



# Recalibrated Equations for Determining Effect of Oil Filtration on Rolling Bearing Life

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An Erratum was added to this report June 2015.

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Page 17, Equation (16b) has been changed to read as follows:

$$\sigma = (\sigma_1^2 + \sigma_2^2)^{1/2}$$

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## Summary

In 1991, Needelman and Zaretsky presented a set of empirically derived equations for bearing fatigue life (adjustment) factors ( $LF$ s) as a function of oil filter ratings. These equations for life factors were incorporated into the reference book, “STLE Life Factors for Rolling Bearings.” These equations were normalized ( $LF = 1$ ) to a 10- $\mu\text{m}$  filter rating at  $\beta_x = 200$  (normal cleanliness) as it was then defined. Over the past 20 years, these life factors based on oil filtration have been used in conjunction with ANSI/ABMA standards and bearing computer codes to predict rolling bearing life. Also, additional experimental studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and type of particle contamination. During this time period filter ratings have also been revised and improved, and they now use particle counting calibrated to a new National Institute of Standards and Technology (NIST) reference material, NIST SRM 2806, 1997. This paper reviews the relevant bearing life studies and describes the new filter ratings. New filter ratings,  $\beta_{x(c)} = 200$  and  $\beta_{x(c)} = 1000$ , are benchmarked to old filter ratings,  $\beta_x = 200$ , and vice versa. Two separate sets of filter  $LF$  values were derived based on the new filter ratings for roller bearings and ball bearings, respectively. Filter  $LF$ s can be calculated for the new filter ratings.

## Introduction

It has long been recognized that lubricant contamination can affect bearing life, reliability, and performance. In 1976, Tallian (Refs. 1 and 2) was the first to publish a systematic study of the effect of contaminated lubrication on rolling-element fatigue life. He presented a probabilistic model (Ref. 1) to predict rolling-element fatigue life under conditions where the elemental surfaces in contact (raceways and balls or rollers) incur progressive damage during stress cycling from contamination in the lubricant. He subsequently correlated his analysis to surface density damage from experimental fatigue life data for ball bearing inner raceways (Ref. 2).

Hirano and Yamamoto (Ref. 3) reported that contaminants added to various lubricants could initiate scuffing in rubbing contacts. Dalal et al. (Refs. 4 and 5) reported that ball bearing lives in excess of 50 times ANSI/ABMA<sup>†</sup> (American National Standards Institute/American Bearing Manufacturers Association) bearing manufacturers’ catalogue calculations (Refs. 6 and 7) were achieved by operating with prefiltered ultraclean lubricant, in which the only source of metallic contamination was the test bearing itself. However, Dalal et al. did not run a control test lot of bearings without filtration to determine the exact life improvement attributable to filtration. They later induced raceway damage in their test bearings with a hardness indenter in an attempt to simulate in-service contaminant-caused

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indentations; although reduced from the ultraclean values, the fatigue lives were still longer than the ANSI/ABMA calculations.

Fitzsimmons and Clevenger (Ref. 8) carried out an extensive test program with tapered roller bearings using a variety of lubricants and contaminants with controlled particle sizes, types, and concentrations. They noted that two-body wear occurs when a hard, rough surface plows a series of grooves in an opposing softer surface. They stated that solid contaminants in lubricants are conducive to three-body abrasive wear, which occurs when particles are introduced between sliding surfaces and abrade material off both surfaces. They added the caveat that a certain amount of abrasive wear may be tolerable, depending on the application. As an example, noise in gears that results from looseness in the bearing system rather than from surface failure may constitute cause for “failure” of the application.

In tapered roller bearings, wear normally occurs on the surfaces where there is combined rolling and sliding contact; for example, between the roller ends and the cone large-end flange. Fitzsimmons and Clevenger found that wear of these surfaces increased linearly with the concentration of hard contaminant particles: In a discussion to the Fitzsimmons and Clevenger paper (Ref. 8) Kimbauer and Ferris reported that 3- $\mu\text{m}$  filtration prevents circulation of the hard particles that cause abrasive wear.

There were two independent investigations to determine the effect of oil filtration on rolling-element bearing life. These were those of Loewenthal et al. (Refs. 9 to 11) from 1978 to 1982 for ball bearings at NASA Lewis (now Glenn) Research Center in Cleveland, Ohio, and Bhachu et al. (Refs. 12 and 13) for roller bearings in 1981 at the Imperial College in London, England. In general, the results reported by both Loewenthal et al. and Bhachu et al. verified the trends of Tallian’s analysis (Refs. 1 and 2).

Subsequent to the research reported above, additional experimental and analytical studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and types of particle contamination (Refs. 14 to 20). Gabelli, Morales-Espejel, and Ioannides (Ref. 20) provide a comprehensive review of the literature, analysis, and a limited database related to particle damage in Hertzian contacts and rolling bearing life ratings. The reported research results from these references were similar to that reported by Tallian (Refs. 1 and 2), Fitzsimmons and Clevenger (Ref. 8), Loewenthal et al. (Refs. 9 to 11), and Bhachu et al. (Refs. 12 and 13). For a defined operating condition and bearing size and type, bearing life was a function of lubricant cleanliness and the number, size, and material properties of particles entering the Hertzian contact of the rolling element and raceway. However, there is conflicting opinion as to whether the elastohydrodynamic (EHD) film thickness to surface composite roughness, or the  $\Lambda$  ratio, mitigates the negative effect of lubricant cleanliness on rolling-element fatigue life. That is, is the effect of contamination on bearing life less severe with increasing film thickness?

There are numerous published papers that have studied the relation of debris dents on the EHD film thickness and rolling-element fatigue. Most of these papers are summarized and discussed by Gabelli, Morales-Espejel, and Ioannides (Ref. 20). Wedeven and Cusano (Ref. 21) and Kaneta, Kanada, and Nishikawa (Ref. 22) experimentally studied the effects of moving dents and grooves on the EHD film thickness. Numerical solutions to the effect of simple debris dents on the EHD film thickness and surface and subsurface stresses are given by Venner (Ref. 23); Ai et al. (Refs. 24 to 26); Nelias and Ville (Ref. 27); Ville et al. (Refs. 28 and 29); and Chapkov Colin, and Lubrecht (Ref. 30). The results of these analyses and experiments suggest that the debris dents reduce the EHD film thickness at the dent site, increasing the contact and subsurface stresses, thereby reducing rolling-element bearing fatigue life.

In 1991, based on the research of Loewenthal et al. (Refs. 9 to 11) and Bhachu et al. (Refs. 12 and 13), Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life adjustment factors ( $LF$ s) as a function of the oil filter ratings ( $FR$ ), where  $\beta_x = 200$ . These equations for  $LF$  were incorporated into the reference book, “STLE Life Factors for Rolling Bearings” (Ref. 32). These equations (Ref. 31) were normalized ( $LF = 1$ ) to a  $FR$  of 10  $\mu\text{m}$  at  $\beta_x = 200$  (normal cleanliness) as it was then defined by ISO Standard 4572 (Ref. 33) and a ratio of the EHD film thickness to surface composite roughness,  $\Lambda$ , of 1.1.

For over 2 decades these life factors based on oil filtration (Refs. 31 and 32) have been used in conjunction with ANSI/ABMA standards (Refs. 6 and 7) and bearing computer codes (Ref. 34) to predict rolling-element bearing life. However, in 1999 filter ratings underwent a revision and improvement (Ref. 35). They are now based on ISO 16889 (Ref. 36), replacing the older and now disavowed ISO 4572:1981 (Ref. 33).

A primary difference between these filter ratings is how particle size is specified and measured. The older filter test method used particle counters calibrated per ISO 4402:1991 (Ref. 37), with a variable material comprising irregularly shaped particles, AC Fine Test Dust (AC FTD). The new filter test method employs particle counters calibration per ISO 11171:1999 (Ref. 38), based on spherical particles traceable to a National Institute of Standards and Technology (NIST) reference material (SRM 2806, 1997) (Ref. 39), providing more accurate and verifiable results (Refs. 35 and 40). In addition, the new filter rating method uses a somewhat different test dust (ISO Medium Test Dust, ISO MTD) (Ref. 41), replacing the no-longer-available dust previously used, AC FTD.

The Needelman-Zaretsky life factors published in the early 1990s (Refs. 31 and 32) do not correspond to this revised and improved system of filter ratings. This paper focuses on the improved methods for measuring filter performance (improved filter ratings), and recalibrated bearing life factors incorporating these new ratings. The improved system of filter rating provides more accurate measurement of industrial filter performance, including performance improvements achieved by advancements in the design, materials, and manufacture of filters over the past 20 years.

In 2007, the ISO Standard 281, “Rolling Bearings—Dynamic Load Ratings and Rating Life” (Ref. 42), was radically changed. It is based on the inclusion of a “fatigue limit” in the bearing life calculations and the addition of a “contamination factor for circulating oil lubrication with on-line filters.” This “contamination factor” is based on the analytical work of Ioannides et al. (Ref. 17) and on a “cleanliness code according to ISO 4406” (Refs. 43 and 44). ISO 4406:1987 (Ref. 43) is the cleanliness code based on particle counters calibrated to the now obsolete ISO 4402 calibration (Ref. 37) that used irregularly shaped AC FTD. ISO 4406:1999 (Ref. 44) uses the approved ISO 11171 calibration method (Ref. 38) based on the NIST spherical-particle reference material. ISO 281:2007 (Ref. 42) incorporates the new ISO 4406:1999 cleanliness code (Ref. 44) but does not specify filter ratings to the levels of contamination.

