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ASLE TRANSACTIONS

Bearing Performance Investigations Through Speed Ratio Measurements

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The performance of ball bearings and their lubricants were studied by monitoring the speed ratio (the ratio of ball pass to shaft speed). Bearing speed ratio measurements were found to provide new insights into bearing and lubricant performance. This paper presents a summary of the development of the theoretical speed ratio equations, describes a unique measurement technique for measuring and monitoring the speed ratio, and presents several applications. Highlighted are experiments revealing the tendency for greases to promote varying degrees of slippage and skidding at the ball-raceway contacts.

INTRODUCTION

Ball bearing rings are of relatively thin cross section, and they deform elastically when the bearing is loaded. Under a pure thrust load, the rings expand equally at each ball position. The contouring of these deformations on the bearing outer ring can be seen with the aid of holographic interferometry, as shown in Fig. 1.

During bearing operation, displacement waves, which are generated by those deformations, travel about the periphery of the rings at the ball passage frequencies and create the bands of fretting corrosion that are sometimes found in the center of bearing rings (Fig. 2). The technique described here for evaluating bearing and lubricant performance is based on the fact that those displacement waves provide an exact measure of the ball pass frequency.

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Ball pass frequency is a function of bearing geometries, clearances, loads and rotational speeds. In any given application, bearing geometry is fixed. Therefore, changes in ball passage are independent of geometry. By dividing the ball pass frequency by the shaft rotational frequency, a bearing speed ratio parameter (BSR) is created which is independent of shaft speed.

THEORETICAL DEVELOPMENT

The orbital velocity of a ball center about the bearing axis is determined by (referring to Fig. 3) the rolling radius R, the ball diameter d, the bearing contact angle β, and the speed of the rotating ring. The rolling radius at any contact angle β (neglecting contact deflections) is given by the following:

$$R = \frac{E - d \cos \beta}{2} \quad [1]$$

At point i, the ball-inner race contact, the linear velocity of a rotating inner ring is:

$$V_i = R\ \omega_i \quad [2]$$

At point O, the ball-outer race contact, the linear velocity is zero for a stationary outer ring. Therefore, the linear velocity of the ball center is:

$$V_b = V_i/2 = R\ \omega_i/2 \quad [3]$$

The orbital velocity of the ball is its linear velocity divided by the radius of motion, or

$$\omega_b = V_b/(E/2) \quad [4]$$

NOMENCLATURE

- A = axial deflection factor
- B = total curvature
- BSR = bearing speed ratio
- d = ball diameter
- E = pitch diameter
- n = number of balls
- P_d = diametral clearance
- R = radius to ball-inner race contact
- V_b = linear ball velocity
- V_i = linear velocity of ball-inner race contact
- β = contact angle
- ω_b = orbital velocity of ball
- ω_i = rotational velocity of inner ring

Fig. 1—Holographic interferogram showing contour of outer ring deformations.

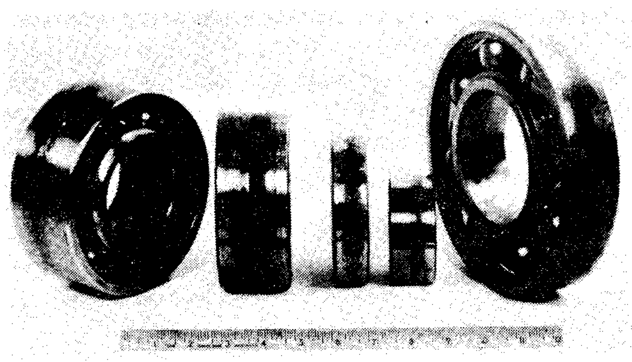

Fig. 2—Bearings with fretted rings

Combining Eqs. [3] and [4] gives:

$$\omega_b = \frac{R\ \omega_i}{E} \quad [5]$$

Substituting Eq. [1] yields:

$$\omega_b = \frac{\omega_i}{2}\left(1 - \frac{d \cos \beta}{E}\right) \quad [6]$$

The speed ratio of a ball bearing (BSR) is defined as the angular velocity of the ball train, multiplied by the number of balls and divided by the angular velocity of the rotating (inner) ring. For a thrust loaded bearing, assuming the individual ball speed equals the retainer speed, the speed ratio is:

