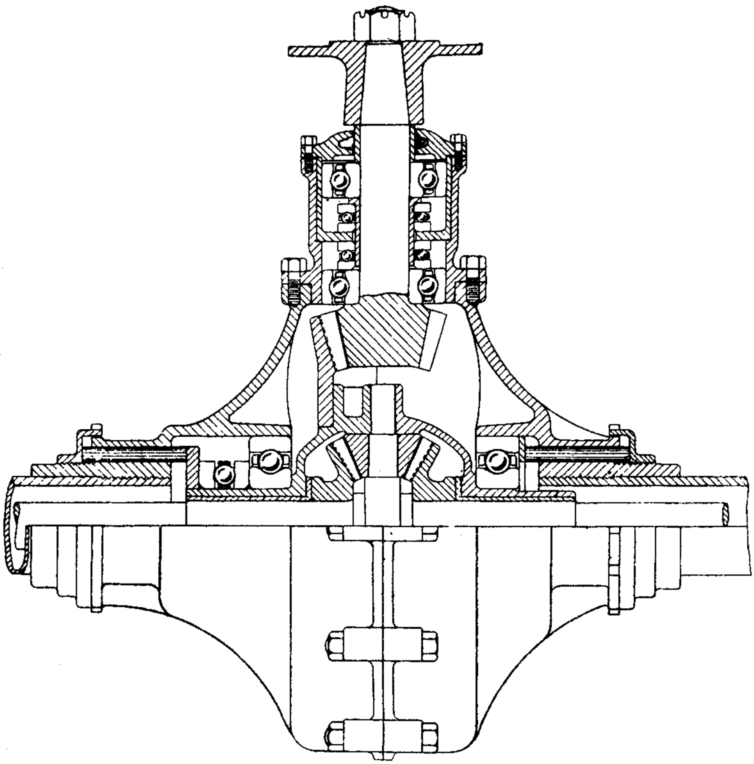

FIG. 17.

by the journal bearings. Instead of packing pieces being used, nuts are provided on the back axle housings, the end movement of which is transmitted to internal washers by means of three pins. By these means it is quite impossible to tell when the thrust bearing is properly adjusted, as the thrust in the opposite direction is taken by the ball journal bearing, and not until considerable side thrust had been put upon the latter would the over-adjustment be observed. Again, any after adjustment of these nuts will tend to throw the bevel gear out of adjustment, and unless these pins are all exactly the same length, the washers will be out of square with the shaft and cause overloading of the balls and races in one spot.

The pinion is mounted with a double thrust washer, but there

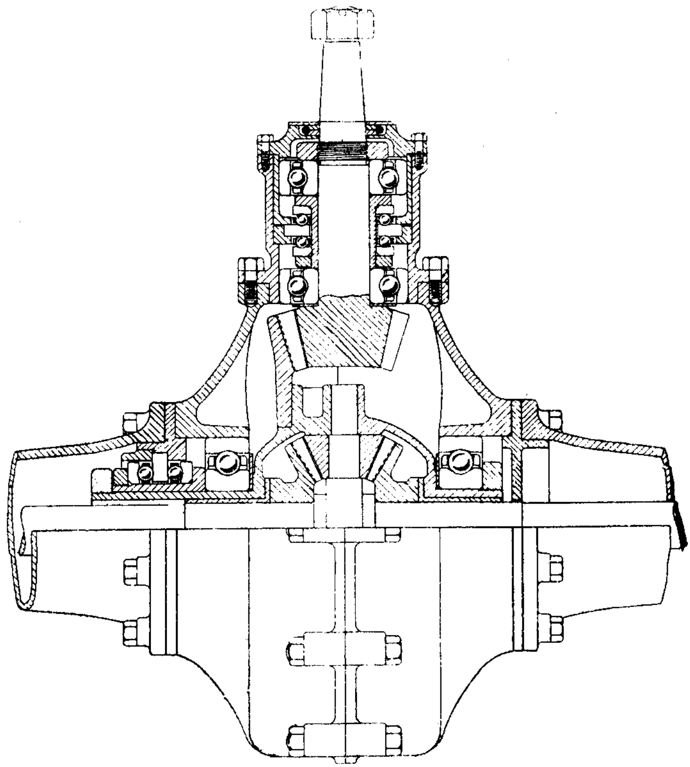

FIG. 18.

is no adjustment, the journal bearings and the double thrust bearing being entirely controlled by the boss of the universal coupling which is mounted on the end of the pinion spindle by means of a taper. It must be evident to everyone that this taper will never ensure the coupling being mounted on the pinion twice alike, and that the thrust bearing will either be greatly overloaded or be too loosely adjusted.

Another design that is often met with is the mounting of the small pinion with a ball bearing outside it. This bearing then has to take most of the load, and, as there is not sufficient room to mount one large enough, it invariably gives trouble.

The correct method of mounting these bearings is shown in Fig. 18. The differential housing is supported as before on two ball journal bearings. The inner races of these are firmly clamped endways by means of the two nuts on the end of the differential sleeve. A double thrust bearing is mounted on one end. The adjustment of this bearing is made independently before assembling, and when properly adjusted the adjustment is firmly locked. The centre race of this bearing, which controls the bevel wheel sideways, is screwed up against a shoulder, the distance of which from the centre line of the pinion can be easily checked and cannot be altered. A similar piece to that which holds the double thrust bearing in the housing is used at the other end of the differential as a means of preventing grease extending along the shaft, and this makes a more or less grease-tight joint for the bevel wheel housing. The bevel pinion is mounted in the same way, and has a double thrust bearing which has been properly adjusted before assembling and locked in adjustment. It is clamped inside the housing by means of a collar and sleeve; the distance from the shoulder on this collar to the centre of the back axle can easily be determined and cannot afterwards be altered. The bevel gear therefore will run in its correct pitch, and there will be no chatter of the wheels tending to interfere with the correct running of the teeth in each other, and wear and noise will therefore be reduced to a minimum. It will be noticed that all the journal bearings have thin steel discs mounted alongside them next to the gear wheels so as to prevent dirt and chips getting into the ball races. All the inner races of the journal bearings, being the revolving members, are clamped sideways on the shaft, and all the outer or stationary races are free to slide endways.

One objection raised to the use of a double thrust bearing on the differential sleeve is that it is unsymmetrical and necessitates the use of different lengths of axle tube. It will be noticed that this design gets over this objection.

It is difficult to understand why double thrust washers are not universally used on both the differential housing on which the bevel wheel is fixed and upon the bevel pinion shaft, as not only is it found in practice that wear is very considerably diminished thereby, but the noise which is now such a very important factor in all parts of a car is also reduced.

Fig. 19 shows bearings as often mounted in a gear box. The driving shaft has the first journal bearing only held on by means of a thrust washer, and the second journal bearing is not clamped upon the shaft at all. The main shaft has its journal bearings mounted in the same way. The adjustment for the thrust bearings is through the outer races of the two outer bearings, through two special washers on to the two single thrust bearings, and then through the inner races of the two journal bearings on to shoulders on the driving and main shafts, the thrust between these two being taken on two hard steel buttons inside the driving shaft. The

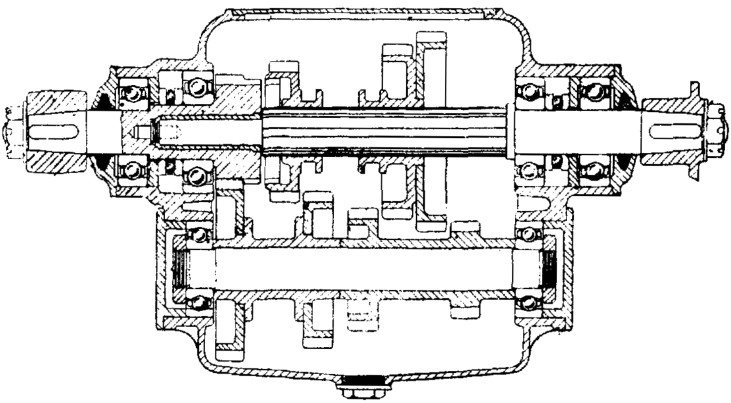

FIG. 19.

adjustment, therefore, of these thrust washers depends entirely upon the fitting of the outer dust caps on the ends of the gear box, and it is impossible to avoid the liability of too tight or too slack. adjustment. This adjustment also necessitates the sliding of the outer races of the two outer bearings in their housings, and unless the inner races are a sliding fit on the shaft to accommodate themselves to the outer races, side thrust will be put upon these journal bearings. Again, the inner races should not be a sliding fit on the shafts because "creep" must result. The two journal bearings on the lay shaft have their inner races properly clamped on the shaft, but any end movement is controlled by these bearings, the end caps on the gear box having internal flanges which abut against the outer races. If either the length of these internal flanges, or the length of the sleeve on which the gears are mounted, is too great, a permanent side thrust will be put upon these journal bearings.

Fig. 20 shows the same gear box with the bearings correctly fitted. Double thrust washers are shown on the driving shaft and the main shaft, both of which can be properly adjusted and the adjustment firmly locked before they are mounted on the shaft. The journal bearings have all their inner or revolving races firmly clamped endways and all their outer or stationary races a sliding fit. On the lay shaft the outer races are a sliding fit in the housing, the shaft being prevented from moving endways by

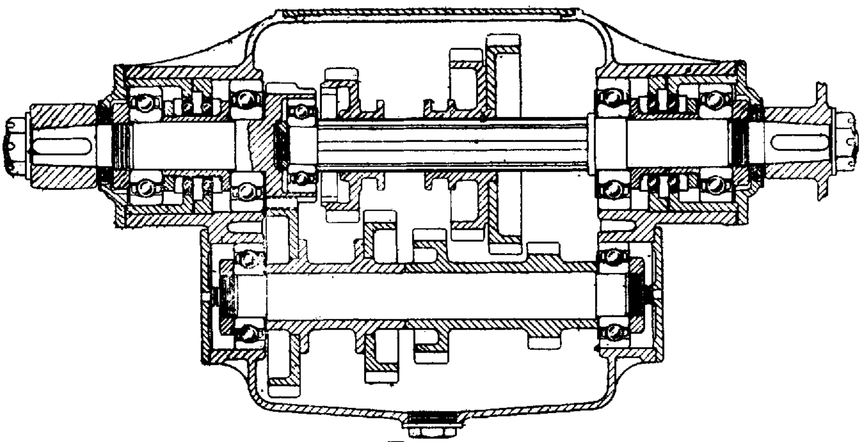

FIG. 20.

means of hard steel buttons between it and the end caps. Steel washers are shown mounted alongside the journal bearings on the interior of the box to prevent dirt and chips getting into the ball races.

Fig. 21 shows ball bearings as often applied to a worm driven differential. Two single thrust washers are shown one on each end of the worm, and the inner races of the journal bearings are held in position by these instead of being firmly clamped. This is a particularly bad form of mounting, as not only is it very difficult to adjust these bearings correctly, but the elongation of the worm and shaft due to the heating of the gear will put excessive overload upon the bearings. It is, of course, specially important with the high speeds and thrusts met with in worm drives to ensure that the thrust and journal bearings are efficiently mounted.

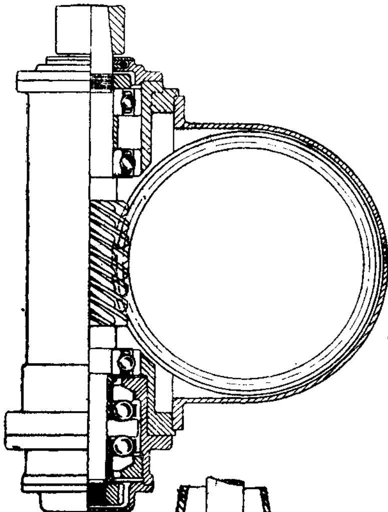

Fig. 22.

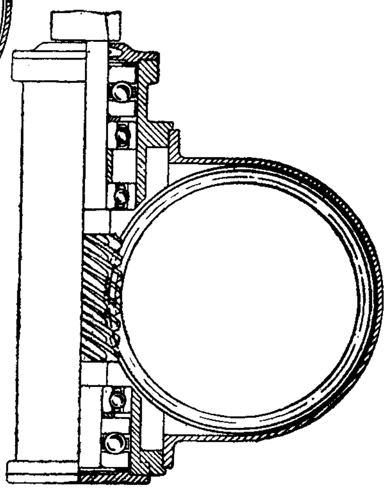

Fig. 21.

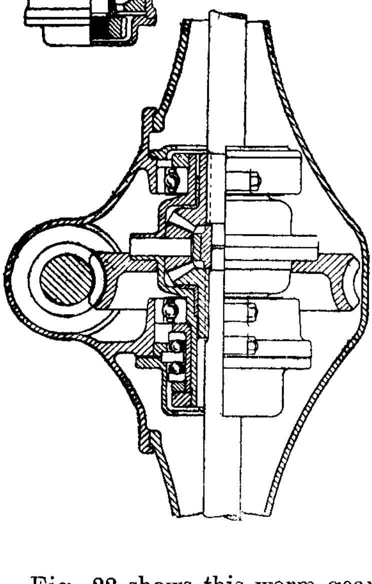

Fig. 22.

Fig. 22 shows this worm gear mounted with the double thrust bearing as recommended. The adjustment is made independently and can be locked permanently in its correct position. This bearing can always be removed and replaced without altering the adjustment. The differential itself is mounted on two single journal bearings and a double thrust washer. This particular design is interesting because, when the axle shafts are withdrawn from the hub end, the whole worm gear can be taken out of the back axle casing for inspection without dismantling it in any way.

Fig. 23 shows the bearings as often applied to front hubs. Here the inner stationary races are clamped endways and the outer revolving ones are a sliding fit in the housing, which, as already

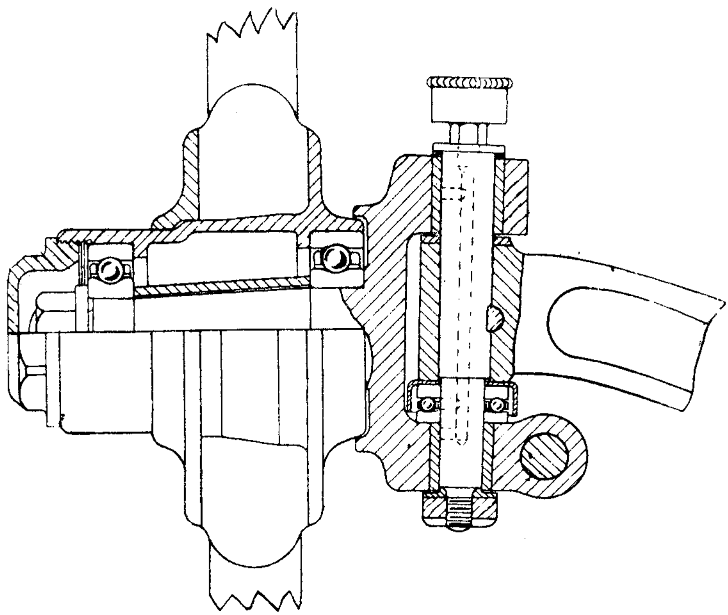

FIG. 23.

pointed out, is not correct. End movement is controlled by means of the two internal flanges which are formed on the inside of the hub between the two bearings. It has frequently been found that the distance between these two flanges is slightly greater than the length of the sleeve separating the two inner races. When these latter are clamped up tight, it therefore forces them together and a permanent side thrust is put upon the bearings. Practically no provision is made for excluding moisture at the inner end of the hub, and trouble soon arises from this cause.

The correct method of mounting these bearings is shown in Fig. 24. The inner race of the outer bearing only is clamped on the shaft, the other inner race being a sliding fit thereon. Both the outer races are firmly clamped endways in the hub.

Fig. 25 shows the same bearings with a double thrust bearing mounted between them. In this case both the outer races of the journal bearings are held firmly and clamped endways in the hub, the inner races being a sliding fit upon the axle and sleeve.

In these applications it is most important to lock the end disc

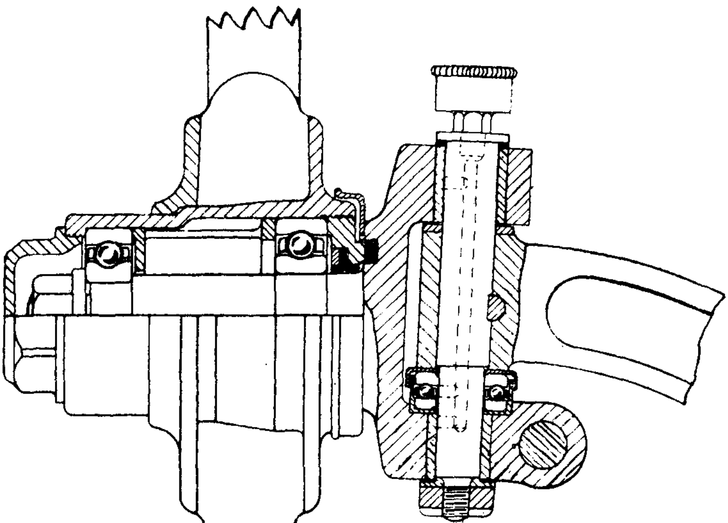

Fig. 24.

clamping nut firmly in the hub, and also to see that the hub is not too thin at this point. One method is to employ three cheese-headed screws and allow the heads to project well into the hub as shown in the figure, and to fit a split pin passing through the hub shell into the nut. If the hub shell is made too thin at this point it is liable to expand and allow the threads to become loose.