Based on the above discussion it became the objective of the work reported herein to (1) review methods and data for determining the effects of lubrication oil particle quantity and size for calculating the bearing  $L_{10}$ , (2) experimentally calibrate older filter ratings that used AC FTD to new filter ratings using the NIST traceable particle counting calibration and ISO MTD, (3) recalibrate the Needelman-Zaretsky equations for determining effect of oil filtration on rolling-element bearing life to new filter ratings per ISO 16889 (Ref. 36), (4) determine filter ratings and related life factors based on new cleanliness codes per ISO 4406:1999 (Ref. 44) for ISO 281:2007 (Ref. 42), and (5) compare recalibrated filter life adjustment factors to cleanliness ratings presented in ISO 281:2007.

## Nomenclature

$C_1$	empirically determined constant
$C_2$	empirically determined exponent
$C_u$	fatigue load or stress endurance limit of the bearing material
$D_p$	bearing pitch diameter, mm
$E_x$	particle removal efficiency, $(1 - 1/\beta_x) \times 100$ , percent
$e_C$	lubricant contamination or oil cleanliness factor
$FR$	filter rating, micron
$h$	elastohydrodynamic (EHD) film thickness, $\mu\text{m}$
$h_c$	central or average elastohydrodynamic (EHD) film thickness, $\mu\text{m}$

$L_{10}$	bearing 10-percent or catalog life or the operating time that is exceeded by 90 percent of a group of bearings of a given type, hours or millions of inner-ring revolutions
$L_{50}$	bearing 50-percent or median life or the time that is exceeded by 50 percent of a group of bearings of a given type, hours or millions of inner-ring revolutions
$LF$	life adjustment factor for effect of filtration on bearing $L_{10}$ life
$m$	Weibull slope or modulus
$ND_x$	average number of particles downstream of the filter whose particle sizes are greater than $x$ $\mu\text{m}$
$NU_x$	average number of particles upstream of the filter whose particle sizes are greater than $x$ $\mu\text{m}$
$P$	applied dynamic equivalent (applied) load to the bearing, N
$x$	particle size, $\mu\text{m}$
$a_{\text{ISO}}$	life adjustment factors $LF$ in ISO Standard 281:2007
$\beta$	filtration factor or filter rating, $NU_x/ND_x$
$\beta_x$	old filter rating, micron
$\beta_{x(c)}$	new filter rating, micron
$\kappa$	ratio of the actual viscosity of the lubricant in the bearing to a reference viscosity where $\kappa \approx \Lambda^{1.3}$
$\Lambda$	lubricant film parameter, $h_c/\sigma$
$\sigma$	composite surface roughness, $(\sigma_1^2 + \sigma_2^2)^{1/2}$ , $\mu\text{m}$
$\sigma_1, \sigma_2$	root mean square (rms) surface roughness of contacting bodies, $\mu\text{m}$
$\omega_{1,2}$	rotational speed, rpm
1	body 1 or ball or roller
2	body 2 or bearing raceway

## Failure Morphology

It is generally accepted that if a rolling-element bearing is properly designed, manufactured, installed, lubricated, and maintained, “classical” rolling-element fatigue is the failure phenomenon that limits bearing life (Ref. 45). Rolling-element fatigue is extremely variable but is statistically predictable depending on the steel type, steel processing, heat treatment, bearing manufacturing and type, and operating conditions. This type of fatigue is a cycle-dependent phenomenon resulting from repeated stress under rolling-contact conditions and is considered high-cycle fatigue. Sadeghi, et al. (Ref. 46) provide an excellent review of this failure mode.

Rolling-element fatigue can be simply categorized as either surface or subsurface initiated (Fig. 1). The subsurface-initiated fatigue failure is that referred to as ‘classical’ rolling-element fatigue. The fatigue failure manifests initially as a pit or spall that is generally limited in depth to the zone of the resolved maximum shearing stresses and in diameter to the width of the contact area (Fig. 1) (Ref. 45). Figure 1(a) illustrates the sequential progression of a subsurface-initiated crack from a nonmetallic inclusion such as a hard oxide inclusion that acts as a stress raiser. With repeated stress cycles the crack propagates to form a crack network that reaches the surface, resulting in a spall shown sequentially in the bearing race track. At this point the bearing is no longer fit for its intended purpose and should be removed from service. Bearing rating life  $L_{10}$  as defined by the standards ANSI 9–1990 (Ref. 6), ANSI 11–1990 (Ref. 7), and ISO 281:2007 (Ref. 42) is based on subsurface origin (classical) rolling-element fatigue (Ref. 45).



When the bearing is operated under conditions that deviate from the rating or reference condition, the fatigue origin can be of “surface” origin as illustrated in Figure 1(b). In this instance the spall can initiate from a defect or stress raiser on and/or near the surface of the bearing raceways and/or rolling elements (Refs. 45, 47, and 48). In the instance of surface-initiated fatigue spall from hard-particle contamination, the load zone of the race and/or rolling element is indented by the contaminant (Fig. 2). The indent acts as a stress raiser from which the crack initiates and then propagates to form a crack network into the subsurface region of resolved maximum shearing stresses (Fig. 1(b)). A spall similar in appearance to that of the surface-initiated spall is formed. The characteristic difference between the surface- and subsurface-initiated spalls is that surface-initiated spalls result in a “arrowhead-type” geometry at the leading edge or point of origin on the rolling-element or raceway surface (Fig. 1(b)).

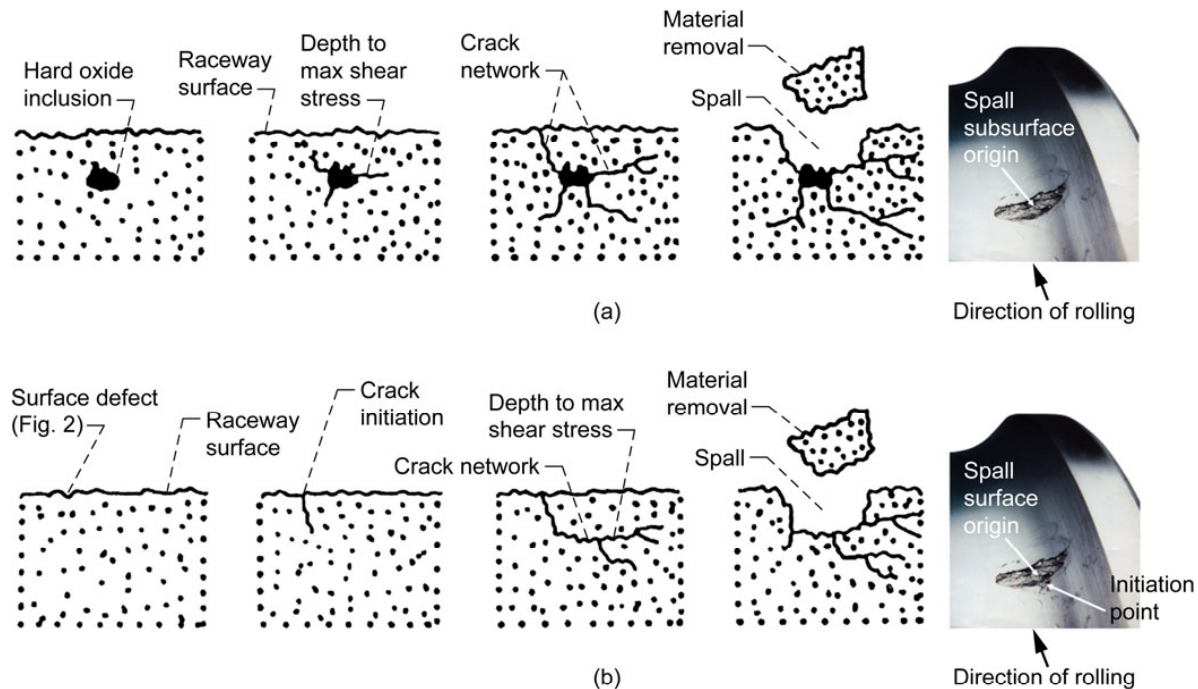


Figure 1.—Subsurface- and surface-initiated rolling-element fatigue failures. For subsurface-initiated failure, spall begins in subsurface region of maximum resolved shearing stresses. For surface-initiated failure, spall begins at surface defect or discontinuity that acts as a stress raiser. (a) Subsurface-initiated spall from hard oxide inclusion. (b) Surface-initiated spall from debris indentation.

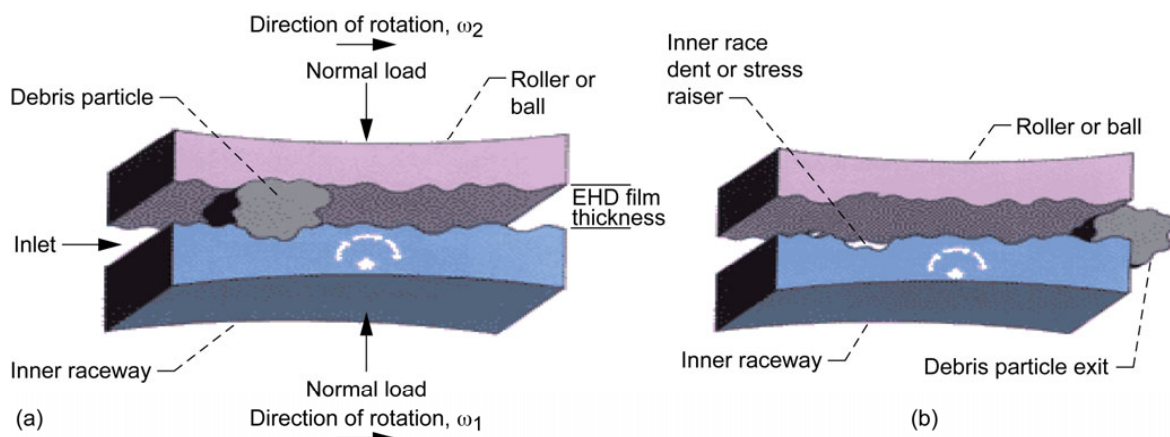


Figure 2.—Stress raiser from indentation of debris particle entering Hertzian contact zone between rolling element and raceway.

