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EFFECT OF FILTRATION ON ROLLING-ELEMENT-BEARING LIFE IN A CONTAMINATED LUBRICANT ENVIRONMENT

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SUMMARY

Fatigue tests were conducted on groups of 65-millimeter bore diameter deep-groove ball bearings, under four levels of filtration with and without a contaminated lubricant. The baseline test series used a noncontaminated, MIL-L-23699 type, test lubricant installed in a recirculating lubrication system containing a 49 micron absolute (30 microns nominal) full flow filter. Four series of tests were conducted in which contaminants, similar to that found in gas turbine engine filters, were injected into the test filter's supply line at a constant rate of 125 milligram per hour per bearing. The test filters had absolute particle removal ratings of 3, 30, 49, and 105 microns (0.45, 10, 30, and 70 microns nominal), respectively. Test conditions included a bearing shaft speed of 15 000 rpm, a radial load of 4580 newtons (1030 lb) yielding an inner-race maximum Hertz stress of 2410 MPa (350 000 psi) and a bearing lubricant supply temperature of 347 K (165° F). The bearings operated with a full EHD lubricant film thickness to composite surface roughness ratio of 3.3

Bearing life and running track condition generally improved with finer filtration. The bearings tested in contaminated oil with 3- and 30-micron absolute filters had statistically equivalent lives, approaching those of the baseline, noncontaminated lubricant bearings.

Surface distress and wear of the test bearings running track markedly increased with coarser filter size. The bearings tested with 105-micron filtration experienced failure from gross wear without spalling. The weight loss of unfailed bearings tested with 3-, 30-, 49-, and 105-micron filters were, respectively, 1.9, 3.2, 4.2, and 89 times the weight loss of the baseline bearings.

Fatigue failures were both surface and subsurface initiated with a trend toward more surface initiated failures with coarser filtration. An increase in Weibull slope with coarser filtration was also observed.

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INTRODUCTION

It is well recognized that fatigue failures which occur on rolling-element contacts are a consequence of competitive failure modes developing primarily from either surface of subsurface defects (refs. 1 to 5).

Subsurface initiated fatigue, that which originates slightly below the surface in a region of high shearing stress, is generally the mode of failure for properly designed and well maintained rolling-element bearing and gear systems. This failure mode, in essence, determines the upper bound of rolling-element service life provided that all other surface related failure mechanisms have been circumvented.

Surface initiated fatigue, often originating at the trailing edge of a localized surface defect, is the most prevalent mode of fatigue failure in machinery where strict lubricant cleanliness and/or sufficient elastohydrodynamic film thickness are difficult to maintain (see appendix for additional background information). Nicks and scratches arising from machining or handling, debris dents and surface layer inclusions act as localized stress raisers from which fatigue cracks can propagate.

Furthermore, in rolling-element systems where significant amounts of combined sliding and rolling contact occur, such as for the contact between gear teeth and the cone-rib contact in a tapered roller bearing (ref. 6), the presence of debris in the oil can lead to gross wear damage and failure.

Presently there is a scarcity of test data relating the quantitative effects of filter performance on rolling-element component life. The objective of this study is to quantify the effects of filtration level on the service life of rolling-element bearings which operate in a controlled contaminated lubricant environment.

In order to accomplish this objective, fatigue tests were conducted on a multiple bearing fatigue test machine with 65-millimeter-bore-diameter deep-groove ball bearings that were lubricated with MIL-L-23699 qualified, neopentylpolyol (tetra) ester oil. Four levels of filtration were investigated utilizing static filters with absolute particle removal ratings of 3, 30, 49, and 105 microns. The results of these tests were compared to a controlled baseline series of tests in which no contaminants were added to the test lubricant. Test conditions consisted of a bearing shaft speed of 15 000 rpm, a radial load of 4580 newtons (1030 lb) and an oil in temperature of 347 K (165° F). The test environment has been designed to simulate an aircraft-type lubrication system in that it contains multiple bearings, pumps, valves, and other hydraulic components commonly found in such systems.

TEST MATERIALS

Test Bearings

The test bearings were ABEC-3 grade, deep groove ball bearings with a 65-millimeter bore diameter, a 90-millimeter output diameter, a 13-millimeter width and contained 18 balls having a 7.94-millimeter (5/16-in.) diameter. A photograph of a disassembled test bearing appears in figure 1. The inner and outer races as well as the balls were manufactured from a single heat of carbon vacuum-degassed (CVD) AISI 52100 steel. The nominal hardness of the races and balls were Rockwell C 62±1.0. The ball retainer was a riveted, two-piece, machined, inner-land riding cage made from silicon-iron bronze. The raceways were ground to a nominal surface finish of 0.1 microns (4 μin.) rms and the balls to a surface finish of 0.05 microns (2 μin.) rms.

Test Lubricant

The oil utilized for this test program was a neopentylpolyol (tetra) ester for which a substantial amount of test data is available (ref. 7). This oil is qualified for use in jet-engine lubrication systems under MIL-L-23699 specifications. Its properties are given in table I.

Test Filters

All of the test filters used during the course of this study were of porous-depth media construction. Manufacturer's specifications of these commercially available test filters along with the test series number in which they were used are presented in table II.

The absolute removal ratings shown in table II have been determined from traditional filter test methods per MIL-F-27656 where a known distribution of graduated glass beads are fed "once through" the test filter. The largest diameter of glass bead that passes through the filter on to a microscopic slide establishes its absolute removal rating. This rating method can give misleading results since it does not accurately take into account the construction of the filter media nor does it reflect the time dependent performance of the filter in a recirculating lubrication system.

The work performed at Oklahoma State University (refs. 8 and 9) was fundamental in pointing out the dynamic nature of filter performance. This research has led to a standard test procedure (ANSI B93.31-1973) known as the multipass filter test in which a test dust slurry is continuously recirculated through the test filter while taking upstream and downstream samples to determine the filter's instantaneous efficiency (ref. 9). Unfortunately filtration ratio charts for the test filters used in this study are presently unavailable.

Test Contaminants

The contaminants used in this study consisted of a mixture of carbon dust, Arizona coarse test dust, and stainless steel powder. The composition of this contaminant mixture, as listed in table III, was similar to the composition of particulate matter found in the lubricant filters of typical aircraft gas turbine engines, based on a field survey of 50 JT8D commercial engines. Analysis of debris from the engine lubrication system showed that up to 90 percent of the particles were carbonaceous in composition with the remainder being primarily silaceous and metallic. To give a physical appreciation of the fineness of the test contaminant mixture in which nearly all of the particles are less than 40 microns in size; the period at the end of this sentence is greater than 700 microns in diameter.

APPARATUS AND PROCEDURE

Bearing Fatigue Test Machine

A plan view of the bearing test machine utilized in this investigation is shown in figure 2. Two identical bearing fatigue testers, containing four test bearings each, were operated concurrently. Each bearing tester is driven by a quill shaft. The high-speed jackshafts are driven by flat belts from a single low-speed jackshaft which in turn is driven by a flat belt from the output shaft of the 37.3-kilowatt (50-hp) variable speed drive (eddy current clutch and ac motor). The use of the variable speed drive prevents the bearings from undesirable high acceleration during startup.

Figure 3 shows a cross section of the bearing fatigue tester. Each tester consists of a shaft on which four 65-millimeter bore diameter bearings are mounted. The bearings are loaded in a radial direction only, by a hydraulic cylinder. The radial load is transmitted to the two center bearings through a wiffletree which divides the applied load equally and compensates for small differences in bearing internal clearance. The radial load is reacted by the two outboard bearings as shown. Thermocouples were mounted to the outer wall of each of the test bearings. Each test bearing had its own calibrated oil jet.

A schematic of the test lubrication supply system is shown in figure 4. It simulates an aircraft type of lubrication system in that it contains multiple bearings, pumps, hydraulic valves, heat exchanger, and other hydraulic components which are commonly found in such systems.

The lubrication supply system (see fig. 4) delivers oil to the test bearings at a constant rate of 136+6.8 kg/hr (300±15 lb/hr). The flow to each bearing is metered by a fixed orifice in the oil jet that was calibrated to give the proper oil flow. The oil is directed at the contact land between the bearing inner race and the cage. Excess oil supplied by the supply pump is bypassed back to the supply tank by the pressure regulating valve that maintains the pressure upstream of the filter at 0.62 MPa (90 psi). The total oil flow required for the eight test bearings, 1090 kg/hr (2400 lb/hr), passes through the test filter. A pressure reducing valve maintains the bearing lubricant supply pressure at a constant 0.28 MPa (40 psi).

From the test bearings the oil gravity drains into a collector pan from where it is returned to the oil supply tank by the scavenge pump. On this return line is an oil-water heat exchanger that regulates the oil in temperature to the bearings at 347 K (165° F). An automatic oil sampler for taking periodic oil samples is located between the scavenge pump and the heat exchanger. The oil sampler removes a small cross section of oil from the scavenge line each time it cycles.

For the contaminated lubricant tests, the contaminant injection system shown in the upper right corner of figure 4 was used. The test contaminant and replenishment oil are mixed together in an oil slurry form and added to the contaminant mixing tank. The hydraulic intensifier injects approximately 12 milliliters of contaminant slurry per stroke into the oil supply line ahead of the test filter during each cycle. The injection cycle rate is controlled by an on-off timer.