For back axles a usual method of mounting is shown in Fig. 26. Here the inner stationary races are clamped and the outer revolving ones are allowed to slide. This, as before mentioned, is not correct, and they should be fitted as shown in Fig. 27, where the two outer races are held and clamped endways and the inner races are allowed to slide on the axle sleeve. In this case the back axles

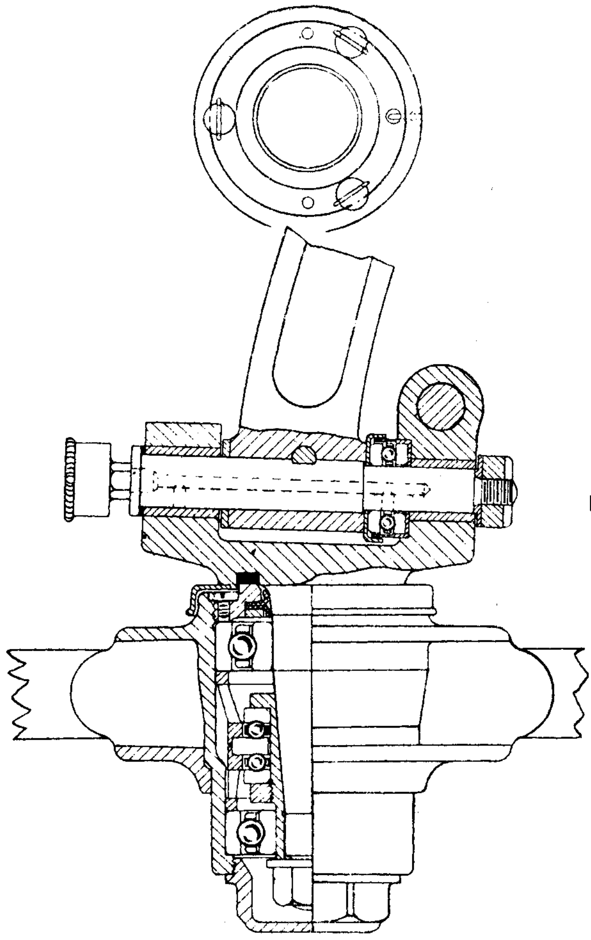

FIG. 25.

are located in the differential and the thrust is controlled by the double thrust bearing on the differential sleeve.

A bad design of double thrust bearing is shown in Fig. 28, where a distance piece is used to separate the two revolving washers. It is necessary to make this piece sufficiently exact in length to ensure that when the nut is screwed up tight the bearings will be in proper adjustment. For the reasons already stated this is a very difficult thing to do, and would necessitate an individual piece for each bearing, thus making interchangeability impossible. The latter is, of course, necessary, as the parts are loose and handled separately. If this distance piece be

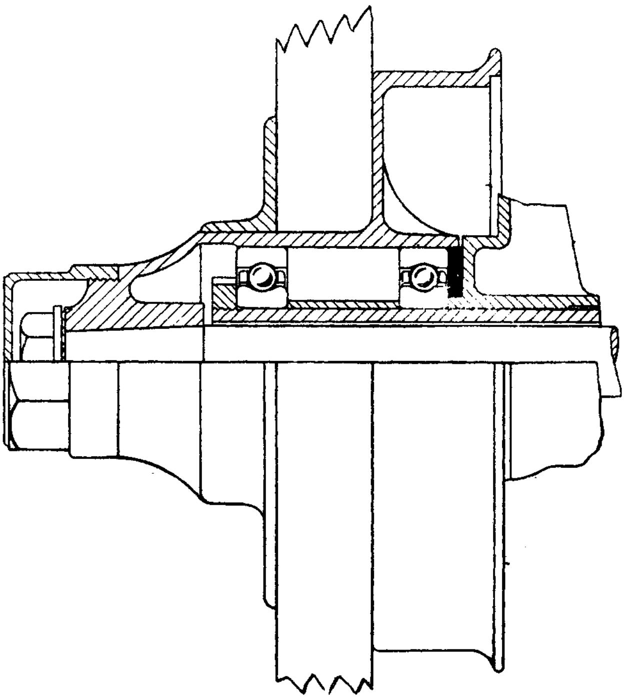

Fig. 26.

used in connection with a bearing having two rows of half-inch balls with ten balls in each row, and if it be one-thousandth of an inch too short, it will put a permanent load of half a ton on the bearing. It will thus be seen how important it is to have correct adjustment, which can only be obtained with a bearing mounted as shown in Fig. 29.

A single thrust bearing is sometimes used for taking thrust in both directions as shown in Fig. 30. It is mounted with the

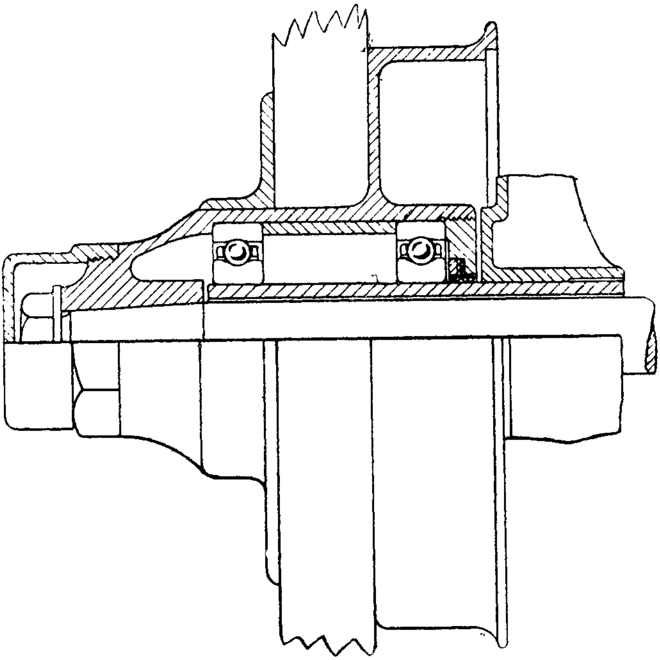

FIG. 27.

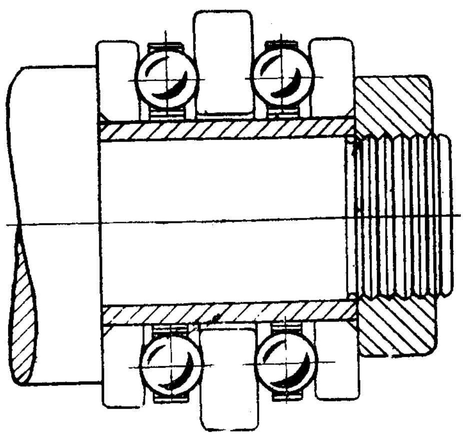

FIG. 28.

washers an easy sliding fit in the housing and clear of the shaft. There is also a clearance or slop between the two shoulders in the

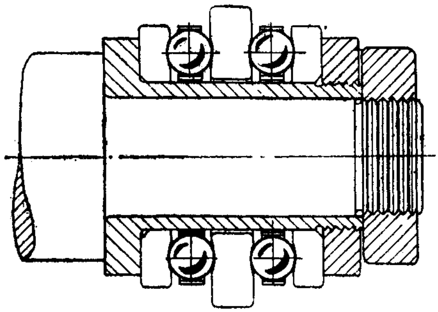

FIG. 29.

housing and the two shoulders on the shaft. When the thrust on the shaft is from left to right, as shown in Fig. 30, the washer A

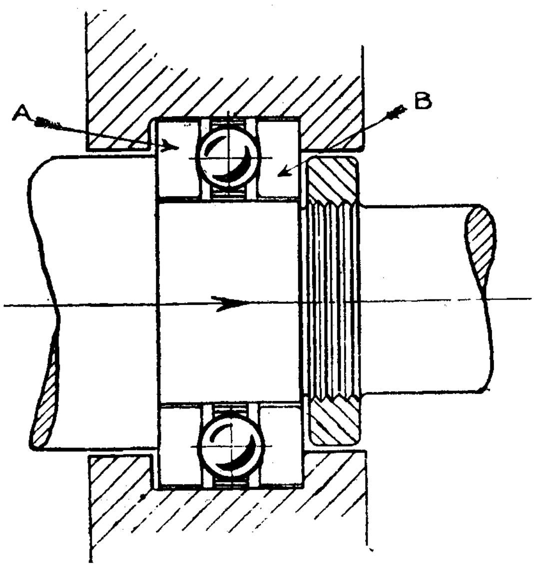

FIG. 30.

revolves with it, and the washer B is stationary against the fixed abutment in the housing. When the thrust of the shaft goes from right to left, as shown in Fig. 31, it moves along in that direction, taking the washers along with it, so that the washer B now revolves with the shaft and the washer A is stationary against the opposite fixed abutment in the housing. The drawback to this arrangement is that it is liable to be badly adjusted owing to the fact that, in the ordinary sense of the word, it should be fixed out of adjustment or with slop between the ball races, and that the washers, which have to be held in position by the housing, have, at the same time, to revolve freely in it. It also permits of a considerable side shake which is objectionable, and the shock caused by the reversal

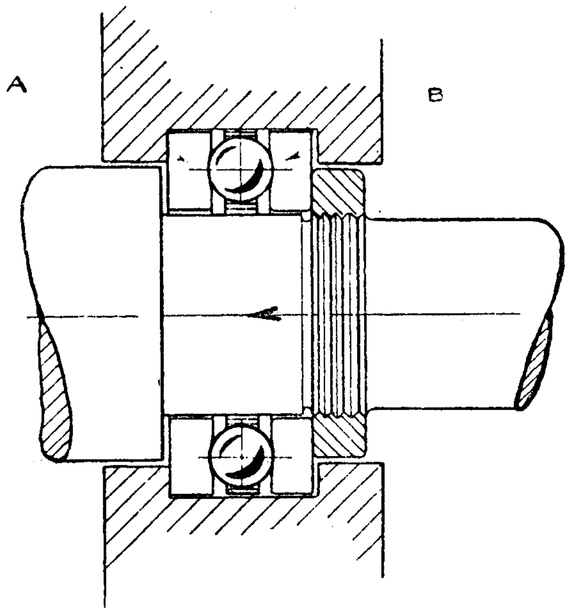

FIG. 31.

of the thrust tends to overload the bearing. Any dirt in this bearing would quickly cause the housing to wear away and the washers to become loose and out of concentricity.

The form of single row double thrust bearing shown in Fig. 32, although not correct in theory—in so much as the load is not carried across the ball in the direction of the load—is found, however, to work well in practice when used to locate a shaft against light intermittent loads in either direction. The ball races are so designed that the balls are quite free radially, and therefore cannot carry any journal load, but when side thrust comes upon the inner race from right to left the balls are caught between the races at A and B, there being a few thousandths of an inch clearance between the races at C and D. When the thrust reverses in the direction from left to right the load is taken across the races at C and D, the balls then having a clearance across A and B. In

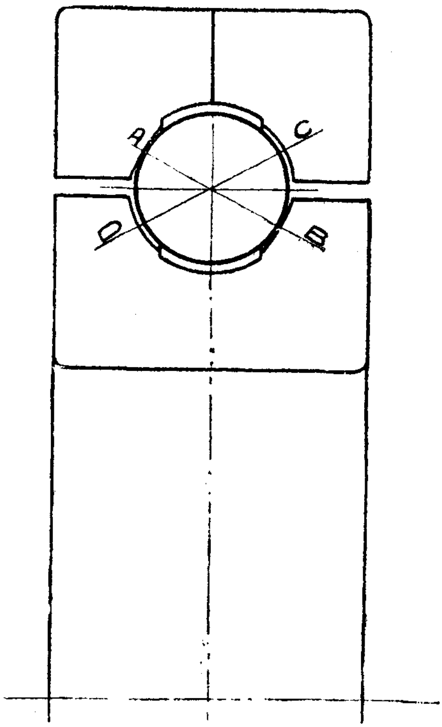

FIG. 32.

the figure this clearance is shown of considerable dimension for diagrammatic purposes. It is actually only a few thousandths of an inch.

When designing any special form of ball race, nothing should be left on in the form of a boss or lug, nor should anything be taken off in the shape of a keyway which will interfere with the continuity of the race. Any alteration in the cross-section of a ball race is bad, as it is impossible to ensure its keeping true to shape after it is finished. For the same reason nothing should be ground off any part of a ball race after it is made, as the least thing will throw the ring out of truth. Very light sections should also be avoided, as they are so easily distorted, and, being glass hard, they are liable to crack through.

Coming now to the assembling of the bearings in the assembling shops, it is important that ball bearings should not be driven into position through the balls and races, that is to say, when mounting the inner race on a shaft, the outer race should not be touched. Noisy bearings are generally caused by the indentations made in the races by the balls when the bearings have been forced into position through the medium of the balls. The same remarks apply, of course, to dismantling the bearings. Great care should be taken not to expose the bearings to dirt or moisture. It is unfortunately a not infrequent thing to observe ball bearings taken from the boxes in which they are packed and allowed to lie about on a dirty bench or even on the floor. Owing to the grease with which they are covered, they very easily pick up any dirt which may be about, and when once this gets into the ball races it is not an easy matter to get rid of it. The bearings should be kept in their boxes until the very last minute, should be assembled directly, covered as much as possible with grease, and boxed in so that dirt cannot settle upon them.

An excessive amount of wear in the bearings of gear boxes is frequently met with, and this has in some cases been directly traced to emery being left in the boxes from the operation of grinding in the gears in position. Slave races are, of course, used for this purpose, and are removed from the boxes when the grinding is finished. The boxes are then carefully washed in a number of different baths of paraffin, and then allowed to drain and dry. This, nevertheless, is not sufficient, as, however carefully it is done, emery is still found clinging to the walls of the box and in the pockets and corners. They should be finally washed by a good force of water from a hose pipe to drive away all grains that are left. A good method is, after washing, to paint the inside of the box as well with a suitable varnish, which fixes any grains of emery that have been left, and prevents their eventually getting into the bearings. This will also fix flakes of aluminium which come from the castings. Small chips from the gear wheels getting into the bearings also tend to lap out the races and balls, and this should be prevented by protecting them with a thin steel disc or washer. There are two ways of doing this. One is to have the washer mounted in and forming part of the bearing itself, and the other is to have a separate steel washer mounted on the shaft or in the housing, as shown in Fig. 33. The former is undoubtedly the neater of the two methods, but it is more expensive, and has the disadvantage that other bearings of the same size on a different part of the car are not interchangeable with it.

The proper inclusion of oil and grease, and exclusion of dirt and moisture, is now by far the most important point to be carefully watched in connection with ball bearings, and too much stress cannot be laid upon the need for every attention being given

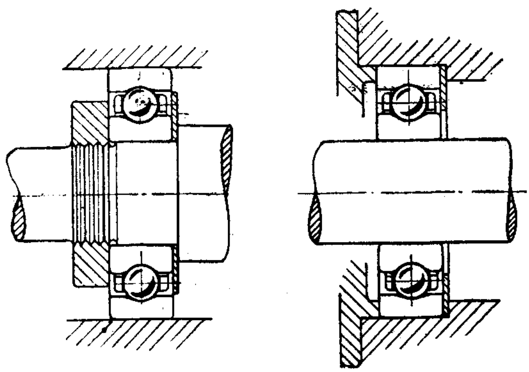

FIG. 33.

to the details of mounting in this connection. The method employed by the Wolseley Company is shown in Fig. 34. This consists of a split gunmetal disc which is held together by means of a spring so that it hugs the shaft. Two flanges on the disc are made an easy fit on either side of a cover formed in the end of the housing; this retains the oil and prevents the possibility of dirt and moisture getting in. Another arrangement devised by Mr. Barty, the engineer to the F.I.A.T. Motor Cab Company, is shown in Fig. 35; it consists of white metal segments which are cut so as to overlap one another. The exterior surface is formed in the shape of a V, round which a coiled spring is sprung. The washer is inserted in a grooved channel in the cover, and the spring acting in the V not only hugs the washer against the shaft, but tends to expand the parts of this washer against the sides of the channel in the housing. Both these methods have now been in service a considerable time and have worked very satisfactorily.

With front hubs something more than this is required. The usual method of washing down the wheels of a car is either by a hose or buckets of water, and the force of the water is sufficient to drive its way past the ordinary leather washers. A guard such as is shown in Figs. 24 and 25 is also necessary to protect the joint from any sudden rush of water, all that can get past this being in the form of a trickle down the outside of the hub. The ordinary form of leather or felt washer, or the rings just mentioned, are

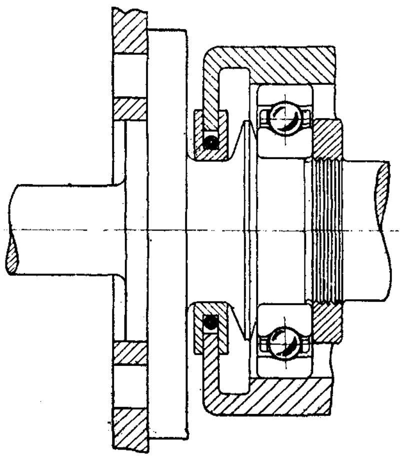

FIG. 34.

then sufficient to prevent the water getting into the ball races. With hubs it should be possible to fill the whole of the space which is not occupied by the bearings with grease, and not only to quite fill the space, but also to force in from the outside end of the hub a little more, so that it actually starts to exude from the joint at the inner end. If this is carefully attended to it should be quite impossible for water to find its way into the hub. A quite considerable amount of trouble is caused at the present time through want of sufficient attention to this detail. Another reason why hubs should be filled up with grease is that it prevents condensation from the air on the ball races owing to differences of temperature, as there can then be no air present.

Some designers of motor cars may complain that many of these suggestions are useless refinements and not necessary, but on the other hand, those who are in more personal touch with repair shops, and more particularly those whose experience has been gained in keeping a large fleet of cars in full commission, will

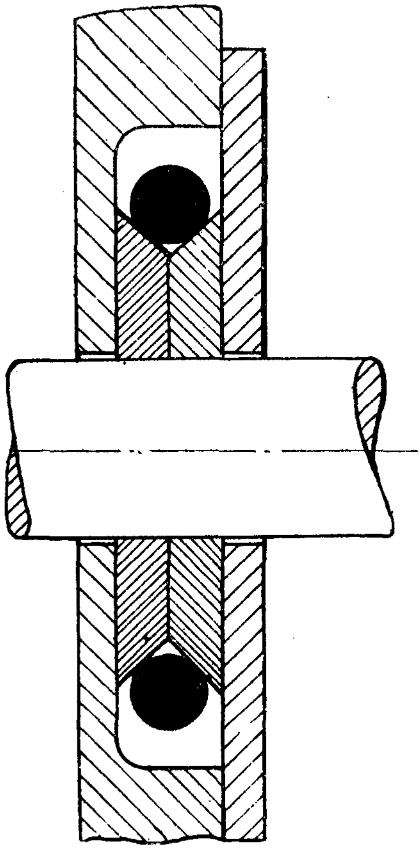

Fig. 35.

agree that they are necessary. This experience must be of value in designing commercial vehicles where the working conditions are so much more severe and reliability is of so much importance.

