NASA/TM—2009-215285

Interference-Fit Life Factors for Roller Bearings

Fred B. Oswald and Erwin V. Zaretsky
Glenn Research Center, Cleveland, Ohio

Joseph V. Poplawski
J.V. Poplawski & Associates, Bethlehem, Pennsylvania

November 2009

NASA STI Program . . . in Profile

Since its founding, NASA has been dedicated to the advancement of aeronautics and space science. The NASA Scientific and Technical Information (STI) program plays a key part in helping NASA maintain this important role.

The NASA STI Program operates under the auspices of the Agency Chief Information Officer. It collects, organizes, provides for archiving, and disseminates NASA's STI. The NASA STI program provides access to the NASA Aeronautics and Space Database and its public interface, the NASA Technical Reports Server, thus providing one of the largest collections of aeronautical and space science STI in the world. Results are published in both non-NASA channels and by NASA in the NASA STI Report Series, which includes the following report types:

- TECHNICAL PUBLICATION. Reports of completed research or a major significant phase of research that present the results of NASA programs and include extensive data or theoretical analysis. Includes compilations of significant scientific and technical data and information deemed to be of continuing reference value. NASA counterpart of peer-reviewed formal professional papers but has less stringent limitations on manuscript length and extent of graphic presentations.

- TECHNICAL MEMORANDUM. Scientific and technical findings that are preliminary or of specialized interest, e.g., quick release reports, working papers, and bibliographies that contain minimal annotation. Does not contain extensive analysis.

- CONTRACTOR REPORT. Scientific and technical findings by NASA-sponsored contractors and grantees.

- CONFERENCE PUBLICATION. Collected papers from scientific and technical conferences, symposia, seminars, or other meetings sponsored or cosponsored by NASA.

- SPECIAL PUBLICATION. Scientific, technical, or historical information from NASA programs, projects, and missions, often concerned with subjects having substantial public interest.

- TECHNICAL TRANSLATION. English-language translations of foreign scientific and technical material pertinent to NASA's mission.

Specialized services also include creating custom thesauri, building customized databases, organizing and publishing research results.

For more information about the NASA STI program, see the following:

- Access the NASA STI program home page at *http://www.sti.nasa.gov*

- E-mail your question via the Internet to *help@sti.nasa.gov*

- Fax your question to the NASA STI Help Desk at 443–757–5803

- Telephone the NASA STI Help Desk at 443–757–5802

- Write to:
  NASA Center for AeroSpace Information (CASI)
  7115 Standard Drive
  Hanover, MD 21076–1320

NASA/TM—2009-215285

Interference-Fit Life Factors for Roller Bearings

Fred B. Oswald and Erwin V. Zaretsky
Glenn Research Center, Cleveland, Ohio

Joseph V. Poplawski
J.V. Poplawski & Associates, Bethlehem, Pennsylvania

Prepared for the
63rd STLE Annual Meeting & Exhibition
sponsored by the Society of Tribologists and Lubrication Engineers
Cleveland, Ohio, May 18–22, 2008

National Aeronautics and
Space Administration

Glenn Research Center
Cleveland, Ohio 44135

November 2009

Level of Review: This material has been technically reviewed by technical management.

Available from

NASA Center for Aerospace Information
7115 Standard Drive
Hanover, MD 21076–1320

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161

Available electronically at http://gltrs.grc.nasa.gov

Interference-Fit Life Factors for Roller Bearings

Fred B. Oswald and Erwin V. Zaretsky
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

Joseph V. Poplawski
J.V. Poplawski & Associates
Bethlehem, Pennsylvania 18018

Abstract

The effect of hoop stresses in reducing cylindrical roller bearing fatigue life was determined for various classes of inner-ring interference fit. Calculations were performed for up to 7 fit classes for each of 10 bearing sizes. The hoop stresses were superimposed on the Hertzian principal stresses created by the applied radial load to calculate roller bearing fatigue life. A method was developed through a series of equations to calculate the life reduction for cylindrical roller bearings. All calculated lives are for zero initial internal clearance. Any reduction in bearing clearance due to interference fit would be compensated by increasing the initial (unmounted) clearance. Results are presented as tables and charts of life factors for bearings with light, moderate, and heavy loads and interference fits ranging from extremely light to extremely heavy for bearing accuracy class RBEC−5 (ISO class 5). Interference fits on the inner ring of a cylindrical roller bearing can significantly reduce bearing fatigue life. In general, life factors are smaller (lower life) for bearings running under light load where the unfactored life is highest. The various bearing series within a particular bore size had almost identical interference-fit life factors for a particular fit. The tightest fit at the high end of the tolerance band produces a life factor of approximately 0.40 for an inner-race maximum Hertz stress of 1200 MPa (175 ksi) and a life factor of 0.60 for an inner-race maximum Hertz stress of 2200 MPa (320 ksi). Interference fits also impact the maximum Hertz stress-life relation.

Introduction

Rolling-element bearings often utilize a tight interference fit between the bearing inner ring and shaft or between the outer ring and housing bore to prevent fretting damage at the interface. American National Standards Institute/American Bearing Manufacturers Association (ANSI/ABMA) standards (Refs. 1 and 2) as well as catalogs of bearing manufacturers specify suggested fits for various operating conditions. Any fit must be based on the most severe operating conditions expected, including highest speeds and highest vibration levels.

A tight fit of the bearing inner ring over the shaft reduces internal bearing clearance. The clearance change can be compensated for by adding initial internal clearance to the bearing. However, the interference fit of the inner ring over the shaft also adds a hoop stress on the bearing inner ring. Czyzewski (Ref. 3) showed that the hoop stresses are generally tensile (designated by a plus (+) sign) and can negatively affect fatigue life.

Coe and Zaretsky (Ref. 4) analyzed the effect of hoop stresses on rolling-element fatigue. Their work was based on the analysis of Hertzian principal stresses from Jones (Ref. 5) and the Lundberg-Palmgren bearing life theory (Lundberg and Palmgren (Ref. 6)). Coe and Zaretsky (Ref. 4) superimposed the hoop stresses on the Hertzian principal stresses whereby the shearing stresses in the stressed volume of the contact between the rolling element (ball or roller) and inner race of the bearing are increased. The increased maximum shearing stress at a depth below the contacting surface due to hoop stress is

$$\left(\tau_{\max}\right)_h = \tau_{\max} + \frac{\sigma_h}{2} \tag{1}$$

where τ_{max} is the maximum shearing stress, $(\tau_{max})_h$ is the maximum shearing stress including the effect of the hoop stress, and σ_h is the hoop stress.

Coe and Zaretsky (Ref. 4) applied Equation (1) to modify the Lundberg-Palmgren life equation as follows:

$$L_{10} = \left[\frac{\tau_{\max}}{\left(\tau_{\max}\right)_h}\right]^{c/e} \left[\frac{C_D}{P_{eq}}\right]^p \tag{2}$$

where c is a stress-life exponent, e is the Weibull slope or modulus, C_D is the bearing dynamic load capacity, P_{eq} is the equivalent bearing load, and p is a load-life exponent.

The Coe-Zaretsky analysis assumed for simplicity that all components (inner race, rollers, and outer race) are affected equally by the inner-race interference fit. This assumption results in a conservative prediction of bearing life. A more rigorous analysis should apply the life reduction due to hoop stress to only the inner race without modifying the lives of the outer race and rolling-element set.

Subsequent to the Coe-Zaretsky analysis, Zaretsky (Ref. 7) and (Zaretsky, Poplawski, and Root (Ref. 8)) developed a procedure (Zaretsky's Rule) for separating the lives of bearing races from the lives of the rolling elements (considered as a set). This procedure allows for calculating the reduction of bearing life from an inner-ring interference fit without affecting the rolling-element set and outer race lives, thus providing a more accurate bearing life analysis.

In view of the aforementioned, the objectives of this work were to (a) independently determine the lives of the inner races, outer races, and roller sets for several classes of radially loaded, cylindrical roller bearings subject to inner-ring interference fit; (b) calculate the reduction in cylindrical roller bearing fatigue life due to interference fit of the inner ring; and (c) develop life factors applied to the bearing life calculation for interference fits according to the ANSI/ABMA standards for shaft-fitting practice.