The test stand instrumentation included an accelerometer system which detects bearing failures as well as the standard protective circuits which shutdown the drive system, if any of the test parameters deviate from the programmed conditions.

Parameters monitored and recorded during the test were rig shaft speed (bearing inner race speed), oil flow to each tester, test bearing outer race temperature, lubricant supply and scavenge temperature, and rig vibration level. The rig shaft speed was measured with a magnetic pickup and 60-tooth gear. The oil flow was measured with a turbine-type flow meter. All temperatures were measured with Chromel-Alumel-type thermocouples.

Test Conditions

The operating conditions selected for this study were representative of those which might be encountered by rolling-element bearings subject to a heavy duty application, not unlike those found in helicopter gearboxes or large ground vehicles.

Test conditions consisted of a bearing shaft speed of 15 000 rpm and a radial load per bearing of 4580 newtons (1030 lb). This radial load resulted in a theoretical maximum contact stress of approximately 2410 MPa (350 000 psi) on the inner race. The temperature of the lubricant into the test bearings as well as the sump temperature was maintained at 347 K (165° F). Based on the average outer race temperature of 361 K (190° F) the theoretical isothermal elastohydrodynamic minimum film thickness at the inner race contact from Cheng's formula (ref. 10) is 0.373 microns (14.7 μin.) at these test conditions. This resulted in a λ ratio, the ratio of minimum film thickness to composite surface roughness, for the test bearing of 3.3 which provides a full EHD lubrication condition (ref. 11).

During the contaminated lubricant phase of the test program, test contaminants of the compositions previously described were periodically (approximately every 10 min) injected into the test filter lubricant supply line in 12-milliliter quantities, containing approximately 170 milligrams of contaminant. This produced a contamination rate of 125 milligrams of contaminants per hour per bearing or roughly a level teaspoon of contaminant powder every 16 hours. Based on the aforementioned test filter's oil flow rate, the contaminant injection rate resulted in an average concentration level supplied to the test filter of 0.9 milligram per liter. The amount and concentration of contaminants actually reaching the test bearings were some function of the filter's rating and efficiency, generally being only a minute fraction of the ingestion rate as determined by particle counts taken downstream of the filter.

Experimental Procedure

The test study was divided into five test series. In each series up to 32 test bearings were tested under a different level of contamination. Each set containing eight bearings was tested sequentially until a specific number of bearing failures had occurred or until each of the unfailed test bearings had received their designated number of test hours. Each new test series was initiated with eight new test bearings installed in clean test heads. After proper installation of the test bearings, new test oil that had been prefiltered through a 10-micron absolute filter was metered into the system. The stand lubrication system was then given time to warm up. The stand was brought up to test conditions within an hour in gradual increments of speed and load.

Test stand accelerometers were used to indicate test bearing failure. After a high vibration indication, the appropriate test head was disassembled. The test bearings were ultrasonically cleaned in a solvent and weighed. The test bearing or bearings that had the greatest weight loss, generally several times greater than the others, were tentatively identified as failed. These bearings were later confirmed as failed by inspection after disassembly. In addition to bearing weight changes, the radial clearance changes of the test bearings were also recorded but were not useful in identifying bearing failure. The failed bearing was replaced by a slave bearing which was not considered in any subsequent statistical bearing failure analysis. To continue the test, the replacement bearing along with the remaining unfailed test bearings were reinstalled and testing was resumed.

Experience with the lubrication system of actual aircraft gas turbine engines has shown that the viscosity and total acid number of the oil remains relatively constant due to the periodic replenishment of the oil consumed during operation with fresh oil containing the necessary additives. To simulate this oil management practice, a certain percentage of test oil was drained daily from the test sump and replaced with an equal amount of fresh oil. This replacement oil had the same concentration of contaminants, if any, as the test oil at the time of replenishment. Initially an oil replacement rate of 20 percent was selected but the constancy of the oil's viscosity and acid number permitted a 5 percent rate later in the study.

The baseline test series, designated as test series I, established the nominal fatigue life of the test bearings for a clean lubrication system with no external contaminants present in the oil. A filter with a 49-micron absolute particle rating removal was incorporated into the test lubricant system in keeping with some of the finer aircraft oil filters current in service.

Test series II to V were conducted in the same manner as test series I with the exception that filters of different ratings were installed into the lubrication system (see table II) and test contaminants of a prescribed composition (see table III) were metered into lubrication line ahead of the test filter. As in the case of test series I, periodic oil samples were taken from the lubrication line downstream of the test bearings and subjected to viscosity, total acid number, and particulate oil analyses.

The statistical methods of reference 12 for analyzing rolling-element fatigue data were used to obtain a plot of the log log of the reciprocal of the probability of survival as a function of the log of bearing test hours to failure (Weibull coordinates).

RESULTS AND DISCUSSION

Fatigue Tests

Fatigue life results of the bearings tested are summarized according to lubricant filtration level in table IV and their failure distributions are presented in figure 5.

In test series I, thirty-two bearings were fatigue tested to provide baseline data for this study. Ten of the 32 bearings experienced fatigue failure prior to the 2000-hour suspension time. All of the remaining bearings except one which was suspended after 27 minutes of operation for rough operation completed the test undamaged. One of the ten fatigue failures, a premature failure at 14 hours from the first group of bearings tested, was thought to be test rig related and consequently treated as an early suspension in accordance with reference 12.

At these test conditions the Anti-Friction Bearing Manufacturers Association (AFBMA) catalog 10-percent life is 47 hours. The experimental 10-percent life of the baseline bearings was 672 hours or more than 14 times the AFBMA forecasted life. Using the bearing life adjustment factors of reference 11, the predicted 10-percent life of the test bearings can be increased by a factor of 5.2 to 245 hours, still 2.7 times less than that measured. According to reference 11, data currently available is not statistically sufficient to assign a value for the potential fatigue life performance improvement of carbon vacuum degassed (CVD) steel relative to air-melted steel. However, some of the data presented in reference 11 suggests the certain vacuum degassing methods can show fatigue performance approaching that exhibited by consumable electrode vacuum remelting (CVM) processing which is assigned a life improvement factor of three. Consideration of this factor would bring the predicted test bearing life more in line with that determined experimentally.

In comparing the results shown in table IV and figure 5, good fatigue lives, approaching those of the noncontaminated lubricant, baseline bearings, were obtained in tests with 3-micron (series II) and 30-micron (series III) absolute filtration in a contaminated lubricant. The experimental lives of bearings tested with these filters were statistically equivalent. These lives were approximately 80 and 40 percent of the baseline bearings' lives at the 10- and 50-percent life levels, respectively. However, the experimental life differences among these three groups of bearings varied from being statistically insignificant at the 10-percent life level to highly significant at the 50 percent level as indicated by the confidence numbers tabulated in table IV. The fatigue lives obtained with the test bearings from the 49 micron filter test series (series IV) with a contaminated lubricant were approximately 50 and 25 percent of the baseline bearings (series I) 10 and 50 percent lives, respectively.

It is not conclusive that the fatigue life values determined from test series I represent the "upper limit" in terms of bearing life had even stricter lubricant cleanliness conditions been maintained as suggested by the research of reference 13. In reference 13, a single test bearing was tested with an isolated closed loop lubrication system to avoid the introduction of metallic wear debris from any component other than the test bearing itself. However, the practicality of maintaining such an "ultra" clean lubrication system in practice is questionable.

In tests conducted with the 105-micron filter in contaminated oil, all 16 test bearings experienced gross wear which precluded spalling. The rivets on three test bearing cages failed causing cage separation, presumably due to buildup of debris in the ball pockets and high ball-race traction forces. Excessive vibration from the remaining bearings forced test suspension in less than 500 hours of testing.

It is interesting to note, from the fatigue life results shown in table IV, that the value of Weibull slope increases substantially with coarser test filters. According to the contaminated lubricant, fatigue life model advanced in reference 14, the Weibull slope or life dispersion parameter should increase for bearings experiencing progressive surface damage while running. Damage accumulating at a linear rate would be expected to double the actual slope of rolling-element life associated with a bearing which had an unchanging number of pre-existing surface defects. Quadratic damage accumulation would be expected to triple it. Although no correlation with damage accumulation rate has been made, the Weibull slope from test series II is nearly twice that of the baseline, noncontaminated lubricant test series and slightly greater than three times the baseline series in the case of test series III and IV. This seems to lend some qualitative support to the contaminated lubricant, fatigue life model of reference 14.

Effect of Contaminants on Test Bearing Condition

Post test examination of the raceways of the long-lived, suspended test bearing from test series I, II, and III showed great differences in the degree of surface distress in relation to filtration level. These differences in running track appearance are distinctly greater than that which might have been anticipated from their relative fatigue life rankings. Figure 6 shows representative macro and scanning electron microscope (SEM) photographs of test bearing inner races that were suspended from test without fatigue failure. Included for comparison are photos of an untested bearing. It is apparent from these photos that the amount of surface distress progressively increases with coarser filter size as evidenced by the intensity and width of the wear track coupled with the increasing absence of grinding marks. The original grinding marks clearly present on both the untested and clean lubricant test bearings in the SEM photographs are still visible on the bearings from the 3-micron filter tests after nearly 1200 hours of testing but are only faintly visible on bearings tested with the 30-micron filters. The grinding marks on the bearings from the 49- and 105-micron filter tests are completely worn away and only a matted surface remains. A macroscopic view of this progressive surface distress is quite apparent from figure 6 where the width and intensity of the wear track is barely perceptible in the case of the bearing from the 3-micron filter tests, but is readily apparent for the 30-micron filter test bearing and quite extensive for the bearings tested with 49- and 105-micron filters.

