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ABSTRACT

A joint program was undertaken by the NASA Lewis Research Center and the Army Aviation Systems Command to restore by grinding those rolling-element bearings which are currently being discarded at aircraft engine and transmission overhaul. Three bearing types were selected from the UH-1 helicopter engine (T-53) and transmission for the pilot program. Groups of each of these bearings were visually and dimensionally inspected for suitability for restoration. A total of 250 bearings were restored by grinding. Of this number, 30 bearings from each type were endurance tested to a TBO of 1600 hours. No bearing failures occurred related to the restoration by grinding process. The two bearing failures which occurred were due to defective rolling elements and were typical of those which may occur in new bearings. The restorable component yield to the three groups was in excess of 90 percent.

INTRODUCTION

The last three 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

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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 (refs. 1 to 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 (refs. 6 and 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 (refs. 8 to 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 (ref. 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 (ref. 12). Hence, the probability of subsurface fatigue can be greatly minimized within current 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 (refs. 13 to 15). As a result, in aircraft engine and transmission applications, a large number of bearings are discarded at overhaul or during periodic maintenance. This results in millions of dollars in bea ıngs which have to be replaced anually. In addition, critical alloying elements such as molybdenum, manganese, chromium, nickel, and vanadium are consumed.

In order to reduce costs and conserve materials by reusing bearings, bearing rework has been performed both in private industry and within the U.S. Government on a limited basis. Basically, bearing rework as currently practiced comprises the following steps (ref. 16):

1. Replace defective balls or roller sets with new balls or rollers that match the original parts with respect to size, tolerance, finish, sphericity, hardness, and material. The bearings are then reassembled and checked for radial and axial clearance and, where required, contact angle.

2. Defective retainers are replaced. Alternately, silver plating from original retainers are removed, and the retainers are nondestructively inspected and replated to original specifications.

3. Ball and roller raceways are polished to remove superficial corrosion or other shallow surface defects. The reworked bearing surfaces are then checked for profile, concentricity, surface finish, and metallurgical integrity.

4. The inner- and outer-diameter surfaces of the bearing rings are ground and replated to original drawing specification.

5. Duplex and multiple stack bearing sets are rematched in accordance with original specifications using individual bearings from rejected or failed sets.

The aforesaid method retains subsurface damage in the used bearing raceways which may propagate to fatigue failures and thus limit the reworked bearings' life and reliability. Reworking of any kind which involves significant removal of material (and thus this damage) from bearing raceways has been avoided in previous refurbishing or reworking techniques.

The program reported herein was undertaken to investigate the endurance characteristics of aircraft turbine engine and transmission bearings whose raceways were restored by grinding. The primary objectives were to (1) determine the yield of bearings suitable for restoring by grinding; (2) determine the endurance characteristics of restored bearings under simulated operating conditions; (3) determine post-test condition of the restored bearings; and (4) establish a specification for bearing restoration by raceway grinding.

BEARING SELECTION FOR RESTORATION

Three bearing types were selected from the UH-1 helicopter engine and transmission for the program of restoration by grinding. The criteria for selection were (a) bore of 50-mm or larger and (b) acquisition cost to the Government of $50.00 or greater. Based on information furnished by the Corpus Chrisi Army Depot, the bearings selected had among the highest replacement rates during normal maintenance and overhaul. The bearings and their Federal Stock and component engine part numbers are listed in table I.

The 210-size (50-mm bore) split inner-ring ball bearing is shown in figure 3. 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 the T-53 gas turbine engine. Engine operating conditions for the bearing include a speed of 24 000 rpm and a thrust load of 450 pounds. Bearing oil-in temperature in the engine is approximately 200° F and oil-out temperature is between 250° to 300° F. A total of 150 of these bearings were sent by the overhaul depot to be inspected for restoration.

The 111-size (55-mm bore) cylindrical roller bearings shown in figure 4 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 the T-53 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 200° F and oil-out temperature is approximately 350° F. A total of 121 of these bearings were sent by the overhaul depot to be inspected for restoration.

The third bearing type was the triplex ball bearing set shown in figure 5. Eighty-six sets of these bearings were sent to be inspected for restoration. These are 7216-size (80-mm bore) angular-contact ball bearings which are mounted on the input bevel gear pinion shaft of the UH-1 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 3175 pounds and a thrust load of 4274 pounds at a speed of 6600 rpm. All the bearings in the set are match ground with a 100-pound preload. There are two manufacturers of this bearing. However, each manufacturer's design is similar which allows for complete interchangeability.

INSPECTION OF REMOVED BEARINGS

A total of 529 bearings of the three types were subjected to inspection to determine suitability and yield for restoring by grinding. These included 150 split inner-ring ball bearings (210-size), 121 cylindrical roller bearings (111-size), and 86 triplex sets of the angular-contact ball bearings (7216-size). The inspection procedure consisted of visual and dimensional inspections including ring diameters, flatness, roundness, hardness, magnaflux for flaws, raceway surface conditions (spalling, wear, etc.), and general suitability of all components for restoring by grinding. The restorable bearing yield percents are given in table II for the three bearing types.

Of the 150 split-inner-ring ball bearings inspected, 145 were judged restorable for a yield of 96.7 percent. On the basis of restorable components, the yield was 98.2 percent. The general appearance of this group of bearings was excellent. Raceway tracks were light and balls and separators were in good condition except for some oil varnish deposits on separator lands and ball pockets. The five bearings that were judged non-restorable had defects including fatigued raceways, corrosion, or bore and/or outside diameters (O.D.) greater than 0.001 inch out of tolerance. The incidence of bore and O.D. variance from print tolerance was high at 47.3 percent. Some of this variance is explainable by gaging discrepancies in cases were 0.0001 inch off-print dimension was noted. However, 28 of these bearings had rings out of tolerance from 0.0002 to 0.001 inch but were judged restorable since these bore and O.D. surfaces could be plated and reground to print dimensions. Only one bearing was non-restorable solely because of out-of-tolerance rings. Fifty of the 145 restorable bearings were randomly chosen to be restored by grinding.