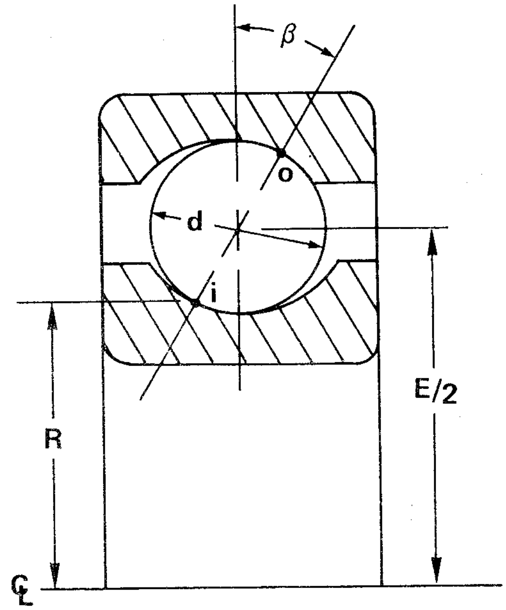

Fig. 3—Bearing geometries illustrated

$$BSR = \frac{n\ \omega_b}{\omega_i} \quad [7]$$

Substitution of the expressions for ω_b gives the final expression for the speed ratio in thrust load:

$$BSR = \frac{n}{2}\left[1 - \frac{d \cos \beta}{E}\right] \quad [8]$$

The determination of BSR is considerably simplified when the bearing is supporting a pure axial load. In that case, normal ball loading is constant about the pitch circle, and, therefore, ball and retainer velocities are practically identical at all times. Furthermore, Jones (*1*) has provided equations and charts from which the operating contact angle is readily determined. The procedure which is valid for the low speed case where contact angles are equal at both races, is as follows:

1. Compute the free contact angle from

$$\beta_o = \cos^{-1}\left[2\frac{Bd - P_d}{2\,Bd}\right] \quad [9]$$

2. Calculate the operating contact angle from

$$\beta_1 = \tan^{-1}\left[\frac{A + \sin \beta_o}{\cos \beta_o}\right] \quad [10]$$

where A, which is a function of load, is determined from chart 59 (*1*).

3. Calculate speed ratio

$$BSR = \frac{n}{2}\left[1 - \frac{d \cos \beta_1}{E}\right] \quad [11]$$

BSR INSTRUMENTATION

The instrumentation required for measuring *BSR*, illustrated in Fig. 4, comprises: a sensor to monitor the shaft rotational frequency, a sensor mounted perpendicular to the bearing outer ring to monitor outer ring displacements, a band pass filter and a counter. The band pass filter is sometimes required to separate rotational frequency harmonics from the outer ball pass signal. The magnitude of bearing outer ring deformations over each ball is approximately 100×10^{-6} inches pk-pk at 600-lb thrust (2). To date, fiber optics proximity sensors have been used for measuring these minute displacements. The installation of one of the fiber optic probes is shown in Fig. 5.

EXPERIMENTS

Bearing System Performance

Electric motor bearing systems are commonly designed as "fixed-free" arrangements; that is, one bearing is fixed or clamped axially, while the other bearing is free to float axially in its housing to compensate for thermal expansions of the shaft. Often, the floating bearing is preloaded with a wave-spring washer so as to minimize ball skidding. When a machine is being assembled, bearings can stick or get hung up in their housings, if the fits are too tight or if the bearing gets misaligned. When this happens to the floating bearing, thermal expansions can degrade system performance and cause early bearing failure.

It can be determined whether or not a bearing is floating by observing its *BSR* during a startup (thermal transient). If a bearing is floating and is loaded by a preload spring, its *BSR* will be unaffected by thermal shaft growths and should, therefore, remain constant. On the other hand, if the floating bearing is jammed, the *BSR* of both bearings in the system will vary as a thermal transient is experienced.

In a laboratory test, a bearing which was designed to float was loosely fitted in its housing and *BSR* recorded during the thermal transient which followed a startup. The duration of the thermal transient was approximately one hour, as is seen from the bearing temperature versus time curve in Fig. 6. Also in that figure, the curve of outer ring motion represents the axial movement of the floating bearing outer ring in its housing. This was measured by a dial indicator mounted on the housing. The shape of this curve shows that the bearing did float in its housing as the system experienced the thermal transient, which is an indication that the bearing thrust load remained constant throughout the experiment. Referring now to the bearing speed ratio curve, two observations can be made. First, the bearing speed ratio settled close to the design value in the steady state following the thermal transient. This is an indication of a properly installed bearing and properly operating bearing system. Secondly, there was a dip in the speed ratio

