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EFFECTS OF SURFACE REMOVAL ON ROLLING-ELEMENT FATIGUE

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The Lundberg-Palmgren equation was modified to show the effect on rolling-element fatigue life of removing by grinding a portion of the stressed volume of the raceways of a rolling-element bearing. Results of this analysis show that depending on the amount of material removed, and depending on the initial running time of the bearing when material removal occurs, the 10-percent life of the reground bearings ranges from 74 to 100 percent of the 10-percent life of a brand new bearing. Three bearing types were selected for testing. A total of 250 bearings were reground. Of this number, 30 bearings from each type were endurance tested to 1600 hr. No bearing failure occurred related to material removal. Two bearing failures occurred due to defective rolling elements and were typical of those which may occur in new bearings.

INTRODUCTION

The last four decades have seen a significant increase in the severity of applications in which rolling-element bearings are expected to function reliably and with long life. Rolling-element bearings are now required to operate at much higher speeds and, to a lesser extent, higher temperatures than in the early 1940's. The increased speed and temperature requirements originated principally with the advent of the aircraft gas turbine engine. Its development, coupled with the appearance of a variety of high-speed turbine-driven machines, has resulted in a wide range of rolling-element bearing requirements for main-shaft, accessory, and transmission applications.

Classical rolling-element fatigue which is of subsurface origin has been considered the prime life limiting factor for rolling-element bearings although actually less than 10 percent of them fail by fatigue. With proper design, handling, installation, lubrication, and system cleanliness, a rolling-element bearing will eventually fail by fatigue. Because fatigue results from material weaknesses, research to improve material quality has been a continuing activity. The remaining 90 percent of the failures are due to causes such as lubricant flow interruption, lubricant contamination, lubricant deterioration, excessive dirt ingestion, improper bearing installation, incorrect mounting fits, mishandling of bearings prior to installation, installing a contaminated bearing, manufacturing defects, ring growth in service, and corrosion.

Nonmetallic inclusions are one cause of classical rolling-element fatigue (1-5). Basic inclusion types include sulfides, aluminates, silicates, and globular oxides. These inclusions may act as stress raisers similar to notches in tension and compression specimens or in rotating beam specimens. Incipient cracks emanate from these inclusions (Fig. 1), enlarge and propagate under repeated stresses forming a network of cracks which form into a fatigue spall or pit (Fig. 2). In general, the cracks propagate in a plane approximately 45° to the normal; that is, they appear to be in the plane of maximum shearing stress. In addition to nonmetallic inclusions, large carbides can act as stress raisers and nucleate fatigue cracks (6,7).

One method for increasing rolling-element bearing life, reliability, and load capacity is to eliminate or reduce nonmetallic inclusions, entrapped gases, and trace elements. Melting steel in a vacuum provides large life improvements (8-10). Double vacuum-melted bearing steel, now commercially available, is processed with the first heat being vacuum induction melted (VIM). The material is subsequently vacuum arc remelted (VAR). The result is a material with marked reductions in nonmetallic inclusions, gas content, and trace impurities (11). Tests with 120-mm bore ball bearings made from VIM-VAR AISI M-50 steel produced fatigue lives at least seven times that achieved by vacuum arc remelted steel (12). Hence, the probability of subsurface fatigue can be greatly minimized within current time between overhaul (TBO) intervals and bearing steel state-of-the-art.

Failure by the other modes enumerated above are for the most part nonpredictable and tend to be surface as opposed to subsurface originated. In general, these failures due to surface originated defects, occur much earlier than those failures due to classical rolling-element fatigue (13-15). As a result, in aircraft engine and transmission applications, a large number of bearings are discarded at overhaul or during periodic maintenance.

It has been a practice in commercial and military aircraft application that bearings removed at overhaul be refurbished. Bearings are disassembled, cleaned, and visually and dimensionally inspected. If no major imperfections are found, the bearings are reassembled, lubricated, and packaged for further service.

In some cases new rolling-elements are inserted (16).

Bearing regrinding or restoration represents a logical extension of the practice of bearing refurbishment. Restoration entails grinding the races and other critical surfaces of used bearings to their original characteristics and dimensions. Grinding can remove raceway imperfections to significant depths below the surface. In addition, this stock removal is accomplished by grinding races on the same machines and with the same controls that are used in the manufacture of new bearing raceways. The result is geometrical accuracy and surface finished identical to new bearing raceways (17).

The objectives of the work reported herein were: (1) to determine analytically the additional life potential of refurbished and reground rolling-element bearings; (2) determine the endurance characteristics of reground bearings under simulated operating conditions; and (3) determine post-test condition of the reground bearings.

SYMBOLS

a	semiwidth of Hertzian contact ellipse
c	exponent
e	Weibull slope
F	probability of failure
h	exponent
L_{10}	10-percent life of brand new bearing
L_{50}	50-percent life of brand new bearing
l	length of rolling track, m (in)
q	maximum Hertzian contact stress, N/m^2 (psi)
R_{10}	10-percent life of refurbished and restored bearing
S	probability of survival
V	stressed volume, m^3 ($in.^3$)
x	fraction of stress volume removed
z	depth under surface, m (in.)
z_0	depth to maximum orthogonal reversing shear stress, m (in.)
η	number of stress cycles
	lubricant film parameter h/σ
σ	composite surface finish, $\sqrt{\sigma_1^2 + \sigma_2^2}$, μm (μin.)
σ_1,σ_2	surface finish of raceways and rolling-elements, respectively, μm (μin.)
τ	subsurface orthogonal reversing shear stress, N/m^2 (psi)
τ_0	maximum value of τ, N/m^2 (psi)

Subscripts

n	new material
r	refurbished and restored bearings
t	time of restoration
u	used material
0	corresponding to $\eta_r = 0$

RESTORATION PROCEDURE

When a bearing raceway is damaged by fatigue spalling, it is not considered for restoration by grinding. If, however, there is superficial damage to the bearing raceways, caused by dirt or fatigue debris from the rolling elements, raceways can often be restored by grinding. In general superficial damage extends <0.05 mm (0.002 in.) from the surface.