Nomenclature

b	semiwidth of Hertzian contact area, m (in.)
C_D	dynamic load capacity, N (lbf)
C_0	static load capacity, N (lbf)
c	stress-life exponent
D	diameter, mm (in.)
E	elastic (Young's) modulus, MPa (psi)
e	Weibull slope
k	conversion constant in Equation (17)
L	life, millions of inner-race revolutions or h
L_{10}	10-percent life or life at which 90 percent of a population survives, millions of inner-race revolutions or h
$(LF)_h$	life factor for hoop stress
LR	life ratio defined in Equation (13)

P	radial load on bearing, N (lbf)
$\mathcal{P}_i$	pressure between shaft and inner race due to interference fit, MPa (psi)
p	load-life exponent
R	radius of curvature, mm (in.)
S	stress, MPa (ksi)
$S_{\max_{IR}}$	maximum Hertz stress on inner race, MPa (ksi)
$S_{\max_{OR}}$	maximum Hertz stress on outer race, MPa (ksi)
S_t'	tangential stress including hoop stress superimposed on Hertz stress, MPa (ksi)
u	dimensionless depth below surface to maximum shear stress (= z/b)
z	distance below surface to maximum shear stress due to Hertzian load, m (in.)
Δ	diametral interference, mm (in.)
ν	Poisson's ratio
σ	stress, MPa (ksi)
τ	shear stress, MPa (psi)
$(\tau_{\max})_h$	maximum shear stress modified by hoop stress, MPa (psi)

Subscripts

adj	adjusted life
eq	equivalent
h	hoop stress (in tangential or *x*-direction)
IR,OR	inner or outer race of bearing
n or *z*	normal direction
R	roller set
S	shaft and inner-ring bore
t or *x*	tangential direction

Enabling Equations

Subsurface Shearing Stresses

A representative cylindrical roller bearing is shown in Figure 1. The bearing comprises an inner and outer ring and plurality of rollers interspersed between the two rings and positioned by a cage or separator.

Figure 2(a) is a schematic of the contact of a cylindrical roller on a race. Figure 2(b) shows the principal stresses at and beneath the surface. From these principal stresses the shearing stresses can be calculated. Four shearing stresses can be applied to bearing life analysis: the orthogonal, the octahedral, the von Mises, and the maximum. For the analysis reported herein, only the maximum shearing stresses are considered.

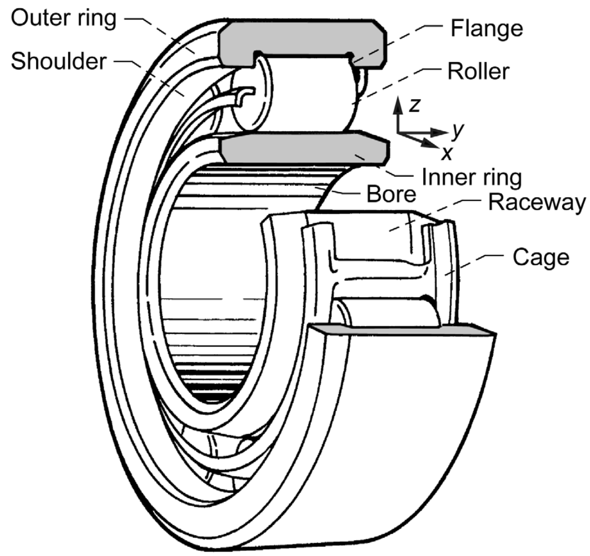

Figure 1.—Cylindrical roller bearing.

The maximum shearing stress is one-half the maximum difference between principal stresses:

$$\tau_{\max} = \frac{\sigma_z - \sigma_x}{2} \tag{3}$$

Coe and Zaretsky (Ref. 4) showed that the subsurface shear stress in a cylindrical roller bearing due to Hertzian loading as a function of z, the depth below the surface, can be expressed in terms of the maximum Hertz stress $S_{\max}$ and the non-dimensional depth below the surface u:

$$\tau = S_{\max}\left(\sqrt{1+u^2} - u - \frac{1}{\sqrt{1+u^2}}\right) \tag{4}$$

where $u = z/b$ and b is the semiwidth of the Hertzian contact area (Fig. 2(a)). By setting the derivative of Equation (4) with respect to u equal to zero and solving by iteration, the maximum shearing stress from Hertzian loading occurs at $u = 0.786152$. Substituting u into Equation (4) gives the maximum shear stress due to Hertzian loading:

$$\tau_{\max} = -0.30028\ S_{\max} \tag{5}$$

Coe and Zaretsky (Ref. 4) considered effects due to the hoop stress from an interference fit of the inner ring on the shaft and the effects due to rotation superposed on the Hertz stress. Their analysis showed that the additional effects cause little change in the location of the maximum shear stress on the inner-race surface; the variation in u due to the added hoop stress is only 0.074 percent.

The principal stresses in the tangential direction and the effect of the added hoop stress are illustrated in Figure 3 where S_n is the normal stress, S_t is the tangential stress due only to Hertzian loading, and S_t' is the tangential stress including hoop stress superimposed on S_t. The maximum shear stress is one-half the difference between S_n and S_t'.

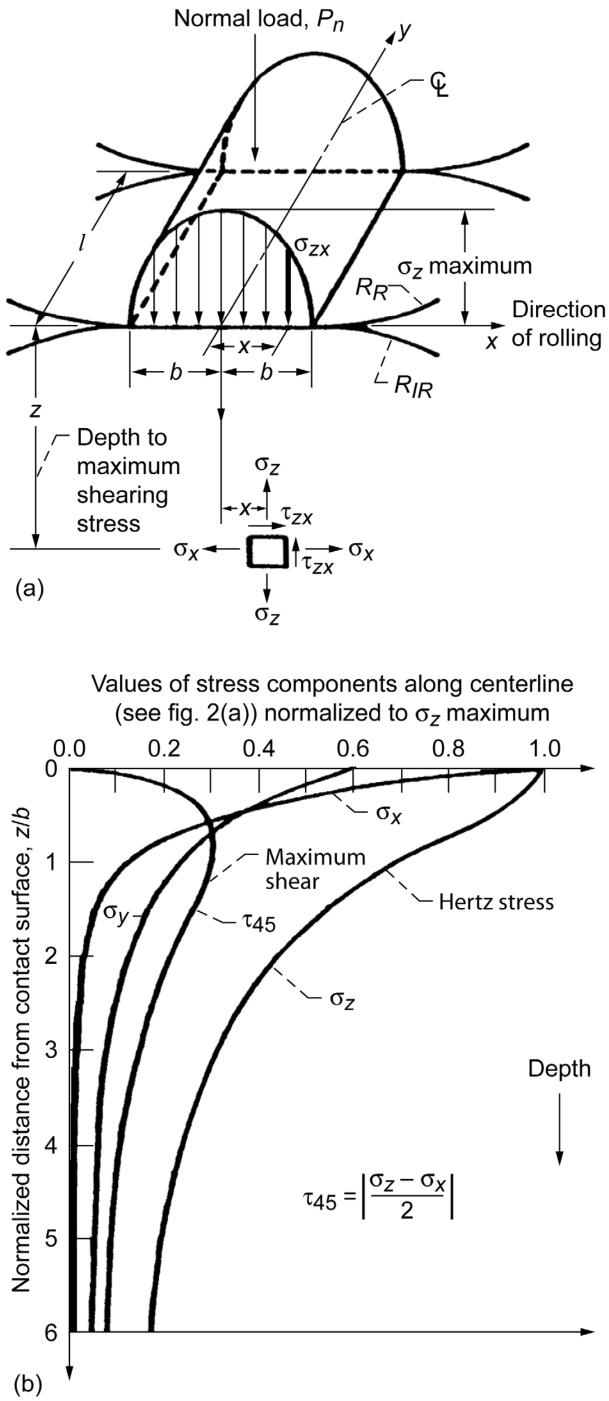

Figure 2.—Subsurface stress field under line contact. (a) Hertz stress distribution for roller on raceway showing principal stresses at depth *z* below surface. (b) Distribution of principal and shearing stress as function of depth *z* below surface.