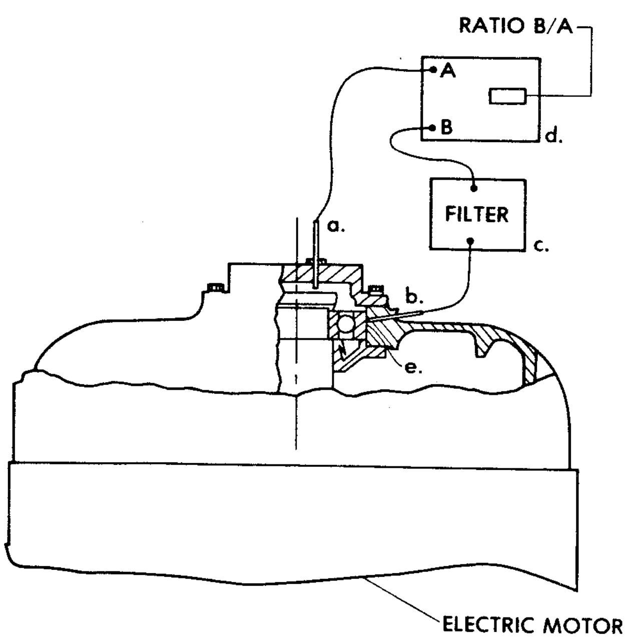

Fig. 4—Speed ratio instrumentation

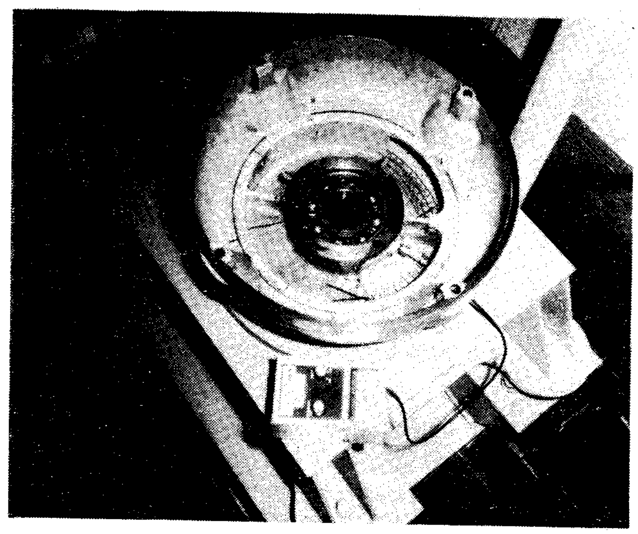

Fig. 5—Installation of fiber optics probe in bearing housing

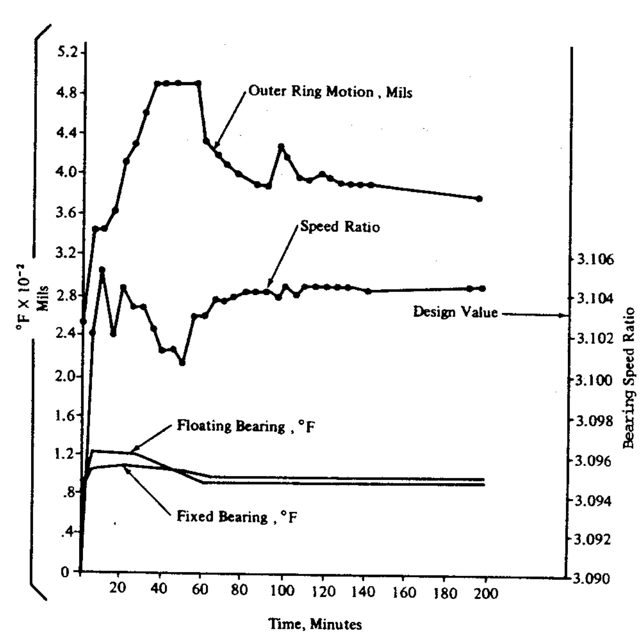

Fig. 6—Bearing speed ratio: Loose bearing-housing fit

curve in the transient state. This dip was a result of the reduction of bearing internal clearance created by the generation of heat at the ball raceway contacts which caused the bearing to expand from within. The speed ratio is proportional to the bearing clearance and load. The observed changes of bearing speed ratio in the presence of a constant bearing load indicated the original bearing clearance was reduced by four ten-thousandths of an inch during the thermal transient.

The floating bearing was removed and replaced by one that had a tight housing fit. Data were again recorded during the thermal transient which followed the startup. This time, the duration of this transient was approximately four and one-half hours. Referring to the two displacement curves of Fig. 7, it was observed that the outer ring motion did not follow along with the shaft growth. It was stuck in its housing. Consequently, as the shaft expanded against the stuck bearing ring, the bearing load was increased. Referring to the bearing speed ratio curve, the *BSR* varied considerably during the thermal transient, which suggested that the bearing was not floating. Secondly, the *BSR* did not settle to the design value, which is an indication of improper bearing loading.