The process of bearing restoration by grinding was first reported in References (17) and (18). Rejected bearings are disassembled, the components are visually inspected, and the hardnesses of the bearing races are measured. The components that are determined to be restorable are dimensionally inspected. Where necessary, the bearing faces, bores, and outer diameters are ground and either nickel or chrome plated to a thickness that will allow the surfaces to be reground to the original blueprint dimensions. Both inner and outer raceways are ground to a depth of at least 0.05 mm (0.002 in.) but not more than 0.15 mm (0.006 in.), which removes all superficial damage and a large portion of the fatigue damaged stressed volume. The surface is finished to its original blueprint specification. The bearing is then refitted with new rolling elements having a diameter equal to the diameter of the elements previously contained in the bearing plus twice the depth of regrinding. The bearing separator is stripped of its silver plating, where applicable, inspected for cracks, and replated. The new rolling elements are placed within the separator and the bearing is reassembled.

For the cylindrical roller bearings the procedure is the same except that the roller length as well as the roller diameter are increased by a value of twice the depth of regrinding.

For ball bearings the effective race curvature is the same as the original dimensions within significant mathematical values. The original values of contact angle, resting angle, and radial clearance are the same. Although the restored bearing contains oversize balls and oversize raceways, the total effective geometry of the bearing has not been changed, and consequently, the contact stress level will be essentially identical to that of the original bearing.

FAILURE THEORY

The accepted method for calculating rolling-element bearing fatigue life is based on the Lundberg-Palmgren life formula (19-21). This formula describes the functional relations among the probability of survival, the number of stress cycles to failure, stress, stressed volume, and depth of occurrence of the maximum orthogonal reversing shear stress. The statistical dispersion in life satisfies the Weibull distribution as follows:

$$\log \frac{1}{S} \propto \frac{\tau_0^c n^e}{z_0^h} V \tag{1}$$

where $V = 2\,alz_0$

In the present case a comparison of the life of restored or refurbished bearings to the life of brand new bearings is required. For the purpose of comparing the failure distributions of the two groups, a constant of proportionality is not required in equation (1).

The basic failure theory will now be applied to analyzing the reliability of refurbished and restored bearings (22). Let it be assumed that when refurbished and restored bearings are put back into service they are run under the original service conditions. The maximum value of the orthogonal reversing shear stress is unchanged, but its location is shifted deeper into the original bearing material according to the amount of material removed during grinding. (See Fig. 3.) Therefore, the stressed volume of a restored bearing is composed of a newly stressed portion and an older portion with a history of stress cycles on it. The probability of survival of the restored bearing is equal to the product of the probabilities of survival of the new material with that of the used material.

$$S_r = S_n S_u \tag{2}$$

The removal of a material layer of the stressed volume in the fractional amount x exposes a new layer of material. The probability of survival for this layer is based on equation (1):

$$S_n = \exp\left[-x\left(\frac{n_r}{L_{10}}\right)^e \ln\frac{1}{0.9}\right] \tag{3}$$

where L_{10} is the 10-percent life of the original bearing and n_r is the additional number of stress cycles after restoration.

Next, consider the used material portion of the stressed volume. If the cycles of stress accumulated before restoration are denoted by n_t, then the following equation gives the probability of survival for the used portion of the material in the restored bearing:

$$S = \exp\left[-(1-x)\left(\frac{n_t + n_r}{L_{10}}\right)^e \ln\frac{1}{0.9}\right] \tag{4}$$

The probability of survival of the used material is S_0 when $n_r = 0$ (i.e., as soon as the bearing is started in operation again). However, $S_0 < 1$ according to equation (4). This means that there is a finite probability of failure F_0 immediately on starting the restored bearing:

$$F_0 = 1 - S_0$$

The physical reasons for this are simply that the material has already endured n_t stress cycles and that some of the bearings have been damaged. However, it must be assumed that such bearings will be detected during inspection and scrapped. The normalized probability of failure, which includes the effect of having discarded the bearings expected to immediately fail due to incipient cracks, is given by the following equation:

$$F_u = \frac{F - F_0}{S_0} \tag{6}$$

The expressions

$$S_u = 1 - F_u \tag{7a}$$

$$S = 1 - F \tag{7b}$$

and equations (5) and (6) may be used to write the normalized probability of survival for the used material as follows:

$$S_u = \frac{S}{S_0} \tag{8}$$

From equations (2) to (4) and (8) the probability of survival for the restored bearing is written as a function of the time at which restoration occurs and fraction of stressed volume removed.

$$S_r = \exp\left(\left(\ln\frac{1}{0.9}\right)\left\{(x-1)\left[\left(\frac{n_t + n_r}{L_{10}}\right)^e - \left(\frac{n_t}{L_{10}}\right)^e\right] - x\left(\frac{n_r}{L_{10}}\right)^e\right\}\right) \tag{9}$$

Equation (9) gives the probability of survival that would be expected if an endurance test were run on a group of restored bearings.

RESTORED BEARINGS AND PROCEDURE

Bearing regrinding

Bearings rejected for reuse in application were disassembled into its component parts. These components were visually inspected, and the hardness of the bearing races were measured. The bearing components were either put aside for regrinding or scrapped.

Those components determined to be restorable were dimensionally inspected. Where necessary, the bearing faces, bores, and outer diameters were ground and either nickel or chrome plated to a thickness that will allow the surfaces to be reground to the original print dimensions.

Both inner and outer raceways were ground to a depth not exceeding the maximum depth of the maximum resolved shearing stress under their maximum loaded condition but not less than 0.05 mm (0.002 in.). The surfaces finish was maintained to its original print specification. The bearing was then refitted with new rolling elements of a diameter equal to the diameter of the elements previously contained in the bearing plus twice the depth of regrinding.

For the ball bearings the effective race curvature was identical to the original dimensions within significant mathematical values. The original values of contact angles, resting angle, and radial clearance remained unchanged. Although the restored bearing contained oversize balls and oversize raceways, the total effective geometry of the bearing had not been changed, and consequently, the contact stress level and calculated bearing life of the restored bearing were essentially identical to that of the original bearing. The bearing separator was stripped of its silver plating, where applicable, inspected for cracks and replated. The new, oversized rolling elements were placed within the separator, and the bearing was reassembled.

For cylindrical roller bearings the procedure as outlined above was the same with the exception that the roller length as well as the roller diameter were increased by a value twice of the depth of regrinding.