These bearings are made of AISI 52100 steel, which may be expected to experience dimensional changes in turbine-engine mainshaft positions where soak back temperatures near 400° F may be expected. Generally, the use of AISI M-50 steel for mainshaft bearings is recommended.

The 121 cylindrical roller bearings consisted of 54 from manufacturer A and 67 from manufacturer B. The number of bearings judged restorable were 52 and 62, respectively, for yields of 96.3 and 92.5 percent, respectively and an overall yield of 94.2 percent. On the basis of restorable components, the yield was 96.7 percent. The general appearance of these bearings was excellent. Raceway tracks were light, and the rollers and separators were in good condition except for some oil varnish deposits on the separator lands and roller pockets. The seven bearings that were judged non-restorable had defects including fatigue spalling, corrosion, or surface distress on the raceways and damaged O. D. surfaces of outer rings. Seventeen of the total group of 121 bearings had bore or O. D. dimensions out of print tolerance, but only three of these discrepancies were greater than 0.0001 inch.

The bearings chosen for restoring by grinding were 50 randomly chosen bearings from the 62 restorable bearings all from manufacturer B.

The total of 86 triplex sets of the angular contact ball bearings consisted of 71 sets from manufacturer C and 15 sets from manufacturer D. The number of sets judged restorable were 46 and nine, respectively, for yields of 64.8 and 60.0 percent, respectively, and an overall yield of 64.0 percent. The total number of bearings considered restorable was 207, for an overall bearing yield of 80.2 percent. On the basis of components, that is, rings and cages, the overall restorable component yield was 91.6 percent.

The general appearance of this group of bearings was much poorer than the other two groups of bearings inspected. There was a high incidence of corrosion and rusting in the raceways. Most of this apparently occurred after removal from the transmission and could be attributed to insufficient protection during or prior to storage. Many of the raceway tracks appeared to have experienced wear or a change in surface characteristics associated with operation in dirt or debris contaminated lubricant. The incidence of spalling fatigue failures was much higher than with the other two groups. Some of this spalling may have been related to debris damage.

Of the 48 bearings that were judged non-restorable, 37 had corrosion pitted or rusted raceways, two had separator damage, and nine had raceway spalling fatigue failures. Three bearings from two triplex sets were not returned for inspection by the overhaul depot, thus accounting for the discrepancy in quantities from the total of 86 triplex sets. The spalling fatigue failures occurred in seven triplex sets, for an 8.1 percent failure rate. Bearings from 24 triplex sets or 27.9 percent had corrosion pitted or rusted raceways.

Only 25 of the 255 bearings inspected had bore or O. D. dimension discrepancies greater than 0.0001 inch, but none of these were bad enough to be judged non-restorable.

The 50 triplex sets chosen for restoring by grinding included sets from both manufacturers C and D.

The restorable yield on the basis of components (rings and separators) was greater than 90 percent for each bearing type inspected. For the split-inner-ring ball bearings the restorable component yield was 98.2 percent; for the cylindrical roller bearing, 96.7 percent; and for the triplex angular-contact ball bearings, 91.6 percent.

PROCESS OF RESTORING BY GRINDING

Figure 6 shows the relative magnitude of shearing stress as a function of depth below the surface. In the zone of maximum resolved shearing stress, which generally occurs from 0.002 to 0.006 inch below the surface for most rolling-element bearing applications, there exists the maximum orthogonal shearing stress, the maximum octahedral shearing stress, and the maximum shearing stress on a 45° plane. Due to the cyclic nature of the shearing stresses, a fatigue crack will eventually occur within this zone and propagate into a crack network. This network will eventually lead to spalling and failure of the raceway bearing surface whereby the bearing is no longer suitable for the purposes for which it was intended.

In addition to this failure phenomena, damage to the surface can occur which is caused primarily by dirt and foreign object or wear debris within the lubrication system and in some instances by corrosion of the raceway surfaces. This surface damage, in the form of indentations or scratches, acts as stress raisers on the bearing raceway surfaces, and also initiates fatigue cracks which propagate into crack networks and eventually leads to spalling.

Where a rolling element such as a ball or a roller fails by spalling, debris damage generally occurs to the raceways. In addition, where an inner or outer raceway fails there is generally debris damage to the rolling elements and the non-failed race. These failure phenomena make the bearing unusable both for continuous application and for subsequent reuse. However, most of the debris damage occurs at depths less than 0.002 inch below the surface of the rolling-element raceways.

Another problem in the operation of rolling-element bearings is growth of the bearing race rings. As an example, the inner or outer races, can grow due to metallurgical transformations or due to hoop stresses during operation. This growth results in the bearing being not reusable after removal from its application.

For aircraft applications, bearings removed at overhaul are usually cleaned and visually inspected for defects and dimensional conformance to print.

Bearings are usually not taken apart, except those which by design are separable. Therefore, impending or minute fatigue spalling cannot be detected except by hand, feel, or noise testing. Both of the latter techniques are very subjective and can only be used with any degree of success by a bearing manufacturer and only after establishing performance yardsticks on large production runs of new bearings.

In general, the rejection rate of bearings removed from turbine engines and transmission at overhaul is approximately 50 percent. Based upon experience, less than 10 percent of the bearings removed have failed due to fatigue. Therefore, 90 percent of those rejected bearings can be restored by grinding the raceways and thus reused since the raceway damage causing rejection is usually much more superficial than fatigue failure damage.

In order to restore bearings economically, it is necessary to start a production run of components when a minimum economic lot size of components has been accumulated. A lot is then processed utilizing process-travelers and manufacturing drawings in same manner as a manufacture of new bearings. Some quantity of bearings received for restoring have non-restorable components. These are scrapped. The restorable components are stored in a "parts bank" from which interchangeable components will be drawn to make an economic lot size for restoring.

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

Those components 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 print dimensions.