The progressive increase in surface distress with an increase in filter size is indicative of a degradation of the local EHD film thickness. Debris particles often breech the clearance between contracting elements causing an interruption of the protective film which in turn leads to surface distress. The size of the particles which penetrate the contract are on the order of the EHD film thickness which is, in this case, 0.34 microns and, therefore, are extremely difficult to filter out. However, occasionally larger particles can be entrained, judging by the size of the debris dents, and these would be more amendable to filtration. Debris dents themselves can cause substantial drops in local EHD film thickness (ref. 15) contributing to metal-to-metal contact. A loss of film thickness will inevitably lead to surface distress, causing an increase in tangential shear forces on or near the surface and hasten the onset of surface initiated fatigue failure.

Corroborating this apparent increase in bearing surface damage with filter size is the increase in bearing average weight loss of the suspended test bearings as listed in table V. On the basis of grams per 100 test hours, the bearings tested with 3-, 30-, 49-, and 105-micron absolute filters had, respectively, 1.9, 3.2, 4.2, and 89 times the weight loss of the baseline bearings tested with the noncontaminated oil. It is instructive to note from table V that failed test bearings generally experienced a weight loss several times that of an unfailed bearing. Thus, taking individual bearing weight measurements of thoroughly cleaned test bearings from a group containing suspected failures, became a reliable means of selecting the failed bearing or bearings from those unfailed without resorting to disassembly.

Evidently, the observed increase in surface distress with filter size is a direct consequence of the number and size of debris particles suspended in the oil passing through the test bearing contacts. Table VI shows the results of particulate oil analyses on selected oil samples. These samples were taken by the automatic oil sampler previously described at a point downstream of the test bearings and scavange pump. These samples were extracted at various times during the course of the test. The particle count values listed in table VI are a time related measure of the filter's efficiency coupled with the induced debris generation rates of the test bearings and scavenge pump. The absolute accuracy of the particle count readings should not be given undue emphasis in view of the difficulties typically encountered in obtaining unbiased oil samples and the delicate nature of the particle count method itself. Excluding the occasional numerical inconsistencies, taken as a whole, table VI shows a trend of decreasing particle levels with finer filtration. In addition, the debris levels show, for the most part, no significant increases with time. This indicates that the filter is performing stably and that the debris generation rates associated with the bearings and pump are relatively constant. An appreciation of the filter's effectiveness can be gained by comparing the particle count level obtained during tests to that obtained during calibration when contaminants were introduced into the system without the filter being present. In nearly all cases the efficiency of the filter to remove particles of 5 microns or larger in terms of the percentage of incoming particles caught by the filter is in excess of 99 percent. During calibration of the contaminant injection system, it was determined that particulate oil analysis could account for 80 percent of the total amount of contaminants added to the oil on a weight basis. Furthermore, the distribution of particles in terms of the percentage of particles of a given size range found in the oil samples approximated that of contaminant test mixture (compare tables III and VI).

Failure Modes

Metallurgical examination of test bearing failures were made with the intent of classifying the type and origin of failure. In most cases, the spall, which propagates rapidly under the heavy test loads predominantly in the rolling direction, would remove evidence of a surface defect that might precipitate failure. In several cases where bearings were suspected of subsurface initiated failure, metallographic cross sections were made through the spall area. These examinations failed to uncover any metallurgically anomalous conditions such as faulty microstructure, nonmetallic incursions or carbide agglomeration which would conclusively verify subsurface origin. Microscopic examination of the spall morphology and the areas adjacent to the failure site generally gave the best indication of the most probable failure mode.

Figures 7, 8, and 9 show selected macro and SEM photographs of representative spalling failures from three of the test series. Although no subsurface originated defects could be identified, the spalling picture in figure 7, from the baseline, noncontaminated lubricant test series was probably subsurface initiated. The absence of surface distress preceding the spall area coupled with the steep slope at the point of spall initiation are indicative of this type of failure (ref. 3).

The failures examined from this test series appear to be predominantly subsurface originated with a few completely unclassifiable. It is possible that some of these indeterminant failures could have nucleated from debris dents, acting as stress raisers, since dents were occasionally found on unfailed bearings as in figure 6(b). Many of these debris dents were significantly larger than the "absolute" pore size of the filter, suggesting that debris particles can break loose from the line downstream of the filter and/or be generated with the bearing assembly itself. It is not uncommon to find debris dents which are larger than the filter's openings on the raceways of test bearings which have been protected by full flow filtration (refs. 3 and 16). A large lubrication system with many components, such as the one used in the course of this study, inevitably contains many debris generators, notwithstanding the bearings themselves. Despite attempts to carefully control lubricant cleanliness, it is not uncommon to find a surprisingly high number of surface defect related fatigue failures on bearings from laboratory controlled tests, particularly when the high cleanliness of the bearing material greatly lessens the chance for subsurface-type failures.

Although not conclusive, the failure mode associated with the test bearings appears to shift from predominantly subsurface initiated to predominantly surface initiated with increasing filter size. Figure 8 shows a probable surface defect initiated spall on the outer race of a bearing from test series II. The probable spall initiation site, a small pit followed by an arrowhead-shaped spall pattern has been identified. This shallow spalling pattern is characteristic of a point surface originated spall (ref. 2). The small pit is surrounded by a network of microcracks suggesting a localized collapse of the elastohydrodynamic (EHD) film thickness, ostensibly from debris particles interrupting the film. Many of the spall failures from series II and a majority of those examined on bearings from test series III originated from the raceway surface.

In the case of the bearings from tests with the 49 micron absolute filter, series IV, all fatigue failures appear to be the direct result of severe surface distress. The extensive microcracking, or frosting in the running track of the bearing inner race, shown in figure 9, in all likelihood is a direct consequence of an interruption of the local EHD film by debris particles. A loss in protective film thickness permits metal-to-metal contact with a substantial increase in tractive forces leading to plastic flow and microcracking as shown. Failed bearings from this test series typically had multiple patches of shallow microspalls containing larger pits in various stages of development. The microcrack network would eventually coalesce to form an incipient spall that would grow in size. The variation in the width of the microspalled area within the running track shown in figure 9(a) is probably due to ball instability as the ball skips over debris clinging to the raceway.

No fatigue failures were encountered in test series V with 105 micron absolute filtration through several hundred hours of testing. The wear rates associated with these bearings were so high, that it was unlikely for a fatigue spall to develop to an appreciable size before being worn away by the abrasive contaminant trapped between the ball and raceway (see fig. 6(f)).

An observation which can be made after examining the condition of the bearing surfaces from this study is that a significant degree of surface distress can be expected if debris particles are able to penetrate the contact, irrespective of the initial EHD film thickness. To provide maximum component life, a concerted effort must be made to prevent contaminants from gaining entry into the system through proper sealing and secondly, to provide an effective means of removing particles that have become suspended in the oil through fine filtration.

SUMMARY OF RESULTS

Fatigue tests were conducted on groups of 65-millimeter-bore-diameter deep-groove ball bearings with a noncontaminated lubricant and with a contaminated lubricant under four levels of filtration. The bearings were manufactured from a single heat of vacuum-degassed AISI 52100 steel. In the baseline test lubricant, a MIL-L-23699 qualified, neopentylpolyol (tetra) ester was prefiltered prior to installation into a recirculating lubrication system containing a 49-micron absolute, (30 micron nominal) full flow filter. In the remaining series of tests, contaminants, of a composition similar to that found in filters from aircraft gas turbine engines, were injected into the test filter's supply line at a constant rate of 125 milligrams per hour per bearing. The test filters, of porous depth media construction, had absolute particle removal ratings of 3, 30, 49, and 105 microns (0.45, 10, 30, and 70 microns nominal), respectively. Test conditions included a bearing shaft speed of 15 000 rpm, a radial load of 4580 newtons (1030 lb) producing a maximum Hertz stress of approximately 2410 MPa (350 000 psi) on the bearing inner race. The temperature of the lubricant into the test bearing and the sump temperature were maintained at 347 K (165° F).

1. Bearing life and running track condition generally improved with finer filtration. However, statistically similar fatigue results were obtained in bearing tests with 3- and 30-micron filtration in a contaminated lubricant, approaching those obtained in the baseline, noncontaminated lubricant tests at the 10-percent life level. Experimental 10-percent lives from contaminated lubricant tests with 49-micron absolute filtration were approximately half those of the baseline, noncontaminated lubricant tests.

2. Surface distress on the running tracks and wear of the test bearings progressively increased with coarser filter size. Gross wear with the 105-micron absolute filter precluded the onset of rolling-element fatigue.

3. Bearing life dispersion parameter or Weibull slope increased with coarser filtration.

4. The probable failure mode appeared to shift from predominantly subsurface initiated fatigue for the bearings from the baseline, noncontaminated lubricant tests filter to surface initiated fatigue for the bearings tested in a contaminated lubricant with 49-micron absolute filtration.