Test Bearings

Three bearing types were selected for regrinding and testing. The bearings were selected based upon their high replacement rates during maintenance and overhaul. The bearings are identified in Table 1 together with their specifications.

The 210-size (50-mm bore) split inner-ring ball bearing is shown in Fig. 4(a). These bearings are made from AISI 52100 steel by a single manufacturer. The bearing cage or separator which is made from silver-plated bronze is of a one-piece design and is inner-land riding. These are mainshaft bearings from the front compressor position of a gas turbine engine. Engine operating conditions for the bearing include a speed of 24 000 rpm and a thrust load of 2002 N (450 lb). Bearing oil-in temperature in the engine is approximately 366 K (200 °F) and oil-out temperature is between 394 to 422 K (250 to 300 °F).

The 111-size (55-mm borc) cylindrical roller bearings shown in Fig. 4(b) are manufactured by two separate manufacturers. The design of each manufacturer is sufficiently different whereby interchangability of the bearing components is not possible. The bearings are manufactured from AISI M-50 steel. The bearing cage or separator is a one-piece, silver-plated steel, inner-land riding design. These bearings are from a gas turbine engine rear compressor position. Operating loads are nominal for this bearing. They are considered to be under a light radial load. The maximum bearing speed is 24 000 rpm. Bearing oil-in temperature in the engine is approximately 366 K (200 °F) and oil-out temperature is approximately 450 K (350 °F).

The third bearing type was the triplex ball bearing set shown in Fig. 4(c). These are 7216-size (80-mm bore) angular-contact ball bearings which are mounted on the input bevel gear pinion shaft of a helicopter transmission. The bearings are manufactured from AISI M-50. They have a one-piece, silver-plated steel, inner-race riding cage. Operating conditions in the helicopter for the triplex set include a radial load of 14 123 N (3175 lb) and a thrust load of 19 012 N (4274 lb) at a speed of 6600 rpm. All the bearings in the set are match ground with a 445-N (100-lb) preload. There are two manufacturers of this bearing. However, each manufacturer's design is similar which allows for complete interchangeability.

TEST PROCEDURE

Endurance tests were performed in order to evaluate the process of restoring by grinding on each of the three bearing types and determine that the restored bearing will provide lives at least as long as the desired time between overhaul of 1600 hr. Speed, load, and lubrication conditions were chosen to be representative of each bearing application. The test conditions are shown in Table 2.

Each of the bearing types were tested in test heads specifically chosen for the particular speed and load conditions. The test facilities were capable of continuous running with test interruption only due to bearing failure or inadvertent test facility malfunctions. The lubrication systems for the three facilities had many common features. Each system used MIL-L-23699 Type II ester, from single lubricant batches. The test bearings were lubricated by jets. The lubricant was recirculated through appropriate heat exchangers to maintain the desired flow rates and lubricant-in temperatures.

In order to simulate lubricant replenishment due to leakage and evaporation in engine and gearbox lubrication systems and periodic lubrication changes, the test facility lubrication systems were periodically drained and refilled with new lubricant. Also, the lubricant was changed each time at new bearing or bearings were put on test.

RESULTS AND DISCUSSION

Life Analysis

Parametric results were generated using equation (9). For computation two restoration times were arbitrarily chosen: the 50-percent life ($n_t = L_{50}$) and the 10-percent life ($n_t = L_{10}$). The results are plotted on Weibull coordinates (Fig. 5) with the fraction of stressed volume removed during restoration as a parameter. Weibull coordinates (23) are the log-log of the reciprocal of the probability of survival plotted as the statistical percentage of specimens failed (ordinate) against the log of time to failure (abscissa). The calculations were performed for $x = 0$, 1/4, 1/2, 3/4, and 1, assuming a Weibull slope of 10/9.

The extreme values of $x = 1$ and $x = 0$ are special classes. For $x = 1$ the entire stressed volume layer is removed, and, therefore, an essentially brand new bearing is the result. The failure rate in this case is Weibull in nature and plots as a straight line on Weibull coordinates. For $x = 0$ no grinding is performed; the failed or damaged bearings are merely culled from the total population at time n_t. (This condition is that of a refurbished bearing.) The remaining bearings are put into service again. The results for $x = 0$ agrees with the result presented in Reference 24 for this special case.

For values other than $x = 1$ the failure rate is non-Weibull because the plot is nonlinear. There are proportionately more early failures than would be the case for a true Weibull distribution. In general, for increasing amounts of material removed in restoration, the theory predicts closer approximation to the new bearing failure rate. The theory also shows that if a given amount of material is to be removed in restoration, then longer bearing life will be achieved with an earlier refurbishment, say at the 10-percent life rather than the 50-percent life.

The ratio of the 10-percent life of restored bearings to the 10-percent life of brand new bearings is shown in Fig. 6. As would be expected, the shortest life conditions occur when the bearings are restored at the 50-percent level with a minimum amount of stressed volume removed.

Endurance Testing

Thirty of each of the three bearing types were chosen at random from the groups of bearings restored by grinding. These bearings were then tested for endurance in their respective facilities for a duration of 1600 hr at conditions representative of the specific application. The objective of the tests was to demonstrate the capability of the bearings restored by grinding to operate satisfactorily for the desired time between overhaul of 1600 hr. This is in contrast to bearing fatigue testing which is designed to run at conditions chosen to accelerate spalling fatigue failures.

In each of the three sets of tests, none of the restored bearings experienced failures which could be related to the restoring process. Twenty-eight of the 30 7216-size angular-contact ball bearings, 29 of the 111-size cylindrical roller bearings, and all 30 of the 210-size split-inner ring ball bearing reached the desired 1600 hr time without failure.

Eight 210-size split-inner-race ball bearings were tested simultaneously with two bearings in each of four test heads. In order to complete the 1600 hr duration for 30 bearings, the final test setup utilized only three of the four test heads. The lubricant volume in the sump was adjusted accordingly.

Subsequent to the successful completion of the 1600 hr tests, the bearings were disassembled and visually inspected. The raceways had visible running tracks typical of bearings running under thrust load. All race surfaces and ball surfaces were generally discolored from heat and lubricant staining. The raceway running tracks were discolored to a lesser extent. The condition of all contacting surfaces including raceways, balls, and cage surfaces was excellent, with no indications of any detrimental effects of the restoring process or damage from the endurance testing.