As regards the running of ball bearings, provided that they have been properly mounted and safeguarded from dirt and moisture, the only question that remains is that of lubrication. It may seem out of place to make any mention of this, because a ball bearing is supposed to be nearly frictionless, and therefore to require no lubrication at all. The only sliding friction that can be present is between the balls and the separating cage, and as the load between these is practically negligible, lubrication should be a small matter. Were lubrication of the bearing necessary, it would be an extremely difficult thing to provide, as it would necessitate the provision of a medium between the balls and the races at their point of contact, and it is very doubtful whether any lubricant could be found—with the possible exception of graphite—which could be retained between these surfaces owing to the great pressure which must be present at what is, theoretically, an infinitely small contact point. A lubricant, however, must be present to keep the hard steel surfaces in a greasy condition. The grinding process leaves the surfaces of the steel entirely unprotected and very susceptible to external chemical influences. Water and acid in every shape must be excluded, and the greases employed should be such as do not readily absorb moisture from the atmosphere. For this reason only a pure mineral grease should be used. Many of the so-called cup greases and solidified oils contain saponifiable matter which readily absorbs moisture from the ordinary atmosphere. Not only does the presence of a small amount of moisture in the grease tend to create rusting in the ordinary sense of the word, but it also produces a different form of oxidisation which it is difficult to account for. This effect—samples of which are on the table—often looks similar to that obtained by dilute acid, and goes on even when the bearings are wrapped in oiled paper and kept in dry stores. It produces an eating away of the surface of the steel around the point of contact of the ball on the ball race, and must play some part in destroying the surface of the ball bearings when they are working under ordinary conditions. The etching effect under these conditions is so fine as to be difficult to detect, but it must account for some of the wear which takes place in bearings where the grease used has absorbed moisture. This effect does not happen with pure mineral grease. It is also well to avoid the use of some of the so-called solidified oils, as they not only contain saponifiable matter, but also an added body which is introduced to form a thickening. This has been found to consist of such material as French chalk, and in one case where ball bearings were found to wear out rapidly, the grease was analysed and found to contain as much as 75 per cent of this material. Needless to say it acts as a fine lapping compound and soon wears away both the balls and races.

Graphite is another material which is to be avoided unless it can be absolutely proved to be of the very finest. A large number of the graphite greases on the market, which are no doubt perfectly suitable for other purposes, are extremely injurious to ball bearings, quickly lapping down the races and balls.

As already pointed out, care should be taken to see that not only the ball bearings themselves, but the housings surrounding them, are crammed full of suitable grease, so filled in that it exudes from the joints and helps to make them dirt and air tight. Examples of badly worn and damaged races are shown upon the table, with a description of the cause.

The ideal is always impossible to attain, but it should be approached as nearly as possible. It is a well-proved fact that a ball bearing running under the nearest ideal conditions—say on an electric motor in a clean shop—will run for years, day and night, at high speeds without showing any signs of wear whatever. Such conditions it is impossible to obtain in motor car practice, but the aim should be to approach this as nearly as possible.

Accidents must happen, and some ball bearings will fail and require renewal. This latter should be done as promptly as possible. A point that will assist this to an enormous extent is standardisation and the reduction of the number of types and sizes to a minimum. When a bearing that is not a standard size and type fails and a new one has to be made specially, the vexatious delay that results cannot but have a bad effect on the type of car that requires it. It is recognised throughout the world that all component parts of machinery should be as far as possible of standard sizes and interchangeable. It is difficult to do this with certain parts, but it would appear as though ball bearings lend themselves very well to this ideal. This would tend greatly to simplify the work of making parts interchangeable and of reducing the stocks to be carried not only in the manufacturing shops. but also in repair works and garages.

In conclusion, my thanks are due to Messrs. C. A. Barrett, W. B. Mair and A. Haskins for valuable assistance, and to the Hoffmann Manufacturing Company, Limited, for the loan of models and samples and for the preparation of the diagrams.

THE DISCUSSION.

Mr. D. J. SMITH, in opening the discussion, said: I think that the importance of this paper will be realised when it is remembered that 90 per cent of the bearings of the modern motor car are ball bearings. I agree with the author that the modern ball bearing is practically indestructible if it is properly fitted, properly lubricated and properly treated, but all those who repair motors know very well that the ideal method of fitting and ideal treatment are very seldom met with. There are one or two subjects that I think the author has hardly dealt with sufficiently in the paper, and among these are the effect of moisture on ball bearings. On p. 232 he points out that moisture is a very great danger to the life of ball bearings, but I hardly think it is realised that moisture can still exert a most harmful effect even in a bearing that is extremely well lubricated by being run in an oil bath; if any water is present, the wear is enormous and very rapid, and of this I have actual examples in my possession.

There is one point which the author has not mentioned at all, and that is that in any position where ball bearings are liable to come into contact with water, the admixture of 10 per cent of lard oil to the other lubricant practically stops the ill effects of the water. I do not know how that has been discovered, but the effect is certain, and perhaps the author will be able to tell us the reason. I, myself, think that the lard oil forms an impervious layer round the balls, and it is thus impossible for the water to attack them, and the water does not become emulsified with the lard oil as it does with lubricating oil.

Mr. MAX LAWRENCE: May I ask where the ball bearing referred to above was used?

Mr. D. J. SMITH: It was the outer bearing of a rear axle, and although, when taken out, it was well lubricated and showed no signs of having been without lubrication, yet it was evident that water had been able to enter the bearing in washing down the car.

Mr. MAX LAWRENCE: Was it originally a good fit?

Mr. D. J. SMITH: Yes, it was quite correct. In fact, the bearing had to be punched out under considerable pressure, and when the new bearing was fitted the shaft was in perfect alinement.

A MEMBER: Was the outer race revolving?

Mr. D. J. SMITH: No, and the inner race was fixed on the shaft by a keyway.

A MEMBER: Was the alinement of the shaft perfect when the load was on or off?

Mr. D. J. SMITH: When the load was on. In this particular axle there was a tension rod right underneath by which the tension could be adjusted so that the shafts were quite in alinement.

This is not an isolated case, and I have seen at least a dozen quite as bad.

There is another matter dealt with in the paper which is a very important one and one which affects the whole of the motor car industry, namely, the question of lubrication. I am afraid that many lubricating oils are treated with some substance—I do not know enough about the lubricating oil trade to say what it is—but there is some substance added to the lubricating oil which undoubtedly attacks ball bearings badly. Quite recently I had a sample sent to me of a composition of borax powder which had about the same lubricating qualities as emery, the object of which was stated, in the accompanying letter, to be to increase the consistency of lubricating oil. How much was to be put in the oil I do not know, but of course any degree of consistency could be obtained by adding enough. I sent a sample to Messrs. Hoffmann, and their report on it was just as damaging as I had expected. I should be interested to know how many oils are treated in this way, although I think the motor trade is largely to blame for this, because if an oil traveller comes along and offers a sample of oil at 1*s*. 9*d*. and the next man offers it at 1*s*. 6*d*. the trade always, or nearly always, takes the cheaper, and they must be prepared for trouble in consequence. In a letter which I contributed to the technical press a little while ago I suggested that the Society of Motor Manufacturers and Traders should adopt a standard to which all motor lubricating oils should conform.

Very few builders of cars now make their own ball bearings, but they buy them from one or two large manufacturers, so that we can generally assume the quality of the material and the workmanship to be excellent. I think the chief trouble is the manner in which they are fitted.

(Mr. D. J. Smith.)

The method in which ball bearings are fitted to a great many front hubs, shown by the author as an example to avoid, cannot be too strongly condemned. Undoubtedly in all these hubs a thrust bearing is required. The damage that can be wrought by having bearings so fitted without thrust bearings is very great. I have a large number of discarded bearings in my possession, among which I may specially mention one which was fitted on the front hub of a motor cab without any thrust bearing as dozens of them still are, and which, when taken out, showed very little wear up and down in the ring but a large amount laterally due to the thrust which the bearing was never intended to take. In fact, the thrust became so great that it burst the outer ring which had not been revolving in its housing. In another bearing taken from the same position but fitted with a thrust bearing, I found absolutely no sideplay whatever, yet the bearing had run a considerably greater distance than the previous one. It is very disappointing to find so many makers even of modern cars fitting their bearings on front wheels without thrust races. If this paper will only impress on designers of cars the necessity for treating ball bearings properly, which they do not seem to have appreciated up to now, especially in relation to hubs, it will have been exceedingly valuable.

Dr. H. S. Hele-Shaw: During the last five years I have been doing work, the success of which has depended almost entirely upon the effective use of ball and roller bearings, and I can say that no advance in engineering detail has been more remarkable than the gradual and successful progress towards the production of ball and roller bearings, particularly the former, which has fulfilled the required conditions of working under considerable loads, and being practically indestructible. I remember seeing, when a boy, an old roller bearing that had been put on a water wheel at the suggestion of my great-grandfather, and which replaced the oak shafting, working in a cast iron bearing. This attempt must have been considerably more than fifty years ago, and, no doubt, there were many others, but it is certain that it was only quite recently that the progress in our knowledge of materials has enabled successful ball and roller bearings to be made. Now one of the things that we have learnt more about in recent years is the nature of the structure of metals, thanks to the progress of microphotography, and we can understand from this, and the sort of diagrams that Dr. Rosenhain has rendered us familiar with, why it is that only a really first class kind of steel, properly hardened, is suitable to withstand the enormous pressures of rolling contact in a ball bearing. It is true that for many years ball bearings have been quite satisfactory under the comparatively low pressures employed in bicycles, but I am, of course, talking now of the severe loads in many modern applications. One thing that we have learnt, and it is emphasized in the paper, is the necessity of having a curved surface, and not a flat surface, for the ball to roll upon. Of course the latter is theoretically correct from the point of rolling, but the pressure at the point of contact is so enormous that we can easily see why the crystalline structure of the metal in contact will only enable a small load to be borne in such a case, owing to the great pressure per sq. in. Another point referred to in the paper which practice has taught us, but which theory would never have done, is the entire disappearance of very light cages. It is not very long since it was generally considered that a very light cage was quite enough to keep the balls supported and in their respective places. We now know that when the load actually comes on the bearing the forces transmitted by the cage may be very considerable, and that the cage requires to be just as carefully made and proportioned to sustain these loads as the rest of the bearing.

Although ball and roller bearings are tending to general standardisation, there are, of course, certain differences in the bearings adopted by different makers. Thus the Skefko ball bearing, of which a figure is shown, has certain important advantages, and there is no doubt that the Skefko bearing will, to a certain extent, stand a side thrust, but I have also found in practice that the Hoffmann bearings actually work very well with a small amount of side thrust, although of course they are not supposed to take any side thrust at all. There is again a very interesting question which is raised by the peculiar design of the Skefko bearing as against the Hoffmann, in which the bearing area on the inside and outside race is equalized in the former; on the other hand, it does not appear why the additional bearing area on the Hoffmann, with the race curved in cross-section, is not an advantage. There are, of course, many other excellent bearings on the market, and the competition is now so keen that the user can rely with confidence on the results as long as he keeps to the conditions laid down by the makers. It is only fair to the makers to say that a number of the failures of ball bearings are due to the

(Dr. H. S. Hele-Shaw.)
user wilfully violating the conditions of load and speed given, in order to economise by using a smaller size of bearing than should properly be adopted.

One thing is remarkable, and that is the great difference in the design of roller bearings. On the one hand very long rollers are now being used satisfactorily under heavy loads. Messrs. Hoffmann are adopting a very short type of roller, and this has given excellent results under the trying conditions in which I have applied it. We have advanced much more near to finality in regard to ball bearings than roller bearings, and most engineers have in times past experienced such trouble with the latter that it is with difficulty that they can be persuaded to use them, and, because of their present expense, it will be some time before the use of roller bearings becomes general.

Mr. Max Lawrence: I am very much interested in the paper, but I have come to ask for a little more information. We are shown in the paper methods in which we must not house bearings, and I should certainly think that it is obvious that some of these illustrations do not show the right way if a bearing is required to wear for any length of time. I should like to ask the author if he can give us some figures in support of some of the criticisms he has made as to the shape of the path on which the balls roll. He compares them and says that the best form of path is made to a radius of 5 per cent to 10 per cent larger than that of the ball. Could he tell us what the load per in. diameter per ball should be for the steels that we are using at the present time; I mean carbonising steels, fairly deeply carbonised and hardened in the ordinary way. He says that cupping has a marked effect, but I should like to know what the effect is in figures. I have made a large number of ball bearings in which there was no cupping at all, and, theoretically, of course, presuming that an incompressible steel has been obtained, both in the race and the ball, and that it is not loaded beyond a certain amount, and that it is properly hardened, there really ought to be no difference between the 5 per cent or 10 per cent cupped race and the flat race, and I should like to know if experiments have been carried out and what their results have been.

A point upon which I entirely disagree with the author, if I understand him aright, is the question of tolerances and clearances desirable and necessary between a bearing and its housing mentioned on page 209. He recommends tolerances of half a thousandth of an inch upon the shaft and upon the bearing, and he makes some remarks as to overload. Most of us remember that a shaft and a hole of the same size will not marry, and that there must be a difference of size, and that if a slightly tapered plug is once entered into a hole, both being truly an inch in diameter, they will seize, and if they were chemically clean I doubt if they could be got apart again. Then the author goes on to explain the proper fit between a ball and its bearing, and to show how the maker has already provided the proper clearances between the two races and the ball; thus if he marries his male shaft on the female race with half a thousandth overplus, perhaps he will get a difference on the average, not by having a maximum marrying a minimum, but by having a minimum marrying a maximum. Certainly, unless the housings yield—for the shaft will not yield—with half a thousandth and unyielding surfaces such as hardened steel races and balls, I should think that the overload of a ton on a circular race would be far exceeded. The amount of crushing that is necessary and desirable in marrying ball bearings, in my opinion, is far greater than can be measured, and we use a slang term in the works which is known as "feelth." In order to work a grinding machine to the necessary degree of accuracy to manufacture ball bearings, you must have this new sense. Some men never have it and they do not know what I am talking about, but a grinder, that is, a man who can manufacture with this kind of accuracy, must have this sense. The bearings must be let on the shafts and into the housings with a certain amount of physical effort which can only be ascertained by experience.

I am very much interested to learn the author's explanation of the creeping between a slack ball bearing and the shaft upon which it is housed. It comes to me as quite a new explanation of that phenomenon. I have always found that if the shaft upon which the ball bearing is housed is a hardened shaft, a good deal more slackness is permissible than if the shaft is made of mild steel and not hardened. I had a very notable instance of this, because, through a regrettable accident, quite a number of shafts in motor cars for which I was responsible were made too small in one dimension—I will not say whether it was a drawing office mistake or a constructional mistake—but anyway the result was that a certain amount had to be turned off and a collar pressed upon the shaft. The section of these collars was $\frac{1}{4}$ in. by $\frac{1}{4}$ in.,

(Mr. Max Lawrence.)
and the width of the ball bearing was slightly larger, and I had the misfortune to see all these shafts back again with the washer made like a U leather so that I thought it was made of tin and did not know how it got there. The defect was that the race had cut its way into the soft collar, and as I had personally inspected a number of these shafts and knew that the race was a correct and proper fit upon the soft washer, and it had cut it out absolutely in a U shape to the depth of $\frac{1}{4}$ in., it was a most extraordinary experience, and therefore I think there must be some other forces at work than those suggested by the author as the cause of the eating of a race into its shaft. Dr. Hele-Shaw just gave me an idea of what the Skefko ball bearing was likely to do, and the author criticises it by saying that it is not useful in that the shape of the path does not give any advantage from the extra number of balls that can be used under the load. Most shafts, while transmitting power, and if housed in ball bearings, have a certain amount of deflection when in use. Take the shaft in a gear box which is subject to the thrust from the gear into which it meshes. Certainly in some shafts there is a certain amount of deflection in the centre, and that would bring about the undesirable circumstance, mentioned by the author, of the shaft not being in line. Of course, it is entirely a question of trial and error, whether it is better to employ a bearing such as the Skefko which will allow this bending of the shaft to take place without bringing extra load on the bearings or whether the surface of the bearing should be made so large as to cause no anxiety about the deflection of the shaft. The slope of the ball at the edge or at the middle on the lubricated surface does not matter much, in my opinion. If the author could give us some figures relative to the duty that can be allowed on flat surfaces and curved surfaces, I think it would help us to solve the problem ourselves.

