Rolling Contact Bearings With Special Reference To Oil Lifting Equipment

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The use of rolling contact bearings in oil lifting equipment is no novelty; but, at times, it is desirable to review the salient factors relating to the bearings themselves, and their selection, application, and maintenance. Although application and maintenance are the subjects of prime interest to the operator, some knowledge of the bearings is extremely useful in analysing bearing trouble.

Affecting the efficiency of the subsurface pump are two factors: the valve seat and the ball which seats against it. The ball must be able to resist the corrosive effect of the fluid and also the constant pounding against the seat, and to seat properly it must also be finished with sufficient accuracy.

To provide corrosion resistant qualities, the balls are made of stainless or similar materials. Generally, two types of stainless steel are most frequently found, namely AISI Type 440 and AISI Type 329. The principal ingredients in Type 440 are chromium, carbon, manganese and silicon added to iron. In the Type 329 the principal alloying elements are chromium, nickel, manganese, molybdenum, silicon and carbon. In each case the elements have been listed in the order of percentage of each used.

Type 440 stainless steel balls are relatively hard with Rockwell C readings of 60-63 for balls up to and including 1/2 in. Larger balls have a hardness of 55-60 Rockwell C. These hardness measurements are made on flat surfaces, but if the hardness reading is made on the balls directly, a somewhat lower reading will be obtained. The balls are available in various grades and the grade determines the accuracy to which the ball is finished. For balls generally used in valves, the diameter tolerance per ball may be as small as 00005 in. or as large as .0002 in. depending upon the grade to which it is manufactured. Balls made from Type 440 find their principal use in valve applications.

Type 329 stainless steel balls are not quite as hard as are those made from Type 440: the hardness is about 45-52 Rockwell C. These balls, being generally used in valve applications, are finished to a tolerance of .0002 in. in size range 5/8 in. to 2 in. and a tolerance of .0003 in. in the range of 2-1/8 in. to 2-1/2 in. Type 329 steel is corrosion resistant to a higher degree against a wider range of materials than Type 440 steel.

Information relating to the manufacturing tolerances is available (1).

BALL BEARINGS

The basic difference between the ball bearing and the roller bearing is in the guiding of the rolling element. If a ball is placed in the groove in the bearing ring, it is free to rotate about any of its many axis. In fact, in some bearing types, spinning of the ball may occur almost continually so that the axis about which the ball rotates is constantly changing. In the roller bearing, however, it is necessary that the axis of the roller at all times maintain a definite relation with respect to the center line or axis of the bearing, and which is the same as the axis of the shaft on which the bearing is mounted. To maintain the desired position of the roller in the bearing, the roller must be guided by flanges or rings which may be attached to the outer or inner ring of the bearing. When a guiding ring is used it is allowed to float between two rows of rollers which are further guided by the raceways themselves. Guiding of rollers will be referred to in connection with various types of roller bearings and also in connection with their lubrication.

Although ball bearings are not found in oil lifting they have general application as electric motors, centrifugal pumps, etc. Before 1900 several types of ball bearings were developed in Europe. To place the balls in the grooves of the inner and outer rings, various designs were used. In some of these, the balls were inserted through holes in the rings and raceways; in others, the balls were introduced through slots in the sides of the grooves. This arrangement is still used in some designs known as the filling slot type, (Fig. 1), and such bearings are useful where the load is principally

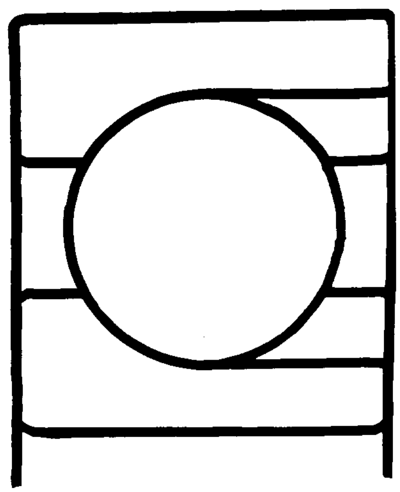

FIG. 1

radial in direction. However, if the thrust load, applied along the axis of the shaft, is substantial, there is a tendency for the balls to be forced against the side of the ball groove, and interference with the filling slot may cause noisy operation and early failure.