Both inner and outer raceways are ground to a depth not exceeding the maximum depth of the maximum resolved shearing stress under their maximum loaded condition but not less than 0.002 inch. The surfaces finish is maintained to its original print specification. The bearing is 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 ball bearings the effective race curvature is identical to the original dimensions within significant mathematical values. The original values of contact angles, resting angle, and radial clearance remain unchanged. 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 and calculated bearing life of the restored bearing will be essentially identical to that of the original bearing. The bearing separator is stripped of its silver plating, where applicable, inspected for cracks and replated. The new, oversized rolling elements are placed within the separator, and the bearing is reassembled.

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

Referring to figure 6, the location of the zone of maximum resolved shearing stress in the inner and outer raceways after regrinding is displaced by a distance X, and the stress is redistributed accordingly. The dimension X should be not less than 0.002 inch nor more than the original maximum depth of the zone of maximum resolved shearing stress at the maximum load condition for the bearing. The bearing kinematics, internal clearances, and contact loads during operation remain unchanged. A new volume of material is being stressed which should result in a life or probability of survival equivalent to the bearing's original life or survival probability.

ENDURANCE TESTING

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 hours. Speed, load, and lubrication conditions were chosen to be representative of each bearing application. The test conditions are shown in table III.

Test Apparatus and Procedure

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.

Split-inner-ring ball bearing test head. - A sectional view of a test head used for the 210-size split-inner-ring ball bearings is shown in figure 7. Four of these test heads were used, each one driven through a quill coupling from a support spindle driven by flat belts and 25 horsepower electric motors. One of each pair of test bearings was mounted in a floating housing and was thrust loaded against the other with a calibrated hydraulic cylinder.

Lubrication is provided by jets located between the test bearings. Two test lubrication systems, separate from the support bearing lubrication systems, were used, each supplying lubricant to two test heads (four test bearings). Each test lubrication system contained a heated 4-gallon sump, a 1.7-gallon per minute at 100 psi supply pump, a nominal 10-micron absolute filter, and a turbine type flowmeter. Scavenge from the test housings was by gravity drain lines. Bearing outer-race temperature and lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded on a strip chart recorder. Oil flow rates, spindle speeds, load system pressure, and lubrication system pressures were periodically recorded. Continuous, unattended 24 hours per day, 7 days per week operation was accomplished.

Cylindrical roller bearing test head. - A section view of a test head used to test the 111-size cylindrical roller bearings is shown in figure 8. Three of these test heads were used, each containing four test bearings and two support ball bearings. The test spindles were driven by 7.5 or 10 horsepower electric motors and a geared speed increaser. Power was transmitted from motor to gear box by pulleys and V-belts. Flat belts transmitted power from the gear box to the test spindle. Each pair of test bearings was radially loaded by a spring scale and lever arm arrangement.

Lubricant was supplied to the test bearings through jets located between each pair of test bearings. Each test head has a separate test lubrication system with a heated 4-gallon capacity sump, a supply pump, a water-cooled heat exchanger, a nominal 10-micron filter, a volumetric flow meter, and a scavange pump. A separate lubrication system was used for the support bearings.

Test bearing outer-ring temperatures, oil inlet and outlet temperatures were measured with thermocouples and were continuously recorded on strip charts. Machine vibration was continuously monitored and in conjunction with several automatic pressure and temperature monitoring and safety devices, the test rigs were capable of 24 hours per day, 7 days per week, unattended operation.

Angular contact ball bearing test head. - A section view of a test head used to test the 7216-size angular-contact ball bearings is shown in figure 9. Four of these test heads were used, one on each end of two spindles driven through V-belts with 20 horsepower electric motors. A pair of test bearings were mounted in each test head in a back-to-back arrangement. The bearings were thrust loaded against each other by four bolts with a fifth strain-gaged bolt used to set and monitor the thrust load magnitude. The desired radial load was applied to each pair of bearings with a dead weight load through a 10 to 1 lever arm.

Lubrication is provided by jets located between each pair of test bearings. A single lubrication system supplied lubricant to all eight bearings tested simultaneously. This lubrication system was separate from the system supplying lubricant to the load bearings. The system for the test bearings contained a heated 12-gallon sump, a 25-gallon per minute supply pump, a constant pressure bypass system, a water-cooled heat exchanger, a nominal 10-micron full flow filter, and a 30-gallon per minute scavange pump. Lubricant temperatures were monitored by thermocouples, and flow rate was monitored by calibrated visual gages.

Continuous, unattended 24-hour per day, 7 days per week operation was accomplished utilizing a computer control system. Test bearing outer-ring temperature, thrust load, and machine vibration were continuously monitored. Lubricant inlet and outlet temperatures, oil flow rate, and spindle speed were periodically monitored.

Test Results

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 hours 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 hours. 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 thirty 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 hour time without failure.

Results with 210-size split-inner-race ball bearing. - The bearings in this group were from a helicopter turbine engine, front compressor bearing position. Eight bearings were tested simultaneously with two bearings in each of four test heads. In order to complete the 1600 hour 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 hour 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 252° to 275° F and averaged about 260° F. Oil-out temperature ranged from 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 (275° F) (also most discolored bearing) and the bearing with the lowest outer-race temperature (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. (It is recommended and is common practice to use AISI M-50 for all turbine engine main shaft bearings.)

Results with 111-size cylindrical roller bearings. - The bearings in this group were from a helicopter turbine engine, in the rear compressor bearing position Twelve bearings were tested simultaneously with four bearings in each of three test heads. To complete the 1600 hour 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 220° to 245° F. The oil-out temperature ranged from 205° to 220° F.

Twenty-nine test bearings completed the desired 1600 hour duration. One bearing suffered a failure after only 16.3 hours. 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 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 hour 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 (6×) 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 the pitting, as measured from surface profile traces and metallographic sections, was typically less than 0.0005 inch. 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 4 or 5 μin. RMS in all cases. The roller cylindrical surfaces measured from 4 to 6 μin. RMS. The inner raceway surfaces ranged from 3.5 μin. to 17 μin. RMS. The engine manufacturer's drawing for this bearing specifies 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 10 μin., surface distress was observed on the rollers. On two bearings, where inner raceway finishes were 6 and 7 μin. RMS, some roller surface distress was observed. On the other bearings, where inner raceways varied from 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 27 μin. An accepted criterion for the effectiveness of the EHD film thickness in given conditions with given bearing surfaces is the ratio of EHD film thickness to the composite surface roughness. This ratio is often referred to as film thickness parameter or Λ. The composite surface roughness 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 10 μin. RMS or greater, Λ was 2.4 or less. In the worst cases, Λ was as low as 1.5. Where Λ is less than 3, it should be expected that some asperity contacts will exist (refs. 17 and 18). The detrimental effects of this surface-to-surface contacts is 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 these 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.