Mr. B. W. SHILSON: The author gives, on pages 210 and 211, two methods which are employed for securing and adjusting a thrust bearing. Fig. 14 is given as an extremely bad example, and the author suggests using the form shown in Fig. 15, with a sleeve having a solid collar so that a true surface may be obtained when adjustment is made; in Fig. 29, page 225, however, a double-thrust type of bearing is referred to as the only correct or perfect form, but here again there is shown that same nut for backing up, and I should like to know in what respect that nut is better than the nut shown in Fig. 14. In Figs. 30 and 31 another form of thrust bearing is shown which is fitted into a housing (apparently that housing is split, although the diagram does not show it), and this bearing is somewhat harshly criticised because of its having to be an easy sliding fit; when the thrust is transferred from one side to the other, a very serious shock is imposed on the bearing which tends to overload it. In Fig. 32 there is another case which, to my mind, is exactly similar; here we are told that when the load is in the direction DC there is "a clearance of a few thousandths of an inch" across AB, and that when the thrust is taken in the other direction that clearance also changes over. To my mind that "few thousandths" is rather excessive and would not be necessary in the bearing shown in Fig. 31. If the slack there is harmful, I cannot see why it is less harmful in the type of bearing given in Fig. 32. I consider that bearing is very similar, from the thrust point of view, to the cycle type of bearing which is very harshly criticised in the earlier part of the paper, because the thrust is not carried straight across the ball, whereas in Fig. 31 the thrust is carried directly across the diameter. This bearing, Fig. 32, the author points out, is not correct in theory but works very well in practice. In dealing with the bevel axle several interesting points arise. In Fig. 16 the bearing is mounted with a thrust on each side with packing washers; this, as will be admitted, is a perfectly legitimate way of setting up any mechanism which is liable to errors, but I have known a case of a number of 45 h.p. cars, in which the only bearings that held the differential were, one of the full ball type, $\frac{1}{2}$ in. wide, on the one side, and two of the same type on the large bevel wheel side, there being no thrust bearing whatever. Strange to say, this bearing has not given trouble. According to theory it should have given continuous trouble, but as a matter of fact the cases in which trouble has arisen can be counted on one hand. With regard to the other examples, the author points out that in Fig. 17 the adjustment pins should all be of exactly the same length, but I do not see the advantage of this as they are backed up with a screwed nut. The construction is quite wrong, since, as the author points out and condemns in another part of the paper, that screwed nut is bound to throw these pins out and to give them different relative lengths. The example in Fig. 18, which is strongly recommended, is rather interesting, because a double thrust bearing is shown on one

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side of the differential, and a double thrust bearing on the bevel shaft; as the author says, this is a machining operation which "can be accurately ascertained and is unalterable," but I cannot see that the exact distances are so easily obtained. I think a very much better method of construction would be to use these double thrusts as shown, plus the packing washers, so that adjustment could be made in the shaft for setting these bevels up, because, as all makers of bevel axles know, it is impossible—at all events so far as I know—to cut bevel gears, and set them up so that they will run quietly. They always have to be adjusted to the warping of the gears, and, therefore, even if such machining accuracies were obtainable, I do not think it would be of much advantage. In Fig. 18 there are five sources of error altogether. Working to the left from the centre of the case, there is first the face on the case itself; secondly, there are two faces on the packing washer; thirdly, there is the distance from the centre to the top of the case; fourthly, the distance from the bottom face to the face inside the housing; and fifthly, the faces on the packing washers again. I think, therefore, that if these were machined with allowances made for the employment of packing washers a very much more satisfactory job would result. In Fig. 19 is shown a gear box construction, which I know is adopted to a very large extent. The lay shaft is here positioned up to the covers, and, as the author points out, this is apt to be very harmful, because of the dimensions being wrong and the bearings being pinched up, but there are fewer sources of error from this than there are in the bevel axle just referred to, and therefore there ought to be less chance of trouble. I think, however, that a better way still would be to leave one cover as it stands, and put a securing ring on the inside of the bearing, leaving the other perfectly free to float in the direction of the axis of the shaft. This would not cause very much thrust to come on that bearing, and I think that the same principle could be applied to the first motion shaft, because, as the author points out, such a type of bearing as this is perfectly suitable for the front wheel bearings of cars of moderate size. I should like to know, in passing, what the author considers as the dividing line between a car of "moderate size" and one of greater power. In Fig. 19 the first motion shaft is shown running in a plain bearing, but when we come to the "ideal" box, Fig. 20, it is shown running in a ball bearing; this is a case where of all others one would wish to put a ball bearing if possible, so that the author has leaned a little in his own direction, perhaps, when he shows a plain bearing in the one case and a ball bearing in the other. I do not think it is necessary to go to the expense of putting double thrust bearings on the first motion shaft, but by locking one of the journal bearings, the shaft could then take care of itself. In Fig. 20 the lay shaft is shown with two thrust buttons to take up the thrust, but the same trouble is likely to arise there as is referred to in connection with the worm axle, namely, the elongation of the shaft due to heat caused by the work done in the box. This taking up of the thrust by means of buttons requires to be done as carefully as in some of those other cases which have been condemned, and, as I previously remarked, if one of these bearings were locked I think it would more effectively overcome the trouble and very effectively take off any thrust from that shaft. The author points out that it is very desirable to keep dirt of all sorts from getting into the thrust bearing, particularly in the case of front hubs. I should like to know whether it is desirable to run the ball bearings of a worm shaft in grease or to let them be lubricated from the axle, the oil from which is liable to contain small particles of metal in suspension. Would the grease type of lubrication be sufficient for these bearings; that is, would the author recommend the bearing being practically shut off and filled up in the manner which he recommends for the front hubs. In dealing with front hubs, the author recommends the form shown in Fig. 24 as being preferable to that shown in Fig. 23. I think that if the outer flange (that is the flange securing the outer ball race) shown in Fig. 23 were removed, so that the bearing was able to move lengthwise, then, as a form of bearing housing, it would be preferable to that shown in Fig. 24, because the outer bearing is made somewhat smaller than the inner bearing, and is probably quite sufficient; that is, both are probably stressed to the same point because one is further from the load than the other. Assuming that they are both equally loaded, it is better to fix the larger bearing as I have suggested for Fig. 23 than the smaller one, as the author does in Fig. 24, because the larger one is better able to take the load, which would give a smaller percentage increase than in the case when it is taken on the small bearing. Fig. 24 is also interesting from another point of view. The bottom bearing for the swivel pin is in a bush, which appears

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in the drawing, to be clamped up with a nut, this clamping being transmitted through the thrust bearing to the axle body. I should have thought that the author would have recommended for this a method which would be less liable to overload the bearing, especially as in some cases he has taken such very great precautions. In this thrust bearing, however, we have quite different conditions; it is liable to be wrongly adjusted unless the packing washers are made very carefully and the chances of water getting to that bearing are also very much greater.

Mr. J. D. Roots: I agree with the dictum ascribed to Professor Stribeck on page 196, that is to say, that the ball contacts should be "in a line at right angles to the axis of rotation of the bearing and parallel with the load," and I agree particularly with the latter part of the statement. It seems to me that in the case of an ordinary ring bearing which gets a slight amount of side thrust upon it, it is the wedging action of the alternate twist given to the ball at the point of contact with the race that abrades, or tears out minute pieces of metal from, either the ball or the race, whichever may be the softer or more friable.

I see there is a reference to early motor cars on page 198, and I think it might interest the members of this Institution to know something of the difficulties which were encountered in making the first and second motor cars, and in making the first commercial motor vehicle in Great Britain. At the very outset I sent round a design similar to that now in use, to all the chief makers of ball bearings, to endeavour to get a bearing of this design, and everyone advised me not to attempt to use that type of bearing, and advocated the ordinary cycle form, but with a curve on two sides or two flats and a curve on the other side, with means for adjusting. I did my utmost for the first two cars to get a form of bearing of the present type, but the manufacturers were so conservative they would not make them.

For the first commercial motor or van I made up, in chilled cast iron, two races of the present type and purchased the balls for them. Of course, I knew very well that it could not last very long, but at the time I was dealing with an engine of relatively small power, and I was well aware that with that small amount of power I must do my best not to fritter it away in friction between the engine and the wheels. I was also well aware that the bearing would require changing from time to time. I attempted to harden steel races myself, but I found that they warped very badly, and even those which I got from firms who were at that time manufacturing cycle bearings were out of truth as delivered to me, a fact which I found out by spinning them on lathe centres. Eventually I used ordinary cycle type of bearings of larger sizes on the first and second cars which I made. Owing to the chilled cast iron bearings wearing out so rapidly, I used, in some cases, cast steel without any attempt to further harden it, as I found considerable difficulty in hardening the rings evenly. On the van there was a twin cylinder engine $5\frac{1}{4}$ in. by 6 in., and, as we had contracted to run this van for six months for Messrs. Peek, Frean & Co., for the delivery of biscuits, carrying a ton 30 miles each day, I had undertaken a difficult task for the first commercial vehicle. On this van, in using cast iron ball-race rings on all intermediate shafting, I had to replace the bearings about once a month, which was not so bad when it is realised that a gun metal bearing sometimes had to be changed as often as that when subjecting the earlier vehicles to prolonged hard wear. When the chilled cast iron wore away, it wore away in a rather peculiar manner. I put some pieces under a microscope, and found that it had the appearance of thin flakes or scales. I think "friable" is the best word by which to describe this behaviour of cast iron.

Turning to the diagram, Fig. 13, which shows an ordinary single ring ball-race and ball, the curve of the race with the ordinary ball bearing is struck from a greater radius than that of the ball, and as soon as the slightest thrust comes upon it, which must inevitably occur occasionally, a wedging action is set up on the opposite sides of each ball-race, as there is necessarily some tendency to twist the ball. While the present form of single ball bearing ring reduces, as compared with the old cycle bearing, the spinning or wedging action tending to abrade portions of the softer metal, it still takes place. If the ball is softer than the race, then portions of the ball are taken out; on the other hand, if the race is softer than the ball, portions of the race are taken out. The difference really between the one form of construction and the other is that the distance between the opposed twisting and spinning points is greater in the old cycle bearing than in the new single bearing ring.

Referring to the diagram of my bearing, Fig. 36, it will be seen that this has a race that is absolutely circular in section, and has a diameter only just slightly greater, say between one five-

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hundredth and one thousandth of an inch, than the diameter of the ball.

Whenever a thrust load is added to the journal load the ball takes up an intermediate position and adapts itself to the component of the two forces, journal and thrust. For example, if the journal and thrust loads are equal, then the angle made by the race contact lines will be 45°, i.e., midway between the positions of the ball in a bearing with journal load only and one with thrust load only.

A ball ring of this description with ball-race of circular section takes both journal and thrust perfectly, and I believe that it will last longer because there is none of that wedging action which is bound to result even with the best present forms of construction of ball bearing. With a perfectly circular race, only slightly larger in diameter than the ball, it does not matter in which direction the bearing gets the load, whether journal or thrust,

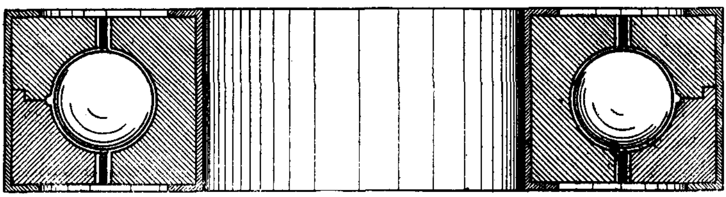

FIG. 36.

and as the component of journal load and thrust varies in direction, the ball must always take up a position corresponding thereto.

I think that the D. W. F. firm have carried out tests in Germany on this class of ball race, but I do not know the results. Fig. 32 in the paper is practically the same thing, so far as the circular form of race is concerned, in spite of the top and bottom grooves. Although apparently for double thrust only, without provision for taking journal and load, this model is capable of taking, and no doubt does take, the last named also. It has a diameter of race which is only just slightly larger than that of the ball. I had the honour of suggesting a similar bearing to Messrs. Hoffmann about two years ago, and I am glad to hear that it has been so successful.

Mr. P. ARMSTRONG: I have perused the paper by the author on the "Causes of Failure in Ball Bearings" and would like to emphasize and comment on a few points.

The principal points upon which I am at variance with the author are on pages 207 and 208. He suggests on page 207 that it is not necessary for thrust bearings to be fitted on the front wheels of moderate sized touring cars. My experience has been contrary to this. He further states that it is necessary for heavy touring cars and motor cabs to have thrust bearings fitted; that being so, I am at a loss to know why he should suggest that the thrust bearing is not necessary for lighter cars.

The best made and best designed journal type ball bearing will only allow an axial load of 10 per cent of its safety load, that is, an axial load of 100 lb. will be equivalent to 1,000 lb. radial load. I will take an example to further show the importance of thrust bearings, and the serious results that may follow their omission. The load on the front axle is, say, 1,000 lb., that is, 500 lb. for each journal and 250 lb. on each ball bearing; I have presumed there are two radial ball bearings in each journal, taking equal load.

The safety load for each ball bearing is, say, 1,000 lb., which is an ample margin; now, if a car is travelling round a curve at such a speed and radius that the centrifugal force will give an axial pressure of 150 lb., fully 100 lb. is taken on the outside wheel, and approximately the whole 100 lb. axial load is taken on the journal ball bearing which has the outer race fixed endwise in the housing. This bearing has now to take the radial load of 250 lb. and an axial load of 100 lb. This latter figure must be multiplied by 10 to obtain the equivalent radial load, which is 1,000 lb.; the total load on this ball bearing is now equivalent to 1,250 lb., which is obviously an overload of 250 lb., which, if continuous, would destroy the bearing; even the short period of loading has a bad effect on the hard steel races and balls, the surfaces now having been unduly stressed and so become less capable of withstanding wear. It will be found by simple calculation that the condition I have outlined exists, and where thrust bearings are not used, the radial bearing is occasionally overloaded by variations in the axial load caused by stones, swerves and wrenching due to curves, passing vehicles, etc.

These points deserve close consideration, and indicate the necessity for fitting all front wheel bearings with thrust bearings.

In Fig. 20, the author depicts a correctly arranged gear case. I agree with the mounting, except in one small detail, namely, the lay shaft, where lateral movement is prevented by means

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of hard steel buttons, which is questionable practice. It would be far better if a sleeve were fitted in one housing, or if the housing were stepped, so that the outer race and the internal flange of the sleeve or the step of the housing could be secured by a cap to prevent lateral movement, thereby allowing the thrust to be taken on one journal bearing instead of on buttons as shown. By this method there is no possibility of permanently loading the journals axially. This, perhaps, is a structural defect, but it should not be allowed to creep in when a correctly designed gear case is depicted.

I regret that the author has not devoted sufficient space to the suitability of the steels used in making both balls and races. Undoubtedly the materials are a crucial point in the efficient life of any type of ball bearing. At the present moment case hardened, or carbonised, material is largely used for ball races and balls. Undoubtedly the early collapse of carbonised ball bearings is due to the minute flaking of the carbonised balls and races.

Whilst watching some experiments conducted with various types and makes of ball bearings and balls we found the steel used by many of the manufacturers to be unsuitable. In testing these ball bearings we had to assume very heavy loads, as with ordinary loads the life of the bearing was far too prolonged, and it was under these extraordinary conditions that the defects in the less efficient materials became apparent.

One of the chief arguments, I believe, for the use of carbonised material, is its exceedingly tough centre, though the surface of the shell remains in a very hard state. Owing to the point contact we found that a very marked deflection takes place in these carbonised materials. Immediately beneath the carbonised surface and against the soft core we found that a slight movement was set up, which caused the two structures in the material to become dissociated. After a few of these balls, made of carbonised material, were run until the breaking down period began to show, the balls were split, and in nearly every instance they showed a complete detachment of the core from the hardened exterior.

On close inspection it was noticed that the adjoining surfaces were quite bright, showing undoubtedly that a slight movement had taken place. Obviously, if the experiment had been prolonged for but a few more revolutions, the balls would all have collapsed. Owing to small vibrations a change of axis of rotation is brought about, thereby allowing the entire surface of the ball to come at some time in contact with the race. What is required of the material for balls is even more necessary in that used for races, which are fixed and always expose the same surface to the element of load attack.

Although creeping cannot take place under the whole of the contacting surfaces of the race, the same detachment is set up, and flaking and minute cracking of the surface of the race quickly takes place, resulting in the speedy and progressive failure of the ball bearing. To overcome this defect, ball bearing manufacturers, not unnaturally, use a tough variety of carbon steel, and it is well known this can only be hardened to but a slight depth. The same defects are again noticeable here as in the carbonised material, but the steel being of combined form does not flake so readily on the deflection of the surface, but this deflection sets up greater friction and burning, due to the sliding movement caused by greater surfaces in contact. It is necessary that the steel used should be a steel that will harden completely through, so that the centre portion will be as hard as the exterior, yet sufficiently tough to withstand the great strains and shocks inherent to the uses of ball bearings, but which will not deflect sufficiently substantially to prevent rolling.

I desire to make a few observations on the remarks made regarding the deflection of balls and races under load. Dr. Hele-Shaw informs us of cases where this deflection is shown to an abnormal extent. I will point out that this deflection does not allow greater loads being taken when a contact of such dimensions is made between balls and races. It would substantially prevent rolling and set up sliding friction, as the diameter of the circle of contact on the outer edge would be less than at the centre, therefore the ball would endeavour to roll at different speeds along the line of contact, resulting in sliding friction in all but one point. The balls and races being under heavy load and of hard material, would quickly burn between the balls and the races where sliding is taking place; also the constant deflection of the balls and races to allow for this great area of contact would, in a relatively short time, set up fatigue in the metal, and the surface would begin to break away, that is, by flaking, and progressive failure of the bearing would be quickly set up.

Again, deflection of balls and races in a marked degree pre-

(Mr. P. Armstrong.)
vents their use in machine tools, owing to the resultant want of alinement, therefore increase in surface of contact due to deflection is not an advantage, but the direct opposite.

Allusion has been made by Dr. Hele-Shaw to the system of internal spherical grinding of the outer race, as adopted by the manufacturers of the Skefko ball bearing, and I would like to make a few observations upon this. The internally spherically ground outer race does not allow a greater load radially and axially as suggested. The load bearing capacity is smaller radially, owing to the greater radius of the race, when compared with the radius of the ball; the assumption that two or three balls carry the load against one in the single-row type is incorrect, as the load is taken by all the balls in the half of the bearing nearest to the load.

It is contended that the inner race of the Skefko type of bearing is constructed so as to allow the contacting points of the balls and races to be such that the tangents drawn from the contacting points are in absolute parallel. This is only true when the bearing is not under load, as deflection or play, however slight, would prevent the tangents being parallel, consequently spin would be present, which varies according to the radial load and the resultant deflection produced, the greater the load, the greater the spin.

Axial load causes exceedingly powerful wedging action, due to the extreme fineness of the wedge, thereby exerting relatively great bursting pressure; therefore axial load will set up an early collapse of the bearing.

Dr. Hele-Shaw: Do you say that there is more wedging action?

Mr. Armstrong: There is distinctly greater wedging action, which will be more easily appreciated when the curvilinear surfaces of the single-row type and the Skefko spherically ground surfaces are compared. In the case of the single-row type, it is general to make the race radius from 5 per cent to 10 per cent greater than the radius of the ball, but in the case of the spherical surface the curvilinear surface in this instance is fully 600 per cent or 700 per cent greater than the diameter of the ball, therefore nearer to the plane surface, which has no resistance axially. Obviously the strength of the bearing is measured by the weakest place.

Mr. A. Ludlow Clayden: The author has not mentioned the effect of speed although he mentions overload. Fig. 37 is a diagram which is plotted roughly from a series of ball-bearing makers' catalogues. The speeds are shown as abscissæ and the safe loads as ordinates. All these bearings are supposed to be approximately interchangeable. This shows the ideas of the different makers as to how the strength of their bearings varies with the speed.

A MEMBER: What is the speed?

MR. CLAYDEN: It starts at 500 and goes up to about 3,000 revs.

