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BALL AND ROLLER BEARINGS IN THE NAVAL SERVICE.

Ball bearings were first extensively used in bicycles and were of the type shown in Fig. I, but their many advantages has led to the extensive adoption of bearings of this type or roller bearings in place of plain journal bearings in the design of light machinery, whilst roller bearings are sometimes used for very heavy work indeed, for example, in the main races of the largest gun turrets. In the naval service, they are principally to be found in electrical, auxiliary and gun mounting machinery.

The advantages of ball and roller bearings over plain journal bearings are as follows :—

(1) **Reduction of Starting and Running Friction.**—It is evident that with true rolling motion, the friction is independent of the viscosity of the lubricant. The function of the lubricating medium is solely to protect the rolling surface against corrosion. As such it is important to note that mineral grease provides the necessary qualities while being free from harmful acids which would cause pitting. When speeds are exceptionally high, oil lubrication is preferable.

The frictional resistance when starting is negligible compared with the best journal bearing, the actual ratio of the starting torques of a shaft supported in ball bearings and plain bearings being of the order of 1 to 800.

Under running conditions the co-efficient of friction varies between 0·0011 and 0·0015. Under maximum load it should not exceed 0·0016 in a well-designed bearing. It is found that friction decreases with increase of load, and may rise to 0·003 when the load is reduced to one-tenth of the maximum. In roller bearings, with pure rolling motion the friction is practically constant at all speeds, but in bearings with end thrust the co-efficient of friction is greater at low than at high speeds.

In comparison with the above, average figures for well lubricated journal bearings are 0·002 to 0·01 depending on the load and the velocity of sliding. The co-efficient of friction in a bearing of the Michell type is generally lower than a ball bearing and varies between 0·0008 and 0·003.

(2) **Increased Life with Less Attention.**—Provided a ball or roller bearing is properly protected against corrosion and grit the determining factor in the life of the bearing will be the fatigue limit of the surfaces in contact. These surfaces it is true, cannot be refitted in the same way as in a journal bearing with anti-friction metal, but it must be borne in mind that from start to finish of the life of the

ball bearing, the alignment and running clearance is maintained; whereas in some plain bearings, periodical refitting and scraping are required. The labour costs involved in these refits must therefore be offset against the relatively high initial cost of a ball bearing.

(3) **Capability of taking High Loads with Small Axial Length.**—An actual example will illustrate the point.

Selecting from the specification of a well-known firm of manufacturers a certain roller bearing has the following dimensions :—

Outside diameter	12 in.
Bore	6 ,,
Width	$2\frac{1}{4}$,,
Maximum steady load at 1,000 r.p.m. =	18,600 lb.

Designing a journal bearing from the above data as regards load, speed, and shaft diameter, we find on working with the usual allowances current in bearing design :—

Shaft diameter	6 in.
Bearing length	15 ,,
Outside diameter	8 ,,

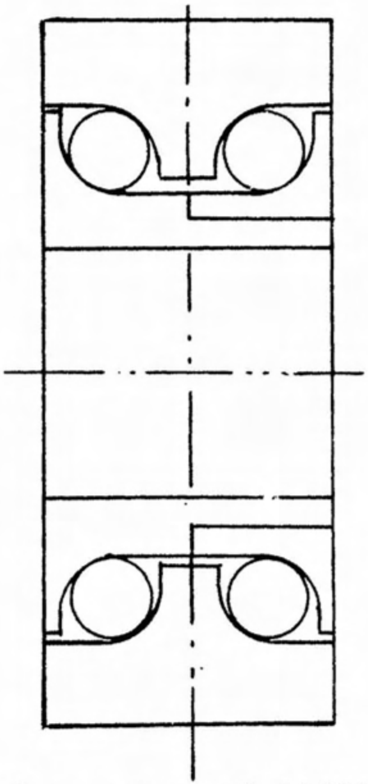
Even if a bearing of the Michell type were used this length would, at the best, be halved. The roller bearing therefore has a large advantage in cases suited to its employment, if space is an important consideration.