SINGLE ROW BALL BEARINGS

About 1903, Conrad invented a method of assembly for the single row ball bearing; this method permitted the balls to be introduced without the use of slots in the grooves. In the so-called Conrad assembly (Fig. 2) the inner ring outside diameter is placed against the bore of the outer ring, and the crescent shaped space remaining between the two rings is then filled with balls of the required size. To give the bearing as much capacity as possible, it will be found that the size and

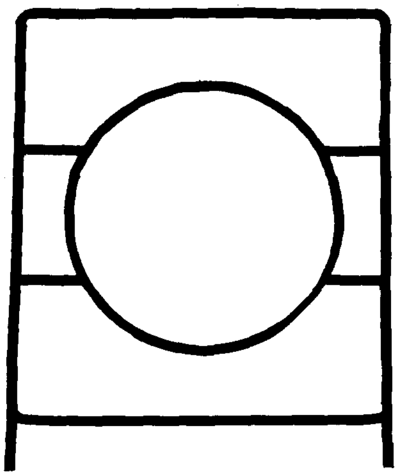

FIG. 2

number of balls are such that it will not be possible to insert the last ball, so to introduce this ball, the outer ring is distorted until it is slightly elliptical in shape. This distortion is accomplished by applyingforce along the diameter which passes through the contact between the inner and outer ring. Then while the outer ring is distorted, the last ball is introduced and the inner ring is moved to a position concentric with the outer ring. The cage or separator, which is generally in two halves, is then applied from each side and the halves are riveted or held together by prongs. Since the bearing does not have any interruption in the ball grooves, it may be used to carry radial load or thrust or any combination of these loads.

ANGULAR CONTACT BALL BEARINGS

To permit the single row ball bearing to carry heavier thrust loads at moderately high speeds, the angular contact type was designed (Fig. 3). In this bearing the line which joins the points at which the balls

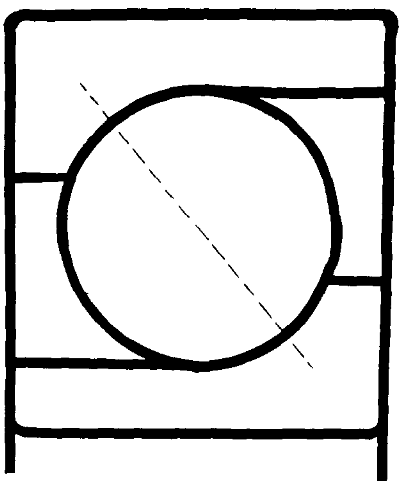

FIG. 3

contact the inner and outer rings is at an angle to the line drawn perpendicular to the shaft axis. Since the thrust load is supported more nearly in line with the direction of the force, the bearing has greater thrust capacity than the Conrad or filling slot single row type. In this arrangement, the contact angle may be any value, but where heavy duty is involved, it is about 30°-40°.

Angular contact ball bearings are frequently mounted side to side so that thrust load in both directions can be carried (Fig. 4). When these bearings are so mounted, it is most important that they be selected for such an

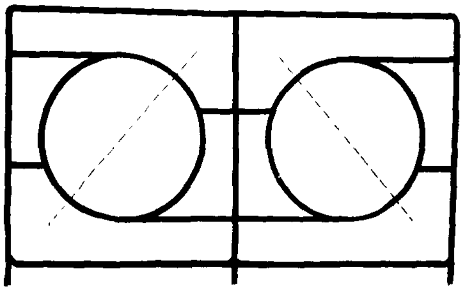

FIG. 4

arrangement. And it is necessary that the bearings be manufactured so that the sides which contact each other have the faces of the inner and outer rings flush. Such bearings carry a special suffix in the bearing number.

In the assembly of angular contact bearings, the outer ring is made with a high shoulder on one side of the groove and a very shallow shoulder on the other side. The inner ring is made similarly or sometimes with equal height shoulders, as in the Conrad assembly. When two bearings are mounted as a pair, either the heavy shoulders of the outer rings or the shallow shoulders of the outer rings may be adjacent. The former arrangement is commonly called the "back-to-back" mounting, while the latter arrangement is called the "face-to-face" arrangement.