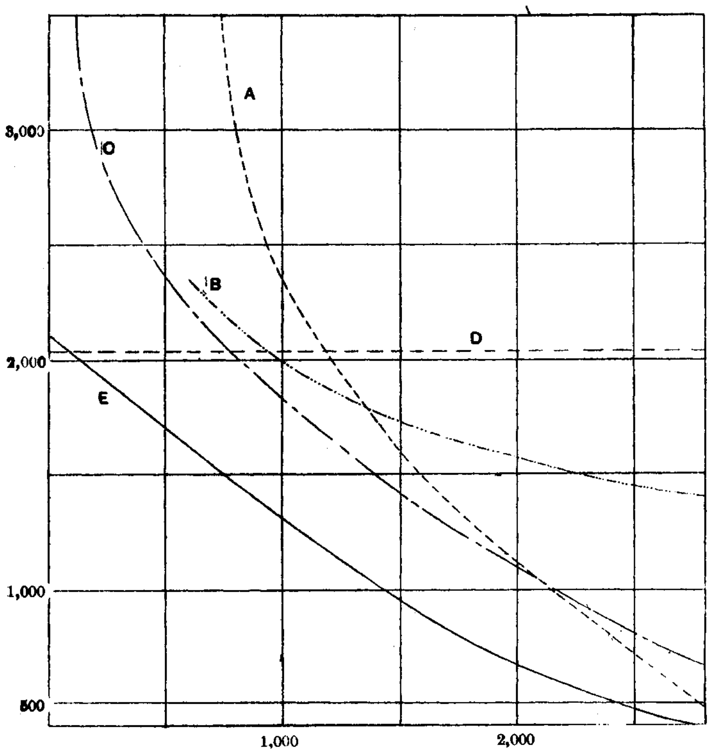

FIG. 37.

per minute. The horizontal line D is obviously ridiculous. Of course these are only catalogue recommendations, and I have not the least doubt that the maker, whose figures are represented by that straight line D, would not recommend the bearing as safe for all speeds. Still, there is such a great difference between the curves, that it seems to me that there must be some law which would determine how the strength varies with increase of

(Mr. Clayden.)
speed under the same load. It would be interesting if the author could give us any information on that point.

Prof. H. S. HELE-SHAW: When the diagram is published shall we have what A, B, C, D, etc., are? I do not see any description in the diagram.

The PRESIDENT: If the curves are obtained from figures published in the catalogues I do not see any objection.

Mr. CLAYDEN: Certainly. A is the Auto Machinery Company's single row journal bearing.

A MEMBER: I should like to point out that there is a great difference in the number of balls as well as a large number of types of that make.

The PRESIDENT: Perhaps when you get these figures you will give the exact particulars from the lists from which you have taken them.

Mr. CLAYDEN: Yes, certainly. The Hoffmann bearing is B, and the Skefko is C. D is the D. W. F., which is obviously not intended to mean anything. E is the Rhineland; it is practically the same curve as the Hoffmann, embodying much the same idea. I really took these bearings because they are all for the same size shaft and they are sufficiently near in widths to be taken as being interchangeable by the ordinary designer who wishes, for any reason, to change the make of bearings used.

Mr. L. H. HOUNSFIELD: The question I intended asking the author is one which Dr. Hele-Shaw commenced but did not finish to my liking, so perhaps he will pardon me if I run through some of his remarks again. He pointed out as an apparent defect in the double race self-alining Skefko bearing that each ball has to run round a conical surface, and as the axis of rotation of the ball is inclined to the axis of its enforced path, there must be a small amount of spinning due to the ball being continuously forced to take an unnatural track. I presume the author will object to this action, and I should like to know whether he considers it serious; if so, does he consider it serious in the bearing shown in Fig. 32, or in the ordinary thrust race where this defect is much more pronounced?

Prof. ARCHIBALD SHARP: I wish to make some remarks on a few points in connection with the theory of ball bearings, which I will put in writing (*a*). The bicycle type of ball bearing has

(*a*) See Communications, p. 269.

been referred to by the author of the Paper and by others during the discussion this evening as defective in design, and I should not like this meeting to disperse with the erroneous impression that this type of bearing is as black as it has been painted. It is a very good type indeed, and although I do not intend to suggest that it should be used indiscriminately by automobile engineers, yet if properly designed and constructed it has none of the vices with which it has been credited to-night (*a*).

Mr. J. F. RONCA: On page 233 the author refers to the use of graphite as a material to be avoided unless it can be absolutely proved to be of the very finest quality. I should like to ask him whether he has found any difference between artificial graphite and natural graphite for use in bearings. All artificial graphite is an electric furnace product, and some of it is a by-product in the manufacture of carborundum, and it may contain small quantities of carborundum and similar abrasive materials. Although the best graphites from the analyst's point of view are generally artificial graphites, I should like to know whether the author has found any of the artificial graphites to be particularly unsuitable on account of the presence of small quantities of powerful abrasives. I have examined some artificial graphites myself which contain very little, usually less than one-fifth of one per cent of ash, and yet this ash is so highly abrasive that it will scratch glass, hard steel, and similar materials very easily.

Mr. T. WATERHOUSE: There are two little points raised in the paper about Skefko ball bearings which I should like to answer. One point was the spin of the ball at the angle, dealt with by Prof. Sharp, but there is another point that he quite forgot, and that is in discussing this it is always to be remembered that there are two rows of balls taking the load all the time in the Skefko bearing, whilst in the other case there is only half the number of balls. The other point raised by the author is the question of iron or steel for cages, and this is a point upon which we have been spending a great deal of time and attention, and we now use a very soft iron, which we find gives precisely the same results as gunmetal, and we have been running them both interchangeably for a great many million revolutions and found the results as to abrasion of the balls identical. The advantage of the iron is, that you get a very much lighter cage and less internal

(*a*) See Communications, p. 269.

(Mr. T. Waterhouse.)

friction in the bearing itself. As regards the fitting of bigger bearings, I think the author's remarks are very valuable, and this paper should be a very useful one for anyone designing motor cars.

There is another point which I should like to raise, and that is, the application of ball bearings to front hubs, as shown in Fig. 23. Theoretically the author is right in Fig. 24, but Fig. 23 is quite the accepted practice in both England and France, and has been for many years, and if properly fitted there should be very little trouble with it, if any at all. The drawback in the instance shown by the author is, that by having his distance piece arranged as shown, any lateral movement must take place between the inner ring and the stub axle, and so wear the axle much more than the corresponding sliding wear between the outer ring and the hub. With the Skefko bearings we maintain that no thrust washer is required at all in the front hubs except on very heavy vehicles like motor lorries, when perhaps a washer is advisable, but for ordinary car practice they are quite good enough, as we have had a very large and satisfactory experience of this.

With regard to the question of material I will not speak about that now, but our practice with the Skefko bearing is to use the very finest steel, to harden it throughout and then temper it, and by that method we have equal expansion and contraction all the way through, because when a bearing is fully loaded there is no doubt that there is a series of waves formed right through the ring of the bearing as has been explained by Mr. Armstrong, and if that is properly divided between the two races, the bearing would last as long as the ball itself. This is really the limit to the wear of the bearing, and we have always found that it is the balls and not the races which fail first.

Mr. Douglas Leechman: I should like to refer to the first paragraph on page 216, where the author deals with the bearings carrying the bevel pinion. The author says: "Another design that is often met with is the mounting of the small pinion with a ball bearing outside it. This bearing then has to take most of the load, and, as there is not sufficient room to mount one large enough, it invariably gives trouble." It seems to me that if that bearing is the one which takes most of the load when it is there, then it is the bearing that is most wanted (or if it is not most wanted, it is at least very badly wanted), and it should not be done away with, but room should be made for a proper one. If the bearing is not there, the bevel pinion is overhung all the time. It is difficult enough to get it to work nicely with the other wheel under any circumstances, and it is most important that it should be supported as thoroughly as possible. I would submit that the right method is not to do away with the bearing because it gives trouble, but to make room for it so that it will wear without giving trouble.

Mr. R. W. A. Brewer: With regard to Fig. 25 and the remarks on page 221 of the paper, can the author tell us whether there has been much trouble caused by the screwed locating nut which is there shown on the innermost side, and which is the only means of attaching the wheel? Has this nut to his knowledge ever come adrift? There appear to me to be many weak points in this design.

Mr. A. Douglas Barty: For the past four years I have been responsible for the running of five hundred motor cabs in London, and these cabs have now done a total of 17,000,000 miles. Ball bearings are fitted to the front wheels, gear boxes, and back axles. I have therefore had some considerable experience as to their maintenance and fitting. The "creep" of both the outer and inner races, which the author mentioned, is very troublesome indeed. To get over this "creep" of the inner ring we have found it necessary to pull the ring up against a shoulder on the shaft with a nut and a fine thread screw, and, in addition, to grind the shaft to the same size as the hole with a limit of 0.0003 in. above the size of the hole. The outer race is more difficult to hold for the reason that it is necessary to allow this race to slide in an endwise direction, and it is, for this reason, impossible to prevent it creeping, even when a very fine limit of fit is made of it.

With regard to the front hubs, as shown in the paper, Figs. 23 and 28, I have tried these but have not found them to be a success. It is necessary, at least for cab work, to have thrust bearings fitted. A cab, as most of you well know, has to turn round in 25 ft. This, in addition to the many turns the driver has to make in traffic and the zig-zag course he has to take through the by-streets to avoid the traffic, throws a lateral load upon the bearings of the front wheels, which those of the journal pattern are quite unable to bear.

It would be interesting to know whether any experiments have been made with the ordinary ball journal bearings loaded in an

(Mr. A. Douglas Barty.)

endwise direction, with a view to finding out the relative life under these conditions as compared with radial loading.

With regard to the handling of ball bearings in the repair shop, I have found that in London there is a great difficulty in getting men who will take sufficient pains to fit them properly. The elderly men who have been trained in various other branches of engineering do not appear to appreciate that the ball bearing is a very delicate piece of mechanism which requires to be kept clean. The younger men who are being trained to the work of erecting seem to be able to remember better what they are told, and it is to the young men we have to look in the future for the correct fitting of ball bearings. I think it is to be regretted that the author did not read his paper two or three years ago in the interest of the young men, many of whom I know are taking great interest in the subject.

I may mention that we have been in touch with the Hoffmann Company from the beginning, and they have been able to help me in quite a number of ways. They have been generally correct in their theory as to the cause of failure, and any alterations we have made to their suggestions have invariably been followed by better results.

From the experience I have gained during the past four years in the use and maintenance of ball bearings, I have seen that by far the greater part of the trouble we experienced has been caused by dirt and water getting into the races from outside. I have, therefore, tried to find a remedy. This has resulted in the packing rings described by the author in his paper. I have found that these rings last twelve months with hardly any appreciable wear; at the end of this time it is necessary only to strip the joints of the rings to enable the spiral spring to keep them tight on the shaft. The saving of grease by this method has been found to amount to about 20*s*. per cab per annum. The actual saving effected in ball races has not yet been ascertained, although it will, no doubt, be very considerable.

COMMUNICATED.

Mr. T. Waterhouse wrote: It is very pleasing to note Dr. Hele-Shaw's appreciation of the beautiful and unique feature of self-alinement, which is characteristic of the double row bearings which he introduced into the discussion. He expressed, however, a little uncertainty as to the relative amount of "abrasion" attendant on the slip or imperfect rolling, due to the axis of rotation of the balls not being parallel to the axis of the inner race. It is well known that this slip must be present in every form of ball bearing employing an elastic material for balls and races (while

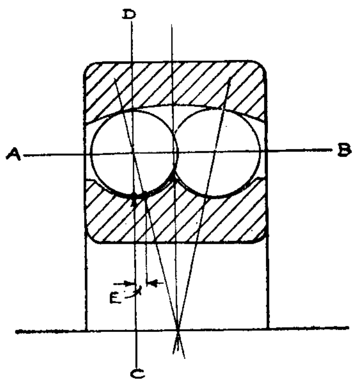

Fig. 38.

loaded), to an extent governed by the load, provided that the materials are not stressed to an amount beyond which the deflection exceeds that at which the elastic limit is reached.

A false impression may easily be obtained by taking only a cursory glance at the diagram of the bearing in question. It may appear that each of the balls has two separate and distinct axes of rotation, namely, one as shown by line AB in Fig. 38, and one as shown by line CD. If, however, this were the case, disastrous spinning of the balls (measured—when assuming the outer race to be stationary—by the periphery of the smallest diameter of the inner race divided by twice the distance E and multiplied

(Mr. T. Waterhouse.)

by the shaft revolutions), with its consequent abrasive effect would result, and the bearing would run only a very short time before collapsing.

The true case is very different, as will readily be seen from

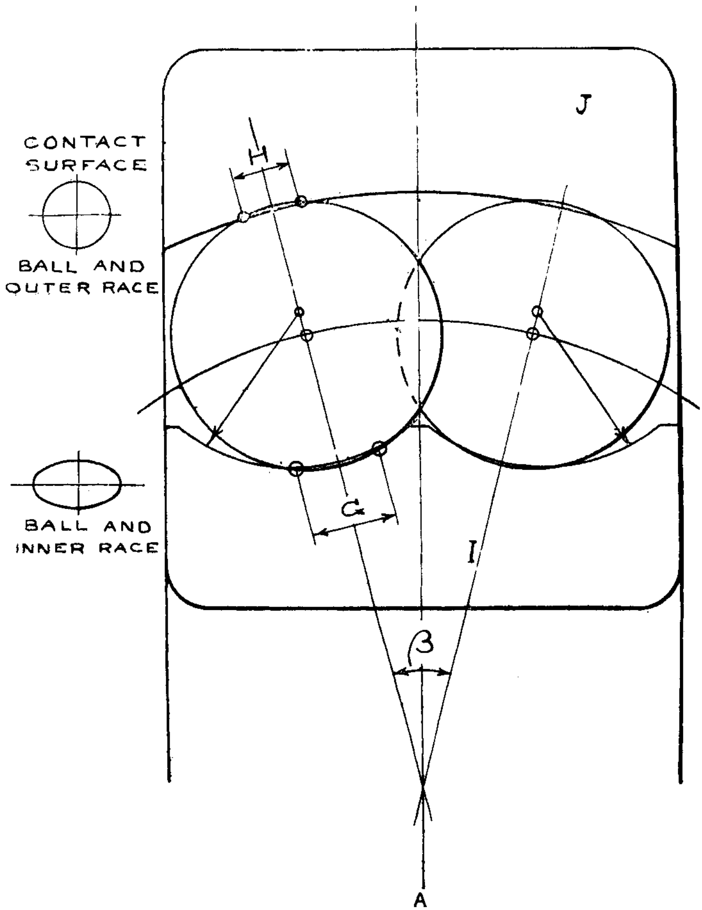

Fig. 39.

Fig. 39, from which it is clear that the conical surfaces of contact, when the bearings are fully loaded, are extremely small although the materials are elastic; they are, too, exactly on opposite sides of one diameter of the ball, and being parallel to the tangents to the ball surface are therefore parallel one to the other.

If the areas of contact on one row of balls on the outer and inner races be exaggerated for the purpose of example, and developed as shown in Fig. 40, both must be concentric, and embraced by

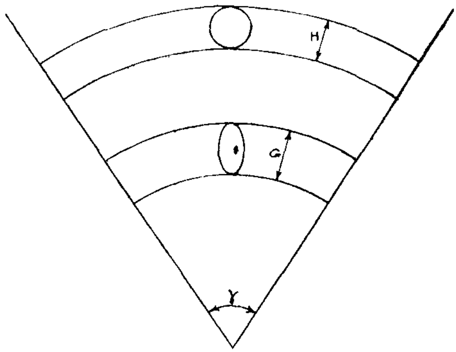

FIG. 40.

the same angle γ. It is unnecessary to demonstrate that A is very approximately equal to $\pi\beta$. (Fig. 39).

Again, assuming the outer race to be stationary, the actual amount of spin relative to this race on the axis AJ (Fig. 39)

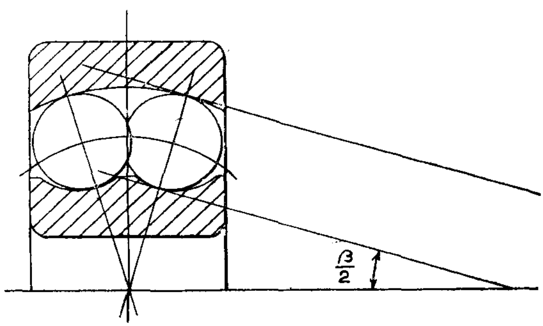

FIG. 41.

is measured by 360° divided by $\pi\beta$, where β is in degrees. This amount obviously is but a very small fraction of one revolution; it corresponds to one complete revolution of the ball about the main axis of the bearing, and also approximately to two com-

R 2

(Mr. T. Waterhouse.)

plete revolutions of the inner race. This slip is, however, absolutely negligible, having no more effect than the slip of a loaded elastic ball rolling in a curved race.

Some rather disparaging remarks were made during the discussion upon the thrust carrying capacity of the bearing, the suggestion being made that the wedging action exceeds that of a single row bearing. That this is absolutely false has been proved by experience and can be demonstrated as follows. The crushing force due to thrust load upon a ball in any radial ball bearing is in inverse proportion to the sine of the angle which the plane of contact makes with the main axis as shewn in Fig. 41.

In this figure a relatively large angle already exists, and while

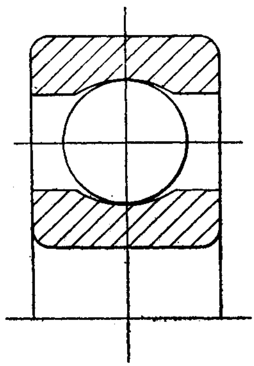

FIG. 42.

the bearing is not loaded in any way it enables it to take full advantage of this feature in resisting safely any end thrust which may be applied. In the single row bearing (Fig. 42), the plane of contact between the balls and the races is parallel to the main axis, and assuming the materials to be inelastic, the crushing load on the balls with the slightest thrust load is infinitely great. It may be argued that the elasticity of the materials permits the planes to become inclined to the main axis (Fig. 43), but it is perfectly safe to say that before this inclination could reach an appreciable amount, sufficient to take even a slight thrust load, the materials would already be stressed to many times their safe working values.

It should be clear from the foregoing that the spherical outer race double row bearing is theoretically able to carry a much higher thrust load than any single row bearing, an amount which has been found by experience to be from 20 to 35 per cent of the listed radial load.