Results with 2716-size angular-contact ball bearing. - The bearings in this group were from triplex sets of a helicopter transmission input pinion bearing. The tandem pair and preload bearing arrangement is shown in figure 5. Thirty 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 hour 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 185° to 195° F. Oil-out temperature ranged from 170° to 180° F.

After 1122 hours 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 hour 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.

General Comments

The results of tne program indicate that bearing restoration is technically feasible. The experience gained from testing the three groups of bearings indicate that the performance of the restored bearings are comparable to brand new bearings. It was originally speculated that infant mortality of the restored bearings would be eliminated or at least minimized, because those bearing raceways which would have been inherently defective have been eliminated during field operation. However, the one ball failure and one roller failure experienced on this program indicate that infant mortality can still be a problem because of potentially defective new rolling elements. This problem would be no greater than that currently being experienced. In other words, while restored bearings can be expected to be comparable to newly manufactured bearings, it would be unrealistic to assume that their group life potential would be greater.

A necessary element in the bearing restoration process is to assure proper quality control of the restored bearings. Further, it is necessary to assure manufacturing consistancy comparable to newly manufactured bearings. An additional element on the bearing restoration process is the assurance that a potential vendor's restoration process will not adversely affect the life or quality of a bearing. As a result of the aforementioned AVSCOM and NASA jointly developed a proposed "Specification for Restoring Bearings by Grinding" which is included as an appendix to this report.

The specification incorporates both the technical and quality assurance aspects which have been defined both in the bearing restoration program and recent advances in rolling-element bearing state-of-the-art. The raw material and melting specifications for rolling elements calls for VIM-VAR AISI M-50 and VIM-VAR AISI 52100 assuring a longer rolling-element life potential then is currently being obtained. Minimum grinding depths are established but the exact amounts are left to the discretion of the vendor. Hardness and surface finish specifications are also tightened from previous AVSCOM practice. Research performed by the NASA indicates both factors to be critical to rolling-element bearing life and reliability. Results of the tests reported herein indicate that surface distress due to marginal elastohydrodynamic film thickness with the existing bearing surface finishes is an inherent defect in current bearing design specification as practiced by AVSCOM bearing suppliers.

The use of AISI M-50 steel is specified in aircraft bearings instead AISI 52100 because of the improved ability of the AISI M-50 steel to retain its hardness at elevated temperatures. Unfortunately, when current bearing specifications are such as to set a minimum hardness of Rockwell C-59 at room ambient temperature, the hardness of the material at bearing operating temperatures of 250° F falls below acceptable values. This minimum hardness defeats the use of the AISI M-50 steel and can result in premature bearing damage and failure. As a result, the hardness criteria for the proposed specification is tightened from that of current AVSCOM practice.

Material and part traceability is another important factor. The ability to trace defective critical bearings in a system can be extremely important when failures are experienced in the field. Traceability is not now being practiced by the U.S. Army. A defective group of bearings or improperly processed or heat treated steel can occur both in new, refurbished, or restored bearings. Where the failure mode can cause a loss of an aircraft, it becomes extremely important to be able to remove all such defective bearings from the military aviation system. Presently, this cannot be accomplished.

Original equipment bearing vendors have a proven capability manufacturing acceptable bearings for a given application. Not all potential vendors for the restoration of aircraft quality bearings have this proven record of bearing manufacturing performance for aircraft quality bearings. This does not mean that these vendors should be excluded from the aircraft bearing restoration. As a result, the specification (appendix) requires that a prospective vendor endurance test a 30 bearing lot of aircraft ball bearings restored in accordance with this specification to qualify to restore ball bearings only. In addition, in order to qualify for roller bearing restoration, the specification calls for the vendor to endurance test a 30 bearing lot aircraft roller bearings. Once the vendor performs these tests, further testing will not be required. The original equipment bearing vendor is exempt from this testing only for those bearings which he has originally manufactured and subsequently restored. Should this vendor decide to restore bearings of another manufacturer, the vendor would be required to endurance test bearings to ensure the necessary life potential.

Quality control will be the responsibility of the military overhaul facility and the vendor. Current quality control procedures if properly enforced should result in quality equivalent to new bearings being purchased.

Evaluation of restored tapered-roller bearings has not been examined in this program. However, there is no reason to believe that bearing restoration by grinding, as described in this paper, could not be applied to tapered-roller bearings which are manufactured from case-carburized steels.

SUMMARY OF RESULTS

A joint program was undertaken by the NASA Lewis Research Center and the Army Aviation Systems Command to restore by grinding those rolling-element bearings which are currently being discarded at aircraft engine and transmission overhaul. Three bearing types were selected from the UH-1 helicopter engine (T-53) and transmission for the pilot program. These bearings were a 210-size split-inner-ring ball bearing and a 111-size cylindrical roller bearing used on the T-53 engine compressor shaft, and a 7216-size angular-contact ball bearing triplex set from the transmission input pinion shaft. Groups of each of these bearings were visually and dimensionally inspected for suitability for restoration. Restorable bearing yields were determined for each bearing type on the basis of bearings (or sets) and on the basis of components (rings and separators).

Fifty of each of the split-inner ring ball bearing and the cylindrical roller bearing and 50 triplex sets of the angular-contact ball bearing were restored. The restoration included regrinding raceways, plating and regrinding bores, outside diameters, and faces, stripping and replating separators, and installing new oversize balls or rollers while maintaining clearances and external geometries and tolerances in accordance to original drawing specifications.