The extent of discoloration on the bearing components suggests that these bearings were exposed to relatively high temperatures in these tests. The measured outer race temperatures were in the range of 395 to 408 K (252 to 275 °F) and averaged about 400 K (260 °F). Oil-out temperature ranged from 386 to 402 K (235 to 265 °F). The hardness of the inner and outer races of several bearings were measured after disassembly. These bearings included the bearing with the highest outer-race temperature (408 K (275 °F)) (also most discolored bearing) and the bearing with the lowest outer-race temperature (395 K (252 °F)). All hardnesses were in the range from 58 R_C to 60 R_C, which is on low end of the acceptable range for rolling-element bearings. These temperatures are approaching the limits for the AISI 52100 material.

Twelve 111-size cylindrical roller bearings were tested simultaneously with four bearings in each of three test heads. To complete the 1600 hr test time for all 30 test bearings, dummy bearings were used to make up the extra positions in the test head. The measured outer-race temperatures for these bearings were in the range from 377 to 391 K (220 to 245 °F). The oil-out temperature ranged from 369 to 377 K (205 to 220 °F).

Twenty-nine test bearings completed the desired 1600 hr duration. One bearing suffered a failure after only 16.3 hr. Subsequent detailed examination of this bearing indicated that the failure initiated as a roller fatigue spall. Failure detection equipment did not detect the failure and shut down the test rig. It was apparent that the roller spall propagated until more than half the roller surface was severely spalled, eventually causing cage breakup and subsequent severe damage to the other rollers and the raceways. Because of the severely overrun condition of the spalled area on the suspected roller, definite evidence of a material defect was not found. However, scanning electron microscope examination of the spalled area and metallographic sections through the spalled area indicated that the subsurface initiated rolling-element fatigue was the primary failure. Since the microstructure and hardness of the roller, in general, were typical of properly heat treated AISI M-50, it is suspected that a stress concentration such as an inclusion or void was in the critical area. Additionally, there was no evidence of roller skew or unusual end wear on any rollers from this bearing.

It is concluded that this premature failure was not related to the restoring by grinding process. The process includes installing new rollers in the restored bearing, and such new rollers are of a quality which would be installed in new bearings. Thus, such a failure could have occurred in a new bearing as well as in this restored bearing.

The bearings that completed the 1600 hr tests were disassembled and visually inspected. Initial examination with the unaided eye revealed the raceways in good condition, with roller tracks somewhat more apparent on the outer raceways than on the inner raceways. Cages were in excellent condition with the normal light wear or burnishing of the silver plating in the pockets and the inner race riding lands. The rollers nearly all showed some circumferential lines typically observed on tested roller bearings. The roller ends and the inner race flange contacts were in excellent condition revealing no significant abnormal roller motion. However, in a few of the bearings, the cage pocket wear indicated a very slight amount of roller skewing. However, the extent of skewing does not appear to present a problem.

More detailed examination at low magnification (6X) of the surfaces of rollers from several of the bearings revealed shallow surface distress or pitting within the flat length of the roller and generally toward the blend of the flat length and the crown. The depth of pitting, as measured from surface profile traces and metallographic sections, was typically less than 0.013 mm (0.0005 in.). Although the inner and outer raceways showed some isolated evidence of very minor surface distress, the damage was mainly limited to the roller surfaces.

Surface finishes of the rollers and raceways were measured on 10 bearings randomly chosen from the total lot of 30. The outer raceways measured either 0.103 or 0.127 μm (4 or 5 μin.) RMS in all cases. The roller cylindrical surfaces measured from 0.103 to 0.152 μm (4 to 6 μin.) RMS. The inner raceway surfaces ranged from 0.089 to 0.432 μm (3.5 to 17 μin.) RMS. The engine manufacturer's drawing for this bearing specifies 0.254 μm (10 μin.) or better for these surfaces although bearing manufacturers typically finish to better surfaces as indicated by the outer raceway and roller surfaces measured here.

In all cases where the surface finish of the inner raceway equalled or exceeded the specified 0.254 μm (10 μin.), surface distress was observed on the rollers. On two bearings, where inner raceway finishes were 0.152 and 0.178 μm (6 and 7 μin.) RMS, some roller surface distress was observed. On the other bearings, where inner raceways varied from 0.089 to 0.203 μm (3.5 to 8 μin.) RMS, no roller surface distress was observed.

The elastohydrodynamic (EHD) film thickness in the roller raceway contacts was calculated using a high-speed roller bearing computer program for the conditions of these tests. The film thickness at both the inner and outer raceway contacts is estimated to be <0.203 μm (8 μin.).

An accepted criterion for the effectiveness of the EHD film thickness is the ratio of EHD film thickness to the composite surface roughness σ. This ratio is often referred to as film thickness parameter or Λ. The composite surface roughness σ is the square root of the sum of the squares of the surface finishes of the two surfaces in contact. For those bearings where the inner-race surface finish was 0.254 μm (10 μin.) RMS or greater, Λ was 0.68 or less. Where Λ is <1.0, it should be expected that considerable asperity contact will occur (25,26). The detrimental effects of this surface-to-surface contact are expected to be further aggravated by skidding. At the very low radial load of these tests, it is expected that some skidding exists, wherein the rollers are orbiting at a speed less than epicyclic speed. Under these conditions of skidding and low Λ, it may be expected that some surface damage would occur. Thus, the surface distress observed on the rollers was apparently related to the test conditions, and not attributed to the restoring by grinding process.

The load and temperature conditions for these tests are estimates of those that the bearing experiences in the engine. The engine conditions, of course, are neither constant nor easily determined. Whether the engine conditions are such that surface distress would occur, such as that observed on these test bearings, is not known, but in view of those test results, that possibility exists.

With the exception of some inner raceway surface finishes not meeting specifications, the endurance tests revealed no problems related to the restoring by grinding process. Raceway surface finish deviations are occasionally found in new bearings, so it is not a problem unique to restored bearings.

Thirty 7216-size angular-contact ball bearings were chosen at random from the tandem pairs of 30 triplex sets. Eight bearings were tested simultaneously with four bearings required for each test setup. In order to complete the 1600 hr duration for 30 bearings, additional bearings were chosen at random from the remaining bearings.