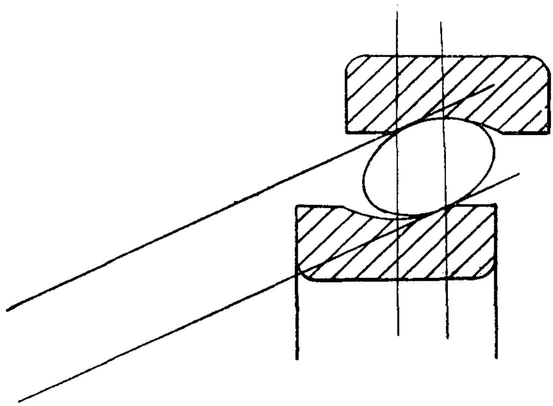

Fig. 43.

Mr. J. V. Pugh wrote: To my great regret I was unable to attend the meeting of the Institution of Automobile Engineers on Feb. 14th, to discuss Mr. Barrett's interesting paper, with much of which I find myself in disagreement.

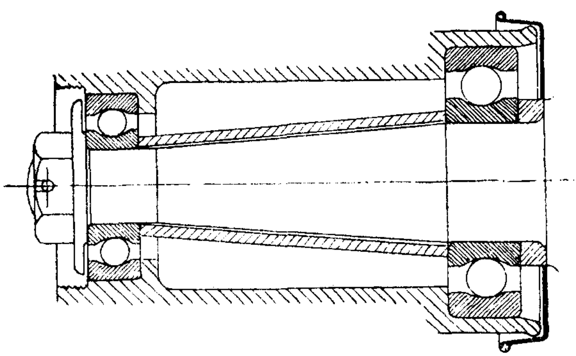

Fig. 44.

He is proposing to the automobile engineers of this country a somewhat complex and costly system of ball bearings as far as the wheels of cars are concerned.

(Mr. J. V. Pugh.)

Most of the points apply chiefly to the front wheel bearings, and so I will confine my remarks to these.

One of the oldest and simplest forms of employing journal bearings in the wheels of motor cars is illustrated in Fig. 44. The inner elements of each of the two bearings are clamped on to the axle by a nut and a distance piece, the distance piece being slightly longer than the distance between the two shoulders of the hub. The inner bearing takes the thrust from the outside, which is always the greater, and the outer bearing deals with the thrust in the other direction. A large washer is placed between the nut and the outer bearing to prevent the wheel coming off in the case of a bearing failure. It will be observed that the arrangement consists of very few parts, and that so long as the nut is kept in position (and it is fixed by the well-understood method of a split pin) the hub cannot possibly come off its axle. The method of keeping dirt and water out of the bearings at the inner end is extremely elementary, but in later examples of this type it appears to have been carefully thought out, and from such hubs of this type as have come under my observation, and from inquiries that I have made, this arrangement gives excellent results.

I give below what I think is a pretty accurate list of twenty-six firms who are using this arrangement.

Argyll; Austro-Daimler; Belsize; Benz; Berliet; Brazier; B. S. A.; Deasy; Delage; Germain; Mercedes; Pierce Arrow; Darracq; S. A. V. A.; Sheffield-Simplex; Singer; Spyker; Stoewer; Sunbeam; Renault; Standard; Vauxhall; Vinot; Vulcan; Wolseley; Calthorpe.

The author, in his paper, condemns this system entirely. I should be the last to urge that the Institution of Automobile Engineers should accept as final the practice of any large number of firms, however eminent, but I do think that we should at least carefully examine the criticisms brought against this practice, and the more complex arrangements which the writer of the paper proposes to substitute for it.

The author condemns this arrangement because:—

(1.) Journal bearings are subject to thrust.

(2.) Neither of the revolving elements is clamped.

(3.) The bearings are not, in his view, sufficiently well protected from water and mud.

Taking these points in order:—

(1.) The assertion that journal bearings are subjected to thrust is not strictly true. The action at the tyre of the largest lateral force that can be applied to a car changes the pressure on the bearing from the vertical line by probably less than 20° (see my article in *The Automobile Engineer*, of October, 1910). No sound theoretical reason has been advanced to show that a journal bearing is unable to withstand a force at 20° to the vertical, or even very much more. There is a line of high speed drills made by Messrs. Alfred Herbert, Ltd., which use journal bearings to transmit the whole of the drill pressure, and I cannot ascertain that they have given the slightest trouble.

I note that it has sometimes been urged that a journal bearing can take direct thrust or direct journal load, but that it cannot take the two together. I believe there is not the slightest ground for this view, which I imagine is based on tests of single bearings in which the load and thrust have been so applied as to produce a twist, that is to say, to throw the axes of the two bearing elements out of line with each other. The outer element approximates too closely to a spherical belt for it to be able to stand a twist of this kind, but where a pair of bearings is used it is quite obvious that there is no tendency whatever to displace the axes of the two elements of either bearing, for the other bearing takes care of this.

(2.) Bearings fitting loosely into their housings will certainly "creep," but the movement is pure rolling with no slipping whatever, and the degree of looseness that is met with in practice is not one which would introduce hammering effects, so that with ordinarily hard hubs there is not the slightest reason to expect that this pure rolling will produce serious wear in the hub during the life of the car. It should be remembered that the movement of the balls in their races is by no means pure rolling, and this circumstance, coupled with the enormously greater area of contact between the bearing and its housing, will far more than counterbalance the greater hardness of the balls and ball races. On the other hand, there is no reason why loose fits should be employed between the bearings and the housings.

(3.) I quite admit that the arrangement for the protection of the bearings appears extremely crude, and it is certainly extraordinarily simple and cheap to make. I think it would be useful if inquiries could be made on a wider scale than can be made by

(Mr. J. V. Pugh.)
a single individual, as to the success of this method of keeping dirt and water out of the bearings.

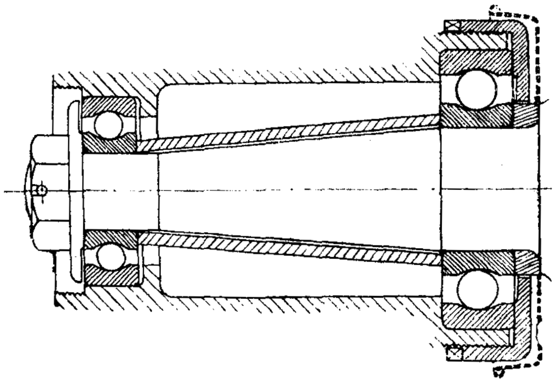

Fig. 45.

There is another series of bearing arrangements, shown in Fig. 45, in which the cap is screwed on to or into the end of the hub to

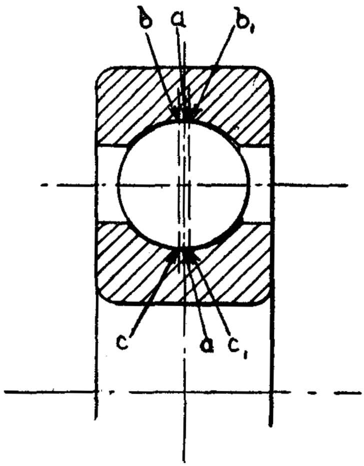

Fig. 46.

keep the dirt out. From a great variety of experiences, I am confident that this practice is not effective unless it is supplemented by the device used in Fig. 44 and shown in Fig. 45 by dotted lines. I think probably the double arrangement is better than the single canister lid, but have not been able to prove

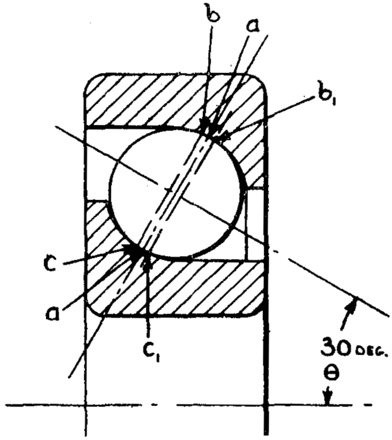

FIG. 47.

this; it has, however, one advantage over Fig. 44, and that is that in addition to keeping water out it can keep in a large quantity of

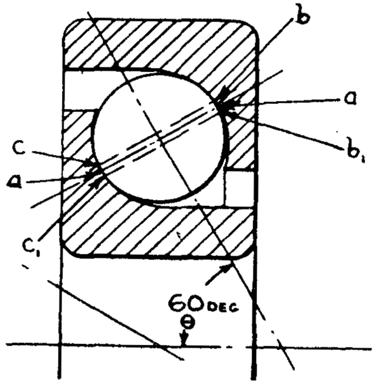

FIG. 48.

an oil or fairly fluid grease. With this style of bearing, the thrust is dealt with by the outer or the inner bearing as in Fig. 44, or one of the bearings is clamped, in which case it is the usual,

(Mr. J. V. Pugh.)
but not the invariable, practice to make the one bearing take the thrust in both directions. This series of bearings is also condemned by the author of the paper.

Now with regard to the designs the use of which the author advocates. These appear to be embodied in Fig. 25, which is provided with two journal bearings, the revolving elements of which are clamped while the stationary elements are free to slide the one on a stub end and the other on a sleeve on the stub end. The first criticism is, of course, that it is more complicated, more costly and heavier. The second, that the space taken up by the sleeve on which the outer of the two journal bearings is mounted takes away from the size of the journal bearing for a given diameter of stub end.

The form of thrust bearing employed is open to two objections urged by the author against the older cup and cone bearing, one, that since it is adjustable it can be wrongly adjusted and ruined, and the other, the degree of slip that takes place.

I enclose a series of sketches (Figs. 46 to 49) based on the

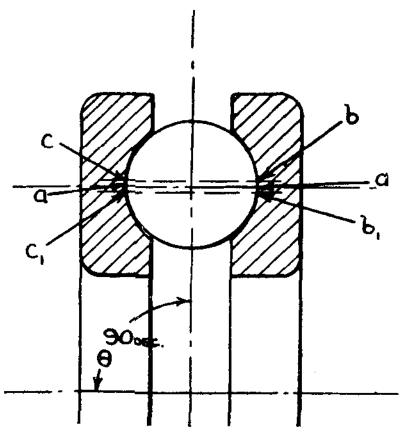

FIG. 49.

ordinary journal bearing 72 mm. outside, and 20 mm. inside, showing the effect of the axes of the ball remaining parallel to the axes of the journal, and at 30°, 60°, and 90° to it, the last being the form of thrust bearing which the author advocates. In each case I have assumed that the radius of the race is about

10 per cent greater than the radius of the ball, and that in conse-

SLIP OF BALL BEARINGS.

Type.	Angle (θ) of axis of Ball.	Slip per rev. of Ball at :—				Total Slip.
		b	b_1	c	c_1	
Journal..	0°	0·00676	0·00676	0·01285	0·01285	0·03922
Cone ..	15°	0·01774	—0·00410	0·03292	—0·00765	0·06241
,, ..	30°	0·02862	—0·01463	0·04962	—0·02548	0·11835
,, ..	45°	0·03924	—0·02456	0·06107	—0·03839	0·16326
,, ..	60°	0·04881	—0·03351	0·06680	—0·04578	0·19490
,, ..	75°	0·05726	—0·04086	0·06726	—0·04802	0·21340
Thrust..	90°	0·06373	—0·04602	0·06373	—0·04602	0·21950

quence the virtual contact between the ball and the race is about $\frac{1}{10}$th of the diameter of the ball.

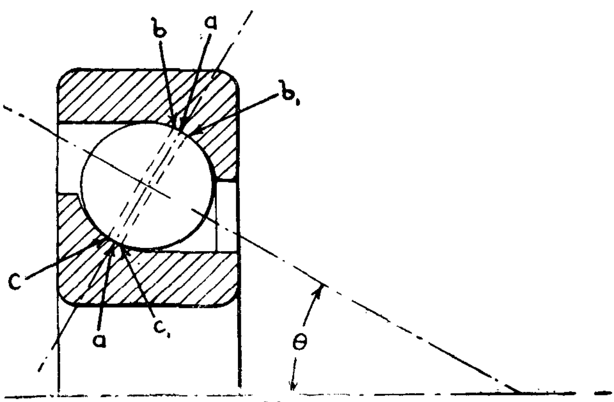

FIG. 50.

I have had calculated out the slip that takes place at each of the four points b b_1 and c c_1 (see Table and Figs. 50 and 51). I

(Mr. J. V. Pugh.)

have assumed that the axis of the ball does not change; this is not, I think, correct, because the ball probably in all bearings constantly changes its axis to a slight degree. This constant change is beneficial, as it distributes the wear, and it does not affect the point that I wish to call attention to, namely, that in

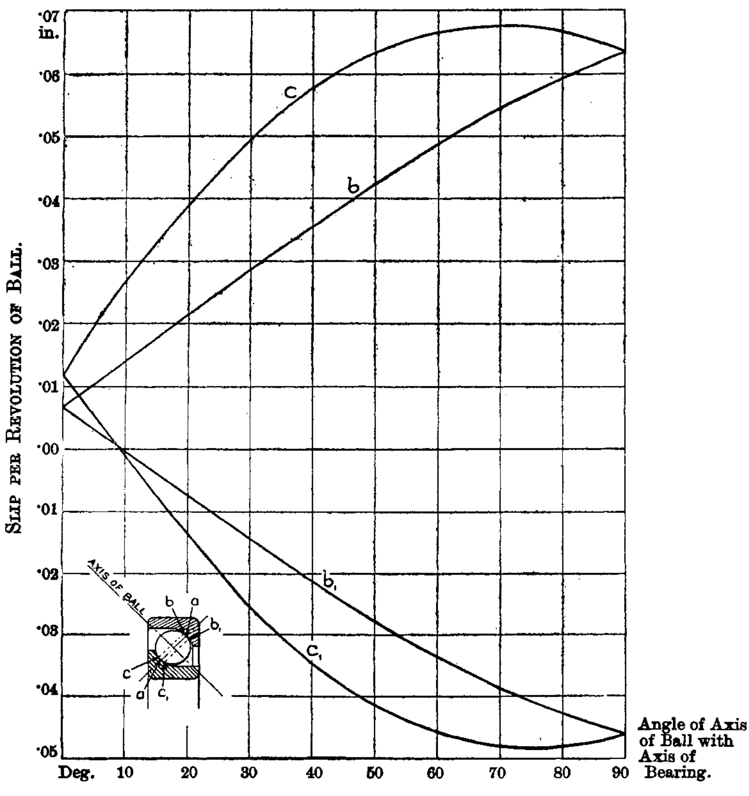

FIG. 51.—Diagram showing Slip on Balls in Bearings.

the thrust bearing the slip is far greater than in the cone bearing which the author condemns on this ground.

He also shows another form of thrust bearing (Fig. 32), of which he speaks highly, but this also is at all times subject to the same disqualification as the cone bearing, namely, the slip of the balls on their races. It will be observed that the total slip in the journal bearing is, as would be expected, less than under other conditions, but the very fact that this slip is always present, even in journal bearings, brings out strongly why it is necessary that a ball bearing should have good lubrication, which can only be secured by forcing lubricant in at one end, and preventing dirt and water getting in at the other end.

There is one further point. The arrangement recommended by the author relies for the security of the wheel on the axle on the dust excluding ring which is screwed into the hub. This ring is locked by three screws, the shanks of which are embedded in the ring, while parts of the heads penetrate into the hub itself. I do not think this plan is sufficiently positive and fool proof; besides which it does not allow for any very fine adjustment. If fresh holes have to be drilled they are liable to be drilled incorrectly, and the engagement of the hub with the head is insufficient for security. I have known of a case where an error of this kind led to an accident. It was not a case of the screw coming out, but of the ring actually turning round in the hub and carrying the screw with it.

The following list gives other makes of cars of which the hubs correspond more closely to Fig. 45 than to Fig. 44; some, however, are rather different, but in all cases there is no provision other than journal bearings to deal with thrust:—

La Buire; Chalmers Detroit; Charron; Clement Bayard; De Dietrich; Delaunay Belleville; Gobron Brillié; Gregoire; Hotchkiss; Leon Bollée; Martini; Metallurgique; Minerva; Mors; Opel; Panhard; Peugeot; Schneider; Unic; Armstrong Whitworth; Bergmann Electric; Dayton; Maudslay; Russell; Clement Talbot; Bianchi; Chenard Walker; Daimler (English); De Dion Bouton; Fiat; Itala; Lancia; Rolls-Royce.

Prof. Archibald Sharp wrote:—

Spinning Motion.—The author rightly calls attention to the so-called "spinning" action that takes place in some ball bearings. He says: "it plays such an important part in connection with the design of ball bearings, and, moreover, its ill effects are so much more pronounced in the larger bearings which are met with in automobile construction. . . . It is found that a bearing designed on this four-point principle very soon wears out, and this is almost entirely due to the so-called 'spinning' action which takes place with this and similar designs." This spinning action exists in all thrust bearings, but as the author

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says nothing in the paper on this point, a short discussion may not be out of place here.

As far as I am aware, I was the first to direct particular attention to the spinning motion in ball bearings, in my book "Bicycles and Tricycles," published by Longmans, Green & Co. in 1896. In a ball bearing, each ball moves in contact with each of the two races. Considering the motion of the ball relative to one of its races, if two bodies move in contact, the relative motion at the point of contact can be resolved into (1) a *translation* in a direction lying in or parallel to the common tangent plane at the point of contact, and (2) a *rotation* about an axis passing through the point of contact. Such rotation can be resolved into two component rotations (a) "rolling" about an axis lying in

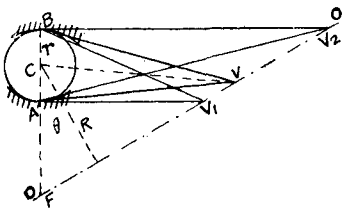

FIG. 52.

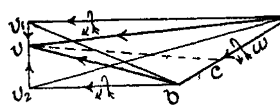

FIG. 53.

the tangent plane, and (b) "spinning" about the axis at right angles to the tangent plane, i.e., about the common normal at the point of contact. Thus, the most general motion of a ball relative to one of its races is compounded of "rubbing," "rolling" and "spinning." Rubbing action existed in many early forms of ball bearings, even now it is to be found in some ball bearings used for the steering heads of bicycles, but it should not be found in any ball bearing of decent design. Spinning and rolling take place simultaneously in the bicycle type of ball bearing, and in all forms of ball thrust bearings.