From the observed changes in *BSR*, it was calculated that the bearing load cycled from 120 to 950 lb and then settled to 250 lb – some 500 lb below the design load.

Ball Skidding Tests

Electric motors with grease-lubricated bearings will sometimes suddenly squeal or howl. These audible noises can develop after months or years of operation, or after only a minute or two. In some instances, this is diagnosed as a failure and the bearings are replaced while, at other times, fresh grease is added to the bearings to quiet them, at least temporarily. The work reported herein was undertaken to develop an understanding of the factors that encourage ball skidding, for skidding or sliding at the ball-raceway contacts leads to objectionable noises and to shortened bearing life.

A schematic diagram of the apparatus is shown in Fig. 8. Bearing speed ratios were measured for the upper bearing which was subjected to thrust loads ranging from a downward force of 350 lb to an upward force of 1600 lb, while operating at 3600 rpm.

Test Bearing Description: size—315, deep groove
retainer—stamped steel
grade—NT-3, super precision
$n = 8$
$d = 1.0625''$
$E = 4.6260''$
$\beta_o = 8.31°$

Oil Lubrication—The BSR performance of a bearing with oil lubrication was obtained to provide a check on the theoretical predictions. The upper motor bearing was equipped with a circulating oil system (2-pint capacity). *BSR* was measured with a full reservoir (2 pints) and with the reservoir empty. For the latter case, only enough oil to keep the ball and raceway surfaces coated (~1/2 oz) was applied to the bearing. The *BSR* performance of the test bearing with oil lubrication is presented in Fig. 9. Measured *BSR*'s were close to