The measured outer-race temperatures for these bearings were in the range from 358 to 364 K (185 to 195 °F). Oil-out temperature ranged from 350 to 355 K (170 to 180 °F).

After 1122 hr with one set of bearings, the inner rings of two adjacent bearings began to turn on their shaft. They were removed and could not be tested further. Their raceways, balls, and cage surfaces were in excellent condition. The bore diameters were within tolerances, but near maximum. This fact, coupled with a shaft size near minimum apparently allowed an undesirable fit situation with this particular bearing pair. Since the bores were within tolerances, this failure could not be directly attributed to the restoring process.

While running the last two bearings in the 30 bearing samples, an additional bearing, one of two which were used only as slave bearings at the opposite end of the test spindle, suffered a severe ball failure. Although this bearing was not one of the original random sample of 30 bearings, it was a bearing from the 30 restored triplex sets. Observation of the failed bearing indicated that the ball failure was due to a metallurgical defect in the ball and had no relation to the restoring process of the races.

Subsequent to the 1600 hr tests, the bearings was disassembled and visually inspected. The inner raceway had visible running tracks typical of bearings run under such conditions for extended times. The outer raceways had visible tracks typical of ball bearings under combined radial and thrust load conditions. The general condition of all raceways, ball surfaces, and cage surfaces was excellent, with no indications of any detrimental effects of the restoring process.

SUMMARY

A life analysis for reground bearings has been developed. In predicting the life of the reground bearing, it was assumed that the probability of survival of the bearing was equal to the product of the individual probabilities of survival for the newly stressed and previously stressed materials. Therefore, the stressed volume removed by grinding and the number of stress cycles accumulated before grinding are variables that affect the life of the reground bearings. A total of 250 bearings were reground from three separate bearing types which were selected for testing. Of this number, 30 bearings from each type were endurance tested for 1600 hr. The following results were obtained.

1. The 10-percent life of reground bearings ranges from 74 to 100 percent of the life of brand new bearings. This life depends on the bearing's previous service time and the amount of material volume removed.

2. The failure distribution for restored and refurbished bearings is non-Weibull with a bias toward more early failures than that of a Weibull distribution.

3. No bearing failure occurred related to material removal. Two bearing failures occurred due to defective rolling elements and were typical of those which may occur in new bearings.

REFERENCES

1. Carter, T.L., A Study of Some Factors Affecting Rolling-Contact Fatigue Life. NASA TR R-60, 1960.
2. Bear, H.R., and Butler, R.H., Preliminary Metallographic Studies of Ball Fatigue Under Rolling-Contact Conditions. NASA TN-3925, 1957.
3. Carter, T.L., Butler, R.H., Bear, H.R., and Anderson, W.J., Investigation of Factors Governing Fatigue Life with the Rolling-Contact Fatigue Spin Rig. _ASLE Trans._, 1958, _1_, 23-32.
4. Jones, A.B., Metallographic Observations of Ball Bearing Fatigue Phenomena. _Symposium on Testing of Bearings_, ASTM, Philadelphia, 1947, pp. 35-52.
5. Parker, R.J., Correlation of Magnetic Perturbation Inspection Data with Rolling-Element Bearing Fatigue Results. _J. Lubr. Technol._, 1975, _97_, 151-158.
6. Chevalier, J.L., and Zaretsky, E.V., Effect of Carbide Size, Area, and Density on Rolling-Element Fatigue. NASA TN D-6835, 1972.
7. Chevalier, J.L., Zaretsky, E.V., and Parker, R.J., A New Criterion for Predicting Rolling-Element Fatigue Lives of Through-Hardened Steels. _J. Lubr. Technol._, 1973, _95_, 287-297.
8. Cobb, L.D., Effect of Vacuum Melting on Bearing Steels, Paper presented at the SAE National Passenger Car, Body, and Materials Meeting, Detroit, MI, Mar. 5-7, 1957.
9. Johnson, R.F., and Sewell, J.F., The Bearing Properties of 1 %C-Cr Steel as Influenced by Steelmaking Practice. _J. Iron Steel Inst. London_, 1960, _196_, 414-444.
10. Morrison, T.W., Tallian, T., Wolf, H.D., and Baile, G.H., The Effect of Material Variables on the Fatigue Life of AISI 52100 Steel Ball Bearings. _ASLE Trans._, 1962, _5_, 347-364.
11. Schlatter, R., and Stroup, J.P., Improved M50 Aircraft Bearing Steel Through Advanced Vacuum Melting Processes. _J. Vac. Sci. Technol._, 1972, _9_, 1326-1333.
12. Bamberger, E.N., Zaretsky, E.V., and Signer, II., Endurance and Failure Characteristics of Main Shaft Jet Engine Bearing at 3X10^6 DN. _J. Lubr. Technol._, 1976, _98_, 580-585.
13. Martin, J.A., and Eberhardt, A.D., Identification of Potential Failure Nuclei in Rolling Contact Fatigue. _J. Basic Eng._, 1967, _89_, 932-942.
14. Littman, W.E., and Widner, R.L., Propagation of Contact Fatigue from Surface and Subsurface Origins. _J. Basic. Eng._, 1966, _88_, 624-636.
15. Leonard, L., Martin, J.A., and Choman, L., Surface and Subsurface Observations of Endurance Tested 6309-Size Bearings. SKF Report AL69M025, SKF Industries, King of Prussia, PA, Oct. 1969.
16. Stanley, D.C., Bearing Field Inspection and Refurbishing. Paper presented at Symposium on Propulsion System Structural Integration and Engine Integrity, Naval Post Graduate School, Monterey, CA, Sept. 3-6, 1974.
17. Hanau, H., Restoration by Grinding of Aircraft Ball and Roller Bearings-A Manufacturer's Viewpoint, _Bearing Restoration by Grinding_, H. Hanau et al, eds., USAAVSCOM-TR-76-27, U.S. Army Aviation System Command, 1976, pp. 1-23.
18. Parker, R.J., Zaretsky, E.V., and Chen, S.M., Evåluation of Ball and Roller Bearings Restored by Grinding, _Bearing Restoration by Grinding_, H. Hanau et al, eds., USAAVSCOM-TR-76-27, U.S. Army Aviation System Command, 1976, pp. 24-42.
19. Lundberg, G., and Palmgren, A., Dynamic Capacity of Rolling Bearings. _Acta Polytech., Mech. Eng. Ser._, 1947, _1_, 1-50.
20. Lundberg, G., and Palmgren, A., Dynamic Capacity of Rolling Bearings. _J. Appl. Mech._, 1949, _16_, 165-172.