(4) **Capability of Taking Combined Thrust and Axial Load.**—Ball bearings of the deep groove type (Fig. 2), are particularly well adapted to take both axial and radial loads. No increase in length is therefore required over the normal race as fitted for pure radial loading.

The path of contact of the balls with the races is moved axially and radially by the thrust load, and the ratio of the axial to the radial load for any given bearing depends on the axial displacement of one race relative to the other. This ratio is approximately 1 : 3 for groove type bearings with continuous thrust load.

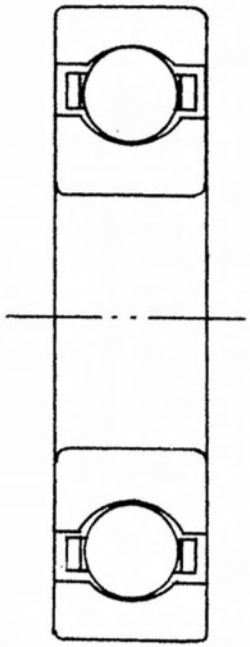
If it is required to carry a larger Radial load, a double row spherical bearing can be employed. Fig. 3. In this design, the permissible thrust load varies with the angle of obliquity of the balls, and is smaller than in the groove-type bearing. Deep groove split bearings are now being developed which are designed to carry a thrust load that is always greater than the radial load. These are described in a later paragraph.

(5) **Ease of Replacement.**—Provided the bearing housings are turned perfectly cylindrical and surfaces forming abutments machined square with the shaft, no fitting is necessary other than initial assembly. Once a ball race has begun to flake, it must be renewed immediately, and it is only necessary to press out the old race and substitute a complete new bearing to finish the job.



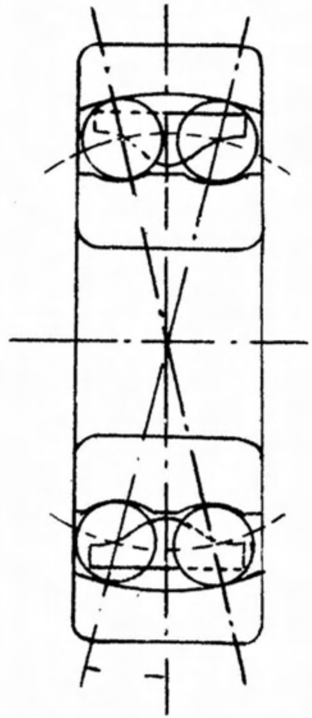
A BICYCLE TYPE BEARING

FIGURE 1.



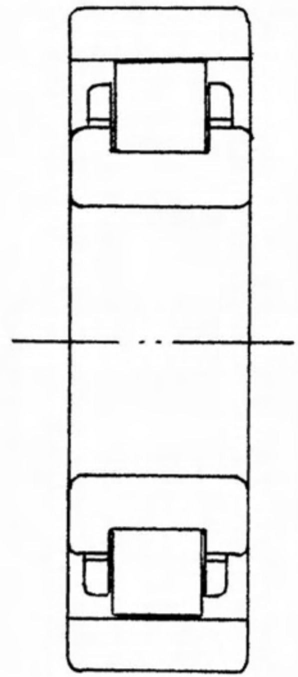
DEEP GROOVE BALL BEARING

FIGURE 2.



DOUBLE ROW SPHERICAL BEARING.

FIGURE 3.



HEAVY DUTY SHORT ROLLER BEARING

FIGURE 4.

Design, Materials, and Manufacture.—The determination of suitable scantlings for a bearing for given loading conditions is largely a matter of practical experience, and firms which specialise in this type of bearing publish tables from which it is possible to select a suitable bearing for the conditions obtaining.

Table I attached, gives typical examples from the tables published by two of the best-known makers, from which it can be seen that the load-carrying capacity of the bearing decreases as the revolutions increase.

Other things being equal, the total load which can be carried by a bearing depends upon the product of the number of balls in the race, and the square of their diameter.

The general design requirements for a good bearing are as follows :—

(1) Annular design for radial loads in preference to cups and cones. *See* Fig. 2.
(2) The use of alloy steels possessing supernormal qualities of toughness and hardness.
(3) Accuracy of dimensions to extremely fine limits in order that each ball should bear its proper share of the load.
(4) Uniformity of material and hardness from surface to centre of balls and races.
(5) Perfectly ground surfaces and an ultra fine mirror-like finish.