Fig. 52 is a half section of a bicycle type ball bearing, C being the centre of a ball and A and B its points of contact with the two races. Draw the tangents at A and B to cut the axis of the shaft at V_1 and V_2 respectively. If pure rolling without spinning exists at A, the motion of the ball relative to the fixed race A is that of a cone enveloping the ball and having its vertex at V_1. Similarly, if pure rolling without spinning exists at B, the motion of the ball relative to the race B is that of a cone enveloping the ball and having its vertex at V_2. These two conditions are mutually inconsistent. The actual motion of the ball will be that of a cone having its vertex at some point V between V_1 and V_2 and passing through the points A and B. In Fig. 53, draw the vector ab parallel to the axis oo. to represent ω the angular speed of the race B relative to the race A. Draw av and bv parallel to VA and VB respectively, meeting at v. Then, the motion of the ball relative to the races being the same as that of the cone VAB, VA is the instantaneous axis of rotation of the ball relative to the race A, VB that relative to the race B, while the vectors av and bv represent the corresponding angular velocities of rotation. For the angular velocity of race B relative to race A is the vector sum, angular velocity of ball relative to race A + angular velocity of race B relative to ball. That is, vector ab = vector av + vector vb. Similarly, draw av_1 and bv_1 parallel to V_1A and V_1B respectively, meeting at v_1; and draw av_2 and bv_2 parallel to V_2A and V_2B, meeting at v_2. Then ABV_2V_1 and v_2v_1ab are similar figures, and therefore v_2v_1 is parallel to AB. Also it can be shown that v lies on v_1v_2. Draw vc parallel to CV to cut ab at c. The vector av is the sum of the rolling av_1 of the ball on the race A and the spinning v_1v of the ball about the axis CA. The vector bv is the sum of the rolling bv_2 of the ball on the race B, and the spinning v_2v of the ball about the axis C_1B. The angular velocity of the ball cage relative to the race A is the vector ac. The angular velocity of the ball relative to the cage is the vector cv.

From Fig. 53, it is evident that the vector v_1v_2, the sum of the spins of the ball on its two races, is equal to $\omega \sin \theta$, where θ is the angle the line of pressure BA makes with the direction of the normal load. If $\theta = 90$ deg., we have the case of a ball thrust bearing in which the sum of the spins is equal to ω. Also from similar figures,

$$\frac{av_1 + bv_2}{ab} = \frac{AV_1 + BV_2}{V_1V_2} = \frac{2\ R \operatorname{cosec} \theta}{2\, r \operatorname{cosec} \theta} = \frac{R}{r}.$$

That is, the sum of the rollings of the ball on the two races is R/r, r being the radius of the ball and R the radius from the axis to the centre of the ball; a result independent of the angle which the line of pressure makes with the axis of the bearing. It will be noticed that the angular velocity of rolling of the ball on the outer race is considerably less than that on the inner race.

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Elastic yielding of Surfaces of Ball and Race.—Without going into a mathematical investigation of the elastic properties of the ball and race, an approximate idea of the extent of the surface of contact in the vicinity of the geometrical point of contact of ball and race, may be obtained as follows:—Let r and r_1 be the radii of ball and race respectively. In Fig. 54 take the axis of X, a tangent at O the point of contact, the axis of Y the normal at O. Let x, y, be the co-ordinates of a point P on the ball circle of radius r, and x_1, y_1, the co-ordinates of a point P_1 on the race circle. Then

$$y = r - \sqrt{r^2 - x^2} \ldots\ldots\ldots\ldots\ldots\ldots \quad (1).$$

Let $x/r = z$; then (1) may be written

$$y/r = \sqrt{1 - z^2} = \tfrac{1}{2}z^2 + \tfrac{1}{8}z^4 + \tfrac{1}{16}z^6 + \tfrac{5}{128}z^8 + \ \ldots\ldots \quad (2).$$

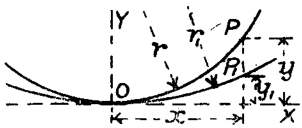

FIG. 54.

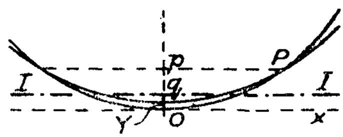

FIG. 55.

$(1 - z^2)^{\frac{1}{2}}$ being expanded by the Binomial Theorem. Similarly, if $z_1 = x/r$, we have

$$y_1/r = \tfrac{1}{2}z_1^2 + \tfrac{1}{8}z_1^4 + \tfrac{1}{16}z_1^6 + \ldots\ldots\ldots\ldots\ldots \quad (3).$$

Let $r/r_1 = k$, then $y_1/r_1 = y_1k/r$, $z_1 = zk$, and (3) may be written

$$y_1/r = \tfrac{1}{2}z^2k + \tfrac{1}{8}z^4k^3 + \tfrac{1}{16}z^6k^5 + \ldots\ldots\ldots\ldots \quad (4).$$

Let $Y = y - y$, then subtracting (4) from (2)

$$Y/r = \tfrac{1}{2}z^2(1 - k) + \tfrac{1}{8}z^4(1 - k^3) + \tfrac{1}{16}z^6(1 - k^5) + \ldots\ldots \quad (5).$$

The various series in equations (2), (3), (4) and (5) are all convergent for values of z less than unity.

If in a single row journal bearing the axial section of the race has a radius of curvature 5 per cent greater than the ball, $k = 20/21$, and (5) becomes

$$Y/r = \frac{z^2}{4z} + 0{\cdot}017z^4 + \ldots\ldots\ldots\ldots\ldots\ldots \quad (6).$$

The load on the ball produces contact of the surfaces in the vicinity of O; let the points P and P_1 be pressed together at the boundary line of the surface of contact, then Y is equal to the sum of the compressions of the surfaces of the ball and race at O. Fig. 55 shows the "geometrical compression," as it may be called, at the point of contact, greatly exaggerated. Draw Pp and Op respectively parallel and at right angles to OX. The actual instantaneous axis of rolling is now displaced from OX to some position II between O and pP. The intensity of the pressure will probably be a maximum at O, and diminish to zero at P, and Oq may be about one-third Op. As the ball rolls forward on the race, its surface at O will rub backwards, its surface at P will rub forwards. Thus even with pure "geometrical rolling" at the point of contact there is a certain amount of rubbing or abrasive action due to the elasticity of the material. The linear speed of rubbing at O is $\omega \times Oq$, $=$ say $\omega y/3$. Probably $Y = r/1000$ is about as great a value as is consistent with durability of the bearing. Substituting this value in (6) we obtain $z = 0{\cdot}20$, that is $x = 0{\cdot}20r$. Substituting in (2) we obtain $y = 0{\cdot}020r$. Therefore the linear speed of rubbing at O is about $0{\cdot}0067r\omega$. If spinning of angular speed ω_s also exists at O, the linear speed of rubbing at P due to the spinning is $\omega_s \times pP = 0{\cdot}20r\omega_s$. Therefore with the values assumed above, for equal angular velocities of rolling and spinning, the rubbing or abrasive action due to spinning is about 30 times that due to the rolling.

Consider now a section of the bearing transverse to the shaft, as in Fig. 3 of the paper. For the inner race, let $r_1 = -4r$, $k = -1/4$, and (5) becomes

$$Y/r = \tfrac{5}{8}z^2 + \tfrac{65}{512}z^4 + \ldots .$$

and substituting $Y/r = 1/1000$ as before, we obtain $z = 0{\cdot}04$, or $x = 0{\cdot}04r$. The surface of contact of the ball with the inner race is approximately an ellipse having its major semi-axis $0{\cdot}20r$, its minor semi-axis $0{\cdot}04r$; the ratio of the diameters being $5:1$.

For the outer race, $r_1 = 6r$, $k = 1/6$, and (5) becomes

$$Y/r = \tfrac{5}{12}z^2 + \tfrac{215}{1728}z^4 + \ldots .$$

and substituting $Y/r = 1/1000$, as before, we obtain $z = 0{\cdot}049$, or $x = 0{\cdot}049r$. That is, the minor semi-axis of the surface of contact at the outer race is longer than that of the inner race. Therefore if the radii of curvature in the plane of the shaft of the inner and outer races are equal, the areas of contact do not greatly differ. If the surface of the outer race is a portion of a

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true sphere, its surface of contact with the ball is a circle of radius $0{\cdot}049r$.

Wedging Action.—The author's remarks on wedging action and axial thrust in ball bearings, and more particularly Fig. 9 of the paper illustrative of his remarks, are somewhat ambiguous, if not misleading. His dictum, "It is quite impossible to design a ball bearing with a single row of balls which will carry more than a small amount of both journal and side thrust, because of the enormous wedging action tending to drive the balls up and into the races and burst them," may be true. But Fig. 9 is misleading in so far as it suggests that the very acute angle between the inner and outer cone surfaces in contact with any one ball gives any wedging action whatever. In this Figure, the two pressures on each ball have a small resultant tending to force the ball towards the thick end of the cone. If this action be resisted by suitable means, Fig. 9 would form, in my opinion, a tolerable, although not a very good, thrust bearing. His statement, "It is this wedging action which destroys the so-called cycle type of bearing, Figs. 11 and 12," is far too sweeping. Neither Fig. 11 nor 12 represents a good design of a bicycle type bearing, but bicycle type bearings are running year in year out without giving any trouble, and assuredly if they receive the same care in design, choice of material, and workmanship that the single row type has received from manufacturers, they may be used for much heavier loads than they are at present subjected to. Probably it is the facility with which single row journal bearings and single or double thrust bearings can be standardised, and applied by designers to widely different uses, that has led manufacturers of ball bearings to develop these types to the neglect or exclusion of the bicycle type. But in my opinion, the bicycle type ball bearing is eminently suitable for the front hub bearings, as I hope to show presently.

In a single row journal bearing with no appreciable slack, the wedging action introduced by side thrust is undoubtedly enormous; the equivalent wedge is of a much more acute angle than that shown in Fig. 10 of the paper, consequently a small side thrust produces an enormous pressure on the balls.

Bicycle Type Bearing.—Assuming that the journal load on a bicycle type bearing taken by at least two balls in each row, the maximum pressure on each ball for a journal load is $W/4 \cos \theta$. Used as a bearing to resist an axial thrust, the load is equally divided between the balls; if there are ten balls in each row, for an axial thrust W_1 the pressure on each ball is $W_1/10 \sin \theta$. Therefore, to produce equal maximum pressures on the balls, $W/4 \cos \theta = W_1/10 \sin \theta$; that is $W_1/W = 2{\cdot}5 \tan \theta$.

The B. S. A. bicycle hub bearings are designed so that the line of pressure BA makes an angle of 20 degrees with the direction of the load on the axle. In the B. S. A. crank axle bearings the corresponding angle is 18°.

Therefore for $\theta = 20^\circ$, $W_1/W = 2{\cdot}5 \times 0{\cdot}364 = 0{\cdot}91$. That is, if it were merely a question of the maximum intensities of pressure on the individual balls, the bicycle type bearing should sustain an axial thrust nearly as great as its journal load. In the case of axial thrust, the load on each ball is constant, in the case of a journal load each ball is only under pressure during a small fraction of its revolution round the axis; and probably this difference in the conditions does affect the result.

In presenting this discussion, I do not intend to suggest that for a shaft to sustain both a direct journal load and a direct *axial* thrust, the bicycle type bearing is as good a design as the type having two single row journal bearings and a double thrust bearing. In fact, I agree with the author that the latter type is superior to the former.

Ball Bearings for Front Axle Hubs.—When a motor car is being steered in a circular path, a radial or "centripetal" force directed towards the centre of the path is required to balance the centrifugal force. Such radial force is supplied by the resistance to side slipping of the tyres. This radial force C applied to the wheel at its point of contact with the ground, is equivalent to an equal parallel force applied at the wheel axle, together with a couple CR, R being the radius of the road wheel. The couple CR can only be resisted by two equal and parallel opposite forces F applied at the journal bearings. In the front axle hubs the distance between two journal bearings is small compared with the radius of the wheel, and F the pressure on the journal bearings due to the side thrust may be much greater than that due to wheel load when the car is running straight.

In a bicycle type bearing with a cone adjustable on the spindle or axle, the virtual length of the bearing is greater than the axial distance between the two rows of balls. Therefore the pressure on the balls due to centrifugal action is less than that in the single row journal bearings of standard type.

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Fig. 56 is a section taken through the axis of the left hand front wheel, showing the forces acting on the front wheel when the car is being steered (a) straight, (b) to the right. F_1 and F_2 are the centres of the two rows of balls of the standard type of bearing, but represent the two virtual points of support of the bicycle type bearing. When the car is running straight the force L applied to the wheel at A, its point of contact with the ground, is vertical, and the journal loads at F_1 and F_2, are inversely proportional to their distances from the vertical AA_1. In the case of the bicycle type bearing, since the pressures on the various balls in a row all pass through F (Fig. 1), the resultant pressures P_1 and P_2 at F_1 and F_2 must make an angle with the vertical equal to or greater than θ. Further, since the forces L, P_1 and P_2 are in equilibrium, they must all pass through some point a on the vertical AA_1. In Fig. 57, draw the vector L vertical to represent the load at A, and from its extremities draw the vectors P_1 and P_2 parallel to F_1a and F_2a respectively. The position of the point a will depend on the initial adjustment of the bearings. If the adjustment is such that there is little or no initial pressure on the balls, a will approach the limiting position a_0, the line F_2a_0 making the angle θ with the vertical. In this case the load at F_2 is taken by one or two balls at the lowest point of the row. If, on the contrary, the initial adjustment has been such as to produce an excessive axial thrust on the bearings, the point a will approach the limiting position a_1, the point of intersection of the vertical AA_1 with the axis of the hub. As the point a in Fig. 56 moves from a_0 to a_1, the corresponding point a in Fig. 57 moves horizontally from a_0 towards the right, its limiting position being at infinity.

If from a the point a_2 be projected horizontally on to the vector L, the point a_2 divides L into two parts respectively equal to the journal loads J_1 and J_2 at F_1 and F_2, for the standard type of hub with double thrust bearing.

Suppose now, the car is being steered to the right, a centrifugal load is thrown on the outer or left hand wheel, which is resisted by the force C parallel to the road surface. The re-action R of the ground on the tyre at A is now the resultant of C and the vertical load L. In Fig. 56, R is drawn to correspond with a value of C equal to one-third L. (It is to be observed that owing to the mass centre of the car being at a considerable height above the ground, part of the load is transferred from the inner to the outer wheel; the limiting case being reached when all the load is on the outer wheel, and the car is on the point of overturning, presuming that the tyre offers sufficient resistance to side slipping.)

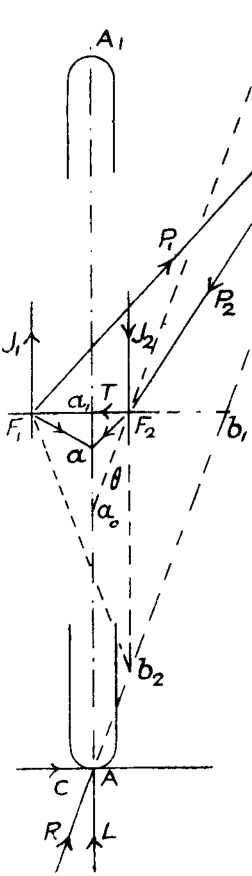

FIG. 56.

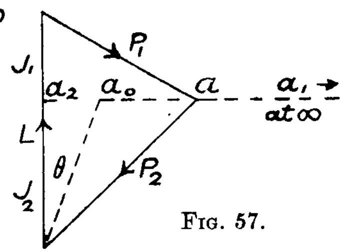

FIG. 57.

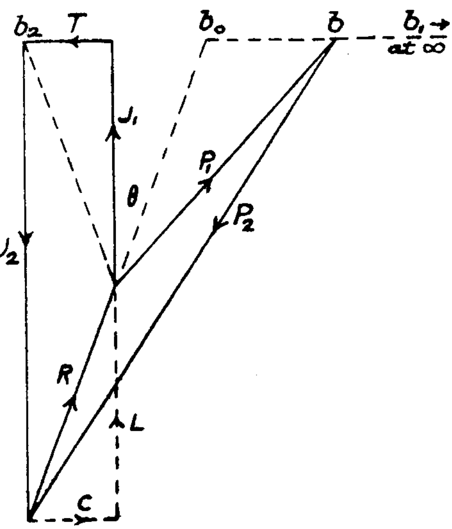

FIG. 58.

Fig. 58 is the corresponding vector diagram of the forces acting on the wheel. In the case of the standard hub with double thrust bearing, the thrust T acts along the axis F_2F_1, the wheel is in equilibrium under the action of the four forces R, T, J_1, J_2, the two latter being the loads on the single row journal bearings

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at F_1 and F_2. The resultant of T and J_1 passes through F_1, the resultant of R and J_2 passes through b_2 the point of intersection of R and J_2. These two resultants must be equal and opposite. In Fig. 7, from the extremities of the vector R draw vectors parallel to J_2 and b_2F_1 respectively, meeting at b_2, and complete the vector diagram by drawing J_1 and T parallel to the corresponding lines in Fig. 5. Note that J_1 and J_2 are very much greater than R.

In the case of the bicycle type hub bearing, the wheel is in equilibrium under the action of the three forces R, P_1 and P_2, the two latter being respectively the resultant loads on the two rows of balls. These three forces must, therefore, all pass through some point b on R produced. In Fig. 58, from the extremities of the vector R draw vectors parallel to F_1b and F_2b respectively, meeting at b. As before, the positions of the point b in Figs. 56 and 58 will depend on the initial adjustment of the bearing. As the point b in Fig. 56 moves along AB, the point b in Fig. 58 moves horizontally, the limiting positions are b_0 and b_1, F_2b_0 (Fig. 56) being inclined at the angle θ to the vertical, and b_1 being the point of intersection of the axis of the hub with the line AB the direction of R. The line b_0bb_1 (Fig. 58) coincides with T produced.