The number, size, and material properties of particles entering the Hertzian contact of the rolling element and raceway impact bearing life. The nature of the particles in the oil is a function of several processes:

1. Manufacturing processes (swarf, chips, and grit)
2. Internal generation, including wear debris and chemical attack of surfaces
3. Ingression from the environment (sand and dust)
4. Maintenance activities (making/breaking fittings and new oil)
5. Lubricant breakdown products (sludges, precipitates, and coke)

Typical particle size distributions from a variety of mechanical systems are shown in Figure 3 (Refs. 31 and 32). The greater number of smaller particles in each of the lubrication systems is due in part to wear mechanisms that generate smaller particles, and in part, to removal processes that tend to remove more large particles than small ones (Ref. 31). Steele (Ref. 49) reported a wide range of particulate levels in unused turbine oils.

Work reported by Tonicello et al. (Ref. 47) showed that for pairs of disks in rolling contact where one pair comprises silicon nitride ( $\text{Si}_3\text{N}_4$ ) against AISI M-50 bearing steel and the second pair comprises AISI M-50 against AISI M-50, and the oil particle contamination is AISI M-50 wear debris, the dent indentation on the raceway of the respective AISI M-50 disk was three times deeper with the  $\text{Si}_3\text{N}_4$  disk than with the AISI M-50 steel disk on a AISI M-50 steel disk (Fig. 4). Greater stress concentration in the Hertzian contact results from deeper dents. It can be reasonably concluded that ceramic balls or rollers can lead to deeper dents in a mating steel raceway (Ref. 47).

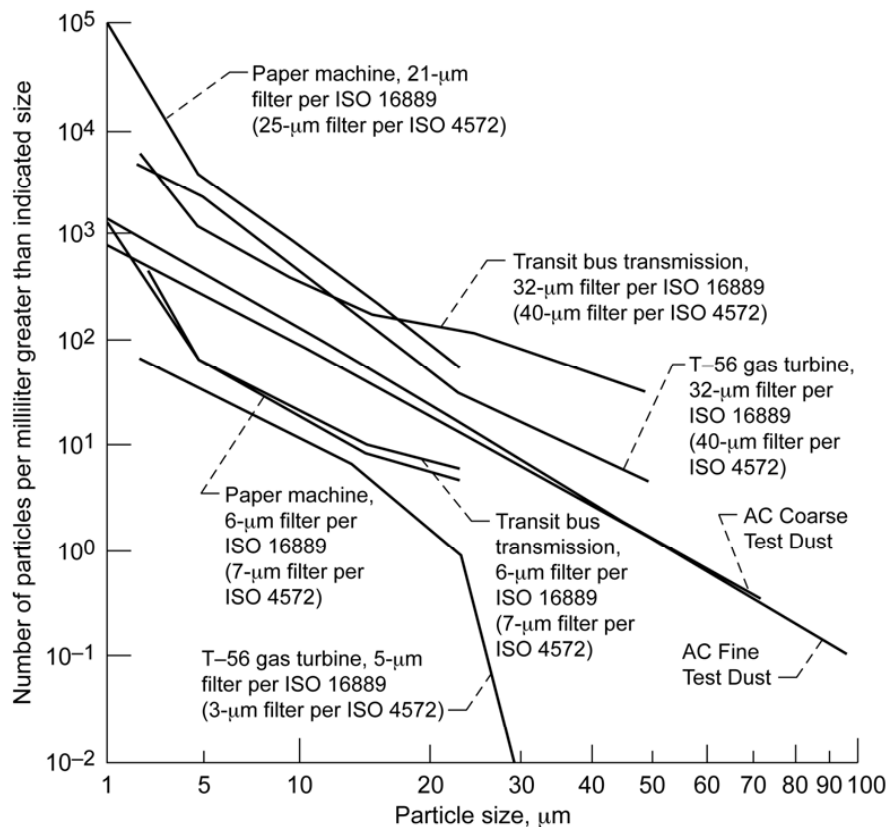


Figure 3.—Comparing effect of oil filtration on particle contamination for mechanical systems according to particle size calibration per ISO 4402 (Refs. 31 and 32).

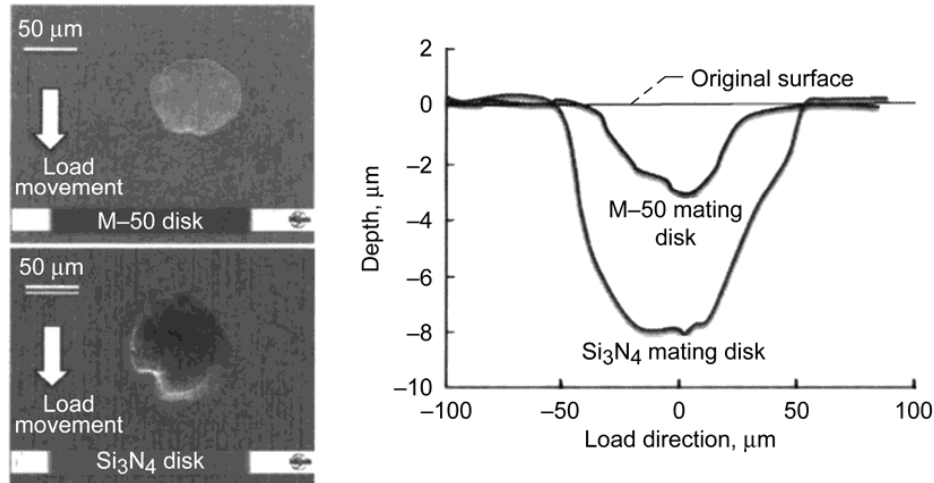


Figure 4.—Dent profiles made on AISI M-50 steel disks by AISI M-50 particles. Mating disk (counterface) is Si<sub>3</sub>N<sub>4</sub> or AISI M-50 steel. (Figure from Tonicello, et al. (Ref. 47).)

Morales-Espejel and Gabelli (Ref. 50) discuss “the different hypotheses available to explain the interaction of sliding (and rolling) with the indentation marks in both gears and rolling bearings” under EHD lubrication. They present theory and analysis that are qualitatively verified by experiment. The micropitting phenomenon occurring around indentation marks are described by them with the same physical model of Morales-Espejel and Brizmer (Ref. 51) that takes into account the progression of surface fatigue induced by locally reduced lubrication conditions.

### Filter Rating Procedure

The basic procedure for rating filters is shown in the flow diagram of Figure 5. During the multipass test, slurry of silica particles is continuously fed into a recirculating system.

Particles flow into the filter, where some are captured, and others return to the reservoir where they continue to recirculate. Throughout the test, particles upstream and downstream of the filter are quantified with electronic automatic particle counters. The filter factor, or filter rating, for particle size  $x$  is defined as the ratio of upstream to downstream counts recorded during the test, denoted  $\beta_x$  and termed the “beta ratio”:

$$\beta_x = (\text{Upstream counts} \geq x \text{ } \mu\text{m}) \div (\text{Downstream counts} \geq x \text{ } \mu\text{m}) = \frac{NU_x}{ND_x} \quad (1)$$

where  $NU_x$  and  $ND_x$  are the average number of particles upstream and downstream of the filter, respectively, whose particle size is greater than  $x \text{ } \mu\text{m}$ .

Originally, automatic particle counters were calibrated using AC FTD, per ISO 4402:1991 (Ref. 37). This material comprises small irregularly shaped particles of silica sand. Although widely used, AC FTD lacked traceability, had batch-to-batch variations, and reported size distributions of dubious accuracy especially below 10  $\mu\text{m}$ . In order to make testing more reproducible, calibration methods were developed traceable to an NIST standard reference material, NIST SRM 2806 (Ref. 39). However, the new calibration method (per ISO 11171:1999) (Ref. 38) reports the size of particles (in microns) as the diameter of an equivalent sphere, rather than the longest dimension as in the original method. These alterations, in effect, changed the “micron ruler.” As shown in Table I (Ref. 38), the particle sizes originally below 10  $\mu\text{m}$  are reported larger, and sizes above 10  $\mu\text{m}$  are reported smaller. For example, a silica particle reported to be 3  $\mu\text{m}$  in size by a particle counter calibrated to the old standard is now reported as 5.1  $\mu\text{m}$  in size using the new calibration standard.

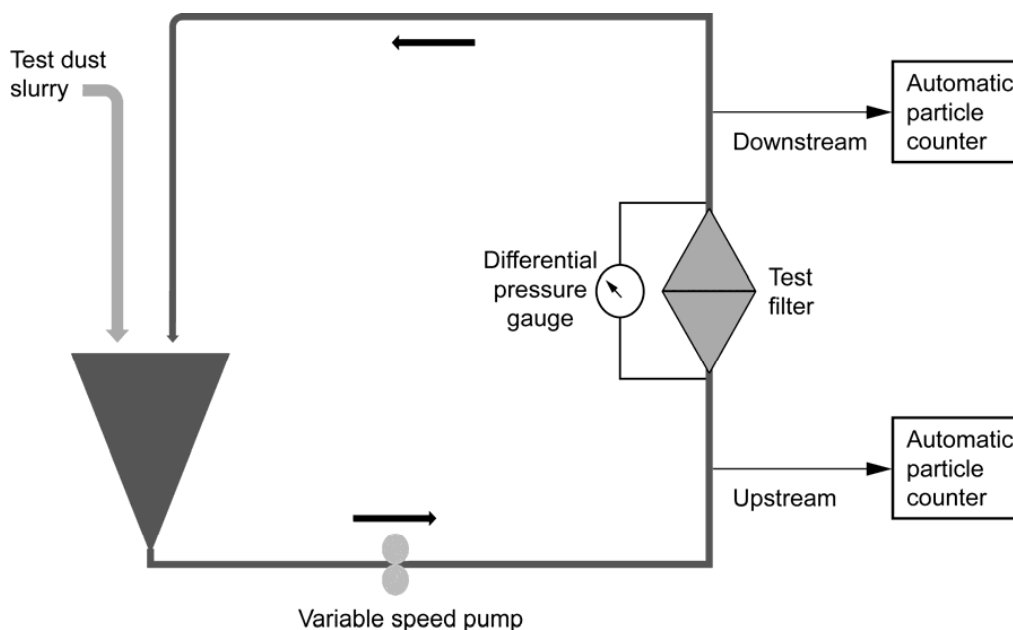


Figure 5.—Basic multipass test procedure for filter rating using silica-based test dust and in-line particle counting.