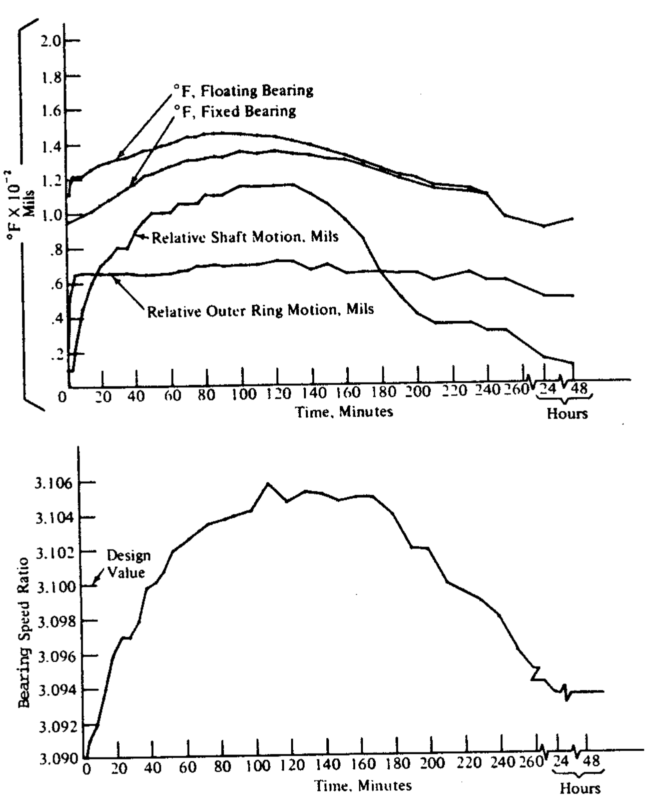

Fig. 7—Bearing speed ratio: Tight bearing-housing fit

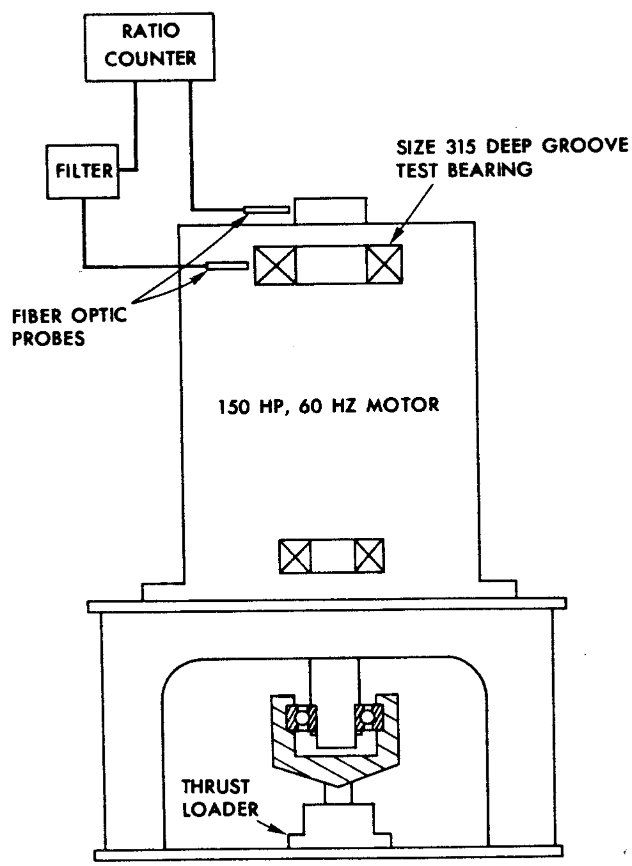

Fig. 8—Schematic diagram of apparatus

the predicted values, with little difference between the two conditions evaluated, thereby supporting the validity of the theory.

Grease Lubrication—Packing Study—Tests were conducted with grease 1 to determine the *BSR* performance at various levels of grease pack. Figure 10 shows the measured *BSR* for a 25 percent (25-gram) pack. These results were similar to those obtained with oil lubrication, excepting that a slight noise came from the bearing as zero thrust was approached.

Through additional testing, wherein the quantity of the grease pack was varied from 2 1/2 percent to the 100 percent, it was established that the *BSR* performance will sometimes deviate significantly from the predicted curve. Illustrated in Fig. 11 is the initial startup performance of grease 1 with three quantities of grease pack: 2 1/2, 25, and 100 grams (100 percent). It was found that there is a range of grease packs where the *BSR* performance is in agreement with the predicted performance, and the 25 percent pack is within that range. When excess amounts of grease were packed into the bearing, the measured *BSR* fell below the theoretical curve, as is illustrated for the 100 percent pack. When small quantities of grease were packed into the bearing, high *BSR*'s exceeding the predicted values were recorded. It was also noted that loud squealing noises always accompanied the rise of *BSR*, whereas there was only a subtle change of pitch of motor noise to accompany decreases in *BSR*.

Grease Lubrication—Load Cycle Study—Additional tests or thrust load cycles were conducted with grease 1 to evaluate how running time and load cycles affect the *BSR* performance. The initial startup data with 100 grams of grease were recorded from −350 to +1600 lb of thrust, which took approximately 30 minutes. Thrust load was then reset at −350 lb. After approximately one hour of operation, thrust load was again gradually increased to +1600 lb as *BSR*'s were recorded. These thrust load cycles were repeated eight times, the results of which are shown in Fig. 12. It was

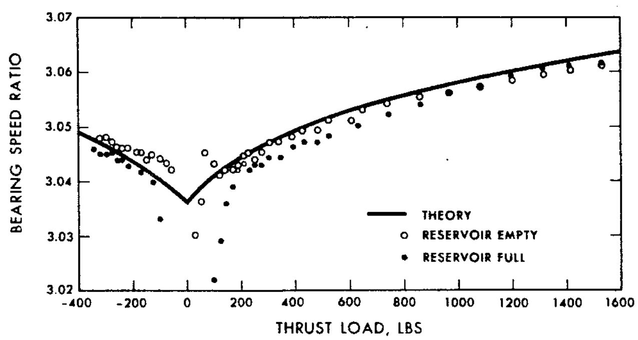

Fig. 9—Bearing speed ratio with oil lubrication (2190 TEP)