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APPENDIX - BACKGROUND

Subsurface originated spalling is the exclusive failure mode considered by the theory of Lundberg and Palmgren (ref. 17). Their analysis has been the accepted basis for bearing service life ratings by the bearing industry. However, this theory is not applicable to surface initiated fatigue which comprises a significant percentage of bearing fatigue failures (refs. 3 and 16). Efforts have recently been made by Tallian to expand fatigue life prediction methods to include surface initiated spalling for rolling-element bearings possessing surface defects (ref. 14).

The presence of contaminants in rolling-element systems will not only increase the likelihood of surface initiated fatigue but can lead to a significant degree of component surface distress. In reference 6, experiments performed on tapered-roller bearings have shown that wear is proportional to the amount of contamination in the lubricant and that the wear rate generally increases as the contaminant particle size is increased. Furthermore, the wear process will continue as long as the contaminant particle size exceeds the thickness of lubricant film separating the bearing surfaces. Since this film thickness is rarely greater than 3 microns (118 μin.) for a rolling contact component, even extremely fine contaminant particles can cause some damage.

The research of reference 18 examined the quantitative effects of oil contaminants on ball bearing life by injecting a mixture of ceramic/silica/iron particles into the bearing lubrication system. Their tests indicate that a continuous contaminant addition rate as small as 10 milligrams during an 8-hour period at a concentration level of 42 milligrams per liter of oil can reduce the bearing life measured in clean oil by 80 to 90 percent. Post test bearing examination revealed that spalling, presumably surface initiated, was the principal failure mode. To put this level of contamination in proper perspective, evaluation of the particulate levels in aircraft turbine engine oils as delivered from various manufacturers was reported in reference 19 to range from a negligible amount to 8.2 milligrams of total sediment per liter of oil with an average value of approximately 2.5 milligrams per liter. In view of this, it is advisable to prefilter the as-delivered oil from the drum through an extra fine filter prior to installation for critical applications.

The potential for a drastic reduction in bearing service life due to the presence of contaminants give great incentive for fine oil filtration, particularly for critical rotating machinery. Aircraft turbomachinery, a case in point, are particularly vulnerable to lubrication or hydraulic system contamination. The common sources of contaminants for aircraft propulsion system are (1) injected air-borne particles, (2) carbonacious particles from the byproducts of combustion (3) particles present in the commercial oil as delivered, (4) particles attached to the internal surfaces of the hydraulic components and (5) particles that are generated during the wear-in period and during lubrication system operation.

Although knowledgeable aircraft lubrication system designers recognize that certain bearing, gearing, and seal problems were related to lubricant contamination, they have, in the past, been reluctant to specify fine filtration levels (less than 10 microns nominal rating) for their system. Until recently, filtration levels for aircraft turbine engine lubrication systems have been mainly limited to metallic screens of 100 to 150 mesh (150 to 110 microns) (ref. 19). However, this situation seems to be changing. Recently, 3-micron absolute filtration was selected for use on the T-700 gas turbine engine which powers advanced helicopters.

Part of this former reluctance to use fine filters stems from the concern that fine lubricant filtration would not sufficiently improve component reliability to justify the possible increase in system cost, weight, and complexity. In addition it is presumed that fine filters will clog more quickly, have a higher clean pressure drop, and generally require more maintenance than currently used filters. The study described in reference 20 demonstrated that these presumptions are not always correct. In this study, tests were performed on a 3-micron absolute main oil filter which replaced the original production 40-micron nominal filter for a helicopter gas turbine lubrication system. The new filter elements provided a much cleaner lubricant with less component wear, while greatly extending the time between filter removals for clogging and oil changes. This was accomplished with a modest increase in filter size and weight and with a new filter clean pressure drop nearly the same as the original production unit.

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TABLE I. - PROPERTIES OF TETRAESTER LUBRICANT

Additives	Antiwear, oxidation inhibitor, antifoam
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flashpoint, K; °F	533 (500)
Firepoint, K; °F	Unknown
Autoignition temperature, K; °F	694 (800)
Pour point, K; °F	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt.%	3.2
Specific heat at 477 K (400° F), J/(kg)(K); (Btu/(lb)(°F))	2340 (0.54)
Thermal conductivity at 477 K (400° F), J/(m)(sec)(K); (Btu/(hr)(ft)(° F))	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

TABLE II. - TEST FILTERS SPECIFICATIONS

Test series[a]	Removal ratings, microns			Clean pressure drop at 19 liters/min (5 gal/min)		Dirt capacity,[d] g	Filter media material
	Nominal[b]	Mean	Absolute[c]	kPa	psi		
I	30	40	49	13.8	2.0	550	Resin impregnated fibers
II	.45	.9	3.0	6.9	1.0	290	Resin impregnated fibers
III	10	20	30	6.9	1.0	270	Resin impregnated fibers
IV	30	40	49	13.8	2.0	550	Resin impregnated fibers
V	70	(e)	105	27.6	4.0	22.5	Sintered square weave wire cloth, stainless steel

[a]In test series I clean oil was used; in all others contaminants were added.
[b]Nominal removal rating based on MIL-F-5504.
[c]Absolute removal rating based on MIL-F-27656.
[d]Dirt capacity based on MIL-F-25682.
[e]Not available.

TABLE III. - BREAKDOWN OF TEST CONTAMINANT COMPOSITION

Constituent	Parts per mixture by weight	Particle distribution
Stainless-steel particles	1	100 percent less than 44 microns
Arizona coarse test dust	10	12 percent less than 5 microns 24 percent less than 10 microns 38 percent less than 20 microns 61 percent less than 40 microns 91 percent less than 80 microns 100 percent less than 200 microns
Carbon-graphite test dust	80	75 percent less than 10 microns 92 percent less than 20 microns 100 percent less than 40 microns
Total contaminant mixture	91	70 percent less than 10 microns 86 percent less than 20 microns 96 percent less than 40 microns 4 percent greater than 40 microns

TABLE IV. - FATIGUE-LIFE RESULTS OF 65-MILLIMETER BORE BALL BEARING TESTS FOR VARIOUS LEVELS OF FILTRATION IN A CONTAMINATED LUBRICANT

[Radial load, 4580 N (1030 lbf); speed, 15 000 rpm, temperature, 347 K (165° F); test lubricant, MIL-L-23699 type.]

Test series	Test filter absolute rating, microns	Test lubricant condition	Experimental hours		Weibull slope	Failure index[a]	Confidence number[b], percent	
			10-percent life, L_{10}	50-percent life, L_{50}			10-percent life	50 percent life
I	49	Clean	672	2276	1.54	9 out of 32	--	--
II	3	Contaminated	505	993	2.78	10 out of 16	63	99
III	30	Contaminated	594	857	5.12	11 out of 16	57	99
IV	49	Contaminated	367	533	5.06	20 out of 32	89	99
[c]V	105	Contaminated	---	----	----	----------	--	--

[a]Number of fatigue failures out of number of bearings tested.

[b]Probability (expressed as a percentage) that fatigue life in contaminated lubricant test series will be less than life with clean oil in test series I.

[c]Test series V was suspended after 448 test hours on each of the test bearings due to excessive bearing wear. No fatigue failures were encountered.

TABLE V. - TEST BEARINGS AVERAGE WEIGHT LOSS

Test[a] series	Test filter absolute rating, microns	Suspended test bearings		Failed test bearings	
		g/bearing	g/100 hr	g/bearing	g/100 hr
I	49	0.0412	0.0031	0.2775	0.0311
II	3	.0548	.0059	.3157	.0390
III	30	.0806	.0100	.1679	.0214
IV	49	.0809	.0130	.3288	.0713
[b]V	105	1.0204	.2757	------	------

[a]In test series I clean oil was used; in all others contaminants were added.
[b]All bearings from test series V suspended due to heavy wear.

TABLE VI. - REPRESENTATIVE RESULTS OF TEST LUBRICANT PARTICULATE ANALYSIS

[*Samples taken downstream of test bearings.*]

Test series	Test filter absolute rating, μm	Test lubricant condition	Test time, hr	Micron size range				
				5 to 15	16 to 30	31 to 50	51 to 100	>100
				Particle count, 10^3 particles/100 ml				
[a]Calibration	---	Clean	0	89.97	2.4	0.5	0	0
[b]Calibration	---	Contaminated	8	287 000	26 700	3 400	187	19.2
			22	969 000	89 100	16 000	462	39.1
I	49	Clean	139	25.0	4.6	1.8	0.4	0.2
			417	23.8	7.1	3.0	.5	.2
			1207	80.0	11.1	.2	.2	.2
II	3	Contaminated	12	129	7	0.7	0.3	0.2
			113	68	5	2.9	.7	.8
			785	145	9	2.7	.5	.3
			876	102	6	1.9	.4	.3
III	30	Contaminated	158	70	8	4.8	1.1	0.6
			164	176	50	25.6	3.7	6.7
			170	152	15	12.5	1.5	1.0
			676	105	19	10.9	2.5	1.7
IV	49	Contaminated	205	1732	79	3.6	0.4	0.2
			412	255	19	6.8	1.0	.2
			663	605	38	8.8	1.2	.4
V	105	Contaminated	116	116 900	4870	287	133	48.2
			202	221 100	5490	549	79	20.5

[a]Unused oil as pumped from tank through 10 micron absolute prefilters.
[b]Contaminant injected into system at 3.1 mg/100 ml/hr without filter and test bearings installed.