The loads on the balls of a bearing are not constant, but intermittent, and in consequence the balls and races are subject to fatigue stresses which are often sufficiently high to limit the life of the bearing; fatigue failure is indicated by flaking of the surface. Roller bearings having a larger area of contact are not subjected to such high stresses for similar loading and therefore have a greater resistance to fatigue than ball bearings, but, theoretically at least, have a higher frictional resistance, although the increase is very small if short rollers are used.

If a bearing is subjected to a simple radial load, a given ball will only bear the maximum load once during each passage round the shaft. If, on the other hand, the bearing is subjected to a simple axial load, the ball will bear the same maximum pressure throughout its passage round the shaft, assuming, of course, that the bearing is correctly lined up in its housing. Hence the safe loads for axial and radial application will differ to a considerable degree, and this difference has been fairly well established by extensive tests. If the thrust load is intermittent, it is generally assumed that bearings of the deep groove type will have the same carrying capacity for all directions of load. If, however, the thrust is steady, a maximum of about 30 per cent. of the rated radial load carrying capacity is allowed. The figure, of course, varies with the designs of different manufacturers, and is greater in a bearing

TABLE I.

Type of Bearing.	Makers.	Outside diameter.	Inside diameter.	Width.	N = 0.	Maximum Steady Load at N. Revolution per Min./Lbs.					
						100	500	1,000	2,000	3,000	4,000
		Ins.	Ins.	Ins.							
Single row deep groove radial ball	S.K.F. ..	4	$1\frac{5}{8}$	$\frac{15}{16}$	3,850	2,530	1,930	1,605	1,250	980	—
		7	3	$1\frac{9}{16}$	10,740	7,130	5,345	4,420	3,345	2,585	—
Single row deep groove radial ball	Hoffmann	4	$1\frac{5}{8}$	$\frac{15}{16}$	—	3,800	2,200	1,760	1,400	—	1,100
		7	3	$1\frac{9}{16}$	—	10,620	6,000	4,930	3,910	—	—
Single row deep groove radial	Hoffmann	12	6	$2\frac{1}{4}$	24,290	24,290	14,000	—	—	—	—
		$21\frac{1}{2}$	12	$3\frac{3}{4}$	59,000	59,000	—	—	—	—	—
Single row short type radial roller	S.K.F. ..	5	$2\frac{1}{4}$	$1\frac{1}{4}$	10,850	7,100	5,580	4,840	—	—	—
		7	3	$1\frac{9}{16}$	18,250	12,540	9,650	8,150	—	—	—
		12	6	$2\frac{1}{4}$	50,000	32,400	26,200	18,600	—	—	—
Single row short type radial roller	Hoffmann	Loads carried may be from 50 to 70 per cent. more than ball bearings of same dimensions									
Single Thrust	Hoffmann	$4\frac{5}{8}$	2	$2\frac{1}{4}$	22,400	6,410	3,000	2,020	—	—	—
		$9\frac{1}{8}$	4	$4\frac{1}{2}$	34 tons	18,130	8,000	—	—	—	—
		$13\frac{3}{4}$	6	$6\frac{3}{4}$	70 tons	33,280	—	—	—	—	—
Single Thrust	S.K.F. ..	$4\frac{5}{8}$	$2\frac{1}{2}$	2	23,900	6,200	3,700	2,675	1,700	—	—
		$9\frac{3}{8}$	5	4	93,200	24,300	13,600	—	—	—	—
		11	6	$4\frac{1}{2}$	118,000	31,400	17,400	—	—	—	—

with no filling slot than in one which has such a slot, as the latter tends to weaken the edge of the race.

The steel most generally employed is a carbon chromium steel to the following approximate specification :—

C	0·95–1·10
Mn	0·2–0·5
S	$\not>$ 0·03
P	$\not>$ 0·03
Cr	1·2–1·50.

Some manufacturers vary the chromium content according to the size of the ball, but there seems little justification for this if proper heat treatment is carried out.