TABLE I.—EXPERIMENTAL COMPARISON OF CONTAMINANT PARTICLE SIZE CLASSIFICATION BASED ON ISO STANDARDS  
[Adapted from Ref. 38.]

Old size, ISO 4572:1981 method (Ref. 33), AC FTD calibration $\beta_x = 200$ , $\mu\text{m}$	New size, ISO 16889:2008 method (Ref. 36), ISO MTD calibration $\beta_{x(c)} = 200$ , $\mu\text{m}$
1	4.2
2	4.6
3	5.1
5	6.4
7	7.7
10	9.8
15	13.6
20	17.5
25	21.2
30	29.4
40	31.7

Along with calibrating automatic particle counters, AC FTD was also used as the test contaminant in the previous version of the multipass test. When it became unavailable in the 1990s, an alternative test material with similar chemical composition and size distributions was adopted, ISO MTD. Although using a slightly different test contaminant influences test results, the changes are less significant than those produced by the new particle size calibration described above. The  $\beta_x$  values obtained via the revised multipass test (ISO 16889:2008) (Ref. 36), using ISO MTD and particle counters calibrated to the NIST standard, are now reported as  $\beta_{x(c)}$  values, where  $x$  is the micron size per Equation (1), and  $c$  emphasizes the new calibration method.

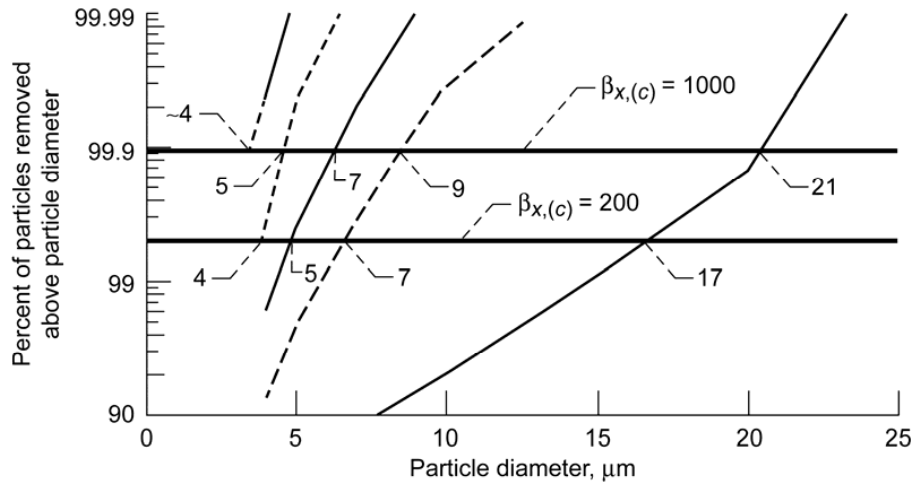


Figure 6.—Representative results from multipass filter testing per ISO 16889:2008.

Twenty years ago leading filter manufacturers rated filters at the particle size  $x$  where  $\beta_x = 200$ , as calculated by Equation (1). For example, a filter rated at  $\beta_5 = 200$  has 1 out of every 200 particles equal to or greater than  $5\ \mu\text{m}$  pass through the filter during testing. However, most engineers and end-users prefer to think of filter ratings as the size where essentially no particles pass through the filter. In an attempt to reach this ideal, many manufacturers now also rate filters at the particle size  $x$  where  $\beta_x = 1000$ . At this higher rating, only 1 particle in 1000 passes through the filter during testing. Using the new ISO standard, modern filter ratings are  $\beta_{x(c)} = 200$  and  $\beta_{x(c)} = 1000$ . Typical removal efficiency results are shown in Figure 6.

The removal efficiency  $E_x$  of any particle size  $x$  can be related to the  $\beta$  factor as follows:

$$E_x = (1 - 1/\beta_x) \times 100 \quad (2)$$

For any filter there is a large particle size above which essentially nothing passes,  $\beta_x \geq 10^4$  and  $E_x \rightarrow 100$  percent. In contrast, there is also a small particle size for which  $\beta_x \rightarrow 1$  and  $E_x \rightarrow 0$  percent, so that nearly all particles this size and smaller freely pass through the filter and accumulate to copious amounts in recirculating systems. For intermediate sizes, a fraction of the particles are captured and the rest pass downstream. As an example, a filter with  $\beta_{10(c)} = 1000$  removes 99.9 percent of all particles  $\geq 10\ \mu\text{m}$  in size during a multipass test.

In summary, the changes to the ISO filter rating standard were as follows:

1. Particle counter calibration
  - a. Changed from AC FTD to NIST calibration
  - b. Increased accuracy and reproducibility
  - c. Changed the “micron ruler”
2. Test contaminant
  - a. Changed from AC FTD to ISO MTD
  - b. Increased accuracy and reproducibility
3. Highest filter rating changed from  $\beta_x = 200$  to  $\beta_{x(c)} = 1000$ 
  - a. Closer to concept of “absolute rating”

Examples:

1. For  $\beta_{5(c)} = 200$ , 1 out of every 200 particles or 5 out of every 1000 particles greater than  $5\ \mu\text{m}$  passes through filter during test.
2. For  $\beta_{5(c)} = 1000$ , 1 out of every 1000 particles greater than  $5\ \mu\text{m}$  passes through filter during test.

## Results and Discussion

In 1991, based on the experimental research of Bhachu et al. (Refs. 12 and 13) and Loewenthal et al. (Refs. 9 to 11), Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life (adjustment) factors ( $LF$ ) as a function of oil filter ratings ( $FR$ ). These equations were normalized ( $LF = 1$ ) to a 10  $\mu\text{m}$  filter rating at normal cleanliness (as it was then defined, where  $\beta_x = 200$  per ISO 4572:1981 (Ref. 33)). The life factor equations were incorporated into the reference book, “STLE Life Factors for Rolling Bearings” (Ref. 32).

The Needelman and Zaretsky oil filtration life factors (Ref. 31) have been used, in conjunction with ANSI/ABMA standards (Refs. 6 and 7) and with bearing computer codes (Ref. 34). Experimental studies made by other investigators verify the relationship between rolling-element bearing life and the size, number, and types of particle contamination (Refs. 1, 2, 4, 5, 8, 14 to 20, 47, and 48). The ISO 281:2007 (Ref. 42) incorporates a rolling-element bearing life factor based on lubricant cleanliness and EHD film thickness based on the work of Ioannides et al. (Ref. 17) but does not relate the lubricant cleanliness to the filter ratings.

Filter ratings have been revised and improved (Ref. 35). They are now based on an upgraded filter rating method per ISO 16889:2008 (Ref. 36), employing particle counts calibrated to an NIST standard (Refs. 36 and 39). The work reported here was undertaken to calibrate the “old” and obsolete filter ratings to the “new” filter ratings, and to recalibrate the previously published bearing life factors based on the old filter ratings to life factors based on the new filter ratings. The revised rolling-element bearing life factors were then compared to the life factors in the ISO 281:2007 Standard (Ref. 42) that are based solely on lubricant cleanliness levels.

### Recalibration of Filter Ratings

Old  $\beta_x = 200$  filter ratings (AC FTD calibration) were converted to new  $\beta_{x(c)} = 200$  filter ratings using Table I and plotted in Figure 7. Although this transformation does not take into account the change in test contaminant to ISO MTD, alterations in counter calibration dominate over the change in test contaminant. The equation relating old  $\beta_x = 200$  filter ratings ( $FR_{\text{OLD}200}$ ) with new  $\beta_{x(c)} = 200$  filter ratings ( $FR_{\text{NEW}200}$ ) is

$$FR_{\text{NEW}200} = 0.722(FR_{\text{OLD}200}) + 2.97 \quad (3a)$$

or

$$FR_{\beta_{x(c)}200} = 0.722(FR_{\beta_x200}) + 2.97 \quad (3b)$$

and

$$FR_{\text{OLD}200} = 1.39(FR_{\text{NEW}200}) - 4.11 \quad (4a)$$

or

$$FR_{\beta_x200} = 1.39(FR_{\beta_{x(c)}200}) - 4.11 \quad (4b)$$

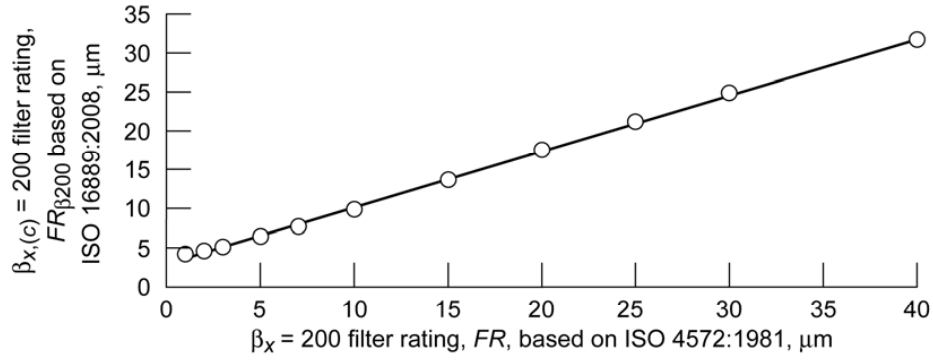


Figure 7.—Comparison of “new” filter ratings (based on ISO 16889:2008 using spherical particle calibration per ISO 11171:1999) with “old” obsolete filter ratings (based on ISO 4572:1981 using AC Fine Test Dust (AC FTD) calibration per ISO 4402:1991).