There are certain precautions which must be observed in mounting double angular contact bearings. In the back-to-back arrangement the outer rings must be checked for squareness before the shaft locknut is tightened. This check can be accomplished by placing a scale across the two outer rings and at several points; and if light can be seen at any place under the scale, the outer rings are tapped lightly until they are square. In some designs, the housing can be used to square the outer rings: the housing is placed over the two bearings before tightening the locknut on the shaft.

When using the face-to-face arrangement, the outer rings of the bearings must be clamped endwise; otherwise, the rings tend to separate under load, and this separation will greatly reduce the bearing capacity, particularly if there is any radial load present. In view of the requirement just mentioned, the back-to-back design is perhaps more foolproof.

When thrust load only is present and the speed is low or moderate, the ball thrust bearing is sometimes used. In this type (Fig. 5) no radial load can be carried.

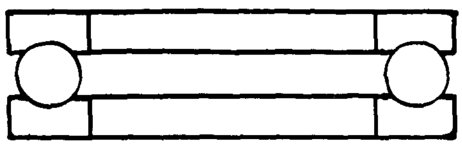

FIG. 5

One bearing ring is mounted on the shaft with a light interference fit, while the stationary ring should be mounted on the support, and there should be some clearance around its outside diameter. With ball thrust bearings it is necessary to provide other bearings to support the shaft radially.

ROLLER BEARINGS
CYLINDER ROLLER BEARINGS

The cylindrical roller bearing (Fig. 6) takes its name from the form of the roller. Theoretically, the roller is a cylinder, but, in practice, the roller is made with a very slight relief near its ends, and this relief prevents concentration of load near the end of the roller and the raceway. The guiding of the rollers is accomplished by flanges which may be integral with the inner or outer ring. In some designs, snap rings in grooves in the inner or outer ring or both are used to guide the rollers. The more accurately the rollers are guided, the more nearly the bearing will live up to its theoretical capacity and life. In a well-made cylindrical roller bearing, the flanges integral with the bearing rings, great care is exercised to see that the ends of the rollers are square with the axis of the roller. Too, the length of the roller and the distance between the flanges must be accurately controlled so the clearance between the roller ends and the flanges can be controlled. Excessive clearance at this point affects the guiding of the roller and permits it to skew.

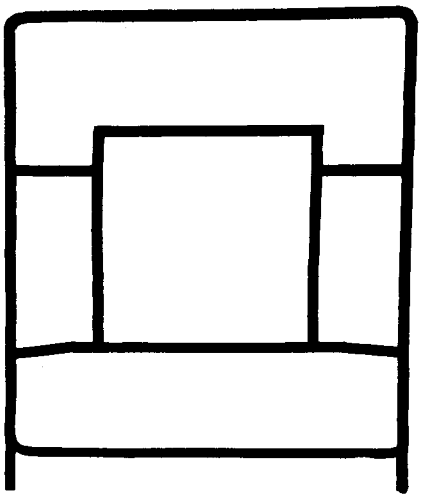

FIG. 6

The cylindrical roller bearing is designed to carry principally radial load. If thrust load is applied, rubbing between the roller ends and the guide flanges occurs with resultant heat. The thrust capacity is quite limited and the thrust load should be applied, preferably, only intermittently. However, one of the main advantages of this type bearing is that, because either the inner or the outer ring can be supplied without flanges, it can easily take care of axial movement of the shaft with respect to the housing. These bearings are frequently used in gear boxes when herringbone gears are used.

TAPERED ROLLER BEARING

In the tapered roller bearing (Fig. 7) the roller is a truncated cone. Since the load line between the roller and the other ring or cup and the load line between the roller and the inner ring or cone meet in an angle, the roller is forced against the flanges on the inner ring. Unlike the cylindrical roller bearing in which guiding is accomplished at both ends of the roller the tapered roller bearing guiding is accomplished entirely at the big end of the roller. These bearings can carry both radial and thrust loads; and as the load on the roller increases, the guiding force at the end of the roller also increases. The load at the end of the roller is carried on a sliding contact; hence the speed of the bearing must be reduced when thrust load predominates or excessive heating will occur.