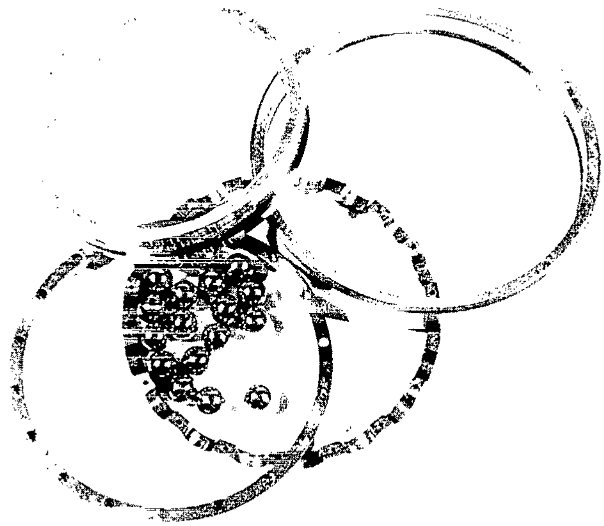


Figure 1. - Test bearing with a 65-millimeter bore diameter.

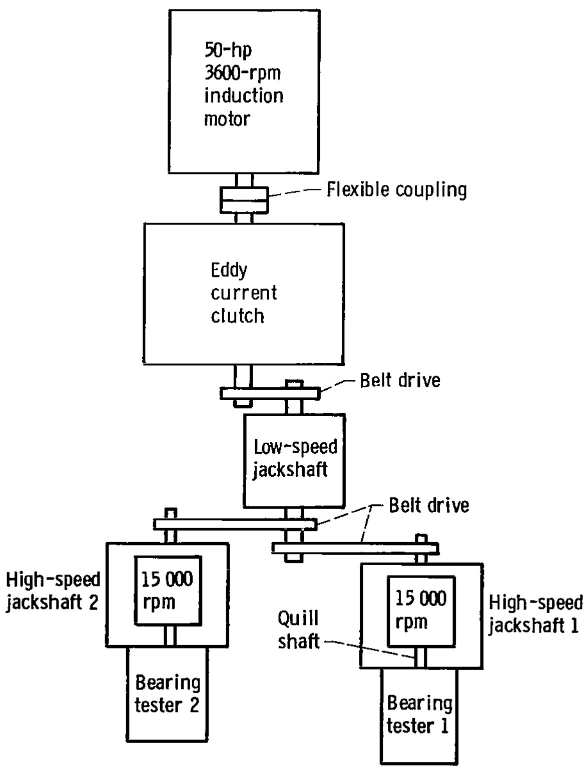


Figure 2. - Bearing test stand.

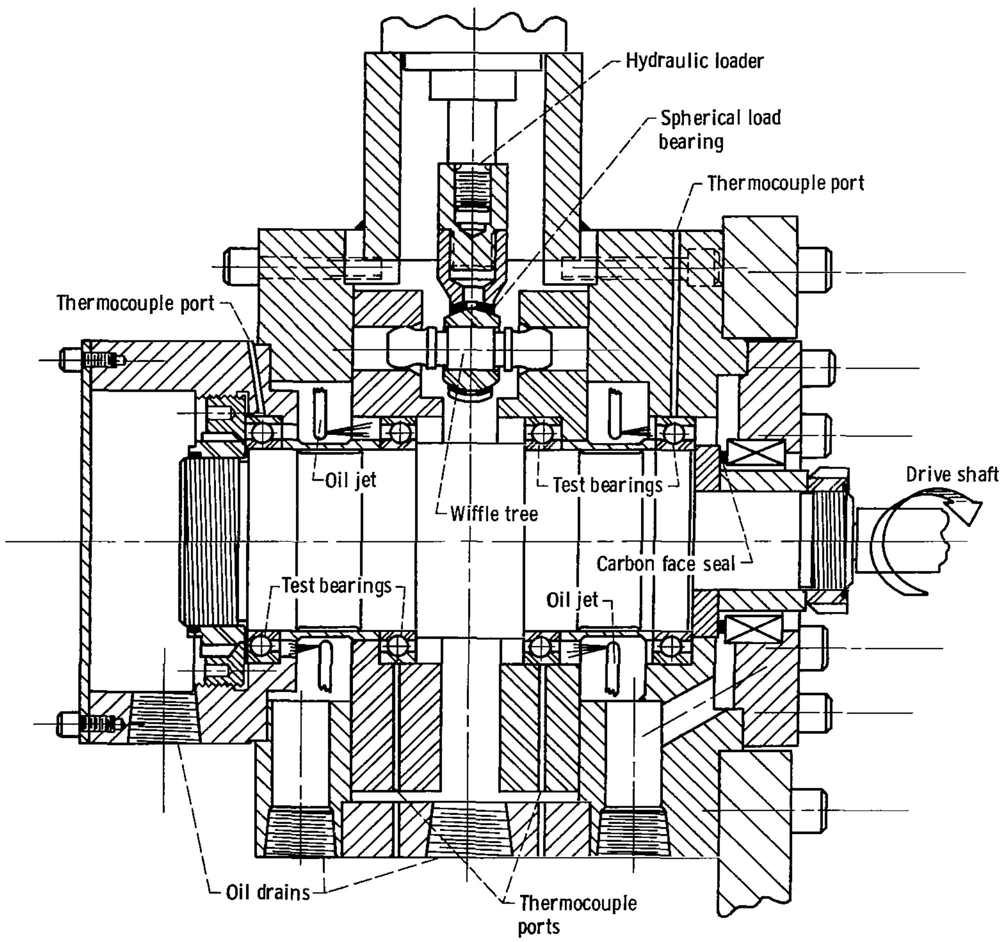


Figure 3. - Bearing fatigue tester.

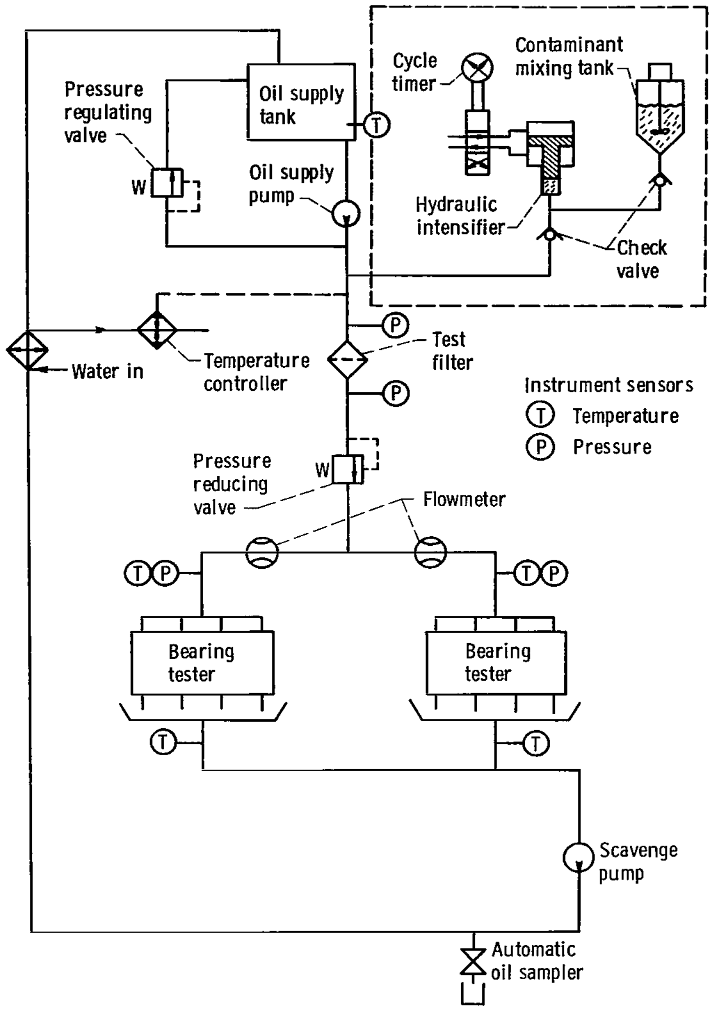


Figure 4. - Lubricant supply system.