Chromium steel is used because it possesses the property of permitting a high degree of penetration of hardening effect. As the carbon percentage is high it is most important to ensure that the free carbides are completely spheroidised.

For very large balls of 4-in. diameter and over, it is sometimes the practice to make them of carburising steel and case harden them. Whatever steel is used, it is absolutely essential that the slag inclusions are reduced to the minimum, and the standard set for this feature in the case of balls and rollers is probably the most exacting of all steel manufacture.

Small balls are cold pressed and larger balls hot forged, and for the former operation the steel should have an ultimate strength of less than 45 tons/in.^{2}

After rough grinding, working strains are relieved by a low temperature anneal at 750° C. Hardening is then carried out in automatic furnaces.

Balls are heated to 820° C. and allowed to remain at this temperature for a given time, in order that all the carbides may pass into solution. They are then quenched in water and the scale is removed.

Tempering is carried out in an electric furnace. A common tempering temperature is 125° C., but the temperature depends on the hardness required. They are then finish ground as described below.

Ball races are treated in a similar manner, and a Brinell hardness figure of 650-700 is obtained.

The ball races are rough turned to size in automatic lathes, and finished in capstan lathes. For cutting the race grooves, circular form cutters are used. After machining to grinding size, the races are heat treated as described above, and scale is removed by sand blasting. They are then ground to finished width, being inspected after each surface is completed. The general tolerance worked to in this dimension is 0·002 in.

The internal and external diameters are then ground. Centreless grinding is used for the latter operation, while for grinding the groove surface, an oscillating grinding wheel is employed, constrained to oscillate on a radius equal to that of the groove. Rubber-bonded wheels are used to obtain a special track finish.

Finally, the race tracks are polished to remove all traces of grinding scratches. This operation is possibly the most important of all, as it has been abundantly demonstrated that the higher the final finish obtained, the greater the endurance of the bearing, other things being equal.

The diametral tolerances worked to are of the order of 0·0005 in. and are measured by special plug and snap gauges. In assembling the races, the necessary working fits are obtained by selection.

The balls and rollers are cold pressed from chrome steel wire in automatic presses. They are ground to within a few thousandths of an inch of finished size and then heat treated.

Grinding is carried out in the case of balls between two revolving discs, one of which carries the grinding surface. By this means a perfect spherical contour is obtained. After heat treatment, the balls are precision ground to finished size and polished. A final inspection follows in which any surface cracks or soft spots are detected and defective balls rejected.

Balls are then graded in steps of 0·00008 in. This is ingeniously done by rolling the balls between inclined knife edges, set at a slight angle. The balls drop through at their correct diameter and are conveyed to bins by tubes.

The cages, whose sole function is to keep the balls equally spaced, are made of a soft material and their mass is reduced to a minimum. They may be either of pressed steel rivetted together in place, or of drilled bronze. A heavy cage tends to damp out noise, but this should not be necessary in a bearing with well-designed grooves.

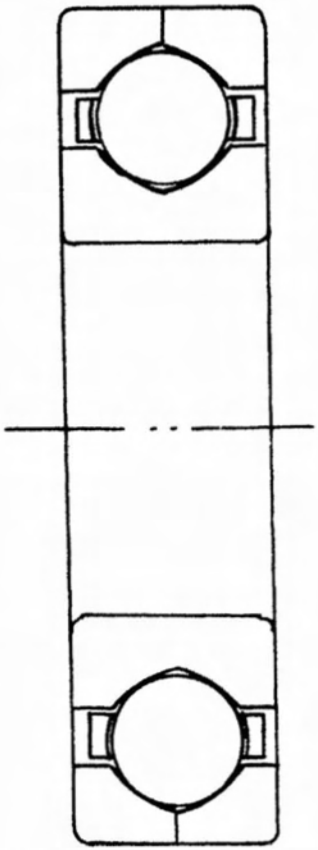
The control of all measurements during manufacture is based on the Johansson Universal Combination Gauge which permits of measurements within a limit of 0·000039 in.