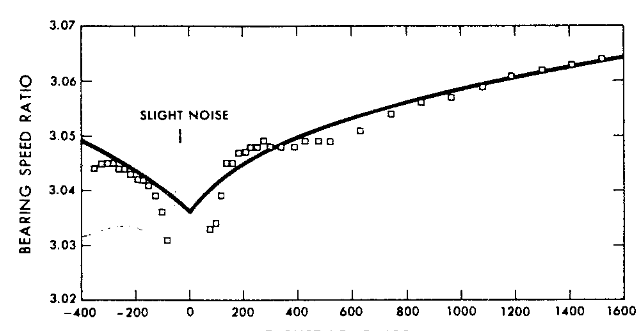

Fig. 10—Grease 1, 25% pack

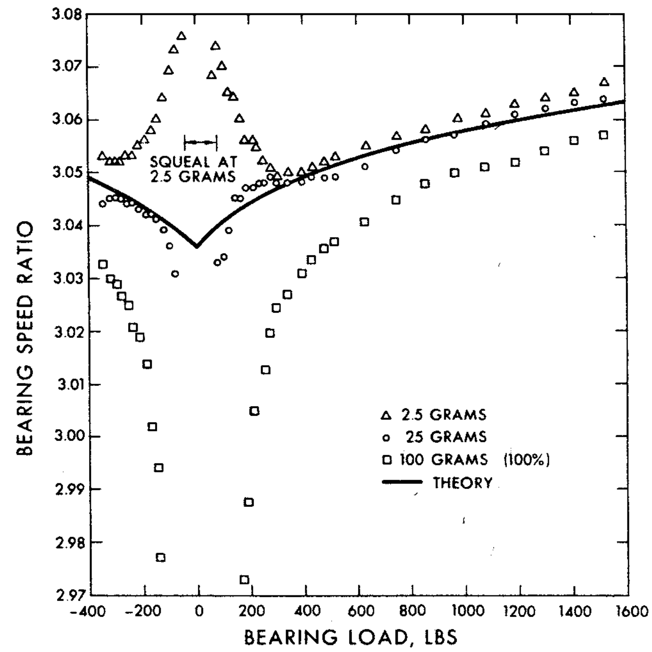

Fig. 11—Initial startup performance of grease 1 at three levels of grease pack.

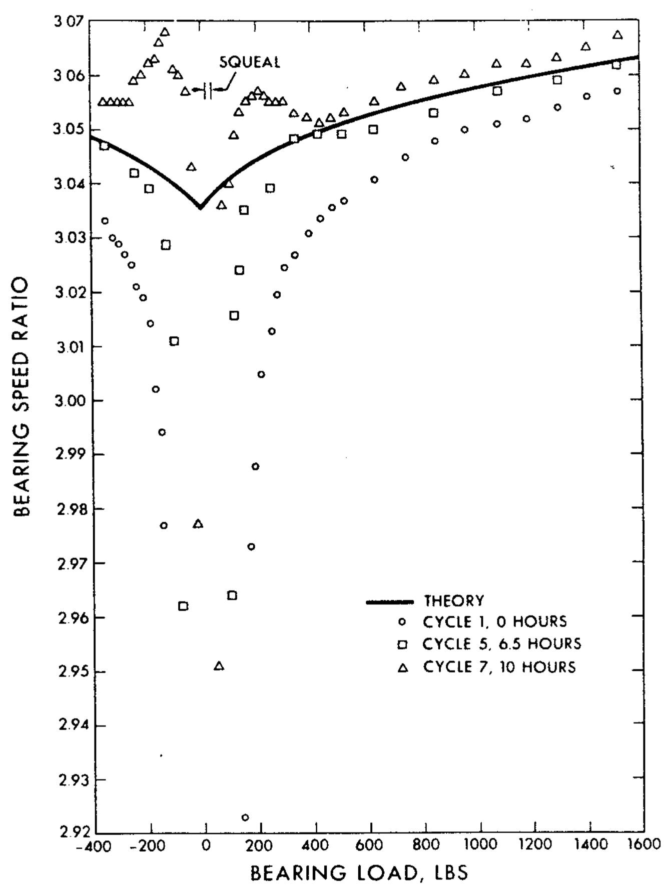

Fig. 12—Load cycle performance of grease 1 at 100 grams

found that, with each successive cycle, there was a shifting upward of the *BSR* curve. After seven load cycles and only 10 hours of running time, the bearing started squealing.

Ball Skidding and the BSR Curve—The *BSR* theory does not consider the effects of slippage at the ball-raceway contacts, and, therefore, the theoretical curve represents the condition where rolling is always present at those contacts. When there is slippage at the inner race contacts, the ball train will slow down, and the *BSR* readings will fall below the predicted value. That is exactly what occurred with the 100-gram grease pack shown in Fig. 11. Below 800-lb thrust, the *BSR* does not parallel the theory curve. Instead, it drops increasingly further below it, as zero thrust is approached. It is suspected that this type of behavior is indicative of ball skidding or journaling at the inner race contacts, even though there were no audible squealing noises. That inner race skidding occurred at very light loads was not surprising. However, it is interesting that the data suggest sliding occurs with as much bearing load as 800 lb.

Observation of the ball train speed on an oscilloscope showed the train to be rotating at a constant speed at most times. In other words, what was observed and what is illustrated in Fig. 11 was not a random stick-slip motion, but rather a continual and constant rate of slip. It is believed such slip arises when the summation of the drag forces on the balls (created by the viscous shearing resistance of the grease) and the ball-cage forces at the leading edge of the ball pocket create a moment on the balls about the outer race contact, which exceeds the traction moment at the inner race contacts.