Endurance tests were performed on 30 of each of the three types of restored bearings to verify that they will satisfactorily operate at speed, load, and lubrication conditions typical of the respective application for a period of 1600 hours. The bearings were subsequently disassembled and examined to determine the success of the restoration and endurance test program. The following results were obtained:

(1) No bearing failures occurred in the 1600 hour endurance tests with all three types of bearings which could be related to the restoration by grinding process.

(2) The bearing failures that occurred, both due to defective rolling elements, were typical of "infant" failures which may occur in new bearings.

(3) The restorable component yield (rings and separators) was 98.2 percent for the split-inner-ring ball bearing, 96.7 percent for the cylindrical roller bearing, and 91.6 percent for the triplex angular-contact ball bearing.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the cooperation of personnel from the Corpus Christi Army Depot for identifying, accumulating, and supplying the used bearings for this program. We would like to acknowledge the cooperation of personnel from Industrial Tectonics, Incorporated, Bearing Division for inspecting and subsequently restoring the used bearings under contract to the U.S. Army Aviation Systems Command. In addition, acknowledgement is given to MAIC Division of Pure Carbon Company, MRC Division of TRW, Incorporated, and SKF Industries, Incorporated, Engineering and Research Center for endurance testing and subsequently examining the restored bearings under contract to the U.S. Army Aviation Systems Command.

APPENDIX

SPECIFICATION FOR RESTORING BEARINGS BY GRINDING

1. SCOPE

1.1 Scope. This specification presents requirements for the restoration by grinding of rolling-element bearings.

1.2 Definitions. For purposes of this specification, the following definitions shall apply:

Rolling-element bearing - A rolling-element bearing is a bearing having an inner race, an outer race, and a plurality of either balls, cylindrical rollers, crowned rollers, or spherical rollers.

Rolling-element - A rolling element is either a ball, a cylindrical roller, or a spherical roller.

Cage or separator - A cage or separator separates positions and/or retains the rolling elements between the inner and outer raceway.

Restoration - The process of grinding raceways and replacing rolling elements in accordance with this specification.

Vendor - Contractor or bidder for restoring by grinding.

Purchaser - U.S. Government or its agencies unless otherwise specified.

2. APPLICABLE DOCUMENTS

2.1 The following documents shall form a part of this specification where applicable to the extent specified herein. Unless a specific issue is specified, the latest revision shall apply.

ORIGINAL MANUFACTURER'S DRAWINGS AND SPECIFICATIONS

AEROSPACE MATERIALS SPECIFICATION

AMS 6490	Steel Bars, Forgings and Tubing 4.0 CR-4.25 Mo-1.0V (0.77-0.85C) Premium Bearing Quality, Consumable Electrode Vacuum Melted
AMS 6444D	Bars, Forging, and Mech. Tubing 1.45 Cr (0.98-1.10C) Premium Bearing Quality, Consumable Electrode Vacuum Melted

AMS 2410E	Plating-Silver, Nickel Strike, High Bake
AMS 2412E	Plating-Silver, Copper Strike, Low Bake
AMS 2406E	Plating-Chromium, Hard Deposit
AMS 2404A	Plating-Nickel, Electroless

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

ASME Life Adjustment Factors for Ball and Roller Bearings - An Engineering Design Guide, 1971.

AMERICAN NATIONAL STANDARDS INSTITUTE

ANSI B46.1 Surface Texture

AMERICAN SOCIETY FOR TESTING MATERIALS

ASTM E18	Rockwell Hardness and Rockwell Superficial Hardness of Metallic Materials
ASTM E112	Average Grain Size of Metals

MILITARY

MIL-B-197	Bearing, Rolling element, associated parts

3. REQUIREMENTS

3.1 <u>Inspection</u>

3.1.1 All bearings for restoring shall be inspected by vendor prior to restoring at vendor's place of business and at vendor's cost.

3.1.1.1 All inspection and test procedures shall be in accordance with ASTM methods unless otherwise agreed upon by the vendor and the purchaser. Where ASTM methods do not exist, the vendor shall submit inspection and test procedures to the purchaser for approval.

3.1.1.2 The vendor shall inform the purchaser of the test and inspection procedures to be used. Once these procedures are established they shall not be changed without prior approval in writing of the purchaser.

3.1.2 All bearing components for restoring except the rolling elements shall be visually, dimensionally, and hardness inspected in accordance with 3.1.1 hereinabove.

3.2 Rejection

3.2.1 All bearing rolling elements shall be rejected without inspection.

3.2.2 Cages or separators not meeting original manufacturer's engineering drawing dimensions or specifications or exhibiting gross wear shall be rejected. All plating shall be removed from separators prior to inspection.

3.2.3 All races having nicks, dents, or other damage on the raceway surface extending to a depth greater than 0.002 inch and/or whose hardness does not conform to original manufacturer's engineering drawing specifications shall be rejected.

3.2.4 Bearings which have been previously subjected to restoration by grinding shall be rejected.

3.3 Raw Materials

3.3.1 AISI M-50 material supplied to this specification for rolling elements shall conform to all requirements of AMS 6490 except as described in the following paragraphs.

3.3.1.1 The AISI M-50 steel supplied to this specification shall have the following composition:

Carbon	0.80-0.85	Molybdenum	4.00-4.50
Silicon	0.15-0.35	Vanadium	0.90-1.10
Phosphorous	0.10-0.25	Nickel	0.10 max.
Sulfur	0.015 max.	Cobalt	0.25 max.
Chromium	0.010 max.	Tungsten	0.25 max.

3.3.2 AISI 52100 material supplied to this specification for rolling elements shall conform to all requirements of AMS 6444D except as described in the following paragraphs.

3.3.3 Materials other than those of paragraphs 3.3.1 and 3.3.2 shall conform to applicable AMS requirements. If, however, AMS requirements do not exist for the material, the material shall conform to the original bearing manufacturer's requirements.

3.3.4 Melting Method - Material shall be produced by vacuum induction melting followed by vacuum consumable electrode remelting. The vacuum consumable electrode remelting process shall be capable of consistently producing ingot that is uniform, yielding a billet free from voids and segregation. Inclusions shall not exceed the limits of AMS 6490. The critical vacuum arc, consumable electrode remelting process variables shall be continuously monitored and recorded by the melter, and in the event that they go beyond the established limits, in an isolated instance, that area of the ingot shall not be used. The established melting practices shall not be changed without informing the purchaser and receiving approval in writing therefor.