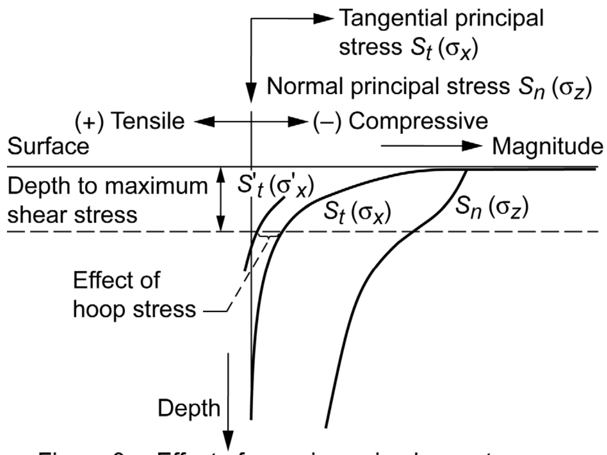

Figure 3.—Effect of superimposing hoop stress on tangential principal stress in direction of rolling under Hertzian contact.

Zaretsky (Ref. 7) gives a simplified procedure for finding the effect due to hoop stress from inner-ring interference fits and inner-ring rotation, assuming the value u = 0.78667. This simplified procedure, when used to calculate the resulting life of a roller bearing, gives results within 1 percent of the value found by iterating for the actual location of the maximum shear stress, even with a very heavy fit.

Zaretsky's procedure requires the contact pressure $\mathcal{P}_i$ between the inner race and the shaft. For the case where both components have the same material properties, $\mathcal{P}_i$ is given as p in Equation (6) below from Juvinall (Ref. 9), where E is Young's modulus, δ is the radial interference, a is the inside radius of the shaft, b is the radius of the inner ring to shaft interface, and c is the outside radius of the inner ring:

$$p = \frac{E\delta\left(b^2 - a^2\right)\left(c^2 - b^2\right)}{2b^3\left(c^2 - a^2\right)} \tag{6}$$

For a bearing race shrunk on a solid shaft, dimensions $\delta = \Delta/2$, $a = 0$, $b = D_S/2$, $c = D_{IR}/2$, and Equation (6) becomes

$$\mathcal{P}_i = \frac{E\Delta\left(D_{IR}^2 - D_S^2\right)}{2D_S D_{IR}^2} \tag{7}$$

Strict Series Reliability and Zaretsky's Rule

Lundberg and Palmgren (Ref. 6) first derived the relationship between individual rolling-element bearing component life and system life. A bearing is a system of multiple components, each with a different life. As a result, the life of the system is different from the life of an individual component in the system. The fatigue lives of each of the bearing components are combined to calculate the system L_{10} life using strict-series system reliability (Ref. 6) and the two-parameter Weibull distribution function (10–12) for the bearing components comprising the system. Lundberg and Palmgren (Ref. 6) expressed the bearing system fatigue life as follows:

$$\frac{1}{L^e} = \frac{1}{L_{IR}^e} + \frac{1}{L_{OR}^e} \tag{8}$$

Zaretsky, Poplawski, and Root (Ref. 8) note that the life of the rolling element set is implicitly included in the inner and outer race lives above. If the life of the rollers L_R (taken as a set) is separated from the race lives, Equation (8) can be rewritten as

$$\frac{1}{L^e} = \frac{1}{L^e_{IR-adj}} + \frac{1}{L^e_R} + \frac{1}{L^e_{OR-adj}} \tag{9}$$

where *adj* in the subscript indicates adjusted lives for the races that will be greater than the corresponding lives in Equation (8). Zaretsky (Ref. 7) and Zaretsky, Poplawski, and Root (Ref. 8) observe that the life of the outer race L_{OR} is generally greater than the life of the inner race L_{IR} and the life of the roller set L_R is equal to or greater than the life of the outer race. In this report, we assume that $L_R = L_{OR}$; thus, Equation (9) becomes

$$\frac{1}{L^e} = \frac{1}{L^e_{IR-adj}} + \frac{2}{L^e_{OR-adj}} \tag{10}$$

We define the ratio of lives between outer and inner races as

$$X = \frac{L_{OR}}{L_{IR}} \tag{11}$$

If we assume that the life ratio X does not change when the roller life is separated from the race lives, Equation (10) becomes

$$\frac{1}{L^e} = \frac{1}{L^e_{IR-adj}} + \frac{2}{\left(X \cdot L_{IR-adj}\right)^e} \tag{12}$$

Bearing Life Factor for Interference Fit

Coe and Zaretsky (Ref. 4) show that the life ratio for hoop stress LR is the ninth power of the ratio of maximum shear stress τ_{max} from Hertz loading alone (from Eq. (5)) to the maximum shearing stress including both Hertizan loading and hoop stress $(\tau_{max})_h$, which can be computed from the simplified procedure of Zaretsky (Ref. 7):

$$LR = \frac{(L)_h}{L} = \left[\frac{\tau_{max}}{(\tau_{max})_h}\right]^9 \tag{13}$$

Equation (13) is based on earlier work by Lundberg and Palmgren (Ref. 6) that uses life exponents for shear-stress that range from 6.9 to 9.3. An exponent of 9 is assumed for the current work. For further discussion of life exponents, see Poplawski, Peters, and Zaretsky (Ref. 13).

Coe and Zaretsky (Ref. 4) applied the life ratio to the life of the entire bearing, which produces an overly conservative estimate for the life of the bearing. Herein, the life ratio will be applied only to the inner race. This new value for the inner-race life $LR \cdot L_{IR-adj}$ is used in the first term on the right-hand side of Equation (12) to calculate the life of the bearing $(L)_h$, including the effects of Hertzian loading and hoop stress:

$$\frac{1}{(L)_h^e} = \frac{1}{\left(LR \cdot L_{IR-adj}\right)^e} + \frac{2}{\left(X \cdot L_{IR-adj}\right)^e} \tag{14}$$

Finally, $(LF)_h$, the life factor for hoop stress, is computed as the ratio of the adjusted life of the bearing $(L)_h$ divided by the original life of the bearing L:

$$(LF)_h = \frac{(L)_h}{L} \tag{15}$$

Determining Life Factor Based on Load and Fit

For most low-speed roller bearing applications (less than 1 million DN, where DN is the inner-ring speed in rpm multiplied by the bearing bore diameter in millimeters), the determination of the appropriate life factor based on roller bearing size, radial load, and interference fit can be related to the bearing static load capacity C_0 without the need to perform extensive calculations.

Rolling element bearing static load capacity was first defined in terms of deformation by Palmgren (Ref. 14): "… the allowable permanent deformation of rolling element and bearing ring (race) at a contact as 0.0001 times the diameter of the rolling element…" For roller bearings, this corresponds to a maximum Hertz stress of 4000 MPa (580 ksi). From Hertz theory Jones (Ref. 5) shows that the relationship between maximum Hertz stress and radial load P for a cylindrical roller bearing is

$$S_{\max} \sim \sqrt{P} \tag{16}$$

Nearly all bearing manufacturers' catalogs provide the static load capacity C_0 for any bearing size. Hence, to determine the appropriate stress at the applied radial load on the roller bearing inner race, Equation (16) can be rewritten as follows:

$$S_{\max} = k\sqrt{\frac{P}{C_0}} \tag{17}$$

where the conversion constant $k = 4000$ for SI units with $S_{\max}$ expressed in megapascals or $k = 580$ for English traditional units with $S_{\max}$ expressed in kips per square inch. Table I (using data from The Timken Company (Ref. 15)) gives the static load capacity for the bearings discussed herein.

TABLE I.—CYLINDRICAL ROLLER BEARING PROPERTIES
[From The Timken Company (Ref. 15).]

ABMA number	Bore, mm	Outside diameter, mm	Number of rollers	Roller diameter, mm	Roller length, mm	Inner-race outside diameter, mm (in)	Static load capacity, C_0, kN (lbf)
1906	30	47	18	5	5	33.75 (1.3287)	14.737 (3 313)
0206		62	12	10	10	37.02 (1.4575)	37.632 (8 460)
1910	50	72	20	7	7	54.00 (2.126)	32.881 (7 392)
1010		80	18	9	9	56.45 (2.2224)	49.446 (11 116)
0210		90	14	13	13	57.65 (2.2697)	77.226 (17 361)
0310		110	10	19	19	61.95 (2.439)	111.010 (24 956)
1915	75	105	24	9	9	81.00 (3.189)	68.796 (15 466)
0215		130	14	18	18	84.50 (3.3268)	149.420 (33 591)
1920	100	140	24	12	12	108.00 (4.252)	123.278 (27 714)
0220		180	14	25	25	115.00 (4.5275)	295.980 (66 539)

The maximum Hertz stress S_{max} value as a function of the applied radial load and static load capacity C_0 is plotted in Figure 4. The appropriate life factor can be interpolated from Tables II to V for the various interference fits.