Conversely, the ball train speed will increase and *BSR* values will rise when balls slip on a stationary outer ring. This condition arises with marginal lubrication when centrifugal forces trap the lubricant at the outer race contacts, thereby starving the inner race and ball pocket contacts. Inner race traction then can exceed outer race traction with sufficient magnitude to cause outer ring slip.

Inner ring starvation can occur with an initial marginal grease pack, or it can develop in a bearing that was initially properly lubricated. For instance, referring to the 2 1/2-gram data in Fig. 11, it is seen that *BSR* begins to rise above the theory at 300-lb thrust and peaks at about zero thrust. It is also noted that extremely loud squealing noises emanated from the motor as zero thrust load was crossed. Such noises also suggest ball skidding occurred.

Attempts were made to observe the ball skidding by means of vibration analysis. An accelerometer was stud-mounted to the bearing housing, and 1/10 octave band frequency charts were recorded. There were broadband vibration level increases whenever squealing was audible; however, it was impossible to detect ball skidding when there was no audible squealing.

Referring to Fig. 12 which shows the effect of load cycling, it is seen that inner race skidding occurred initially, as evidenced by the low *BSR*'s. However, as load cycles were added, grease was worked out of the ball track and the *BSR* performance changed from inner race to outer race skidding by the seventh load cycle. This characteristic of grease 1 to quickly channel and produce outer race skidding was found to be repeatable and consistent.

There is one more characteristic of the *BSR* curve that needs interpretation, and that is the high load portion of the curve. It was seen in Figs. 11 and 12 that some of the *BSR* data are shifted parallel to the theory. Such parallel shifting is interpreted as a change in bearing clearance which offsets the measured *BSR* from the predicted values. For instance, again referring to the 100-gram data in Fig. 11, it is seen that at high thrust loads (800-1600 lb) the *BSR* is parallel to the theory curve, although shifted downward. This downward shift was caused by a decrease in bearing clearance, likely brought on by either temperature gradients produced by churning of the grease, or by an increased contact film thickness, or a combination of both.

Grease Lubrication—Comparison of Several Lubricants—Both the initial startup and the load cycle performance were evaluated for two additional greases. A comparison of the initial startup performance for all three greases and one oil is given in Fig. 13. A 100 percent grease pack was used for all the greases, and the oil reservoir was filled for the oil run.

Notice, first of all at the high loads, that all data fall below the theory line. The downward parallel shift is caused by a reduction of internal bearing clearance. At 1000-lb thrust, the theoretical speed ratio values are given for 0.000- and 0.001-inch internal clearance. For all three greases, the internal clearance was apparently reduced to zero during the startup transient, whereas with oil lubrication, a less drastic reduction in clearance was encountered.

Below 1000-lb thrust, the performance of the greases differ. Inner race-ball skidding begins with grease 1 at approximately 800 lb and becomes quite pronounced below 300 lb. For grease 2 and for oil, inner race-ball skidding occurs between − 100 and + 150 lb of thrust, a much smaller range than is seen for grease 1. It is interesting that grease 2, which performs much the same as oil at light