When there is no centrifugal component C, and no axial thrust, the axial components of P_1 and P_2 are equal, as shown in Fig. 57, and P_1 is considerably greater than J_1. As C gradually increases, the journal load J_2 increases, and J_1 diminishes, until the direction of R passes through F_2 (Fig. 56), when J_1 becomes zero. For greater values of C relative to L the vertical load, J_1 and J_2 increase, their difference being equal to L as shown in Fig. 58. The difference of the axial components of P_1 and P_2 is always equal to C. P_1 and P_2 must always be greater than J_1 and J_2 respectively, provided the points F_1 and F_2 are the same in the two types of bearings. But if the axial distance between the two rows of balls is the same in both types of bearing, the distance F_1F_2 in the bicycle type bearing is considerably greater than the distance between the rows of balls, and the values of P_1 and P_2 may become less than J_1 and J_2. Further, in the bicycle type bearing the pressure P_1 or P_2 is probably distributed over a greater number of balls than is J_1 or J_2 in the single row journal bearing, and the greatest pressure on any one ball may be less in the bicycle type than in the other type. Fig. 59 shows end and side views of one row of balls in a bicycle type bearing, corresponding to the row at F_2 (Fig. 56). The resultant pressure of the balls on the cup or outer race corresponds with the conditions shown in Fig. 58. The greatest pressure will be on the lowest ball, and since P_2 has a considerable axial component, there may be pressure on more than half the number of balls in the row. The relative pressures on the various balls will be somewhat as shown by the vectors in Fig. 59. Fig. 60 shows end and side views of the corresponding vector polygon, which approximates to a portion of a screw. Measuring from the diagram, the greatest pressure on any ball, that on ball No. 56, is between one-fourth and one-fifth the resultant pressure P on the row of balls.

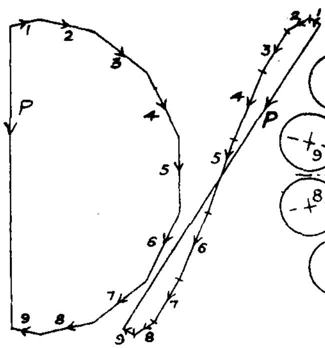

FIG. 60.

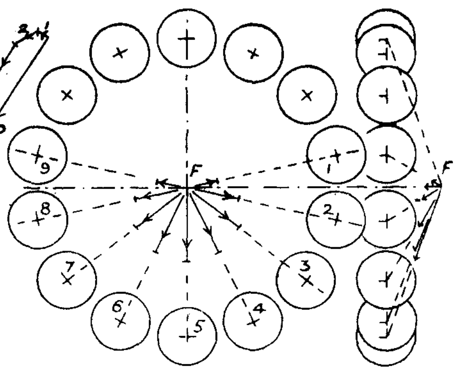

FIG. 59.

The argument may be summed up thus:—In a front axle hub of any type, when the car is being steered sideways, the pressure on the ball bearings is considerably greater than when the car is running straight. In the hub with two single row journal bearings and a double thrust bearing, the journal load is distributed over a less number of balls than in the bicycle type bearing. In the bicycle type bearing the virtual length is greater than in the other type, the number of balls is greater, and the axial component of the pressures on the balls is sufficient to resist the centrifugal load thrown on the wheel, without producing a maximum pressure on any individual ball as great as that produced in the other type.

(Mr. G. F. Barrett.)

Mr. G. F. Barrett, in replying on the discussion and the communications, said and added in writing: I have to thank you, gentlemen, for the very cordial way in which you have taken my paper. One of the principal questions raised this evening has been the advisability of using double thrust bearings in front hubs. As is well known, the firm with which I am connected have always been strongly in favour of this course, but we have met with very considerable opposition from motor car makers. Although it is a fairly simple matter to beard one of them in his den at the works, when I came to think that I should have to stand up in front of the whole lot this evening I thought I must concede something, and that is really the reason why I said that possibly for the front axles of moderate size touring cars they were not necessary. At the same time I am not at all certain that thrust bearings have been proved to be necessary, because there are a great number of motor cars running now, that have been running for a number of years, with only two journal bearings on the front hubs with perfectly satisfactory results. There are, of course, also a number which have worn out, but before saying that this was due to the want of a double thrust bearing, I should like to be sure that the wear was not due to moisture and dirt getting into the bearing.

Mr. Smith said I did not in the paper lay sufficient emphasis on the importance of the question of rust. If this is so, I am sorry, for it is by far the most serious cause of failure in motor cars at the present time, and is one of the things I would particularly ask the motor car engineer to guard against. I regret that I have had no experience of the admixture of lard oil with the lubricant as a rust preventive.

Dr. Hele-Shaw and Mr. Lawrence both raised the question of load carrying capacity. I purposely avoided reference to this question in the paper because I felt that a whole paper could be written on the subject, and a whole evening devoted to its discussion, and that at the end we should not be much wiser. The load carrying capacity of ball bearings is a very difficult thing to determine. Prof. Bertram Hopkinson has said that if two hollow steel balls of the same size and weight as billiard balls approached each other at a speed of 16 ft. per second the impact on these balls at the point of contact would be 280 tons to the square inch. The load a ball bearing will carry depends upon the maximum shock that will come upon the balls and races. When an ordinary plain bearing receives a shock the oil will remain between the surfaces in contact some time before it is squeezed out, and the shock has time to become absorbed before the moving parts get in contact. With a ball bearing there is not that time; the shock comes on the balls immediately. Take a thrust bearing on the end of a worm shaft transmitting 30 h.p. to drive a three-throw water pump, the hammer effect of the water will be sufficient to destroy the ball bearing. Taking the same gear transmitting the same power to drive a motor car, owing to the spring of the chassis, pneumatic tyres, and possibly wire wheels, the same bearing will perform the work quite easily, and a very much smaller one will be sufficient. It will therefore be seen how extremely difficult it is to give anything in the nature of a load table for ball bearings.

Regarding Mr. Lawrence's remarks as to the fitting of the inner races, it is essential that the shaft should be at least half a thousandth of an inch larger than the bore of the bearing. If a ball bearing is fitted on such a shaft and the diameter of the ball race is micrometered after it is on the shaft, it will be found that it has hardly expanded, but if the shaft is made more than half a thousandth, say one thousandth larger, it will be found in that case that the ball race has expanded half a thousandth. It seems that a tightness of fit of half a thousandth of an inch does not expand the ball race.

With reference to Mr. Shilson's remarks about the thrust bearing in Figs. 14 and 15, the mounting of thrust races against nuts screwed on the shaft, as shown in Fig. 14, is not good practice, for, unless the greatest care is taken in machining the threads, the face of the nut is seldom at right angles to the shaft, and when this cannot be secured, Fig. 15 is recommended. The double thrust bearing in Fig. 29 is supposed to have had every care and attention paid to the nuts and threads, this being one of the vital points in connection with its success.

The President: May we take it that the difference is that in Fig. 29 the part in question is made by the ball bearing makers, while the other shown in Fig. 14 is not?

Mr. Barrett: Exactly so. With reference to Mr. Shilson's remarks comparing the bearings shown in Figs. 30 and 32, the difficulty with the former design, as pointed out in the paper, is the adjustment, or, rather, the apparent want of it. When this is correctly done, there are two sets of clearances to be taken up

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on reversal of the load, and the shock must be greater than when only one clearance has to be allowed for as in Fig. 32. The latter design has no adjustment, and the clearance, being controlled by the ground finish of the outer race rings where they abut against each other, can be fixed very exactly. I have in my possession some bearings of this type which have done a lot of work on some of Mr. Barty's cabs in the London streets, and they are in very good condition. It must not be forgotten that this design is only suitable for light and intermittent loads in either direction, and that it carries no journal load at all.

Mr. Shilson's suggestion of using the double thrust bearings, shown in Fig. 18, plus the packing washers, is an excellent one, and would overcome the difficulty which was also raised by other speakers.

As regards the lubrication of the double thrust on the end of worm spindles, this is better shut off and filled up with grease, as is done with front hubs.

The steel buttons shown in Fig. 20 for locating the lay shaft would certainly have to be carefully adjusted if the outer races of the journal bearings were fixed, but as these are allowed free movement endways, and a slight amount of end play of the shaft can be allowed, it is not necessary to have these steel buttons such a very good fit. This method has been largely employed and found satisfactory. It has the advantage of removing all side thrust from the journal bearings.

Mr. Armstrong raises this same point and goes on to discuss the subject of the materials used in the manufacture of ball bearings. As pointed out in the paper, I believe that bad material as a cause of failure in the best ball bearings is very rare. However, new forms of steel are continually being introduced, and the load carrying capacity will no doubt be greatly improved in the future.

With regard to Mr. Root's suggestion that the ball bearing makers were so conservative that they would not adopt his suggestions, I think it was possibly due to the fact that at that time it was practically impossible to make bearings of that type commercially. It has only been in the last few years when modern machine tools have reached the present day fine degree of perfection, that it has been possible to manufacture commercially, ball bearings of the present type.

Mr. Clayden raised the point of the relation of speed to load. This depends to a large extent upon the accuracy of manufacture of the ball races and within the limits of speed mentioned should not in my experience vary as much as is shown by some of the curves produced.

With reference to Mr. Hounsfield's remarks about repairing worn-out ball bearings, provided the bearings are not brittled out and the ball races are not worn down too much, it is possible to just touch up the ball races and fit them with larger balls.

As regards the spinning action of the balls in the Skefko design, this of course exists, but as the angle of the line of load across the balls is not far from being at right angles to the axis of rotation it cannot be of great amount. The spinning action in the bearing shown in Fig. 32 is sufficient to prevent its being used as a continuous thrust bearing in the ordinary way. It is pointed out in the paper that this form of bearing is only suitable for light intermittent loads.

With regard to Mr. Ronca's remarks as to graphite as a lubricant, I should imagine that the Aitchison graphite would not harm a ball bearing. I have tried some graphites and have been unable to detect any wear whatever on the ball races when this has been employed. On the other hand, there are a very large number of graphites on the market which play havoc with ball bearings, and their use cannot be too strongly condemned.

Mr. Brewer asked about the locking nut on the front hub. That is a point which I alluded to and pointed out that care must be taken to see that these nuts are securely locked in position. One has been known to come out, so that care has to be taken to see that they are locked in position by means of the three screws and also a split pin. When such fixing has been used they have not been known to come adrift.

Referring to Mr. T. Waterhouse's communication, he compares the load carrying capacity of the bearing having two rows of balls and with a spherical outer race shown in Fig. 38 to the bearing with one row of balls shown in Fig. 42. As regards pure journal load, the difficulty in the first case is always to insure that two rows carry their proper share of the load. For this to take place it is essential that the outer race should be perfectly free, without any friction whatever, to float opposite to the inner race. In actual practice such a condition must be very difficult to attain, and it must be evident that if the outer race is not

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exactly opposite to the inner one, one row of balls will be loaded and the other row released, so that the journal load will have to be carried entirely by the first row. A very slight difference in alinement will cause this. Again, a slight side thrust will have the same effect, and an alternating thrust or sway of the shaft will first release one row and then the other as the thrust reverses, so that again the journal load will come only on one row of balls.

As regards pure side thrust, Mr. Waterhouse states that this bearing carries more than the single row type, but the latter is not designed to carry any at all, and the former would soon give trouble if any amount of permanent thrust were applied to it. For intermittent side loads it would not be so good as the bearing shown in Fig. 32, as the points of contact are at a greater angle to the load.

As regards combined journal and thrust, it has already been pointed out that under these conditions only one row of balls will be taking the load, and the outer race at that point where the journal load was a maximum would in combination with the thrust load be overloaded.

Referring to Figs. 39 and 40, it is not very easy to calculate the areas of contact, but I have asked Mr. G. J. Wells, Wh.Sc., A.M.I.C.E., to look into this for me, and he gives the approximate areas as follows:—

For a single row bearing having the following dimensions:—

Diameter of outer ball track	$2\frac{3}{4}$ in.
Radius of curvature of outer race	0·275 in.
Diameter of inner ball track	$1\frac{3}{4}$ in.
Radius of curvature of inner race	0·2625 in.
Diameter of balls	0·5 in.

and for a double row bearing of the same dimensions, that is to say, with a spherical outer race of $2\frac{3}{4}$ in. diameter.

If the ball deflects so as to approach the race by 0·0001 in.:—

The area of contact for the single row type is on the outer race 0·00057 sq. in., and on the inner race 0·00063 sq. in.; for the double row type, on the outer race 0·000187 sq. in., and on the inner race the same as on the single row, namely, 0·00063 sq. in.

Under ideal conditions this two row bearing would support the load on three or two balls as against two or one in the single row type.

Taking the load-carrying capacity of the bearings to be proportional to the areas of contact on the weakest part, namely, the outer ball race, and that the load is equally distributed between the two rows of balls in the two row type, the single row will carry the most load in the proportion of 0·00057 to $0{\cdot}000187 \times 2$ or of about $1\frac{1}{2}$ to 1.

The friction of the bearing in the two row type is, within working limits, increased by the larger number of balls.

I have read Mr. J. V. Pugh's communication with interest, and note his criticisms on my paper.

The old and simple form of employing journal bearings in front wheels, as illustrated in his Fig. 44, has, I am aware, been used in the past by a very large number of firms, as indicated in his list, and as far as my knowledge goes, some of these firms still continue to use this arrangement on all their cars, while others are discontinuing its use for the larger cars, and are only using it for the smaller ones. I also know that some of the firms given in this list have had very considerable trouble owing to this method of mounting due to permanent side thrust being put upon the two rows of balls. Considerable difficulty is still being experienced with this type owing to dirt and moisture getting into the bearings.

Referring now to Mr. Pugh's remarks on journal bearings subjected to side thrust: it is of course an advantage to spread the two races as far apart as possible, and therefore resolve the horizontal forces acting on the tyre into nearly vertical components, but this is not found in practice to relieve the journal bearings of side thrust sufficiently to prevent undue wear, except perhaps in the lighter form of touring cars.

The journal bearing in the line of high speed drills referred to, which has to withstand the pure end thrust and no journal load from the drill, does so by virtue of the fact that the end thrust is evenly distributed over all the balls, and therefore is very much less than when subjected to a combined journal and thrust load; in the latter case the journal load is carried on two or three balls, and these are held by this load in the bottom of the tracks. A side thrust coming on to this bearing at the same time would be carried by these same two or three balls. The balls which are not under journal load and which are therefore slack in their races, will not be carrying any of the side thrust.

In regard to the clamping of the revolving elements, it is true

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that the "creep" of bearings is a pure rolling motion, but the action of the outer ball race rolling round in the hub shell produces a knurling action, which is found in practice sufficient to cause the seat in the housing to wear larger in diameter.

With regard to protection of the bearings from dirt and moisture: this is nothing like sufficient, and endless trouble is being continually experienced with bearings shielded only in this way.

I quite agree that the arrangement for excluding water shown in Fig. 45 with the canister lid is better than that shown in Fig. 44, but I do not consider that even this is sufficient to prevent moisture getting in.

With reference to Mr. Pugh's criticisms on the arrangement shown in Fig. 25 in the paper, the first one, that of extra weight and cost, is one which I leave to the commercial policy of motor car manufacturers. If it is proved by actual experience that this arrangement is necessary for the satisfactory running of front hubs, it would appear to me that this objection will be outweighed. The second point is answered in the same way.

As regards the adjustment of the thrust bearing employed: it is true that this is open to the objection that it may be badly adjusted, but in this case it is mounted on an independent sleeve, and can be properly adjusted on the bench and locked in position. This adjustment is not in any way interfered with by the assembling or taking to pieces of the hub afterwards.

I quite agree that there must be a certain amount of slip in ball bearings, but it does not appear to play any part as a cause of failure in them. Mr. Pugh takes the effective contact between the ball and the race at about $\frac{1}{10}$th of the diameter. This may be approximately correct, but it must be evident that the intensity of pressure, as pointed out by Dr. Hele-Shaw at the Meeting, is only at the centre of this band, and the pressure towards the extremities must be very little.

Referring to the double thrust, Fig. 32, I pointed out that this was not correct in theory but was found to work well in practice.

The end disk nut shown in the end of the hub shell locked in position by three screws and a split pin has also been found to work well in practice. No adjustment whatever of this nut is needed. It should be once and for all screwed tight home.

In conclusion, Mr. Pugh's criticisms appear to be based somewhat on theoretical considerations, and, although no doubt this is one way of looking at the matter, yet the recommendations made in my paper are based wholly and solely upon the results of practical experience of ball bearings on motor cars of all types and makes extending over the last eight years.

These points are much more brought home in connection with motor cars used for commercial purposes, taxicabs, delivery vans, omnibuses, etc., as the mileage they run on an average is considerably in excess of that of touring cars, and the experience so gained should be of value to those motor car firms who are desirous of turning out thoroughly reliable cars.

Prof. Sharp endeavours to prove from theoretical considerations that a ball bearing of the bicycle type is suitable for the front axles of motor cars, and is also a suitable bearing for carrying side or axial thrust. He also suggests that the cause of failure of this type of bearing in the past has been that it has not been made of such good materials or of such a high finish as the modern two-point ball bearing.

The Hoffmann Manufacturing Co. of Chelmsford have made and conducted a very large number of experiments to endeavour to get a bearing of this type to stand up satisfactorily against the heavier loads met with in motor car construction, and I know that the material and workmanship have been everything that could be desired. The firm have been quite unable, however, to get this type of bearing to work satisfactorily. Other ball bearing makers have, to my knowledge, also attempted this with the same result.

The President: By the way you have received Mr. Barrett's reply there can be no doubt that this experiment of ours in taking the paper as read has proved successful, and that you would like the same course to be pursued on another occasion. Is that your wish?

The meeting having expressed its approval,

The President continued: Before I sit down there is one remark I should like to make in connection with ball bearings not alluded to by the author, and that is that just prior to the beginning of the automobile industry a large number of the ball bearing makers directed their efforts to making balls by other processes than cutting them from the solid. There were a number of processes by which efforts were made to cheapen manufacture by producing the balls nearly finished as forgings, these forgings

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being formed by rolling, with the result that the ball was weak along one axis; this fact contributed largely to the extra troubles which the ball bearing makers met with in their early days. That method has now been abandoned, and I think there is nothing used except the ball made from the solid. I do not think any of the big firms would produce balls now made by rolling processes. At the same time I think it has been made clear that the troubles met with in practice are more often due to imperfect fitting than to want of care or attention on the part of the manufacturers of the ball bearings.