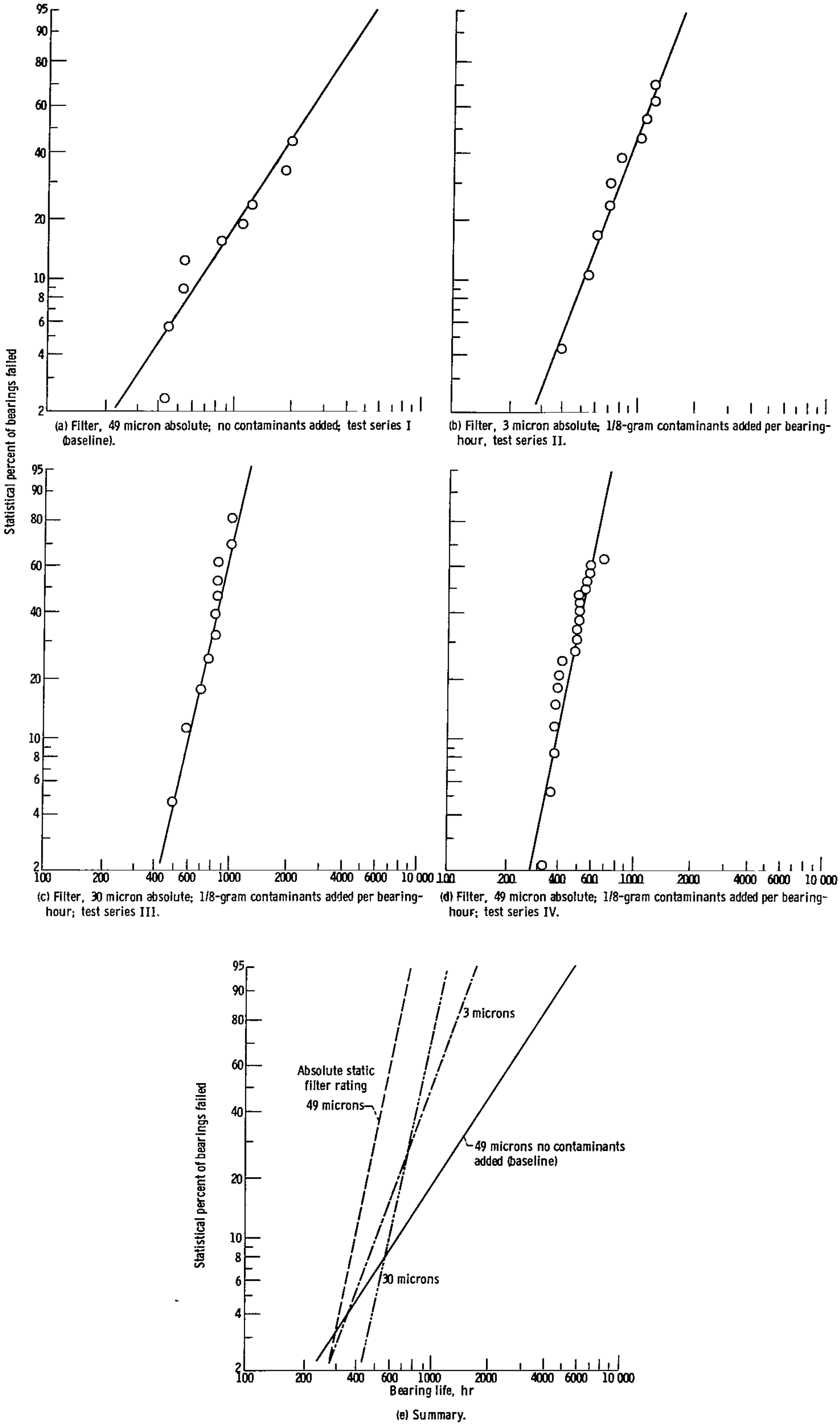
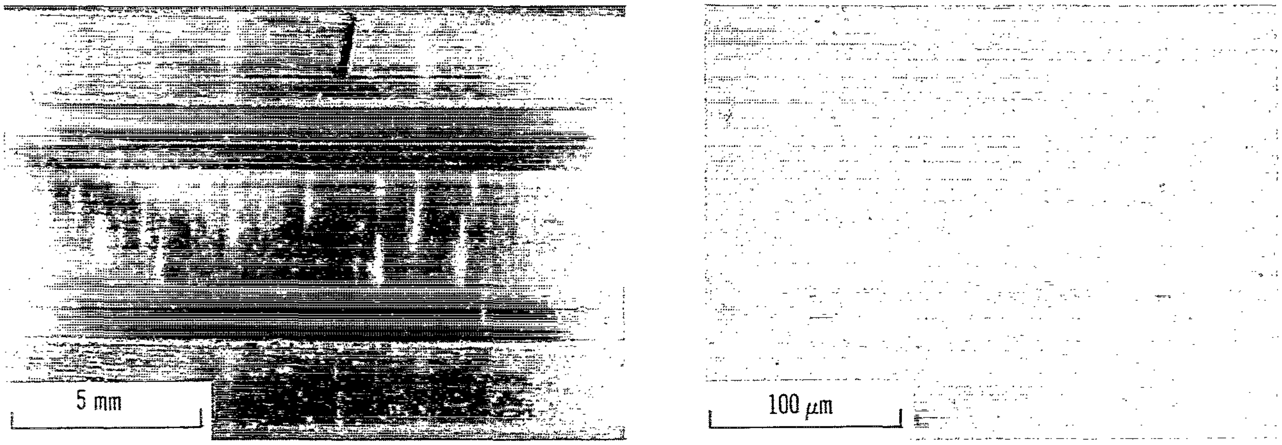
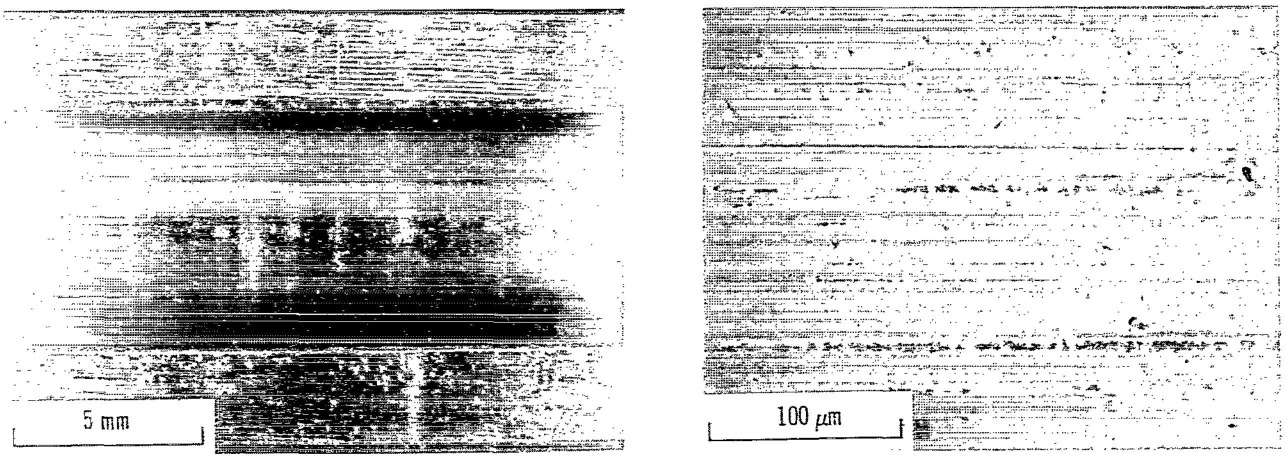


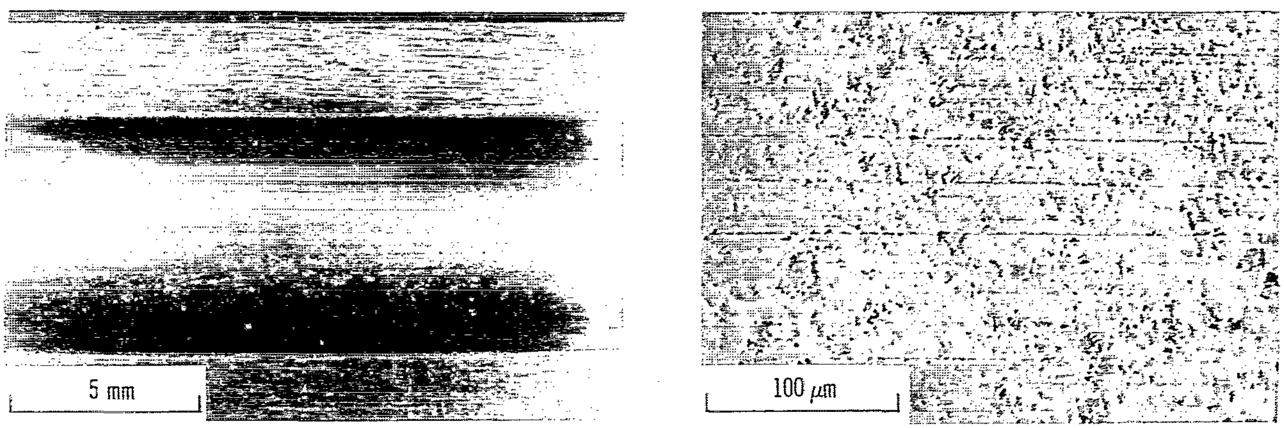
Figure 5. - Effect of filtration level on 65-millimeter bore ball bearing fatigue life in contaminated lubricant. Inner race speed, 15 000 rpm; inner race maximum Hertz stress, 2410 MPa (350 000 psi); lubricant contamination rate, 0.125 gram per hour per bearing.



(a) Untested test bearing.