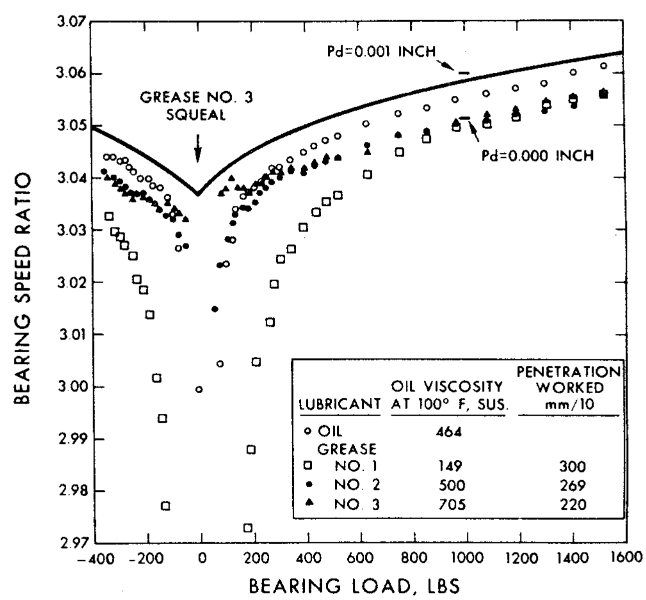

Fig. 13—Comparison of lubricant performance at initial startup

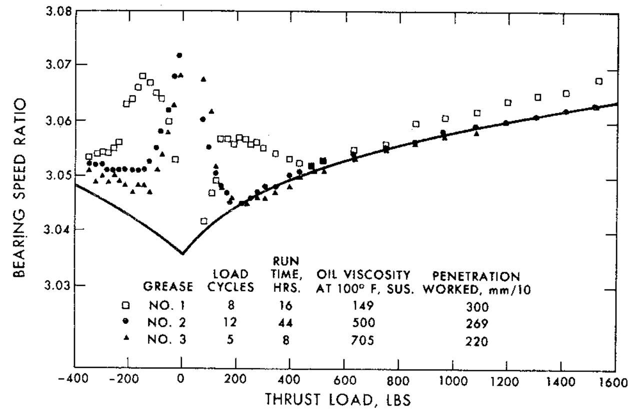

Fig. 14—Comparison of grease performance after the onset of squealing

loads, has a base oil viscosity which is closest to that of the oil used. Grease 3 did not induce inner race-ball skidding as did greases 1 and 2. However, the shape of the curve below 200 lb and the squeal that was heard at 0-lb thrust suggest that outer race-ball skidding was imminent with grease 3 at the beginning of its first run.

All three greases were put through the load cycling test, and with all three greases, similar patterns were observed whereby the initial *BSR* performance degraded to outer race-ball skidding. Figure 14 is a comparison of the performance of the three greases after the onset of squealing occurred. In contrast to Fig. 13, all the data have shifted upward to the theory line or above it. This is so because the bearing was close to being in thermal equilibrium when the data were recorded, and also because there was outer race-ball skidding. The number of load cycles and the total running time of the greases are shown in the figure, along with two characteristics of each grease. It had been seen with the initial startup performance that grease 1 exhibits skidding over a wider load range than either of the other greases. Similarly, in this case, skidding begins at 450 lb with grease 1 and at 200 lb for greases 2 and 3. While the performance of greases 2 and 3 were almost identical after squealing began, it is noted that noise was first heard with grease 2 on the eleventh load cycle at 42 hours, whereas noise was first heard with grease 3 on its first load cycle. Therefore, of the three greases evaluated, grease 2 performed with the least amount of skidding for the longest time.

SUMMARY

This paper has demonstrated the utility of bearing speed ratio measurements. It has been shown how they can be used to inspect newly assembled equipment for proper bearing loads and fits. It has also been shown that *BSR* measurements provide a sensitive technique for measuring ball skidding and for discriminating between inner and outer race skidding. Because the *BSR* is directly relatable to bearing loads and clearances, it provides a view of the internal operation of a bearing, which should prove to be a valuable aid in many areas of bearing research and performance monitoring.

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DISCUSSION

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Mr. Philips' paper is a welcome contribution to at least two problem areas in the theory of the application of rolling contact bearings to rotating machinery. The first problem area is the provision of a method to determine that the bearings is an assembled machine have, in fact, been installed in a manner to allow the bearings to develop their fatigue limited life. The second problem relates to the contact slip between the rolling elements and the rings. The slip is probably a function of ball-cage clearance, angular alignment, lubrication, as well as speed and load. References (*A1*)-(*A6*) attached are papers that are related to this problem.

Mr. Philips' work demonstrates that contact slip is present at speeds of only 3600 rpm even at significant ball loads. This point is not evident in the references cited where speeds an order of magnitude greater are considered. The extension of Mr. Philips' work to correlate the contact film thickness and ball-cage force to the experimentally determined slip would be very helpful. The estimation of the ball-cage force is an especially difficult problem. Can he comment on this point?

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(*A6*) Kodnir, D. S. and Sawvin, L. L., "Determination of Tangential Loads and Coefficients of Friction for Surfaces Working under Contact-Hydrodynamic Conditions." Same as Ref. (*A5*).

AUTHORS' CLOSURE

Mr. Smith points out that slip was observed at significant ball loads for the relatively low speed of 3600 rpm. This is a very real problem with naval machinery wherein reversal of thrust loads brings bearing operation through the zero load zone and, therefore, through the maximum skid zone.

The author does not feel that ball-cage forces are directly relatable to the observed slip. The phenomenon reported here concerns gross slippage of the entire set of rolling elements of a thrust loaded bearing. Inner and outer race traction create a force couple that imparts motion to the train of rolling elements. Drag forces from the lubricant and the retainer generate moments which oppose the direction of the traction couple. Slippage of the ball train occurs when the drag moments exceed the traction moments, or if the traction forces become unbalanced due to lubrication difficulties. The normal ball-cage forces under these conditions are thought to be much less than they would be for a radially loaded bearing. Slippage in pure thrust load is then primarily a function of traction imbalance, which relates to the contact film thickness.