3.4 Process Requirements

3.4.1 All raw material shall be procured only from sources approved by the purchaser.

3.4.2 Vendor may chrome or nickel plate in accordance with the requirements of AMS 2406E or AMS 2404A, the bore surfaces of the inner rings and outside diameter surfaces of the outer rings, face surfaces, and race land surfaces with an alloy to restore these surfaces to original dimensions.

3.4.3 Vendor shall grind and superfinish raceway surfaces and flange surfaces on roller bearing races removing material from said surfaces to a depth not less than 0.002 inch nor more than the depth of maximum shear stress under the maximum operational load conditions for the bearing. The vendor shall define the depth of maximum shear stress. The inner and outer raceways and flanges shall be ground to equal depths.

3.4.4. Vendor shall install the correct number of new rolling elements in the bearings.

3.4.4.1 New balls installed in the bearing shall have a diameter equal to the original ball diameter plus twice the grinding depth specified in paragraph 3.4.3. The original print IRC shall be maintained.

3.4.4.2 New rollers installed in the bearing shall have a diameter equal to the original roller diameter plus twice the grinding depth specified in paragraph 3.4.3. The roller length shall be equal to the original roller length plus twice the grinding depth specified in paragraph 3.4.3. The original print IRC shall be maintained.

3.4.4.3 Material for new balls or rollers to be installed in the bearings shall conform to the specifications in paragraphs 3.3, 3.5, 3.7, and 4.

3.4.4.4 Where specified by the original bearing manufacture, separators shall be plated in accordance with AMS 2410 or AMS 2412 as applicable

3.4.5 Vendor shall inform the purchaser of all manufacturing processes and procedures and inspection and quality control procedures used to produce parts to this specification and receive written approval from the purchaser therefore. Once these practices are established, they shall not be changed without informing the purchaser, and receiving prior approval in writing from the purchaser therefor.

3.5 Furnished Part Requirements

3.5.1 Hardness - All bearing rollers and balls shall have an average hardness of Rockwell C62 to 64. All hardness values of these rolling elements shall be in the range Rockwell C61 to 65. Race hardness shall be according to original bearing manufacturer's specification unless otherwise specified by the purchaser.

3.5.2 Dimensions and Tolerances - All dimensions and tolerances shall be in accordance with original manufacturer's engineering drawings and specifications except as described in the following paragraphs.

3.5.2.1 All balls and roller dimensions shall be in accordance with paragraph 3.4.4.

3.5.2.2 Inner raceways may have diameters less than original specifications by an amount equal to twice the grinding depth specified in paragraph 3.4.2.3. Outer raceways may have diameters greater than original specifications by an amount equal to twice the grinding depth specified in paragraph 3.4.3.

3.5.2.3 Raceway groove radius shall be modified to reflect larger ball diameter while maintaining the same conformity as in the original specifications.

3.5.2.4 Width between flange surfaces of cylindrical roller bearings shall be increased by an amount equal to twice the grinding depth specified in paragraph 3.4.2.3 where applicable.

3.5.2.5 Internal radial and axial clearance, all other applicable internal geometry, and roller-to-race end clearance as applicable shall conform to original bearing manufacturer's specifications.

3.5.3 Surface Finish

3.5.3.1 The surface finish of balls shall be 2 microinches AA or better; ball raceways shall be in accordance with original specifications or better.

3.5.3.2 The surface finish of rollers shall be 6 microinch AA or better on the O.D. and the ends. Roller raceways shall be in accordance with original specifications or better.

3.5.3.3 Raceway surface finish measurements shall be made in the axial direction.

3.5.4 Retained Austenite - The retained austenite content shall not exceed 3 precent as measured in accordance with paragraph 4.3.1 hereinbelow.

3.6 Traceability

3.6.1 Vendor shall assign to each restored bearing a number preceded by the letter "R" and the vendors trade mark or symbol. This number shall be marked on a face of each of the inner and outer races.

3.6.2 Records shall be maintained by the vendor to provide traceability for each new part to its corresponding heat treat lot, and heat of steel.

3.6.3 Records shall be maintained by vendor of all restored bearings by original part number, restored bearing number, serial number, and date of shipment for a period of 15 years from the date of shipment.

3.7 Material Record Requirements

3.7.1 Material identification record shall be retained by the bearing vendor for 15 years from the date of completion of the order. When requested by the purchaser, the records shall be made available for delivery within three working days. This record shall include, as a minimum, the following information:

(a) Forging vendor's certificate of test, as applicable

(b) Purchase order number

(c) Specific heat treatment cycle used (forger and/or bearing vendor)

(d) Numerical results of all required tests and inspections

(e) Macro and micro examination test results

(f) Heat number

(g) This specification number, CLASS and revision number

3.8 Qualification Testing

3.8.1 A lot of ball and roller bearings restored in accordance with this specification, where applicable, shall be tested by the vendor at no cost to purchaser to establish the reliability of bearings restored according to the vendor's manufacturing processes and procedures used to produce parts to this specification except as noted in paragraph 3.8.2 hereinbelow.

3.8.1.2 A lot of bearings for qualification testing shall comprise not less than 30 bearings.

3.8.1.3 Test conditions shall reasonably duplicate in a test rig those maximum speed, load and temperature conditions to which the bearing is subjected in aircraft applications. Bearing lubrication and lubricant shall be similar to that in the aircraft application. Test duration shall be the aircraft TBO time or the bearing L_{10} life calculated in accordance with ASME methods whichever be the shortest duration.

3.8.1.4 All bearings in a lot shall run to the time specified in paragraph 3.8.1.3 hereinabove without failure in the raceways or cage. Should any bearing or bearings fail by lubrication starvation because of test rig failure, said bearing or bearings shall be eliminated from the lot and other restored bearings substituted in their place.