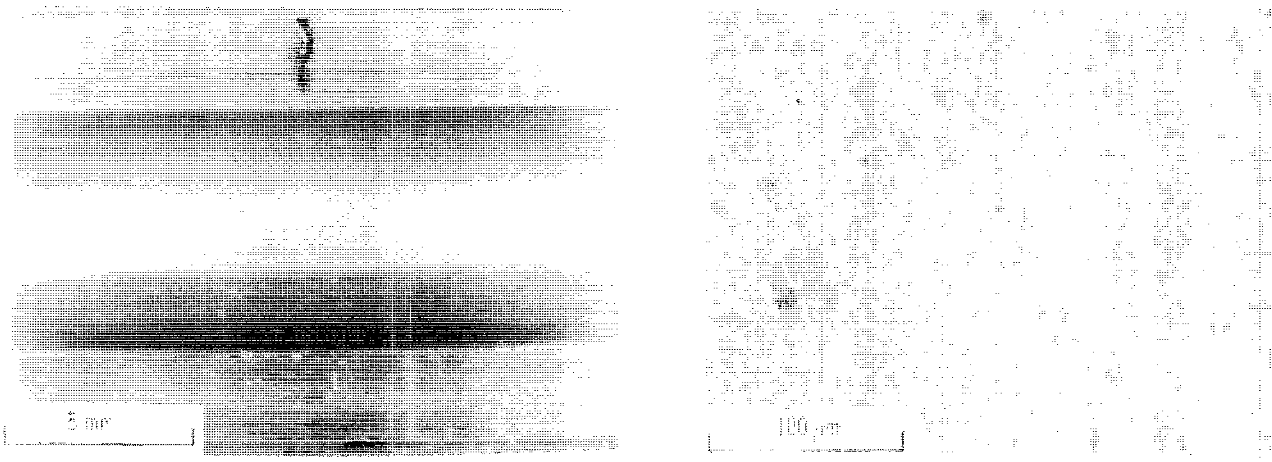


(b) Test bearing suspended after 1206 hours from 49-micron-absolute filter tests with clean lubricant (test series I).

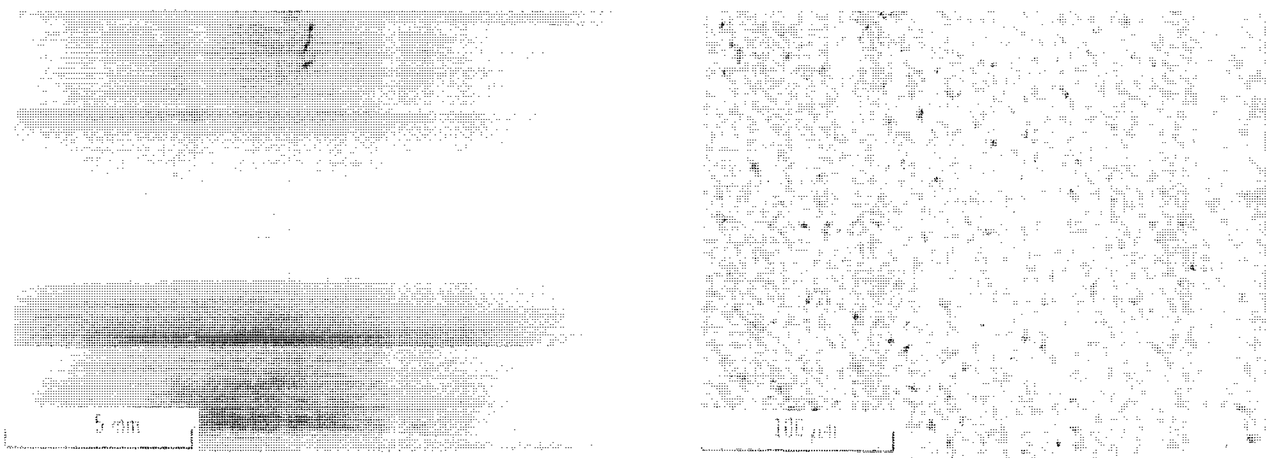


(c) Test bearing suspended after 1172 hours from 3-micron-absolute filter tests with contaminated lubricant (test series II).

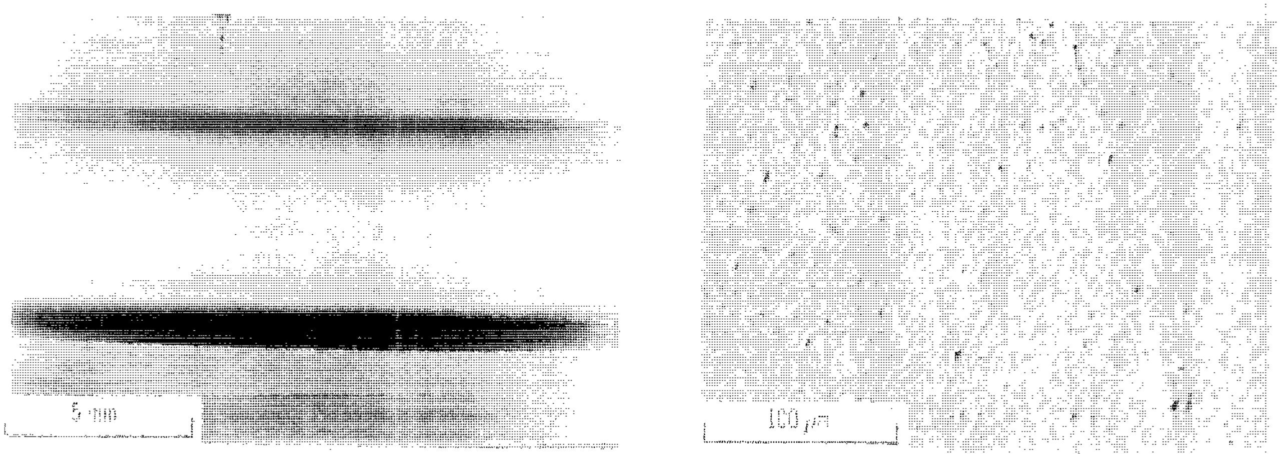
Figure 6. - Macro and SEM photos of test bearings inner race showing progressive surface damage of running track with coarser filter size.



(d) Test bearing suspended after 987 hours from 30.micron-absolute filter tests with contaminated lubricant (test series III).

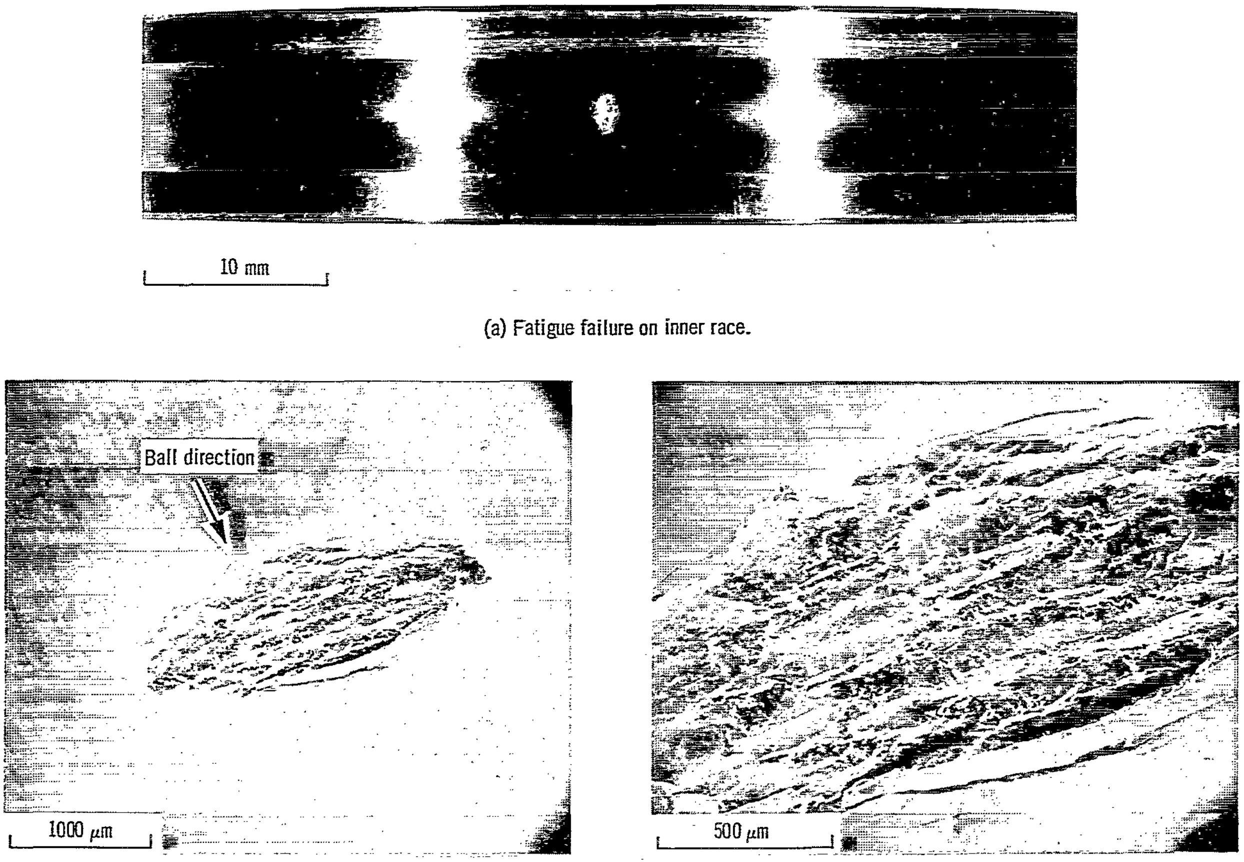


(e) Test bearing suspended after 663 hours from 40-micron-absolute filter tests with contaminated lubricant (test series IV).



(f) Test bearing suspended after 449 hours from 105-micron-absolute filter tests with contaminated lubricant (test series V).

Figure 6. - Concluded.