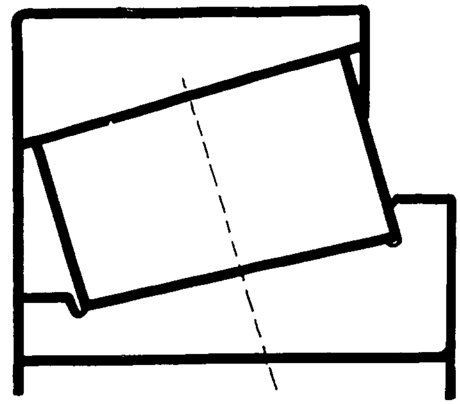

FIG. 7

Tapered roller bearings were developed before the turn of the century and have been extensively used to carry heavy loads at moderate speeds. These bearings are frequently found on the slow speed shaft of the gear box and on other shafts when helical gears are used.

At assembly, the outer rings or the inner rings are adjusted axially until the desired internal clearance is obtained. Since the tapered roller bearing is a separable type bearing, it must be adjusted when it is assembled; but when two bearings are used next to each other, they usually have a common outer ring or common inner ring and frequently the adjustment of the double bearing is permanently made at the factory. When the bearings are used singly, they must be mounted opposed against another bearing of the same type, but like all angular contact bearings, these bearings cannot be used singly to carry radial load. In a rolling contact bearing it is desirable to have at least half of the rolling elements under load. In the tapered roller bearing the adjustment should be made until the axial clearance between the two bearings is between .001 in. and .010 in. For heavy loads at low or moderate speeds, the smaller value should be used; for lighter loads at high speeds, the larger value may be used. These values are approximate guides; and, for specific applications, the bearing manufacturer should be consulted.

SPHERICAL ROLLER BEARING

The spherical roller (Fig. 8) bearing is named after the spherical surface in the outer ring on which the rollers run. The rollers are ground to a radius which is slightly shorter than the radius of the outer ring sphere; hence the load is concentrated toward the center of the roller. The inner ring roller groove is also ground with a radius to give rather close conformance of the roller to the raceway. The spherical roller bearing is a double angular contact type in which the lines of contact of the two roller sets meet at the geometric center of the bearing, and thereby make the design inherently self-aligning, one of the major advantages of

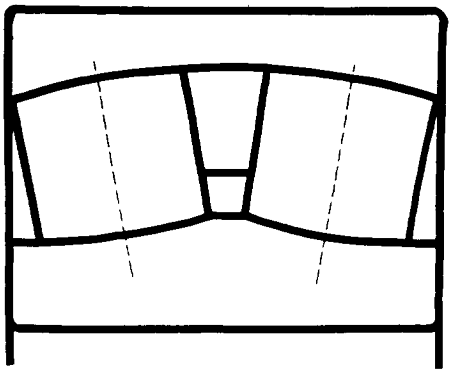

FIG. 8

this bearing type. As a result of this design, the bearing can carry heavy radial loads and moderate thrust loads.

In the original construction of the spherical roller bearing, an integral flange was used around the center of the inner ring. The lines drawn tangent to the contact points between rollers and raceways at the inner and outer rings, formed a cone. As a result of this design, the rollers were forced against the center flange on the inner ring and this flange provided the roller guide. In the latest design of the sperical roller bearing, lines drawn tangent to the rollers at the contacts with the inner and outer rings, are parallel. Furthermore, the center guide flange on the inner rings is replaced by a floating ring which corrects the tendency of the roller to skew The improved design results in increased radial capacity since the length of the roller has been increased and the roller to raceway conformance has also been made closer.

Spherical roller bearings have been extensively used where heavy loads must be carried at moderate speeds, and sometimes at higher speeds, with adequate provision for lubrication. These bearings have been used on the slow speed shaft of gear boxes, at pitman locations,. and at various locations on the pumping structure.

In another design of self-aligning roller bearing, the inner ring is a portion of a sphere and the rollers have a concave shape to conform to the inner ring. The outer ring consists of two parts each with a convex form to conform to the rollers. Since the outer ring is in two halves or two separate rings, some adjustment is required at assembly unless the bearing is obtained from the manufacturer with a spacer to provide required adjustment.