3.8.2 Vendors Exempt from 3.8.1

3.8.2.1 Original vendor of proposed bearings to be restored are exempt from provisions of paragraph 3.8.1 but only to those specific bearing part numbers which said vendor manufactured.

3.8.2.2 Vendors who have restored bearings by grinding in accordance to this specification and which bearings have been tested in accordance with paragraph 3.8.1 inclusive shall be exempt from further testing providing the test results and documentation are made available to purchaser.

4. QUALITY ASSURANCE PROVISIONS

4.1 General

4.1.1 All test procedures except as herein specified shall be in accordance with ASTM methods unless otherwise agreed upon by the vendor and the purchaser in writing.

4.1.1.1 The vendor shall inform the purchaser of the test and inspection procedures to be used. Once these procedures are established they shall not be changed without prior approval of the purchaser.

4.1.2 Samples, representative of the shape and size of each forging and which have been processed through all forging operations along with the parts they represent, shall be tested to show conformance to the requirements of this specification. The frequency and the number and types of tests shall be performed to the requirements of a Quality Control Plan approved by the purchaser.

4.1.3 Finished parts shall be periodically cut-up and tested to the requirements of this specification. The frequency of testing and the number and types of tests shall be performed to the requirements of a Quality Control Plan approved by the purchaser.

4.2 Grain Size

4.2.1 Grain size shall be determined per ASTM E112 on samples representing each heat treat lot of material. For referee tests, grain size shall be determined by comparison of a polished and etched specimen with the chart in ASTM E112.

4.3 Retained Austenite

4.3.1 Retained austenite shall be determined by x-ray diffraction techniques unless otherwise approved by the purchaser. If methods other than x-ray diffraction are used, the method shall be calibrated to and show agreement with x-ray diffraction measurement.

4.4 Hardness

4.4.1 Hardness tests shall be conducted in accordance with ASTM E18.

4.5 Surface Finish

4.5.1 Surface finish shall be determined in accordance with ANSI B46.1.

5. PREPARATION FOR DELIVERY

5.1 Packing

5.1.1 All parts shall be suitably packed in accordance with MIL-B-197 to prevent damage or loss in shipment. (Class of preservation shall be specified by vendor.)

5.2 Marking

5.2.1 Each shipment shall be legibly marked, as a minimum with the purchase order number, manufacturer's name, part name, and part identification numbers.

6. NOTES

6.1 Classification of Characteristics

CRITICAL:	3.2.3, 3.4.3, 3.4.4.1, 3.4.4.3, and 3.5.1
MAJOR:	3.5.2 and 3.5.3
MINOR:	All other paragraphs

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TABLE I. - BEARING IDENTIFICATION AND SPECIFICATION

Bearing	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular-contact ball bearing
Federal stock number	3110-727-3032	3110-071-4568	3110-135-2603
Component part number	1-300-015-04	1-300-176-03 (1-300-176-04)	205-040-246-3
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 (20)	15
Cage material	Bronze	Steel (bronze)	Steel
Cage type	One piece, inner-land riding	One piece, inner-land riding	One piece, inner-land riding

TABLE II. - RESTORABLE BEARING YIELD

	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular contact ball bearing triplex sets
Number of bearings (or sets) for inspection	150	121	86
Number of bearings (or sets) restorable	145	114	55
Restorable bearing yield, percent	96.7	94.2	64.0
Number of restorable components	442	351	701*
Restorable component yield, percent	98.2	96.7	91.6

*The 86 triplex sets included 765 components, since three bearings from two sets were not returned from the overhaul depot for inspection.

TABLE III. - 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, lb	0	100	1905
Thrust load, lb	450	0	2564
Lubricant inlet temperature, $^{\circ}F$	195±5	195±5	150±10
Lubricant flow rate, gpm	0.38±0.043	0.38±0.043	2.1±0.3
Sump capacity, gal	4	4	12
Number bearings per sump	4	4	8
Lubricant change interval, hr	100	100	600

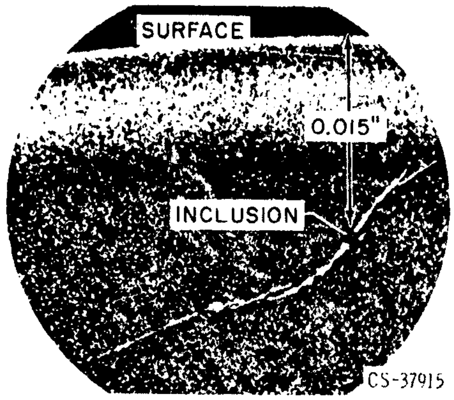

Figure 1. - Fatigue crack emanating from an inclusion.

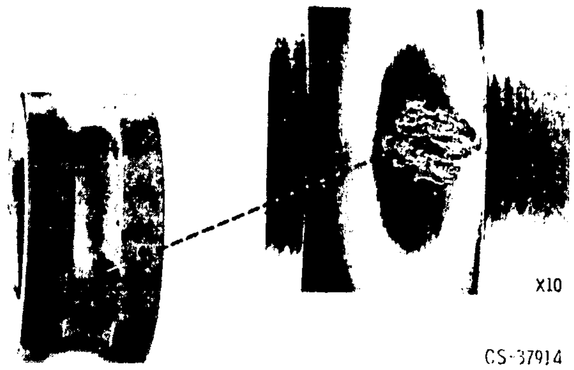

Figure 2. - Typical fatigue spall in bearing race.

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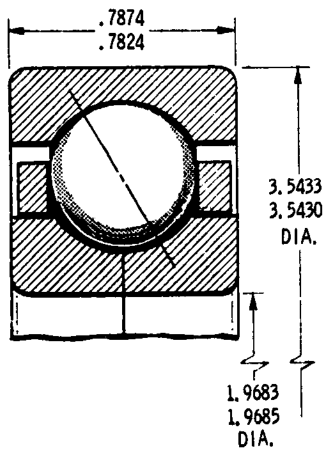

Figure 3. - 210-size split-inner-ring ball bearing.