Multipass filter tests were then performed according to the new ISO 16889:2008 (Ref. 36) method using 25 different filters over a wide range of filter efficiencies from five different manufacturers. This allowed plotting  $\beta_{x(c)} = 1000$  filter ratings with  $\beta_{x(c)} = 200$  ratings. The relationship is plotted in Figure 8, and approximated by the equations

$$FR_{NEW1000} = 1.17 (FR_{NEW200}) + 0.650 \quad (5a)$$

or

$$FR_{\beta_{x(c)}1000} = 1.17 (FR_{\beta_{x(c)}200}) + 0.650 \quad (5b)$$

and

$$FR_{NEW200} = 0.855 (FR_{NEW1000}) - 0.556 \quad (6a)$$

or

$$FR_{\beta_{x(c)}200} = 0.855 (FR_{\beta_{x(c)}1000}) - 0.556 \quad (6b)$$

By cross-plotting the data of Figure 7 with those of Figure 8, a relation between old and obsolete  $\beta_x = 200$  filter rating and new filter rating  $\beta_{x(c)} = 1000$  was obtained (Fig. 9) described by the equations

$$FR_{NEW1000} = 0.848 (FR_{OLD200}) + 4.14 \quad (7a)$$

or

$$FR_{\beta_{x(c)}1000} = 0.848 (FR_{\beta_{x200}}) + 4.14 \quad (7b)$$

and

$$FR_{OLD200} = 1.18 (FR_{NEW1000}) - 4.88 \quad (8a)$$

or

$$FR_{\beta_{x200}} = 1.18 (FR_{\beta_{x(c)}1000}) - 4.88 \quad (8b)$$

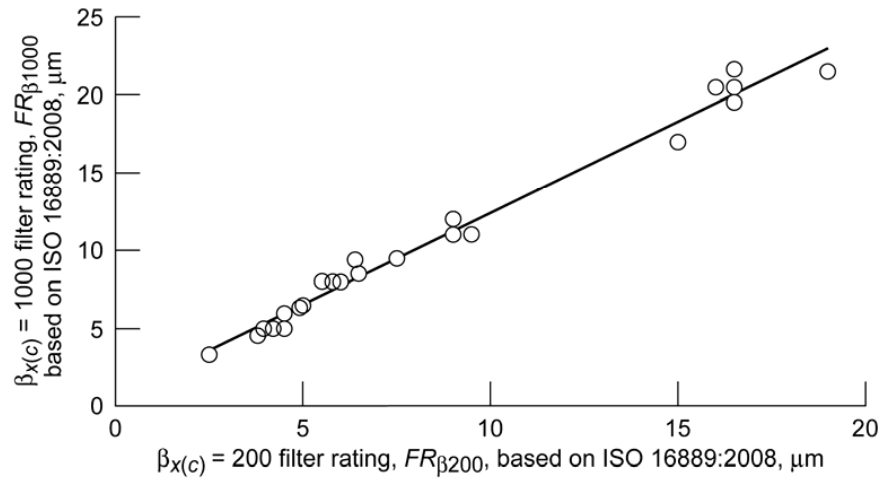


Figure 8.—Comparison of  $\beta_{X(c)} = 1000$  filter ratings with  $\beta_{X(c)} = 200$  filter ratings. Filters tested per ISO 16889:2008.

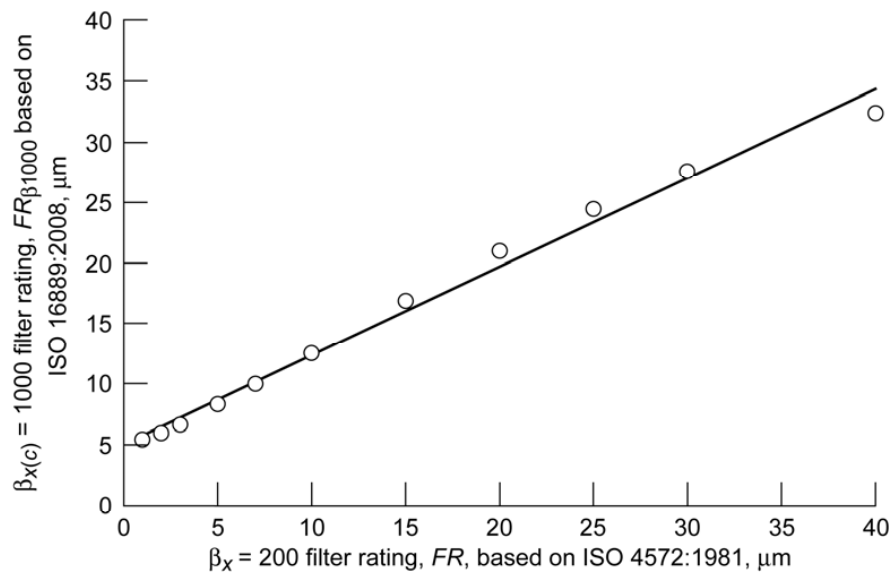


Figure 9.—Comparison between  $\beta_{X(c)} = 1000$  filter ratings based on ISO 16889:2008 and  $\beta_{X(c)} = 200$  filter ratings based on obsolete ISO 4572:1981.



TABLE II.—EFFECT OF OIL FILTRATION ON THE ROLLING-ELEMENT FATIGUE LIFE OF  
25-μm-BORE ROLLER BEARINGS

[Radial load, 2975 N; original surface composite roughness (rms) 0.33 μm. From Bhachu et al. (Refs. 12 and 13).]

Test filter rating ( $\beta_x \geq 200$ ), μm	Composite surface roughness, $\sigma$ after testing, μm	Calculated film thickness, μm	Film parameter after testing, $\Lambda$	Experimental life, millions of inner-race revolutions				Weibull slope, $m$	Failure index <sup>a</sup>	Predicted $L_{10}$ life from Eq. (9), 10 <sup>6</sup> inner-race revolutions
				10-percent life, $L_{10}$	90-percent confidence limits	50-percent life, $L_{50}$	90-percent confidence limits			
40	0.41	0.58	1.4	1.5	1.0 to 2.0	2.1	1.9 to 2.3	5.8	10 out of 10	1.9
25	.36	↓	1.6	2.5	2.0 to 3.0	3.0	2.9 to 3.1	9.4	↓	2.5
6	.32		1.8	4.5	3.4 to 5.8	5.9	5.4 to 6.3	7.0		5.5
3	.26		2.2	8.0	5.5 to 11.6	12.0	10.7 to 13.5	4.7		8.0 <sup>b</sup>
2.5	.22		2.6	6.5	3.6 to 11.8	12.4	9.9 to 15.5	2.9		8.8
Magnetic	.20		2.9	5.0	2.9 to 8.5	10.0	8.3 to 12.0	3.2		---
40	0.41	0.99	2.4	1.8	-----	3.3	-----	3.1	10 out of 10	2.2
3	.26	.99	3.8	9.3	-----	16.2	-----	3.4	↓	9.3 <sup>b</sup>
40/3	.41	.99	2.4	1.7	-----	3.1	-----	3.3		2.2

<sup>a</sup>Number of fatigue failures out of number of bearings tested.

<sup>b</sup>Life prediction normalized to 3-μm filter,  $\beta_x = 200$ .

### Rolling Bearing Fatigue Life Factors

There were two independent investigations to specifically determine the effect of oil filtration on rolling-element bearing life. These were the studies of Loewenthal et al. (Refs. 9 to 11) and Bhachu et al. (Refs. 12 and 13).

#### Roller Bearings

Bhachu et al. (Refs. 12 and 13) used a gear test machine to generate wear debris. The gear wear debris, verified by ferrography to be representative of that found in helicopter gearboxes, was used as the contaminant. Rolling-element fatigue tests were conducted with 25-mm bore roller bearings having a 2957 N radial load. For each test series, gear oil flow was passed through one of four possible filters of different ratings from 2.5 to 40 μm or through an electromagnetic separator and continuously supplied to a parallel roller-bearing fatigue tester. EHD film thickness and  $\Lambda$  values during testing are shown in Table II.

Significantly, tests run with 40-μm filtration for only 30 min before switching to 3-μm filtration showed substantially the same lives as if all running had been with a 40-μm filter. Apparently the early damage could not be healed, at least in these small roller bearings. The test results are also shown in Table II. These results show that life increased with improved filtration.

The original filter ratings were based on the old filter rating method, ISO 4572:1981 (Ref. 33), with  $\beta_x = 200$ . From these roller bearing life data (Table II), it was assumed in (Ref. 31) that the filter life factor takes the following form:

$$LF \approx C_1 (FR)^{-C_2} \quad (9a)$$

$C_1$  is an empirically determined constant and  $C_2$  is an empirically determined exponent. The experimentally determined  $L_{10}$  lives from Table II for filter ratings  $\beta_{x200} = 3$  and 25 μm are 8 and 2.5 million inner-race revolutions. Solving for  $C_1$  and  $C_2$  and normalizing  $LF = 1$  when  $FR = 10$  μm at  $\beta_x = 200$  (normal cleanliness), the following empirical relation was obtained

$$LF \approx 3.5 (FR_{OLD200})^{-0.55} \quad (9b)$$

or

$$LF \approx 3.5(FR_{\beta_x 200})^{-0.55} \quad (9c)$$

Substituting the new filter ratings into Equation (9) from Equations (4) and (8), respectively, allows calculating bearing life factors for new filter ratings:

For  $\beta_{x(c)} = 200$ :

$$LF \approx 3.5 [1.39(FR_{\text{NEW}200}) - 4.11]^{-0.55} \quad (10a)$$

or

$$LF \approx 3.5 [1.39(FR_{\beta_{x(c)}200}) - 4.11]^{-0.55} \quad (10b)$$

For Equation (10) above, where  $FR_{\beta_{x(c)}200} \leq 4$ ,  $LF \approx 2.8$ .