21. Lundberg, G., and Palmgren, A., Dynamic Capacity of Roller Bearings. Acta Polytech., Mech. Eng. Ser., 1952, 2, 1-32.

22. Coy, J.J., Zaretsky, E.V., and Cowgill, G.R., Life Analysis of Restored and Refurbished Bearings. NASA TN D-8486, 1977.

23. Johnson, L.G., The Statistical Treatment of Fatigue Experiments. Elsevier, 1964.

24. Harris, T.H., Rolling Bearing Analysis. 2nd Ed., John Wiley and Sons, Inc., 1984, pp. 443-449.

25. Bamberger, E.N., Zaretsky, E.V., and Anderson, W.J., Effect of Three Advanced Lubricants on High-Temperature Bearing Life. J. Lubr Technol., 1970, 92, 23-33.

26. Li, D.F., Kauzlarich, J.J., and Jamison, W.E., Surface Roughness Effects on Fatigue in Partial EHD Lubrication. J. Lubr. Technol., 1976, 98, 530-537.

Table 1 Specification

Bearing	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular-contact ball bearing
Pitch diameter, mm (in.)	70 (2.756)	72.5 (2.854)	110 (4.331)
Tolerance	ABEC-5	RBEC-5	ABEC-5
Contact angle	27 to 30°	--------------	25°
Bearing steel	AISI 52100	AISI M-50	AISI M-50
Number of rolling elements	14	16	15
Cage material	Bronze	Steel	Steel
Cage type	One piece, inner-land riding	One piece, inner-land riding	One piece, inner-land riding
Conformity, percent			
inner race	51.5	--------------	52
outer race	52	--------------	52
Surface finish μm (μin.)			
inner race	6	10	6
outer race	6	10	6
rolling element	1	6	1
Ball or roller diameter, mm (in.)	12.700 (0.5000)	8.999 (0.3543)	19.844 (0.78125)
Roller crown radius, mm (in.)	---------------	1778 (70)	----------------
Roller length, mm (in.)	---------------	9.000 (0.354)	----------------

Table 2 Endurance test conditions

Bearing	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular-contact ball bearing
Spindle speed, rpm	24 000	24 000	6600
Radial load, N(lb)	0	445 (100)	8474 (1905)
Thrust load, N(lb)	2002 (450)	0	11 405 (2564)
Maximum Hertz stress, N/m^2 (psi)			
inner race	1.018×10^9 (148×10^3)	0.779×10^9 (113×10^3)	2.157×10^9 (313×10^3)
outer race	1.256×10^9 (182×10^3)	$.810 \times 10^9$ (117×10^3)	1.868×10^9 (271×10^3)
Depth to maximum shear stress, Zo, mm (in.)			
inner race	.070 (0.0027)	.033 (0.0013)	.317 (0.0125)
outer race	.125 (0.0049)	.047 (0.0019)	.134 (0.0053)
Lubricant inlet temperature, K (°F)	365±3 (195±5)	364±3 (195±5)	339±6 (150±10)
Lubricant flow rate, m^3/sec (gpm)	$24 \times 10^{-6} \pm 2.7 \times 10^{-6}$ (0.38±0.043)	$24 \times 10^{-6} \pm 2.7 \times 10^{-6}$ (0.38±0.043)	$132 \times 10^{-6} \pm 18.9 \times 10^{-6}$ (2.1±0.3)
Bearing outer race temperature, K (°F)	400 (260)	391 (245)	372 (210)
Lubricant outlet temperature, K (°F)	391 (245)	377 (220)	366 (200)
Λ			
inner race	1.73	.63	1.54
outer race	1.88	.68	1.72
Calculated L_{10} life, hr			
with lubricant factor	11 910	15 000	769
without lubricant factor	5 923	62 464	521
with lubrication and material factors	71 460	90 000	4614
Sump capacity, m^3 (gal)	15.1×10^{-3} (4)	15.1×10^{-3} (4)	45.4×10^{-3} (12)
Number of bearings per sump	4	4	8
Lubricant change interval, hr	100	100	600

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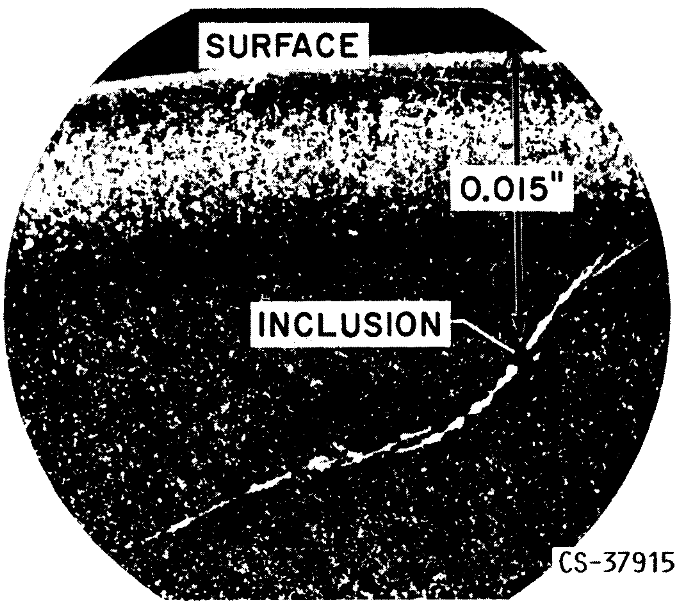

FIGURE 1. - FATIGUE CRACK EMANATING FROM AN INCLUSION.