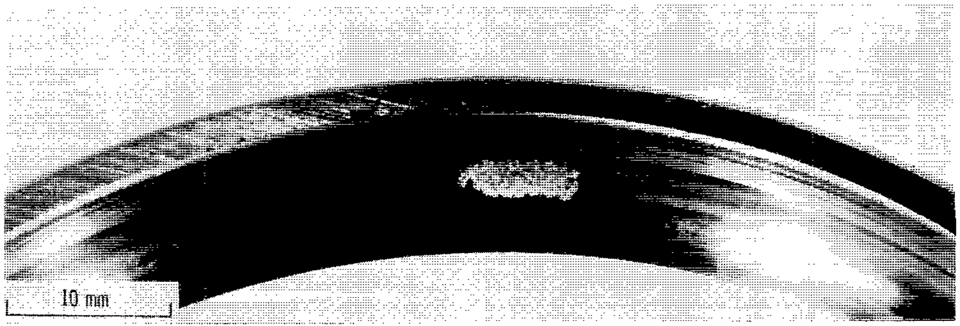


(a) Fatigue failure on inner race.

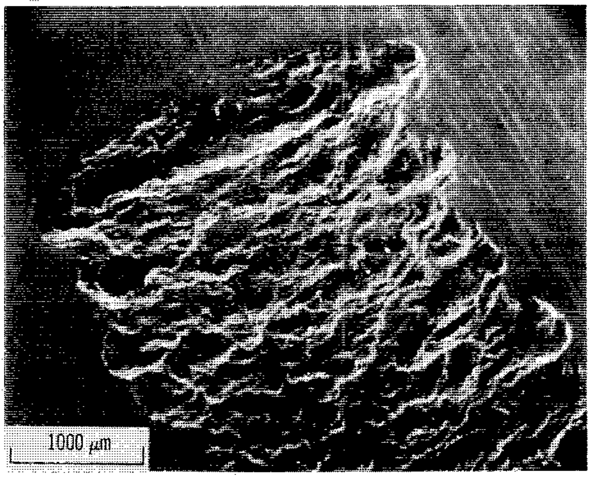
(b) Fatigue spall under low magnification.

(c) Leading edge of spall under medium magnification.

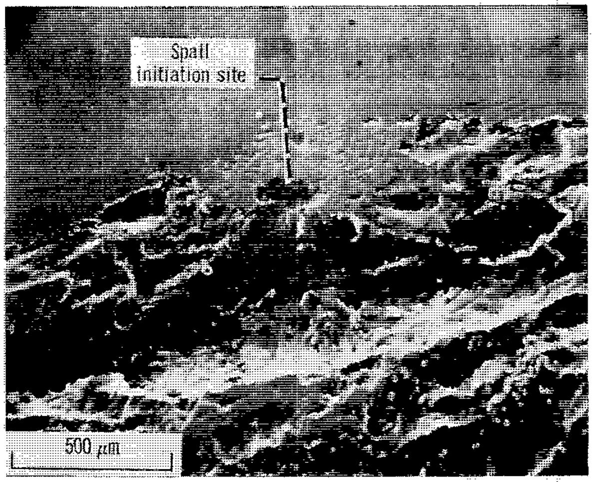
Figure 7. - Apparent subsurface initiated fatigue spall of indeterminate origin on bearing operated in noncontaminated oil with 49-micron-absolute static filter. Running time, 526 hours. Note absence of surface distress or glazing upstream of spall area.



(a) Fatigue failure on outer race.

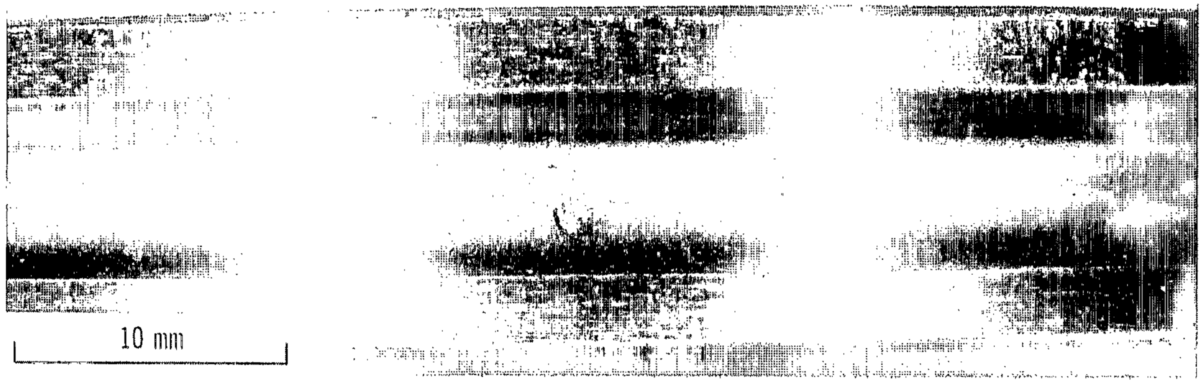


(b) Leading edge of spall under low magnification.

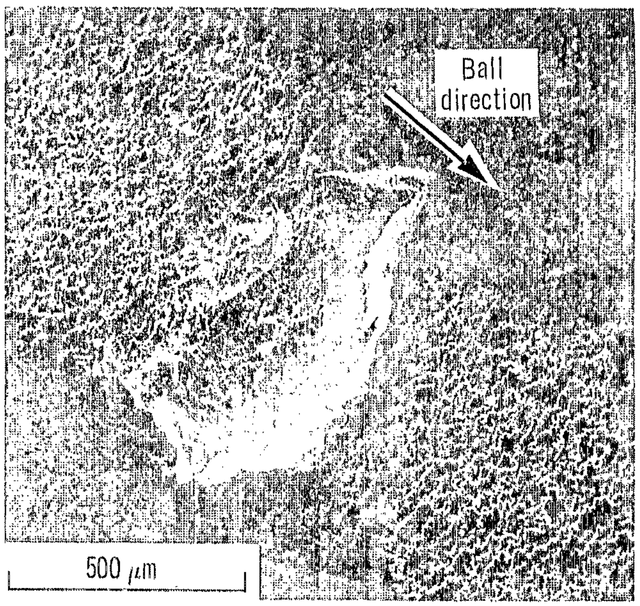


(c) Leading edge of spall under medium magnification.

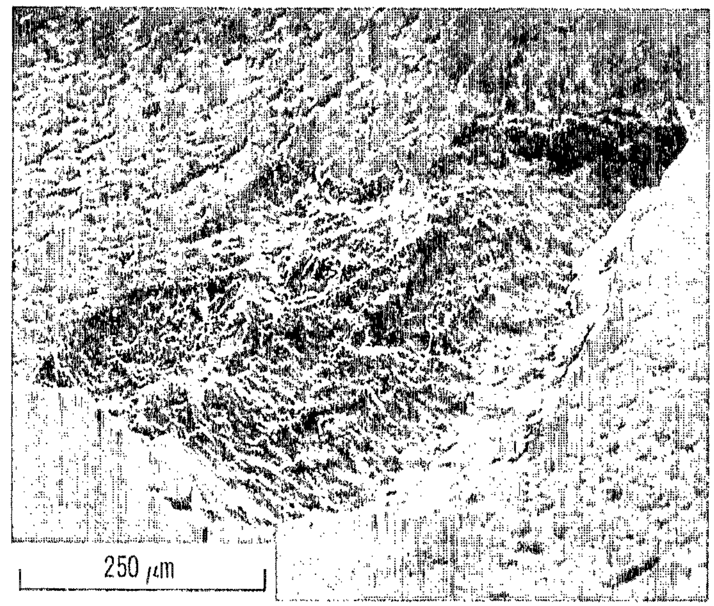
Figure 8. - Surface initiated fatigue spall on bearing operated in contaminated oil with 3-micron-absolute static filter (test series II). Running time, 1172 hours. Note surface microcrack network surrounding likely spall initiation site and arrow-shaped pattern at trailing edge of site.



(a) Inner race, large and small spalls pictured. Frosted area is comprised of vast network of microcracks suggesting collapse of local lubricating film.



(b) Small spall under medium magnification.



(c) Small spall under high magnification.

Figure 9. - Typical surface damage initiated fatigue spalls on bearings operated in contaminated oil with 49-micron-absolute static filter (test series II). Running time, 327 hours. Note network of microcracks contributing to propagation of multiple spalls.

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15. Supplementary Notes

16. Abstract

Fatigue tests were conducted on groups of 65-millimeter-bore ball bearings under four levels of filtration with and without a contaminated MIL-L-23699 lubricant. The baseline series used noncontaminated oil with 49-micron absolute filtration. In the remaining tests contaminants of the composition found in aircraft engine filters were injected into the filter's supply line at a constant rate of 125 milligrams per bearing-hour. The test filters had absolute particle removal ratings of 3, 30, 49, and 105 microns (0.45, 10, 30, and 70 microns nominal), respectively. Bearings were tested at 15 000 rpm under 4580 newtons radial load. Bearing life and running tract condition generally improved with finer filtration. The 3- and 30-micron filter bearings in a contaminated lubricant had statistically equivalent lives, approaching those from the baseline tests. The experimental lives of 49-micron bearings were approximately half the baseline bearing's lives. Bearings tested with the 105-micron filter experienced wear failures. The degree of surface distress, weight loss, and probable failure mode were found to be dependent on filtration level, with finer filtration being clearly beneficial.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
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