NEEDLE BEARING

The needle bearing (Fig. 9) is sometimes regarded as a roller bearing similar to the cylindrical roller bearing. There is, however, a rather important difference. In the cylindrical roller bearing of good design,

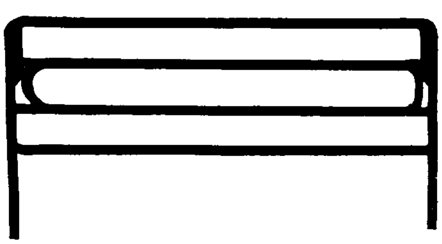

FIG. 9

the length of the roller compared to the roller diameter. should have a ratio not exceeding 2 to 2.5. If the roller becomes excessively long with respect to the diameter, it becomes increasingly difficult to properly guide the rolling elements. In the needle bearing a ratio of length of roller to diameter may be as much as 10. Since it is impossible to adequately guide such a roller by flanges contacting the ends, any guiding must be accomplished by the cage or separator. This type of bearing was originally developed as a high speed-light load carrier. In theory, the rapid rotation of the shaft would carry lubricant into the spaces between the rollers and these wedges of lubricant would act like the oil film in a sleeve bearing. Should a particularly heavy shock load occur, the load would momentarily be carried on the rollers. This action was a decided advantage over the sleeve bearing in which an overload would cause breakdown of the oilfilm and metal to metal contact and severe wear.

The needle bearing is particularly useful where the shaft oscillates in the bearing. Under this condition, the rollers do not have an opportunity to become badly skewed, which might be the case if the shaft rotated in one direction only and the load were moderately heavy. Hence, the needle bearing has been used at the top of the samson post and at other locations where the loads are of an oscillating nature.

CAPACITY AND LIFE CHARACTERISTICS

The capacity and life characteristics of rolling contact bearings were established over 30 years ago. It was discovered that if a group of bearings, at least 30 in number, and made to the same tolerances from the same material, were run under identical conditions, a definite variation of life would be noted. The bearings would not fail at the same life but some would have a relatively short life while others would have a rather long life. It would be highly desirable to make bearings so that they would all fail at about the same time if run under the same load and the same speed. The bearing industry has not yet discovered how such bearings can be made.

To select bearings which will give a reasonable life, it is customary to limit the load and speed so that 90 per cent of the bearings run under the conditions in question, will exceed the minimum acceptable life. To determine the minimum acceptable life, it is necessary to consider the so-called capacity of the bearing and the speed of operation. It is also necessary to have a good estimate of the load applied to the bearing.

To determine the capacity, industry standards have been developed by The Anti-Friction Bearing Manufacturers Ass'n., Inc., with offices at 60 East 42 St., New York 17, N. Y. Established was a formula which considers the material from which the bearing is made, the accuracy of manufacture, the number of rows of rolling elements, the angle of contact, and the number of rolling elements, their effective diameter and their effective length. When these factors are combined, the so-called specific dynamic capacity of the bearing is obtained. This value is that load which the bearing can carry and result in a life of one million rotations. Thus 10 per cent of the bearings may fail before one million rotations and 90 per cent would run beyond one million rotations.

However, if the applied load is less than the specific dynamic capacity figure, the bearing will run longer than one million rotations. For ball bearings the increase in life will be in inverse ratio to the third power or cube of the ratio of capacity divided by load. In other words, if the load on a ball bearing is cut in half, the life will be increased eight times. In the case of roller bearings the power in question is the 10/3 power or as the load is cut in half, the bearing life is increased 10 times.

The variation of life with speed may be regarded as a direct proportion, within the usual limits of speed encountered in usual applications. In other words, if the speed is doubled, the life will be cut in half.

The catalogues of the bearing manufacturers are useful in dealing with problems in which it is necessary to determine the bearing life under specific conditions of load and speed.

THE GEAR BOX

The Gear Box is perhaps the most important link in long satisfactory performance of the pumping unit. From the bearing point of view, the problem is not as simple as it would seem to be on first inspection. The gear box is usually a double reduction unit and the gears are of the herringbone or the helical type. When herringbone gears are used, the bearings on the slow speed shaft are of a type which will locate the shaft axially. Through the gears, the other shafts are located axially so the bearings on the intermediate and high speed shafts can be of the cylindrical roller type or some other type which does not axially restrain the shafts. When helical gears are used, it is necessary to restrain each shaft axially since each shaft is subjected to thrust loads.