As an example of using this procedure, consider a 0210-size bearing with a radial load P = 6.95 kN (1562 lbf). From Table I, the static load capacity C_0 = 77.226 kN (17 361 lbf). Using either Figure 4, with P/C_0 = 0.09, or Equation (17), with $\sqrt{P/C_0}$ = 0.300, and the appropriate value for k yields S_{max} = 1200 MPa or 175 ksi.

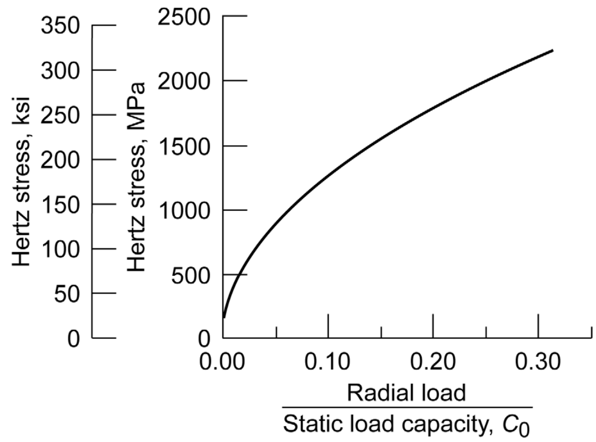

Figure 4.—Relationship between roller bearing maximum Hertz stress and P/C_0 (radial load P divided by static load capacity C_0).

TABLE II.—LIFE FACTORS FOR 30-mm-BORE CYLINDRICAL ROLLER BEARING WITH RBEC–5 TOLERANCES

[Results averaged from ABMA 1906 and 0206 bearings.]

ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor	ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor
j5-min	0.004	1200 (175) 1700 (250) 2200 (320)	1 1 1	k5-min	–0.002	1200 (175) 1700 (250) 2200 (320)	1 1 1
j5	–0.0035	1200 (175) 1700 (250) 2200 (320)	1 1 1	k5	–0.0095	1200 (175) 1700 (250) 2200 (320)	0.77 .83 .87
j5-max	–0.011	1200 (175) 1700 (250) 2200 (320)	0.72 .79 .84	k5-max	–0.017	1200 (175) 1700 (250) 2200 (320)	0.54 .65 .71
j6-min	0.004	1200 (175) 1700 (250) 2200 (320)	1 1 1	m5-min	–0.008	1200 (175) 1700 (250) 2200 (320)	0.83 .88 .90
j6	–0.0055	1200 (175) 1700 (250) 2200 (320)	0.93 .95 .96	m5	–0.0155	1200 (175) 1700 (250) 2200 (320)	0.58 .68 .74
j6-max	–0.015	1200 (175) 1700 (250) 2200 (320)	0.59 .69 .75	m5-max	–0.023	1200 (175) 1700 (250) 2200 (320)	0.40 .52 .61

TABLE III.—LIFE FACTORS FOR 50-mm-BORE CYLINDRICAL ROLLER BEARING WITH RBEC–5 TOLERANCES
[Results averaged from ABMA 1910, 1010, 0210, 0310 bearings.]

ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor
j5-min	0.005	1200 (175) 1700 (250) 2200 (320)	1 1 1
j5	–0.0045	1200 (175) 1700 (250) 2200 (320)	0.99 .99 .99
j5-max	–0.014	1200 (175) 1700 (250) 2200 (320)	0.74 .81 .85
j6-min	0.005	1200 (175) 1700 (250) 2200 (320)	1 1 1
j6	–0.007	1200 (175) 1700 (250) 2200 (320)	0.92 .94 .95
j6-max	–0.019	1200 (175) 1700 (250) 2200 (320)	0.64 .73 .78
k5-min	–0.002	1200 (175) 1700 (250) 2200 (320)	1 1 1
k5	–0.0115	1200 (175) 1700 (250) 2200 (320)	0.80 .85 .89
k5-max	–0.021	1200 (175) 1700 (250) 2200 (320)	0.60 .70 .76
m5-min	–0.009	1200 (175) 1700 (250) 2200 (320)	0.86 .90 .92
m5	–0.0185	1200 (175) 1700 (250) 2200 (320)	0.65 .74 .79
m5-max	–0.028	1200 (175) 1700 (250) 2200 (320)	0.48 .60 .67
m6-min	–.009	1200 (175) 1700 (250) 2200 (320)	0.86 .90 .92
m6	–0.021	1200 (175) 1700 (250) 2200 (320)	0.60 .70 .76
m6-max	–0.033	1200 (175) 1700 (250) 2200 (320)	0.41 .53 .62

TABLE IV.—LIFE FACTORS FOR 75-mm-BORE CYLINDRICAL ROLLER BEARING WITH RBEC–5 TOLERANCES
[Results averaged from ABMA 1915 and 0215 bearings.]

ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor
j5-min	0.007	1200 (175) 1700 (250) 2200 (320)	1 1 1
j5	–0.004	1200 (175) 1700 (250) 2200 (320)	1 1 1
j5-max	–0.015	1200 (175) 1700 (250) 2200 (320)	0.80 .86 .89
j6-min	0.007	1200 (175) 1700 (250) 2200 (320)	1 1 1
j6	–0.007	1200 (175) 1700 (250) 2200 (320)	0.94 .96 .97
j6-max	–0.021	1200 (175) 1700 (250) 2200 (320)	0.71 .79 .83
k5-min	–0.002	1200 (175) 1700 (250) 2200 (320)	1 1 1
k5	–0.013	1200 (175) 1700 (250) 2200 (320)	0.84 .88 .91
k5-max	–0.024	1200 (175) 1700 (250) 2200 (320)	0.67 .75 .80
m5-min	–0.011	1200 (175) 1700 (250) 2200 (320)	0.87 .91 .93
m5	–0.0122	1200 (175) 1700 (250) 2200 (320)	0.70 .78 .82
m5-max	–0.033	1200 (175) 1700 (250) 2200 (320)	0.55 .66 .73
m6-min	–.011	1200 (175) 1700 (250) 2200 (320)	0.87 .91 .93
m6	–0.025	1200 (175) 1700 (250) 2200 (320)	0.65 .74 .80
m6-max	–0.039	1200 (175) 1700 (250) 2200 (320)	0.48 .60 .68
n6-min	–0.020	1200 (175) 1700 (250) 2200 (320)	0.73 .80 .84
n6	–0.034	1200 (175) 1700 (250) 2200 (320)	0.54 .65 .72
n6-max	–0.048	1200 (175) 1700 (250) 2200 (320)	0.40 .53 .61

TABLE V.—LIFE FACTORS FOR 100-mm-BORE CYLINDRICAL ROLLER BEARING WITH RBEC–5 TOLERANCES
[Results averaged from ABMA 1920 and 0220 bearings.]

ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor	ABMA fit class	Clearance, mm	Inner-race Hertz stress, MPa (ksi)	Life factor
j5-min	+0.009	1200 (175)	1	m5-max	–0.038	1200 (175)	0.60
		1700 (250)	1			1700 (250)	.70
		2200 (320)	1			2200 (320)	.76
j5	–0.0035	1200 (175)	1	m6-min	–.013	1200 (175)	0.88
		1700 (250)	1			1700 (250)	.91
		2200 (320)	1			2200 (320)	.93
j5-max	–0.016	1200 (175)	0.84	m6	–0.029	1200 (175)	0.69
		1700 (250)	.88			1700 (250)	.77
		2200 (320)	.91			2200 (320)	.82
j6-min	+0.009	1200 (175)	1	m6-max	–0.045	1200 (175)	0.54
		1700 (250)	1			1700 (250)	.65
		2200 (320)	1			2200 (320)	.72
j6	–0.007	1200 (175)	0.96	n6-min	–0.023	1200 (175)	0.76
		1700 (250)	.97			1700 (250)	.82
		2200 (320)	.98			2200 (320)	.86
j6-max	–0.023	1200 (175)	0.76	n6	–0.039	1200 (175)	0.59
		1700 (250)	.82			1700 (250)	.69
		2200 (320)	.86			2200 (320)	.75
k5-min	–0.003	1200 (175)	1	n6-max	–0.055	1200 (175)	0.46
		1700 (250)	1			1700 (250)	.58
		2200 (320)	1			2200 (320)	.66
k5	–0.0155	1200 (175)	0.85	P6-min	–0.037	1200 (175)	0.61
		1700 (250)	.89			1700 (250)	.71
		2200 (320)	.91			2200 (320)	.77
k5-max	–0.028	1200 (175)	0.70	p6	–0.053	1200 (175)	0.47
		1700 (250)	.78			1700 (250)	.59
		2200 (320)	.83			2200 (320)	.67
m5-min	–0.013	1200 (175)	0.88	p6-max	–0.069	1200 (175)	0.36
		1700 (250)	.91			1700 (250)	.49
		2200 (320)	.93			2200 (320)	.58
m5	–0.0255	1200 (175)	0.73				
		1700 (250)	.80				
		2200 (320)	.84				

Results and Discussion

We applied the analysis described in the previous section to radially loaded cylindrical roller bearings from four bore sizes at either two or four dimension series for each bore size. Each bearing was analyzed at three levels of inner-ring Hertz stress. The dimension series are shown schematically in Figure 5. All bearings are made from AISI 52100 steel and have a "square" cross section, with the roller length equal to the diameter. Properties for the bearings analyzed are listed in Table I.

The calculations were repeated for up to seven fit classes for each bearing, with each fit taken at the tightest, average, and loosest values within the fit class for RBEC–5 tolerance. This required 486 separate analyses. A graphical representation of shaft fits is shown schematically in Figure 6, which was adapted from ABMA (Ref. 1).

Harris (Ref. 16) discusses the effect of surface finish on interference fit as a result of the smoothing of asperities on the surface. He recommends reducing the calculated interference to account for asperity smoothing, depending on the quality of the finish. For very accurately ground surfaces, the recommended reduction is 4 µm (2 µm for each surface, bore I.D. and shaft O.D.). In this work, we have reduced the apparent interference by 4 µm (160 µin.) to account for surface finish effects.

A commercial bearing analysis code (Ref. 17) was used to calculate the unfactored L_{10} lives for the inner and outer races operating without interference fit. The rollers were modeled with an aerospace crown (Poplawski, Peters and Zaretsky (Ref. 18)) chosen to minimize the effect of stress concentrations at

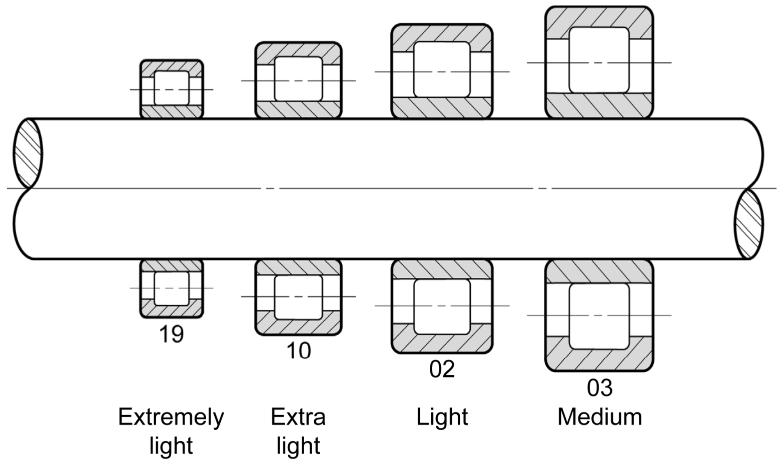

Figure 5.—ANSI/ABMA roller bearing dimension series (adapted from The Timken Company (Ref. 15)).

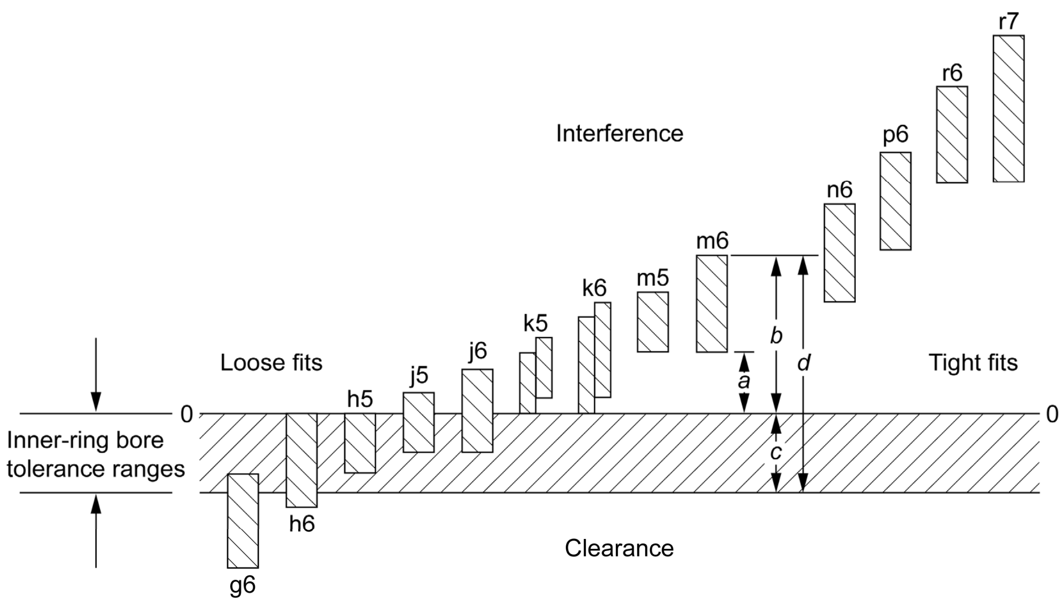

Figure 6.—Representation of ANSI/ABMA shaft fits (adapted from ABMA (Ref. 1)). Dimensions illustrate m6 fit, with loosest fit within tolerance band represented by dimension *a* and tightest fit by dimension *d*. Fit classes j5 to p6 were considered in this report.

the ends of the rollers. The crown has a flat length of 61 percent of the actual roller length and a crown radius approximately 100 times the roller length.

All bearings were modeled at the zero internal operating clearance condition. In the case of inner-ring interference fit, this means the bearings would have an appropriate initial (unmounted) clearance between the rollers and the races.

Each bearing was analyzed for three values of inner-race maximum Hertz stress: 1200, 1700, and 2200 MPa (175, 250, and 320 ksi). The radial load for each bearing was chosen to give the desired stress value. The analysis code estimated the inner and outer race lives using the traditional Lundberg-Palmgren method (Lundberg and Palmgren (Ref. 6)). Therefore, these lives implicitly include the life of the roller set.

Equation (11) was used to find the ratio of the inner and outer race lives for the bearing, and Equation (12) was used to calculate the adjusted value of the inner-race life (with the roller set life separated), assuming that X, the ratio of the race lives, does not change when the roller set life is separated from the race lives.

For this study, data for bearing bore sizes were taken from the table for tolerance class ABEC−5, RBEC−5 (ABMA (Ref. 2)). However, because this reference does not have information for shaft size limits, values for shaft diameter deviations were taken from the ANSI/ABMA shaft-fitting practice table (Ref. 1), which gives tolerance limits for ABEC−1, RBEC−1 quality level for bearing bore sizes from 3 to 1250 mm (0.1181 to 49.2126 in.). The practice of using bore tolerances from an ABEC−5 table and shaft tolerances from an ABEC−1 table is consistent with an example given by Harris (Ref. 16).

The shaft diameter was subtracted from the bore diameter and then 0.004 mm (160 µin.) was added to the difference to account for asperity smoothing. If the resulting fit was positive (indicating clearance), then the interface pressure and thus the hoop stress were assumed to be zero. It the fit was negative, the resulting interference (as a positive number) was used to calculate the interface pressure due to the chosen interference fit in Equation (7).