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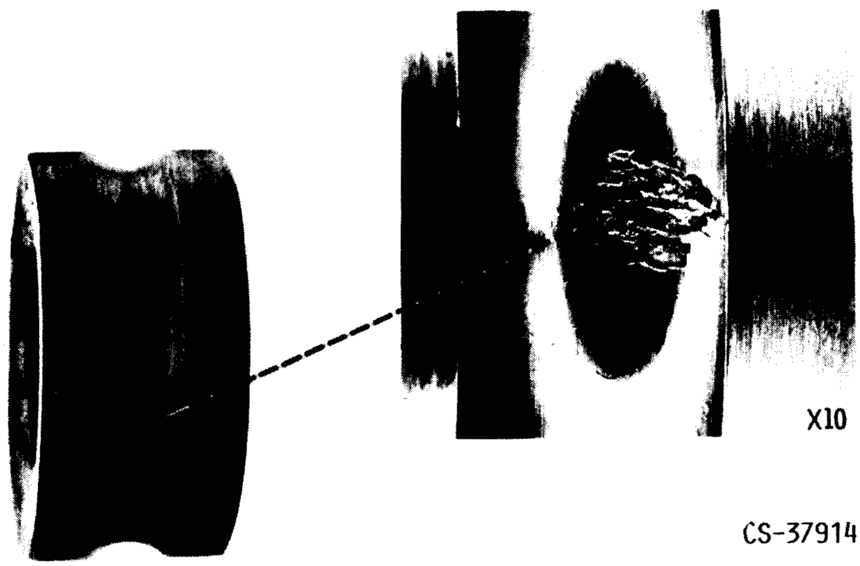


FIGURE 2. - TYPICAL FATIGUE SPALL IN BEARING RACE.

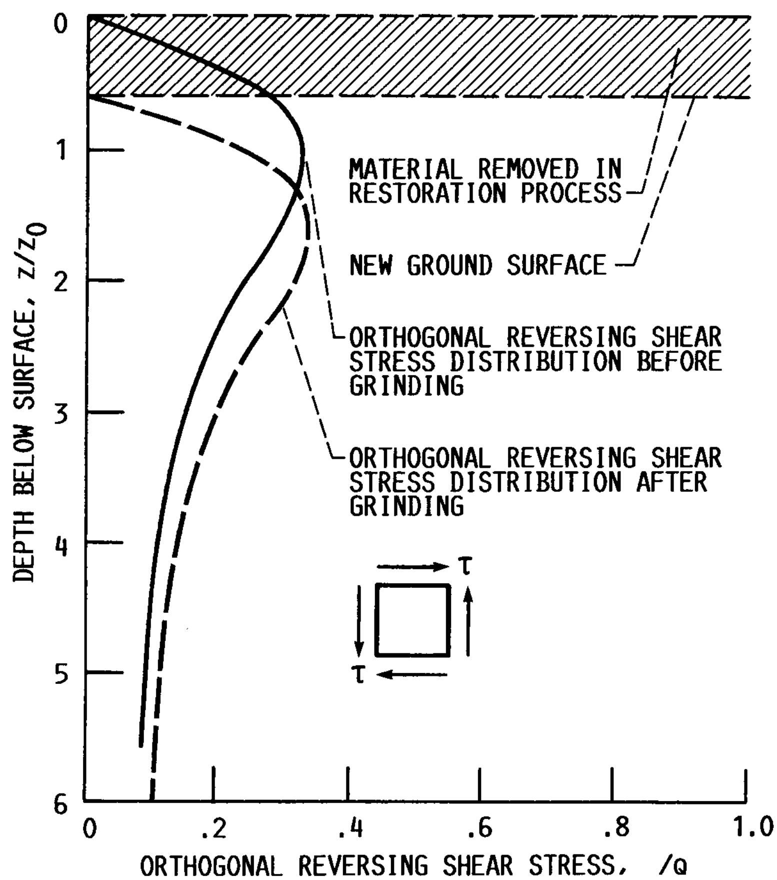
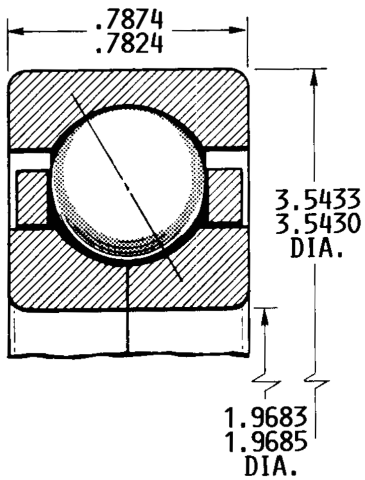
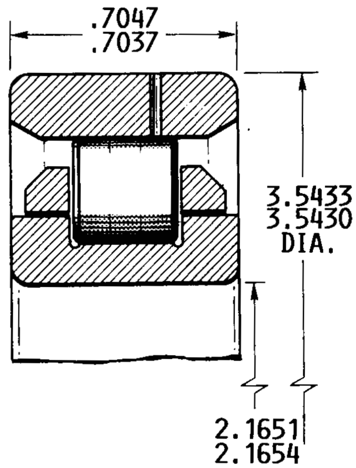


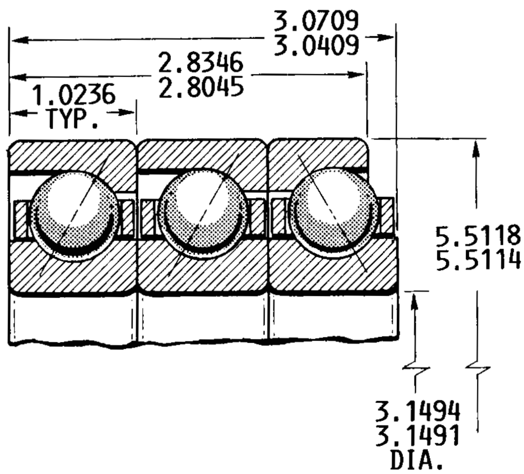
FIGURE 3.- RELATIVE VALUE OF ORTHOGONAL REVERSING SHEAR STRESS AS FUNCTION OF DEPTH BELOW ROLLING-ELEMENT SURFACE. FIGURE SHOWS EFFECT OF GRINDING IN REDISTRIBUTING STRESS.



(A) 210-SIZE SPLIT-INNER-RING BALL BEARING.