To make a complete analysis of the bearing loads in the gear box, a rather lengthy and complicated calculation is required, partly because the torque reverses during part of the pumping cycle. However, for design purposes, it is possible to arrive at a per cent of the peak torque of the unit and use this for calculating the bearing loads. The actual percentage which is used depends upon the type of counterbalance which is employed: roughly speaking, a value of about 65 per cent of the peak torque can be used. If a dynamometer card has been obtained for a specific well, a rather accurate estimate of bearing loads can be made. (For an illustration of the use of the dynamometer card and the resulting torque curve refer to API Standard 11E 9th edition, January, 1961, p. 19). Since the gear box driving the pumping unit frequently operates 24 hr per day and long satisfactory life is desired, the bearings should be selected for a minimum life of 40,000-50,000 hr.

The bearings on the pumping structure, that is, at the top of the samson post and at the equalizer, are subjected to oscillating motion only. For bearings operating under such conditions, it is desirable to select those having a relatively large number of rolling elements of rather small diameter. Such a bearing has a smaller space between the rollers and hence the individual roller path is relatively long and the roller paths of adjacent rollers will overlap, a desirable condition. At the pitman location, a self aligning type roller bearing is desirable to accomodate any angularity between the crank pin and the plane of the pitman movement.

SUCCESSFUL BEARING PERFORMANCE BEARING MOUNTING: SHAFT AND HOUSING

One of the first questions which the designer is confronted with is the manner in which the bearing should be mounted on the shaft and in the housing. There is a simple rule to follow when deciding if the inner ring should have an interference fit on the shaft or a loose fit. First, one must consider the nature of the load and the rotation of the shaft. If the nature of the load is to remain fixed in direction, even though it varies in magnitude, it will be referred to as a stationary load. If the shaft is rotating, it can be said that it is its nature to rotate. If the nature of the load and the nature of the shaft rotation are opposite, then an interference fit of the bearing ring under consideration is required, with the mating member, either shaft or housing. If the nature of the load movement and the nature of the bearing ring movement are the same, then a slightly loose fit of the ring and the mating member is permissible.

To illustrate the foregoing rule, the usual shaft is rotating, as in the gear box, and a load which is constantly in the same direction, as from a gear. In this case the nature of the shaft is to rotate, but the nature of the load is to remain stationary. Under this combination it is necessary to use an interference fit between the inner ring and the shaft. If one considers the outer ring of the same bearing, he finds that the nature of the movement of the outer ring is to be stationary and the nature of the load is also to be stationary. Since the natures are alike, the outer ring can be mounted with a slightly loose fit in the housing. To determine the magnitude of the interference fit or the amount of looseness, reference should be made to the catalogues of the bearing manufacturers. Generally, for small bearings under light load, as in an electric motor, the inner ring carrying a heavy load the interference fit of the inner ring may be several thousands of an inch.

In manufacturing parts, such as shaft seats for bearings, care must be used to finish the seat within the dimension and tolerance specified. If the seat is oversize, the inner ring will be expanded excessively and the clearance between the rolling elements and the raceways may be removed entirely. This expansion is one of the most frequent causes of overheating in rolling contact bearings. It must be remembered that in small ball bearings, the radial clearance or looseness within the bearing may be as small as .0005 in.; hence excessive stretching of the inner ring cannot be allowed. Even in larger roller bearings, where the clearance within the bearing may be as much as .005 in., it is possible to remove all the clearance by mounting on an oversize shaft.

Damage can also be done by excessive out-of-roundness of the housing bore. This situation results in the outer ring being pinched in the housing at assembly or the outer ring being improperly supported and the load being concentrated on a small sector of the outer ring; premature failure is caused.

The second important point in the mounting of bearings is the question of axial location of the shaft through the bearings. When two or more bearings of the self-contained type are mounted on a continuous shaft, only one of the bearings may be located or held endwise at its outer ring. If two bearings are held endwise in the housing support, there is no possibility of the shaft expanding when it warms up during the operation. Therefore, a thrust load will be imposed on both bearings and overheating will soon occur. If the housing bores are undersize or out of round and the outer ring is pinched, the same effect can be obtained even though only one bearing is held axially at its outer ring. In this case the "free" bearing cannot move axially in the housing and overheating will occur.