The simplified procedure described in the next section was used to find the maximum shearing stress $(\tau_{max})_h$, including the effect of Hertzian loading and the effect from hoop stress due to the interference fit on the inner ring. Equation (13) was used to calculate the life ratio (hence the revised life) for the inner race and, finally, the life of the entire bearing was calculated from Equation (14) using the reduced life of the inner race and the original lives of the rollers and outer race.

Analysis of 0210-Size Cylindrical Roller Bearing With m6 Fit

As an example of the methods presented in this report, consider a 0210-size cylindrical roller bearing carrying a moderate load of 6.95 kN (1562 lbf) and a middle-of-the-tolerance-band m6 inner-ring interference fit. From Equation (17), the inner-race maximum Hertz stress S_{max} is 1200 MPa (175 ksi). The analysis code gave lives of the outer and inner races of 14 240 and 2303 million inner-race revolutions. From Equation (8), assuming a Weibull slope of 1.125, the L_{10} life of the bearing is 2068 million revolutions:

$$\frac{1}{L^{1.125}}=\frac{1}{2303^{1.125}}+\frac{1}{14\,240^{1.125}}=\frac{1}{2068^{1.125}} \tag{18}$$

The ratio X of outer race life to inner-race life from Equation (11) is 14 240/2303 = 6.18. The adjusted life of the inner race was found using Equation (12):

$$\frac{1}{2068^{1.125}}=\frac{1}{L_{IR-adj}^{1.125}}+\frac{2}{\left(6.18L_{IR-adj}\right)^{1.125}} \tag{19}$$

The solution gives $L_{IR\text{-}adj}$ = 2535 million revolutions. (Although not needed for this calculation, the adjusted life of the outer race is 15 675 million revolutions.)

The ANSI/ABMA shaft-fitting practice table for ABEC−1, RBEC−1 bearings (ABMA (Ref. 1)) was used to find limiting diameters for a 50-mm (2-in.) shaft with an m6 fit. Shaft fits are illustrated schematically in Figure 6. This figure shows deviations from the nominal bearing bore and shaft diameter of 50.000 mm (1.9685 in.). The shaft deviation can range from 0.009 to 0.025 mm (350 to 1000 µin.) (shown as dimensions a and b in the figure). The bearing bore deviations were found in the table for tolerance class ABEC−5, RBEC−5 (ABMA (Ref. 2)). The bore deviation can range from 0.000 to -0.008 mm (0 to –300 µin.) (shown as 0 and dimension c in the figure).

The loosest m6 fit occurs when the largest bore bearing (50 mm) is mounted on the smallest shaft (50.009 mm or 1.9689 in.), producing a fit of 0.009 mm (350 μin.) tight (before adjusting for surface finish). This fit is illustrated as dimension *a* in Figure 6. The tightest fit is from the smallest bore (49.992 mm or 1.9682 in.) on the largest shaft (50.025 mm or 1.9695 in.), or 0.033 mm (1300 μin.) tight, shown as dimension *d* in Figure 6. The average of these extremes is 0.021 mm (830 μin.) tight. This interference fit was reduced by 0.004 mm (160 μin.) to account for asperity smoothing, assuming smooth-ground surfaces. The resulting middle-of-the-tolerance-band m6 fit is 0.017 mm (670 μin.) tight.

For our example bearing, D_S = 50 mm (1.9685 in.); D_{IR} = 57.65 mm (2.2697 in.); roller diameter D_R = 13 mm (0.5118 in.); S_{max} = 1200 MPa (175 ksi); E = 205 878 MPa (29.86×10^6 psi); Poisson's ratio ν = 0.3; and Δ = 0.017 mm (670 μin.). From Equation (7), the interference-fit contact pressure P_i = 8.67 MPa (1.257 ksi).

Next, the life factor of the inner ring due to the interference-fit stress was calculated by the following the simplified procedure (adapted from Zaretsky (Ref. 7)) to calculate the maximum shear stress including the effect of hoop stress.

(1) Determine the maximum shearing stress τ_{max} from Equation (5), where $\tau_{max} = -(0.3)S_{max}$ = –360 MPa (–52.2 ksi).
(2) Determine the geometry constant B, where $B = D_S/D_{IR}$ = 0.867303 (dimensionless).
(3) Determine m, where $m = P_iB^2/(1 - B^2)$ = 26.3269 MPa (3.818 ksi).
(4) Determine R', where $R' = D_{IR}/d$ = 4.4346 (dimensionless).
(5) Determine K_2, where $K_2 = E(R' + 1)/(4(1 - \nu^2)S_{max})$ = 256.151 (dimensionless).
(6) Assume the value for u to be 0.78667 (dimensionless).
(7) Determine y, where $y = 1 - u/K_2$ = 0.996929 (dimensionless).
(8) Substitute values for S_{max}, m, and y in Equation (20) below to calculate $(\tau_{max})_h$ = –386.489 MPa (–56.056 ksi):

$$(\tau_{max})_h = \tau_{max} - \frac{m}{y^2} \tag{20}$$

(9) Compute the life ratio for the inner ring from Equation (13): $LR = (-360/(-386.489))^9 = 0.5278$.

Using the life ratio from step 9 above in Equation (14), the life of the bearing, including the effect of the interference fit, was calculated to be 1205 million inner-race revolutions:

$$\frac{1}{(L)_h^{1.125}} = \frac{1}{(0.5278 \cdot 2535)^{1.125}} + \frac{2}{(6.18 \cdot 2535)^{1.125}} = \frac{1}{1205^{1.125}} \tag{21}$$

The life factor for hoop stress $(LF)_h$ is

$$(LF)_h = \frac{1205}{2068} = 0.58 \tag{22}$$

Therefore, the middle-of-the-tolerance-band m6 interference fit will reduce the life of this bearing by 42 percent.

Interference-Fit Life Factors for RBEC−5 Roller Bearings

The analysis described in the previous section was applied to 486 separate bearing configurations. It included four bore sizes, up to four dimension series (ranging from extremely light to medium), at three values of inner-ring Hertz stress, and up to seven inner-ring interference-fit classes. Each fit class was evaluated at the minimum, maximum, and average fit level for the RBEC−5 tolerance class.

The shaft interference table from ABMA (Ref. 1) (as illustrated in Fig. 6) shows fit classes ranging from g6 (loose) to r7 (heavy interference). There is no effect on life for the looser fits that produce no pressure at the bore and no values are given for the heavier fit classes for small bearings. Hence, for 30-mm bearings, we calculated life factors for only four fit classes (j5 to m5); for 50-mm bearings, five classes (j5 to m6); for 75-mm bearings, six classes (j5 to n6) and for 100-mm bearings, seven classes (j5 to p6).

All interference fits were adjusted for the effect of asperity smoothing (assuming accurately ground surfaces) by adding 0.004 mm (160 µin.) to the clearance between the shaft and inner ring. If the clearance value was negative (indicating interference), then the resulting pressure was calculated. The results are shown in Tables II to V.

The life factors found in this study range from 1.00 (no effect) where there is no interface pressure to a worst case of 0.36 (64-percent life reduction) for the tightest p6 fit on a 100-mm (220-size) bearing at 1200 MPa (175 ksi) maximum Hertz stress. As should be expected, tighter fits produce smaller life factors (i.e., shorter lives). In general, the life factor is smallest (greatest life reduction) for bearings running under a light load where the unfactored life is highest.