For  $\beta_{x(c)} = 1000$ :

$$LF \approx 3.5 [1.18(FR_{\text{NEW}1000}) - 4.88]^{-0.55} \quad (11a)$$

or

$$LF \approx 3.5 [1.18(FR_{\beta_{x(c)}1000}) - 4.88]^{-0.55} \quad (11b)$$

For Equation (11) above, where  $FR_{\beta_{x(c)}1000} \leq 5$ ,  $LF \approx 3.5$ .

Using Equation (9), the  $L_{10}$  lives for roller bearings were predicted by normalizing Equation (9) to the experimentally obtained  $L_{10}$  life using the 3- $\mu\text{m}$ -rated ( $\beta_x = 200$ ) filter. These results are presented in Table II for comparison purposes.

The posttest inner-raceway measurements for 40- $\mu\text{m}$  ( $\beta_x = 200$ ) filtration showed greater out-of-roundness than in the untested bearing. Less out-of-roundness was observed with finer filtration down to the 8- $\mu\text{m}$  ( $\beta_x = 200$ ) rating. Virtually no out-of-roundness was observed when the 3- $\mu\text{m}$  ( $\beta_x = 200$ ) filter was used. Below the 3- $\mu\text{m}$  ( $\beta_x = 200$ ) level the measurement was similar to that of the unrun bearing. Bhachu et al. (Refs. 12 and 13) suggested that particles smaller than 3  $\mu\text{m}$  ( $\beta_x = 200$ ) were too small to have any effect on roundness and merely passed through the contacts of the rollers and raceways.

There is a strong suggestion from the data of Table II that the lack of contamination contributes to improvement in bearing raceway surface finish during operation. There appears to be a correlation between the lubricant film parameter  $\Lambda$  after testing and rolling-element fatigue life as evidenced by these data.

## Ball Bearings

Loewenthal et al. (Refs. 9 to 11) performed a series of tests to measure the quantitative effects of filtration on rolling-element fatigue life. Four levels of filtration were investigated using full-flow ( $\beta_x = 200$ ) filters rated at 3, 30, 49, and 105  $\mu\text{m}$ . The 3- $\mu\text{m}$  ( $\beta_x = 200$ ) filter used for these tests had been developed to replace the original 40- $\mu\text{m}$  ( $\beta_x = 200$ ) filter for a helicopter gas turbine lubrication system. During service these new filter elements were not only found to provide a much cleaner lubricant with less component wear but contrary to prior belief, greatly extend the time between filter and oil changes, as discussed by Loewenthal et al.

The test bearings were 65-mm deep-groove ball bearings run at 15 000 rpm under a radial load of 4580 N, which produced a maximum Hertz stress of 2410 MPa. The lubricant contaminant rate was 0.125 g/hr per bearing. The test environment was designed to simulate an aircraft lubrication system containing multiple bearings, pumps, and other components commonly found in such systems. Test temperature was 347 K. The test lubricant was a MIL-L-23699 type, which produced a  $\Lambda$  value of 3.3 based on race and ball pretest surface finish measurements. The test contaminant was similar to the particulate matter found in the lubricant filters of 50 IT8D commercial engines (Jones and Loewenthal

TABLE III.—COMPARISON OF BALL BEARING FATIGUE LIFE RESULTS WITH AN ULTRACLEAN LUBRICANT AND WITH DIFFERENT LEVELS OF FILTRATION IN A CONTAMINATED LUBRICANT

[Radial load, 4580 N; speed, 15 000 rpm; temperature, 347 K; test lubricant, MIL-L-23699 type; film parameter  $\Lambda$ , 3.3. From Loewenthal et al. (Refs. 9 to 11).]

Test series (lubricant condition)	Test filter rating ( $\beta_x \geq 200$ ), $\mu\text{m}$	Experimental life, hr		Weibull slope, $m$	Failure index <sup>a</sup>	Confidence number, <sup>b</sup> percent		Predicted $L_{10}$ life from Eq. (12), hr
		10-percent life, $L_{10}$	50-percent life, $L_{50}$			$L_{10}$	$L_{50}$	
Ultraclean Clean (baseline)	3	1099	1741	4.1	5 out of 9	--	--	1099 <sup>d</sup>
	49	672	2276	1.5	9 out of 32	76	--	547
Contaminated	3	505	993	2.8	10 out of 16	93	99	505 <sup>e</sup>
Contaminated	30	594	857	5.1	11 out of 16	96	99	284
Contaminated	49	367	533	5.1	20 out of 32	99	99	251
Contaminated	105 <sup>c</sup>	-----	-----	---	-----	--	--	208

<sup>a</sup>Number of fatigue failures out of number of bearings tested.

<sup>b</sup>Probability (expressed as a percentage) that bearing fatigue life in a given test series will be inferior to the life obtained with ultraclean lubrication. A 90-percent or greater confidence number is considered statistically significant.

<sup>c</sup>Test series was suspended after 448 test hours on each of the test bearings due to excessive bearing wear. No fatigue failures were encountered.

<sup>d</sup>Life prediction normalized to 3- $\mu\text{m}$  filter,  $\beta_x = 200$  and  $L_{10} = 1099$  hr.

<sup>e</sup>Life prediction normalized to 3- $\mu\text{m}$  filter,  $\beta_x = 200$  and  $L_{10} = 505$  hr.

(Ref. 52)). Because this engine has a number of carbon-graphite bearing sump seals, replication of oil contaminants in engines with “windback-type” labyrinth seals demanded the use of a contaminant made of 88 percent carbon-graphite dust, 11 percent Arizona test dust, and 1 percent stainless steel particles.

The results of these tests are summarized in Table III. As with the work of Bhachu et al. (Refs. 12 and 13), improved filtration increased bearing life. However, for the contaminated tests, there appears to be no statistical difference in life obtained between the 3- and 30- $\mu\text{m}$  filters. Because of the severe wear obtained, the contaminated 105- $\mu\text{m}$  filter test series was suspended after 448 hr on each bearing. No fatigue failures were encountered, because of the gross wear of the bearing races. Based upon the test results between the 3- and 49- $\mu\text{m}$  filters the following life relation is suggested from these data for  $\beta_x = 200$  filter rating for ball bearings:

$$LF \approx 1.8(FR_{\text{OLD}200})^{-0.25} \quad (12a)$$

or

$$LF \approx 1.8(FR_{\beta_x 200})^{-0.25} \quad (12b)$$

Using Equation (12), the  $L_{10}$  lives of ball bearings were predicted by normalizing Equation (12) to the experimentally obtained  $L_{10}$  life using the 3- $\mu\text{m}$  ( $\beta_x = 200$ ) rated filter. These results are presented in Table III for comparison purposes.

Substituting the filter rating for  $FR$  ( $\beta_x = 200$ ) from Equations (4) and (8), respectively, into Equation (12), filter life factors can be calculated for the new filter ratings where

1. For  $\beta_{x(c)} = 200$  filter rating:

$$LF \approx 1.8[1.39(FR_{\text{NEW}200}) - 4.11]^{-0.25} \quad (13a)$$

or

$$LF \approx 1.8[1.39(FR_{\beta_{x(c)} 200}) - 4.11]^{-0.25} \quad (13b)$$

for Equation (13), where  $FR_{\beta_{x(c)}200} \leq 4$ ,  $LF \approx 1.6$ .

2. For  $\beta_{x(c)} = 1000$  filter rating:

$$LF \approx 1.8[1.18(FR_{\text{NEW}1000}) - 4.88]^{-0.25} \quad (14a)$$

or

$$LF \approx 1.8[1.18(FR_{\beta_{x(c)}1000}) - 4.88]^{-0.25} \quad (14b)$$

For Equation (14), where  $FR_{\beta_{x(c)}1000} \leq 5$ ,  $LF \approx 1.8$ .

Table IV provides results from Equations (6), (9), and (12) for various filter ratings. The resultant life adjustment factors can be used to adjust the calculated bearing  $L_{10}$  or catalog life to account for filtration level in the lubricant system. These  $LF$  values are normalized to filter ratings ( $FR$ ) of 10  $\mu\text{m}$  at  $\beta_{x(c)} = 200$  and 13  $\mu\text{m}$  at  $\beta_{x(c)} = 1000$  (normal cleanliness) and are independent of the  $\Lambda$  and/or  $\kappa$  values, loading conditions, and bearing size. Based upon the data of Loewenthal et al. (Refs. 9 to 11), it is not recommended to use a life adjustment factor less than 0.5 even when no filter is used. Further, Equations (9) and (12) may reflect the differences in the effect of particle damage between roller and ball bearings.

Technology for improved oil filtration is commercially available. By minimizing the number of harmful particles entering a rolling-element bearing, oil filtration can substantially extend bearing life. In addition to machine-generated wear debris and ambient mineral dusts, all-too-frequent high contamination levels in new oil also requires good filtration.

No reported testing has been performed comparing grease lubrication, which entraps wear debris, with oil lubrication for the same bearings, with or without oil filtration. It is suggested that for long-term application a  $LF = 0.5$  for grease lubrication be considered the same as oil lubrication for bearings without filtration where no periodic regreasing of the bearing occurs. Where the bearing is continuously or periodically regreased, a  $LF = 1$  should be considered.