When the bearings are of the separable type such as tapered roller bearings, it is necessary to locate both bearings at one side of the outer rings; and during the assembly, the proper adjustment must be made as described in the paragraph dealing with tapered roller bearings. If the spacing between tapered roller bearings is rather long and there is a possibility of the shaft running quite warm, suitable allowance must be made in the initial adjustment, to permit the required expansion during the warm up period.

BEARING LUBRICATION

To insure long trouble free operation the bearings must be properly lubricated. It may be true that rolling contact bearings do not depend upon an oil film, as do sleeve bearings, but some lubricant is required. The lubricant carries away the heat generated during operation; and it lubricates the sliding contacts between the rollers and the guide flanges and the cages or separator. Under relatively heavy loads, a film of lubricant between the rollers and the raceways in the load zone is essential.

Generally speaking, oil is the preferred lubricant since the quantity in the housing can be easily controlled and since the oil can be specified and checked to meet the conditions of load and speed and bearing type which prevail. The quantity should be controlled so that the stationary oil level is approximately at the center of the lowermost rolling element. If a higher level is maintained, overheating and loss of oil will result. If a lower oil level is permitted, the operating temperature will be lower, but there is great danger of the bearing running dry with further reduction in level; then rapid failure will occur.

Oils used in rolling contact bearings under average conditions should be of a good quality mineral oil prepared for industrial use rather than for automotive use. The oil must be clean and stored in a clean container until ready for use. The viscosity of the oil should not be less than 70 SSU (Saybolt Universal Viscosimeter) at the operating temperature for ball bearings and 100 SSU for roller bearings. The oil should have a rust and oxidation inhibitor, but other additives are seldom required. Sometimes a compromise must be made in selecting the oil, e.g., for use in the gear box bearings. In this case, the oil is usually selected to properly lubricate the gears and from experience it is known that such an oil will satisfactorily lubricate the bearings.

Grease is frequently used for lubricating bearings where the speed is relatively slow and long periods between addition of lubricant are desired. Greases are specified according to two important characteristics: the consistency of the grease and the viscosity of the oil in the grease. Grease is basically a mixture of a soap and an oil, and the oil is the important factor when rolling contact bearings are involved. The measure of the consistency is the number known as the Worked Penetration of the grease: the higher the number, the softer the grease. Commonly, greases are referred to by numbers such as No. 1 or No. 2, etc. According to the NLGI standards, a No. 1 grease will have a worked penetration of 310-340, while a No. 2 grease will have a worked penetration of 265-295.

For ball bearings, such as electric motor bearings, greases with a worked penetration of around 275 or a No. 2 grease is commonly used. Sometimes where higher temperatures are present, a No. 3 grease with worked penetration of approximately 235 is used. When dealing with roller bearings one must recall the prime difference between the roller bearing and the ball bearing: in the roller bearing one must concern himself with the guide flange for the rollers and the resulting sliding contact. Thus, the grease must be sufficiently soft to enter the bearing and lubricate the contact between the guide flange and the roller end. To insure this lubrication, it is desirable to use a grease with a worked penetration of at least 300, or a No. 1 grease. It is also important that the oil in the grease be of sufficiently high viscosity to meet the requirements previously mentioned, namely, the viscosity should be at least 100 at the operating temperature. Greases can be obtained which have a worked penetration around 300 in which the oil has a rather high viscosity.

When applying grease, great care should be exercised to see that the grease is kept in a clean container and that the application is made with clean hands or tools. One should never leave the grease can open or use a dirty stick in application. The space between the balls or rollers should be packed by hand and the housing should be only one third to one half full, for excessive amounts of grease will cause churning in the housing and resultant overheating which may be so severe that the oil becomes separated from the soap base, after which the dried soap acts like an abrasive and the bearing is quickly worn out. If the grease is applied with a gun, the drain fitting on the housing should preferably be on the side of the housing opposite from the filling fitting. As the grease is introduced, the drain fitting should be removed and grease applied until it comes out of the drain. The bearing should then be operated for some time until the excess grease is allowed to drain from the housing; then the drain plug is replaced.