Figure 7 and Table III show the variation in life factor for 50-mm-bore bearings operating under three levels of Hertz stress at the five fit classes considered (j5 to m6). In the three lightest fits shown (j5 to k5), the minimum fit will produce no interface pressure; hence, the life factor is 1.00. For example, "k5-min" in the seventh row of Table III shows a "clearance" of –0.002 mm (–79 µin.). (This is actually a slight

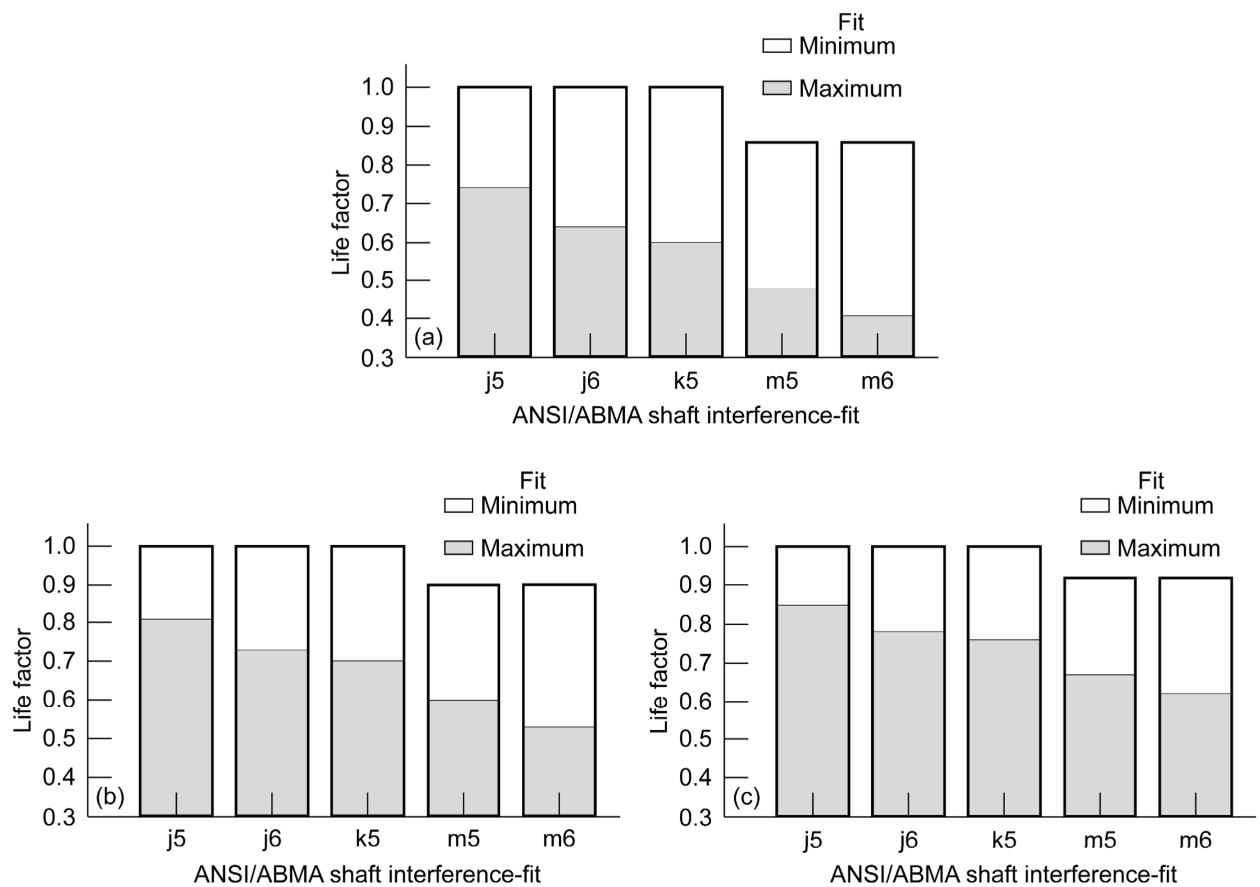

Figure 7.—Life factors for inner-ring interference-fit on 50-mm roller bearings at three maximum Hertz stress levels. Life factors shown at maximum and minimum fit for RBEC–5 tolerance levels. (a) 1200 MPa (175 ksi). (b) 1700 MPa (250 ksi). (c) 2200 MPa (320 ksi).

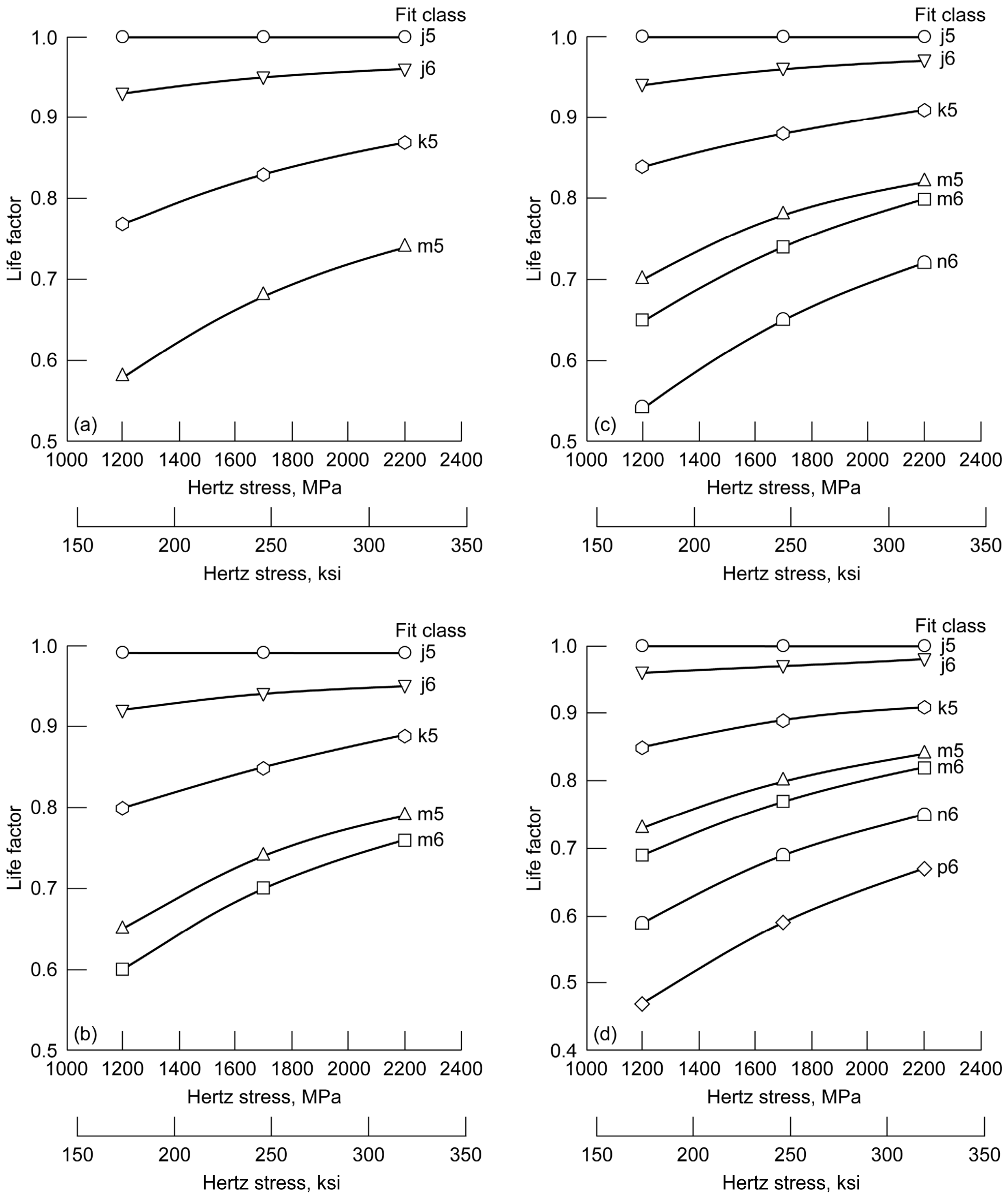

Figure 8.—Life factor versus inner-ring maximum Hertz stress in cylindrical roller bearings for several fit classes. Life factor computed for middle of RBEC–5 tolerance band for each fit. (a) 30–mm bearings. (b) 50–mm bearings. (c) 75–mm bearings. (d) 100–mm bearings.

interference.) However, after adjusting for asperity smoothing, it becomes a clearance of 0.002 mm. There is no interface pressure for a clearance; thus, $(LF)_h = 1.0$. In the bar chart of Figure 7(a), the nonshaded part of the third bar that represents this k5-min fit shows a *LF* of 1.0.

For the heavier fits, the life factor is less, ranging to a low value of 0.41 (59-percent life reduction) for "m6-max", an interference of 0.033 mm (1300 μin.). This fit produces an interface pressure of 14.79 MPa (2.15 ksi) (not shown in the table). For a bearing running at 1200 MPa (175 ksi) maximum Hertz stress, the resulting $(LF)_h = 0.41$. This value is represented by the shaded part of the right-most bar of Figure 7(a).