TABLE IV.—NEEDELMAN-ZARETSKY OIL CLEANLINESS (FILTER)  
LIFE FACTORS ( $LF$ s) BASED ON OIL FILTER RATING ( $FR$ )

Filter rating, $FR$ , $\mu\text{m}$			Life factor, $LF$	
(Old) $\beta_x = 200$	(New) $\beta_{x(c)} = 200$ Eq. (3)	(New) $\beta_{x(c)} = 1000$ Eq. (7)	Roller bearing, Eqs. (9) to (11)	Ball bearing, Eqs. (12) to (14)
3	5	7	2.0	1.4
6	7	9	1.4	1.2
8	9	11	1.1	1.1
10 <sup>a</sup>	10	13	1.0	1.0
12	12	14	0.9	1.0
25	21	25	0.6	0.8
40	32	38	0.5	0.7
49 <sup>b</sup>	38	46	0.4 <sup>b</sup>	0.7
60 <sup>b</sup>	46	55	0.4 <sup>b</sup>	0.6
105 <sup>b</sup>	79	93	0.3 <sup>b</sup>	0.6

<sup>a</sup>Normalized to  $FR = 10 \mu\text{m}$  at ( $\beta_x = 200$ )

<sup>b</sup>For filter ratings at  $\beta_x = 200$  exceeding 40  $\mu\text{m}$ ; with no filtration it is not recommended to use  $LF < 0.5$ .

## Comparison of Filter Life Factors (*LFs*) to ISO Standard 281:2007

In 2000 the International Organization for Standardization (ISO) modified the standard ISO 281:1990, “Rolling bearings—Dynamic Load Ratings and Rating Life,” to include a fatigue limit (Ref. 53). The endurance or fatigue limit as applied to rolling-element bearings is based on the theoretical work presented in 1985 by Ioannides and Harris (Ref. 54). It is a theoretical load or shearing stress (based on a Hertzian contact stress) below which no fatigue failure is assumed to occur, and therefore where fatigue life is infinite. ISO 281:2007 replaced ISO 281:1990 as modified in 2000. The 2007 standard adopted the 1999 approach to bearing life calculations presented by Ioannides, Bergling, and Gabelli (Ref. 55) and includes the effects of lubricant contamination on bearing life.

The ISO 281:2007 Standard (Ref. 42) incorporates a new service life formula that integrates all life adjustment factors *LF* in what is now called  $a_{ISO}$ . The life factor  $a_{ISO}$  includes four interdependent factors: (1) lubrication regime,  $\kappa$ ; (2) lubricant contamination or oil cleanliness,  $e_C$ ; (3) applied dynamic equivalent load (applied load) to the bearing,  $P$ ; and (4) fatigue load or stress endurance limit of the bearing material,  $C_u$ .

$$a_{ISO} = f \left[ \frac{e_C C_u}{P}, \kappa \right] \quad (15)$$

From the ISO 281:2007 Standard (Ref. 42), “...when the lubricant is contaminated with solid particles, permanent indentations in the raceway(s) (and rolling elements) can be generated when these particles are over rolled. At these indentations, local stress risers are generated, which will lead to a reduced life of the rolling bearing. This life reduction due to contamination in the lubricant film is taken into account by the contamination (life) factor  $e_C$ .” In the ISO 281:2007 Standard (Ref. 42), the contamination factor  $e_C$  is given in a table based upon levels of contamination that are not tied to specific filter ratings.

The standard states that the contamination life factor is dependent on the following:

1. Type, size hardness and quantity of the (contaminant) particles.
2. EHD lubricant film thickness (viscosity ratio,  $\kappa$ ).
3. Bearing size (bearing pitch diameter,  $D_p$ ).

The lubrication regime is defined by EHD theory. In the standard it is defined by the parameter  $\kappa$ , the ratio of the actual viscosity of the lubricant in the bearing at operating temperature to a reference viscosity. The reference viscosity is that which would produce a lubricant film thickness equal to the composite surface roughness of the rolling element and the raceway, or  $\Lambda = 1$ . If  $\kappa < 1$ , the contact is in a boundary lubrication regime where the surface asperities of the rolling element and the raceway are in contact. It is preferable to operate the bearing in a lubrication regime where  $\kappa \geq 1$ . As  $\kappa$  increases, bearing life increases. The  $\kappa$  value in the standard is based on what is termed in EHD theory as the lubricant factor, or  $\Lambda$ . The  $\Lambda$  is equal to the EHD film thickness  $h$  divided by the composite surface finish  $\sigma$  of the rolling elements in contact with the raceway:

$$\Lambda = \frac{h}{\sigma} \quad (16a)$$

where

$$\sigma = \left( \sigma_1^2 + \sigma_2^2 \right) \quad (16b)$$

and  $\sigma_1$  and  $\sigma_2$  are the root mean square (rms) surface roughness of contacting bodies. Unfortunately, in the ISO 281:2007 Standard (Ref. 42),  $\kappa$  is based on an undefined lubricant and lubricant properties and an undisclosed composite surface finish,  $\sigma$ . However, from the standard where  $\Lambda$  can be calculated,

$$\kappa \approx \Lambda^{1.3} \quad (17)$$

an approximate correlation can be established between filter ratings and the contamination levels. For example, BFPA/P5:1999 (Ref. 56) correlates ISO contamination levels with  $\beta_x = 75$  and  $\beta_x = 200$  filter ratings. Using a similar approach, we correlated filter ratings at  $\beta_{x(c)} = 1000$  per ISO 16889:2008 (Ref. 36) to the contamination levels listed in ISO 281:2007 (Ref. 42). These contamination levels and filter ratings, together with the life factor  $e_C$ , are shown in Table V. Also, listed are the Needelman-Zaretsky filter life factors at  $\beta_{x(c)} = 1000$  from Equations (11) and (14) for comparison purposes.

The contamination life factors,  $e_C$ , from ISO 281:2007 (Ref. 42) are differentiated by both filter rating and bearing pitch diameter. The Needelman-Zaretsky life factor equations do not differentiate based on bearing size but do distinguish between ball and roller bearing types (Ref. 31).

Although we consider the effect of  $\Lambda$  on rolling-element fatigue life independent of and separate from the filter life factors, the Needelman-Zaretsky life factors (Ref. 31) are normalized at  $\Lambda \approx 1.1$  and  $LF = 1$ .

The contamination life factor  $e_C$  from Table V can be used to adjust the calculated bearing  $L_{10}$  or catalogue life to account for filtration level in the lubricant system. These results are shown in Table VI. The values of  $e_C$  for  $D_p < 100$  mm were used. For the data of Bhachu et al. (Refs. 12 and 13),  $\kappa \approx 2.8$ . For the data Loewenthal et al. (Refs. 9 to 11),  $\kappa \approx 4.7$ . For each respective set of data the EHD film thickness was assumed by us to remain unchanged. Hence, the effect of  $\Lambda$  and/or  $\kappa$  was not factored into the predicted lives shown in Table VI.

The results from the Bhachu et al. (Refs. 12 and 13), and Loewenthal et al. (Refs. 9 and 11), suggest the following: (1) For filtration levels between 4 and 34  $\mu\text{m}$  at  $\beta_{x(c)} = 1000$ , representing “extreme cleanliness” to “typical contamination,” the ISO 281:2007 Standard (Ref. 42) provides a reasonable qualitative estimate of the effect of particle damage on rolling bearing life; and (2) At conditions of severe contamination and above where the filter ratings are  $\geq 35$   $\mu\text{m}$  at  $\beta_{x(c)} = 1000$ , the ISO 281:2007 Standard (Ref. 32) correlates with the Bhachu et al. results. In contrast, the ISO 281 is conservative compared to the Loewenthal et al. test results. This may be attributed to the use in the Loewenthal et al. tests of a contaminant based on carbon-graphite particles that may act as a solid lubricant while the ISO 281 life ratings are primarily concerned with common hard steel contamination that can be found in industrial gearboxes and used by Bhachu et al. in their tests.

Gabelli, Morales-Espejel, and Ioannides (Ref. 20) provide a discussion of the theoretical basis for the calculation of the contamination factor  $e_C$  that correlates with the curves of contaminant life factors versus  $\kappa$  values presented in the ISO 281:2007 Standard (Ref. 42) and the contaminant life factors presented in Table V that are from the standard. According to Gabelli, Morales-Espejel, and Ioannides (Ref. 20), the following variables should apply in determining a contamination factor  $e_C$ : (1) mean bearing (pitch) diameter, (2) level of contamination (filter size), and (3) lubrication rating of the bearing ( $\kappa$  value). Gabelli, Morales-Espejel, and Ioannides reduce the variables by the elimination of the fatigue limit. Their theoretical results were similar to those in Annex A of the ISO 281:2007 Standard (Ref. 42).

There is an issue as to whether the EHD film thickness or  $\Lambda$  ( $\kappa$  value) mitigates the negative effect of lubricant contamination on rolling-element fatigue life. That is, is the effect of contamination on bearing life less severe with increasing film thickness? In order to benchmark their analysis, Gabelli, Morales-Espejel, and Ioannides (Ref. 20) presented endurance data of 172 bearing population samples obtained over several years comprising 14 types and sizes of rolling-element bearings. It was reported by them that “each bearing sample is normally formed of a group of 30 bearings; several thousand bearings were endurance tested for this set of experimental results.”