From time to time it is desirable to clean old grease from the housing and replace it with new grease. To accomplish this exchange the old grease can be flushed from the housing by adding a solvent such as kerosene or very light oil to the grease and running the bearing until the softened grease can be drained from the housing. However, if the grease has been permitted to become very hard in the housing, it will probably be necessary to disassemble the unit and clean the bearing by soaking it in a solvent such as carbon tetrachloride or by immersing it in a hot solution of one of the floor cleaners such as Greasolv. When cleaning bearings as described, they may be blown out with an air hose but be sure that the water is first blown out of the air line. Under no circumstances should the inner ring of the bearing be held in the hand and the outer ring be permitted to spin under the influence of the air stream. This procedure can damage a bearing which has small particles of dirt remaining in it and there is danger of damaging the fingers by the spinning bearing.

APPLICATION OF BEARINGS: INTERFERENCE FIT

To apply bearings to a shaft where an interference fit is involved, two methods may be used.

For smaller bearings, the bearing can be applied in a press. The shaft is supported vertically and the bearing is pressed on its seat by applying force to the inner ring only. To do this it is preferable to slip a tube over the shaft so that the tube contacts the inner ring only. Force should never be applied through the outer ring when pressing the inner ring on a shaft, and all parts around the press, etc. must be clean before the work is started.

The second method, which is usually used when larger bearings are involved, is heating of the bearing and slipping it onto the shaft seat. A suitable tank should be provided with facilities for applying heat to the bottom, and inside the tank, which must be thoroughly cleaned, are placed some wood or fiber blocks upon which the bearing will be placed to prevent the bearing from coming in contact directly with the hot floor of the tank. To permit removal from the hot oil a wire is placed around the bearing. The oil can be any clean oil which can be heated to 200 F. A thermometer is used to prevent overheating of the bearing, and usually a temperature of 200 F is ample to expand the inner ring so that it can be easily slipped onto the shaft. The bearing must never be heated over 250 F; and the bearing must be left in the oil long enough for it to become thoroughly warmed to the desired temperature. When applying the bearing to the shaft the inner ring is squared by tapping slightly around the inner ring with a wooden or brass bar; and as soon as the inner ring is square, it should slide into place. Heavy blows should be avoided since these only cock the inner ring; then there is danger of the bearing cooling and seating before it is in the desired position.

BEARING FAILURES

Bearing failures will eventually occur since there is a limit to the time a bearing will operate before fatigue of the raceways or the rolling elements begins. When early failures or repeated failures in one location are experienced, it is necessary to determine the cause. If the bearing can be removed before it has failed so badly that it has become mutilated, much useful information can be obtained from an inspection of the ball or roller track on the raceways. As soon as a bearing operates for a short time, even under light load, there develops on the raceways, "the ball or roller track" which has a dull appearance compared with the bright appearance of the original grinding of the raceway.

In the usual application, where the load is fixed in direction and the inner ring rotates, the track will extend all around the inner ring and will be of uniform width. In the corresponding outer ring, the load will be concentrated over one sector of the outer ring path while the rest of the past has relatively little marking. In the case of the pitman bearing one sector of the inner ring usually carries a heavier load than does the remainder of the inner ring so one would expect to find a wider load track over the heavier loaded sector. Knowledge of the normal load track is desirable so, when abnormalities are found, their cause can be explained.

Mention has been previously made of the effect of pinching the outer ring in the housing because of improper machining or misapplication of a split housing cap. This effect will be indicated in the track by a widening of the path at the points where pinching occurs.

When normal fatigue begins, the first manifestation is a small crack or breaking of the surface. This crack may be so small that it can only be seen under a glass, but once this has started, it will progress along the raceway or the rolling element. A bearing in which this has happened is unfit for further service.

When a bearing indicates excessive looseness between the rolling elements and the raceways, but is otherwise in good condition, it has been subjected to excessive dirt or abrasive matter in the lubricant. Such bearings are unfit for further service since failure of the cage may be imminent and the capacity of the bearing is reduced by the presence of the excessive looseness.

An excellent booklet with illustrations of different types of failures may be obtained from the American Society of Lubrication Engineers. Failures should also be discussed with the bearing representatives and it is desirable to save some good examples of failures so that an intelligent opinion can be obtained.

Rolling contact bearings, properly selected, applied and lubricated are capable of giving almost any desired life, with a minimum amount of maintenance expense.

BIBLIOGRAPHY

(1) Catalog, Atlas Ball Division, SKF Industries, Inc.