The tolerances worked to are seen to be of very fine order compared with general engineering practice, but from a consideration of the foregoing paragraphs on the elements of design, it is evident that these fine limits are essential if even distribution of loading within the bearing is to be realised.

Service Applications.—Examples in Naval Service cover a large and varied field, including nearly all types of journal and thrust bearings.

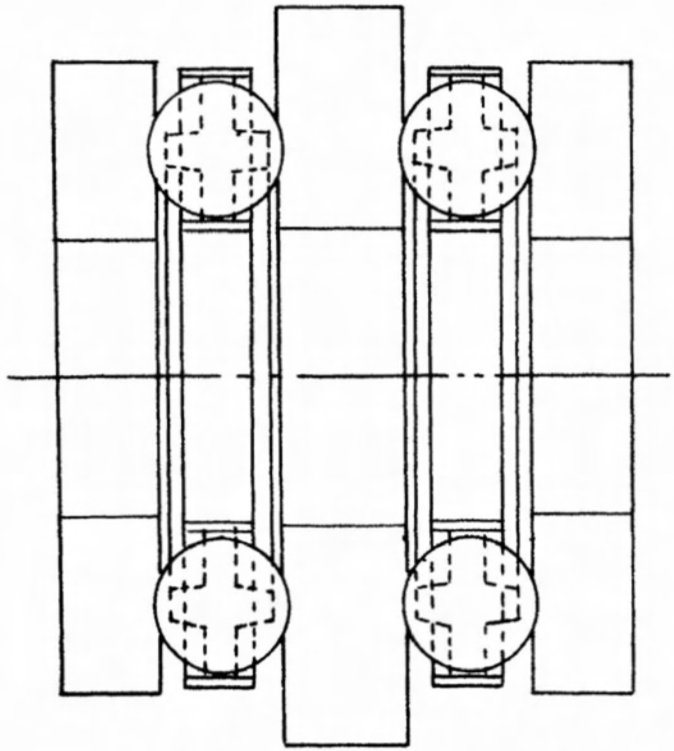
1. **Main Propelling Machinery of Submarines.**—Several thrust races are incorporated in the camshaft drive of submarine engines. As an example may be taken the thrust race on the layshaft.

This is of the double thrust type (Fig. 6) and is subjected to a maximum load of 1,000 lb. at 350 r.p.m. The race employed has a safe working load (as given by the manufacturers) of 1,700 lb. at



DUPLEX BALL BEARING

FIGURE 5



DOUBLE THRUST BALL BEARING.

FIGURE 6.

300 r.p.m. from which it may be seen that a factor of $1\frac{1}{2}$ on the specified load is allowed.

This factor will depend on the space available for mounting the bearing and may be reduced to 1 if there are limitations in this respect.

In the latest submarines electric turning gear is fitted and ball races are incorporated on the motor shaft, intermediate, and main worm shafts. That on the motor shaft is of the deep groove type carrying radial as well as axial load, while the supporting race of the main worm is a self-aligning roller to allow for the disengagement of the turning gear.

2. **Catapults.**—It has been found necessary to employ either roller or ball bearings in almost every case in modern catapults.

The weight and space considerations are of the first importance, and large loads have to be dealt with under these restrictions. In these examples the time of application of the load enters the question as a predominant factor. It must be remembered that the whole operation only occupies a little over one second and that the maximum loads are therefore only borne for a fraction of a second.

For the most part the bearings are of the heavy duty short roller type (Fig. 4) having rollers of length equal to the diameter. High speeds are encountered in some of the bearings, combined with heavy loads, which impose a severe strain on the bearings during the time of operation. All loads are pure radial loads, except in the case of manœuvring gear where single thrust races are employed to locate worm wheels.

The loads worked to for these bearings are in general considerably in excess of the specified loads given by the makers, and this course is warranted by the relatively short time of application of the load. In short, the service is the antithesis of continuous service on which the makers' figures are based.