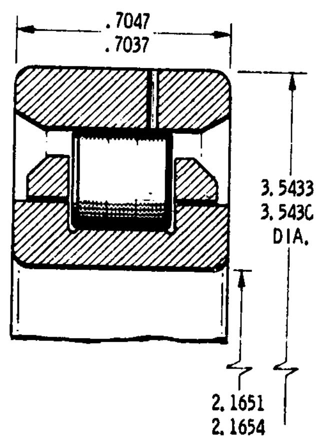

Figure 4. - 111-size cylindrical roller bearing.

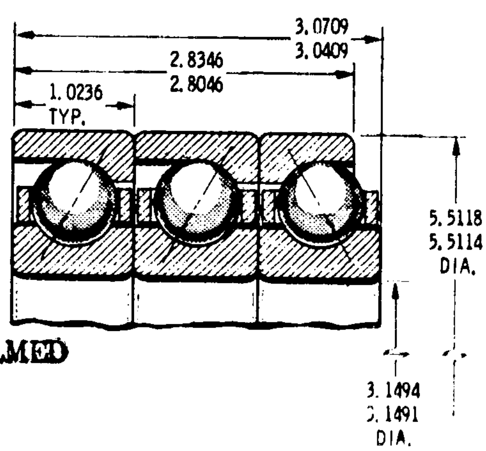

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Figure 5. - 7216-size angular contact ball bearing triplex set.

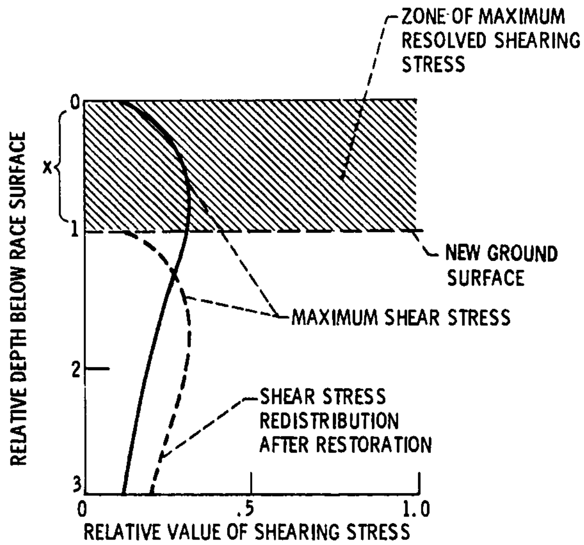

Figure 6. - Relative value of shearing stress as function of depth below rolling-element surface.

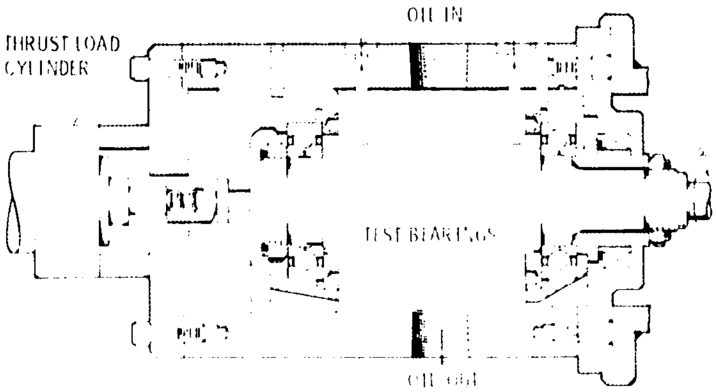

Figure 7. - [illegible]

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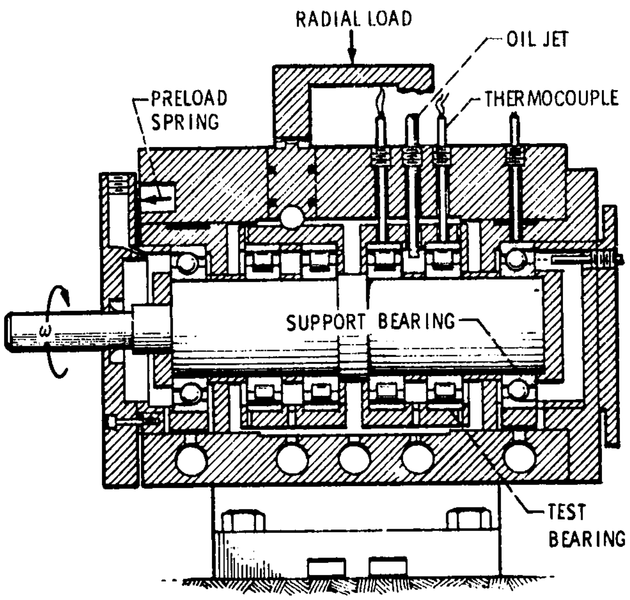

Figure 8. - Test head for 111-size cylindrical roller bearings.

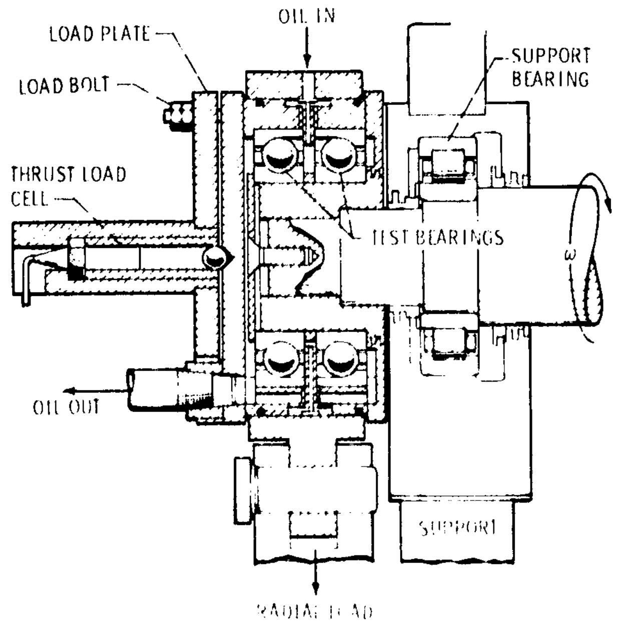

Figure 9. [illegible]