TABLE V.—COMPARISON OF EFFECT OF OIL FILTER RATING ON ROLLING-ELEMENT BEARING FATIGUE LIFE FACTOR ( $LF$ ) BETWEEN ISO 281–2007 (REF. 42) AND NEEDELMAN-ZARETSKY, EQS. (11) AND (14)

Level of contamination <sup>a</sup>	Filter rating, $\mu\text{m}$ , ( $\beta_{x(c)} = 1000$ )	Contamination life factor, <sup>a</sup> $e_c$		Needelman-Zaretsky filter life factor, $LF$	
		$D_p < 100 \text{ mm}$	$D_p \geq 100 \text{ mm}$	Roller bearing	Ball bearing
Extreme cleanliness (Particle size of the order of lubricant film thickness; laboratory conditions)	$\leq 4$	1	1	3.5	1.8
High cleanliness (Oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed)	5 to 9	0.8 to 0.6	0.9 to 0.8	3.5 to 1.3	1.8 to 1.2
Normal cleanliness (Oil filtered through fine filter; conditions typical of bearings greased for life and shielded)	10 to 14	0.6 to 0.5	0.8 to 0.6	1.2 to 0.9	1.1 to 1.0
Slight contamination (Slight contamination in lubricant)	15 to 24	0.5 to 0.3	0.6 to 0.4	0.9 to 0.6	1.0 to 0.8
Typical contamination (Conditions typical of bearings without integral seals; coarse filtering; wear particles and ingress from surroundings)	25 to 34	0.3 to 0.1	0.4 to 0.2	0.6 to 0.5	0.8 to 0.7
Severe contamination (Bearing environment heavily contaminated and bearing arrangement with inadequate sealing)	$\geq 35$	0.1 to 0	0.1 to 0	0.5	0.7 to 0.5
Very severe contamination	None	0	0	0.5	0.5

<sup>a</sup>From Reference 42.

TABLE VI.—COMPARISON OF PREDICTED ROLLING BEARING FATIGUE LIVES  
BASED ON CONTAMINANT LIFE FACTOR,  $e_c$ , FROM TABLE V

Level of contamination (see Table V)	Test filter rating $\beta_{x(c)} = 1000$ ( $\beta_x = 200$ ), $\mu\text{m}$	$L_{10}$ life, inner-race revolutions	Predicted $L_{10}$ life based on contaminant life factor $e_c$ , $10^6$ inner-race revolutions
Roller bearing data from Bhachu et al. (Refs. 12 and 13), Table II			
High cleanliness	6.5 (2.5)	$6.5 \times 10^6$	$\sim 9 \times 10^6$
High cleanliness	7 <sup>a</sup> (3)	8.0 <sup>a</sup>	8 <sup>a</sup>
Normal cleanliness	10 (6)	4.5	$\sim 6$
Typical contamination	32 (25)	2.5	$\sim 3$
Severe contamination	46 (40)	1.5	$\sim 1$
Ball bearing data from Loewenthal et al. (Refs. 9 to 11), Table III			
High cleanliness	7 <sup>a</sup> (3)	$455 \times 10^{6a}$	$455 \times 10^{6a}$
Severe contamination	36 (30)	535	$\sim 56$
Severe contamination	56 (49)	330	$\sim 56$
Severe contamination	116 (105)	403 <sup>b</sup>	$\sim 56$

<sup>a</sup>Normalized to 7- $\mu\text{m}$  filter rating where  $\beta_{x(c)} = 1000$  for high cleanliness level of contamination.

<sup>b</sup>See Table 3. Test series was suspended after 448 test hours on each of the test bearings because of excessive wear. No fatigue failures were encountered.

The Gabelli, Morales-Espejel, and Ioannides data for  $\Lambda$  values varying from 0.4 to 2.9 ( $\kappa = 0.3$  to 4) comprises three contamination levels. They classified their contamination conditions as follows:

1. The first contamination condition was classified as their “standard cleanliness tests.” The filtration level at  $\beta_{x(c)} = 1000$  was  $\leq 7 \mu\text{m}$  and their range for  $e_C$  varied from 0.8 to 1. This would be equivalent to “high cleanliness” in Table V.
2. The second contamination condition was classified as “slight contamination.” This is equivalent to  $\beta_{x(c)} = 1000$  filter range of 15 to  $24 \mu\text{m}$  in Table V. They reported that under the given test conditions the expected contamination (life) factors  $e_C$  can range from 0.3 to 0.5. Their actual life data showed  $e_C$  values that ranged from 0.1 to 0.5.
3. The third contamination condition was classified as “typical to severe contamination.” From Table V this would comprise  $\beta_{x(c)} = 1000$  filter range of  $25 \mu\text{m}$  or greater, although no filters appear to have been used in this test series. Their actual life data showed  $e_C$  values that ranged from 0.01 to 0.3.

The Gabelli, Morales-Espejel, and Ioannides experimental data, if statistically significant, shows a relation between the contaminant life factors  $e_C$  and  $\kappa$  as presented in Annex A of the ISO 281:2007 Standard (Ref. 42).

In 1985 Lorosch (Ref. 14) reported that “The influence of contaminants is great with small bearings and decreases with increasing bearing size. Consequently, large bearings have a larger capacity than calculated.” In other words, for a particular contamination level or oil cleanliness, the effect of lubricant contamination is less severe for larger pitch diameter bearings than for smaller pitch diameter bearings. The effect of bearing size on the contamination factor  $e_C$  is incorporated in Annex A and Table 13 (Table V of this paper) of ISO 281:2007 Standard (Ref. 42). For pitch diameters ranging from 25 to 200 mm, the contamination (life) factors  $e_C$  increased with increasing pitch diameter or bearing size. That is, the larger the bearing the less effect of contamination on the life of the bearing.

The pitch diameters for the bearing tests reported by Gabelli, Morales-Espejel, and Ioannides (Ref. 20) were between 25 and 200 mm. Their data did not show a statistical relation between the contamination levels and bearing size. They explained, “the range of bearing sizes limited the range for comparison with bearing size.” However, the sizes and types of bearings were reasonably representative of those used in most rotating machinery applications.

A comparison of the data in Table II, for the roller bearings from Bhachu et al. (Refs. 12 and 13), and those data for ball bearings in Table III from Loewenthal et al. (Refs. 9 to 11), suggest that the roller bearing lives are more sensitive to changes in contamination level than those of the ball bearing. This is reflected in the Needelman-Zaretsky contamination life factors in Table V.

## Summary of Results

In 1991, Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life (adjustment) factors ( $LF$ ) as a function of oil filter ratings ( $FR$ ). These equations for life factors were incorporated into the reference book, STLE Life Factors for Rolling Bearings (Ref. 32). These equations were normalized ( $LF = 1$ ) to a  $10 \mu\text{m}$  filter rating at  $\beta_x = 200$  (normal cleanliness) as it was then defined and  $\Lambda$  of 1.1. Over the past 20 years, these life factors based on oil filtration have been used in conjunction with ANSI/ABMA standards and bearing computer codes to predict rolling bearing life. Also, additional experimental studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and types of particle contamination. During this time period, filter ratings have also been revised and improved and are now based on particle counts calibrated to a NIST standard reference material in the ISO 11171:1999 Standard (Ref. 38). It was the objective of the work reported herein to (1) review methods and data for determining the effects of lubrication oil particle size for calculating the bearing  $L_{10}$  or catalog life, (2) experimentally correlate older and obsolete filter ratings and the new ISO filter ratings, (3) recalibrate the Needelman-Zaretsky equations for



determining effect of oil filtration on rolling-element bearing life to the new filter ratings, (4) relate the new filter ratings to contamination levels listed in the ISO 281:2007 Standard, and (5) compare recalibrated filter life adjustment factors to those cleanliness ratings presented in ISO 281:2007 (Ref. 42). The following results were obtained:

1. Using two transformations, old obsolete filter ratings can be converted to new ISO filter ratings and vice versa. Approximate equations relating the old  $\beta_x = 200$  *FR* values with the new  $\beta_{x(c)} = 200$  and  $\beta_{x(c)} = 1000$  *FR* values are

$$\begin{aligned} \text{For new } \beta_{x(c)} = 200 \text{ filter rating; } FR_{\beta_{x(c)}200} &= 0.722 (FR_{\beta_x200}) + 2.97 \\ \text{For new } \beta_{x(c)} = 1000 \text{ filter rating; } FR_{\beta_{x(c)}1000} &= 0.848 (FR_{\beta_x200}) + 4.14 \end{aligned}$$

2. Two separate sets of life factors (*LF*) based on lubricant cleanliness for roller bearings and ball bearings, respectively, were derived based on the new  $\beta_{x(c)} = 200$  and  $\beta_{x(c)} = 1000$  ISO filter ratings. These *LF* values are normalized to *FR* values of 10  $\mu\text{m}$  at  $\beta_{x(c)} = 200$  and 13  $\mu\text{m}$  at  $\beta_{x(c)} = 1000$  and are independent of  $\Lambda$  and/or  $\kappa$ , loading conditions, and bearing size. These are:

$$\begin{aligned} \text{For roller bearings and new } \beta_{x(c)} = 200 \text{ filter rating; } LF &\approx 3.5 [1.39(FR_{\beta_{x(c)}200}) - 4.11]^{-0.55} \\ \text{For roller bearings and new } \beta_{x(c)} = 1000 \text{ filter rating; } LF &\approx 3.5 [1.18(FR_{\beta_{x(c)}1000}) - 4.88]^{-0.55} \\ \text{For ball bearings and new } \beta_{x(c)} = 200 \text{ filter rating; } LF &\approx 1.8 [1.39(FR_{\beta_{x(c)}200}) - 4.11]^{-0.25} \\ \text{For ball bearings and new } \beta_{x(c)} = 1000 \text{ filter rating; } LF &\approx 1.8 [1.18(FR_{\beta_{x(c)}1000}) - 4.88]^{-0.25} \end{aligned}$$

3. ISO 281:2007 Standard provides a reasonable qualitative estimate of the effect of particle damage on rolling-element bearing fatigue life for filtration ratings ranging from  $\leq 4 \mu\text{m}$  at  $\beta_{x(c)} = 1000$  (extreme cleanliness) up to  $\beta_{34(c)} = 1000$  (typical contamination). At conditions of severe contamination and above, where the filter ratings are  $\geq 35 \mu\text{m}$  at  $\beta_{x(c)} = 1000$ , the ISO 281:2007 Standard correlated with test results obtained with common hard steel contamination that can be found in industrial gearboxes.

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