Motor Boats.—Ball bearings are employed in the gear box and thrust block of many existing boats, and are being incorporated in all new designs. The bearing subjected to the greatest load in this connection is the thrust race. For this service in the latest designs a deep groove type ball race has been developed which has a circumferential split in the outer race, as shown in Fig. 5. This split, unlike previous split races, is situated at the centre of the race axially. Instead of providing two tracks, one on each race as is usual with a deep groove unsplit bearing, four tracks are provided, two on each race, the centre of the race in way of the joint being relieved. To ensure that the balls make contact with one track only on each race at any time, the manufacturers specify that the radial load should always be less than the thrust load. The carrying capacity of these duplex bearings, as they are termed, is approximately one-third greater than the corresponding deep groove journal bearing.

Self-aligning ball races and deep-groove journal bearings are employed in the epicyclic gear boxes fitted in motor boats. They are not heavily loaded, and serve chiefly as location bearings, enabling the whole assembly to be made smaller and more compact. A thrust race is also employed to take the thrust of the clutch-actuating mechanism in the same manner as in ordinary automobile practice.

Application to Electric Motors.—The advent of ball bearings in electrical machinery design has enabled manufacturers to obtain an increased mechanical efficiency, both at starting and during running. It has also been possible to employ a smaller air gap, as the concentricity of Stator and Rotor is maintained correctly throughout the life of the machine.

Trouble arising from over lubrication of journal bearings and leakage of oil has been eliminated.

In general, deep-groove ball bearings are employed to support the rotor. In vertical motors thrust races are necessary, and self-aligning ball races are often fitted where shafts have a long unsupported length, or with enclosed-type motors where temperature changes will cause appreciable relative axial displacement of the rotor and stator.

In naval service, the present policy is to employ ball or roller bearings in all electrical machinery except certain specified examples, notable among which are main dynamos, steering gear motors, and other motors subject by reason of their position in the ship to severe vibration.

When ball bearings are fitted, provision must be made for taking the thrust of the rotor with the machine tilted to an angle of $22\frac{1}{2}°$. This can be done either by using deep-groove bearings or a separate thrust race.

It is not possible to give any figures for the loads supported by ball bearings in electrical machinery. The necessary type and size of bearing has been evolved for each case from experience, and previous practice is always consulted in new designs.

Gun Mountings.—The most important application of ball and roller bearings in point of size is in the field of gun-mounting machinery. Here bearings are found varying in size from the 16-in. main roller race of over 30 ft. diameter down to minute ball races employed in fire-control instruments.

Roller races are employed in trunnion bearings and a large number are fitted in rotary hydraulic engines in connection with the gun-mounting machinery. Most of these races are of special design, and the material from which they are made and their heat treatment is varied in different cases to produce the required qualities.

Special tests are applied to ascertain the resistance of the balls and races against shock, and in this respect provision is made to obtain the maximum length of contact in roller races. This results in frictional increase, but as speeds are low this is accepted in order to obtain a lower load per unit length of roller.

Ball and roller bearings are also employed in variable-speed gears, centrifugal pumps, control and telegraph shafting and certain isolated cases where the small space available necessitates their adoption.

Upkeep of Ball and Roller Bearings.—The failure of ball or roller bearings is usually attributable to one of two causes :—

(*a*) To the formation of rust on the working surfaces.
(*b*) To the access of dust and grit.

In both cases the effects are similar, rapid wear results, giving rise to noise and vibration, and if the bearing is not renewed eventual complete breakdown by the fracture of the cage or race. The formation of rust usually takes place when the bearing is working in damp and confined spaces where dust and grit may gain access to bearings in open situations.

Most modern bearings are designed for grease lubrication, and an attempt is usually made to arrange that the grease originally packed into the race and thrown out when the machine is started will fill the housing and provide a seal to prevent dirt or moisture obtaining access to the bearing.

There is some shrinkage of the grease and bearings require to be re-charged from time to time, in addition to being well packed in the first instance ; the actual time interval between examinations which is allowable varies in different cases, and can only be ascertained by experience, but bearings liable to excessive damp, such as those adjacent to pumps, etc., should be examined and filled with grease at more frequent intervals than those in drier positions. Covers to keep out dust and dirt increase the life of the bearings considerably, especially in fan and workshop motors, where a large amount of dust is to be expected, and if the bearings are not fitted with suitable covers it saves money and labour to improvise and fit them whenever possible.