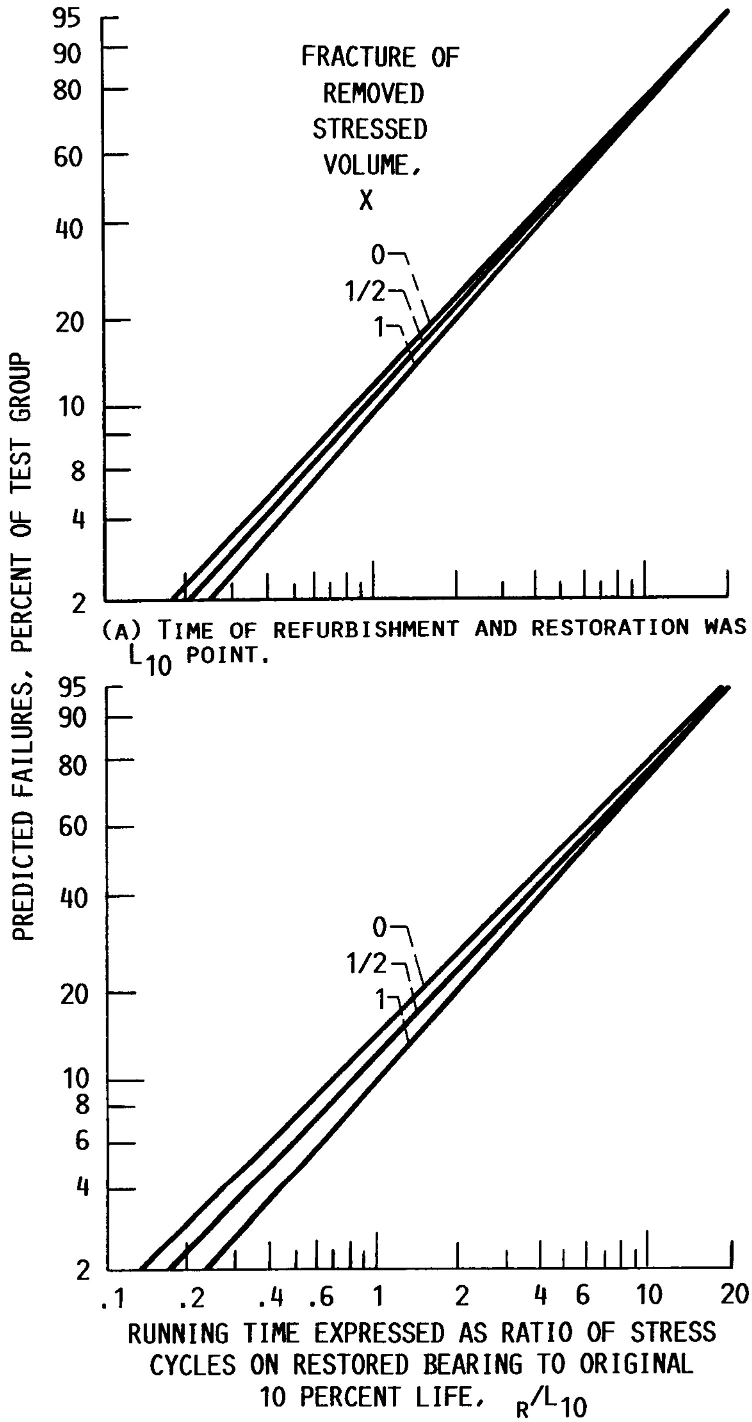


(B) 111-SIZE CYLINDRICAL ROLLER BEARING.



(C) 7216-SIZE ANGULAR-CONTACT BALL BEARING TRIPLEX SET.

FIGURE 4.- TEST BEARINGS.



(B) TIME OF REFURBISHMENT AND RESTORATION WAS L_{50} POINT.

FAILURE 5.- FAILURE RATE OF REFURBISHED AND RESTORED BEARINGS FOR VARIOUS FRACTIONS OF STRESSED VOLUME REMOVED.

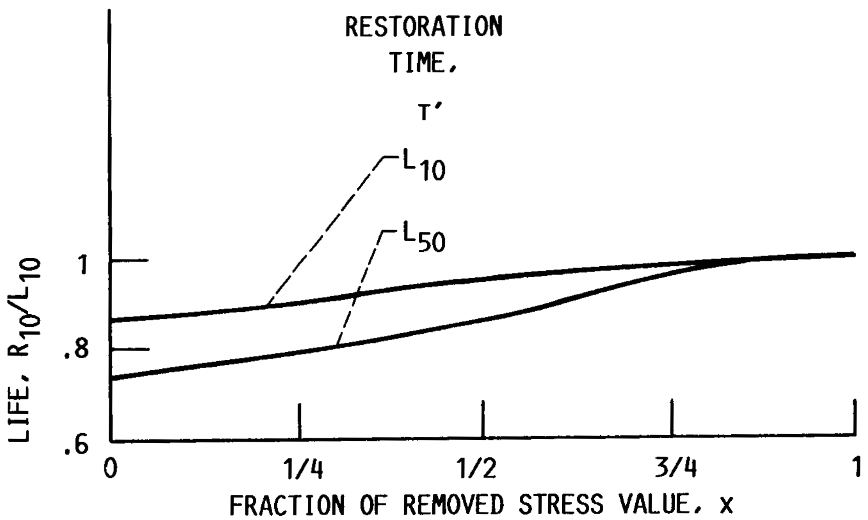


FIGURE 6.- RATIO OF 10-PERCENT LIFE OF RESTORED BEARING TO 10-PERCENT LIFE OF ORIGINAL BEARINGS AS A FUNCTION OF STRESSED VOLUME REMOVED.

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15. Supplementary Notes

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16. Abstract

The Lundberg-Palmgren equation was modified to show the effect on rolling-element fatigue life of removing by grinding a portion of the stressed volume of the raceways of a rolling-element bearing. Results of this analysis show that depending on the amount of material removed, and depending on the initial running time of the bearing when material removal occurs, the 10-percent life of the reground bearings ranges from 74 to 100 percent of the 10-percent life of a brand new bearing. Three bearing types were selected for testing. A total of 250 bearings were reground. Of this matter, 30 bearings from each type were endurance tested to 1600 hr. No bearing failure occurred related to material removal. Two bearing failures occurred due to defective rolling elements and were typical of those which may occur in new bearings.

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