The various bearing dimension series within a particular bore size exhibited almost identical results for the interference-fit life factor for a particular fit, despite significant differences in the interface pressure at the bore required to produce that fit. For example, the four 50-mm bearings analyzed (nos. 1910, 1010, 0210, and 0310) for the average m6 fit of 0.021-mm (827 μin.) interference (before adjusting for finish effect) have fit pressures of 5.00, 7.55, 8.67, and 12.21 MPa (0.725, 1.10, 1.26, and 1.77 ksi), respectively. However, the resulting life factors are nearly identical (0.60, 0.60, 0.58, and 0.60). Therefore, we have combined these results, showing average life factor values for each bore size in Tables II to V.

Interestingly, at a given Hertz stress level, the tightest fits defined in the ANSI/ABMA tables produced very similar life factors, even on different bore sizes and for different fit designations. For example, at the 1200-MPa (175-ksi) maximum Hertz stress level, the tightest (m5) fit on a 30-mm-bore bearing has a life factor of 0.40 whereas the tightest (p6) fit on a 100-mm-bore bearing has a life factor of 0.36. Likewise, for the 2200-MPa (320 ksi) maximum Hertz stress, the m5 fit on a 30-mm-bore bearing has a life factor of 0.61 whereas the p6 fit on a 100-mm-bore bearing has a life factor of 0.58.

Figure 8 shows the middle of the tolerance band life factor for 30-, 50-, 75-, and 100-mm-bore cylindrical roller bearings at the three maximum Hertz stress levels. This plot can be used to estimate the life factor for Hertz stress levels between the values analyzed in this report. However, for conservative design, life factors should be chosen based on the tight end of the tolerance band rather than based on midband values.

Effect of Interference Fit on Stress-Life Exponent

Poplawski, Peters, and Zaretsky (Ref. 13) state that the theoretical relation between maximum Hertz stress and life in a roller bearing (with line contact) is an inverse eighth power:

$$L \sim \frac{1}{S^n{}_{\max}} \tag{23}$$

where L is bearing life, $S_{\max}$ is the maximum Hertz stress, and $n = 8$ is the stress-life exponent.

Interference fits can affect this maximum Hertz stress-life relation. The curves for maximum Hertz stress versus life are shown in Figure 9 for five interference fits on 210-size cylindrical roller bearings. For each interference fit, the Hertz stress-life exponent n was calculated. These values are also shown in Figure 9.

With no interference fit, our calculation resulted in a Hertz stress-life exponent $n = 8.1$, which is close to the expected value $n = 8$. However, with an interference fit at the middle of the m6 tolerance range, the exponent $n = 7.7$. If the results are recalculated based on the tight end of the tolerance range for the m6 interference fit (not shown in Fig. 9), the Hertz stress-life exponent becomes $n = 7.4$. Similar results were obtained for other bearing sizes.

This effect can impact the results of accelerated testing on bearings with a heavy interference fit. If such tests are performed at a high load (thus at high Hertz stress) and then the test results are extrapolated to lower stress levels using the usual stress-life exponent $n = 8$, the predicted value of life may be too high, thus giving a nonconservative design.

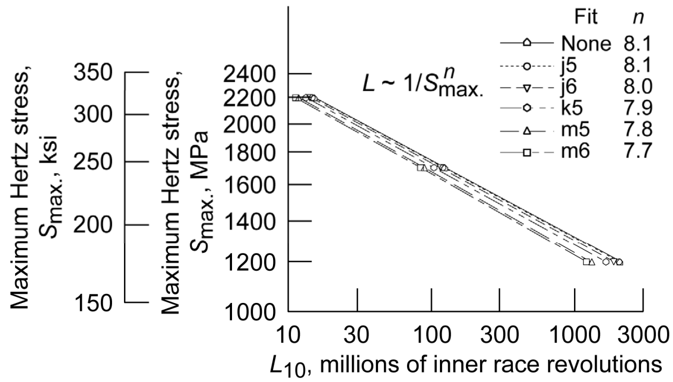

Figure 9.—Relationship between life and maximum Hertz stress for ABMA 0210 cylindrical roller bearings showing effect of interference fit on stress-life exponent *n*.

It is conjectured that the variation in the Hertz stress-life exponent with interference fit may help to explain the load-life exponent $p = 10/3$ reported and used by Lundberg and Palmgren (Ref. 19) for cylindrical roller bearings. Where $p = 10/3$, $n = 6.66$. Unfortunately, they did not report the interference fit that they used in their test bearings.

Summary of Results

The effect of hoop stresses in reducing roller bearing fatigue life was determined for various classes of inner-ring interference fit. Calculations were performed for up to seven interference-fit classes for each bearing series. Each fit was taken at the tightest, average, and loosest values within the fit class for RBEC−5 tolerances, thus requiring 486 separate analyses.

The hoop stresses were superimposed on the Hertzian principal stresses created by light, moderate, and heavy applied radial loads to determine roller bearing fatigue life. The results are presented as life factors for bearings loaded to maximum Hertz stress levels of 1200, 1700, and 2200 MPa (175, 250, and 320 ksi) in up to seven fit classes (extremely light to extremely heavy) and for bearing accuracy class RBEC−5 (ISO class 5). All calculations are for zero initial internal clearance conditions. Any reduction in internal bearing clearance due to the interference fit would be compensated by increasing the initial (unmounted) clearance.

The life factor for interference fit in low-speed roller bearings can be determined through charts or tables from the maximum Hertz stress, which is easily calculated from the applied radial load and the static load capacity. The following results were obtained.

(1) Interference fits on the inner bearing ring of a cylindrical roller bearing can significantly reduce bearing fatigue life. A heavy fit (tight end of m6 tolerance band) on a 210-size roller bearing reduced the fatigue life by 59, 47, and 38 percent from the standard life at maximum Hertz stresses of 1200, 1700, and 2200 MPa (175, 250, and 320 ksi), respectively.

(2) Tighter interference fits produced smaller life factors (i.e., shorter lives). Life factors due to hoop stresses found in this study ranged from 1.00 (no effect), where there is no interface pressure, to as low as 0.36 (64-percent life reduction) for the tightest p6 fit on a 100-mm bore (220-size) bearing with a 1200-MPa (175-ksi) maximum Hertz stress.

(3) The various bearing series within a particular bore size had almost identical interference-fit life factors for a particular fit, despite significant differences in the interface pressure at the bore. Four series (1910-, 110-, 210- and 310-size) of 50-mm-bore cylindrical roller bearings having an average m6 fit of 0.021 mm (827 µin.) interference (before adjusting for finish effect) and producing interference-fit pressures of 5.00, 7.55, 8.67, and 12.21 MPa (0.73, 1.10, 1.26, 1.77 ksi) had resulting life factors of 0.60, 0.60, 0.58, and 0.60, respectively.

(4) In general, the life factor was smallest (greatest life reduction) for bearings running under light load where the unfactored life is highest. For any particular bearing size and interference fit, as the maximum Hertz stress on the inner race was increased, the effect of the hoop stresses on life was reduced, thus increasing the resulting life factor.

(5) The tightest fit at the high end of the RBEC−5 tolerance band defined in ANSI/ABMA shaft-fit tables produced a life factor of approximately 0.40 for an inner-race maximum Hertz stress of 1200 MPa (175 ksi) and a life factor of 0.60 for an inner-race maximum Hertz stress of 2200 MPa (320 ksi).

(6) Interference fits affected the maximum Hertz stress-life relation. With no interference fit, a Hertz stress-life exponent of $n = 8.1$ was found, which is close to the accepted value of $n = 8$. With an m6 middle-of-the-tolerance-range interference fit, the exponent was $n = 7.7$; with an interference fit at the tight end of the range for the m6, the Hertz stress-life exponent became $n = 7.4$.

References

1. American Bearing Manufacturer Association (ABMA) (2001), "Shaft and Housing Fits for Metric Radial Ball and Roller Bearings (Except Tapered Roller Bearings) Conforming to Basic Boundary Plan," ANSI/ABMA-7:1995(R2001), The American Bearing Manufacturers Association, Washington, DC.
2. American Bearing Manufacturer Association (ABMA) (2001), "Radial Bearings of Ball, Cylindrical Roller and Spherical Roller Types—Metric Design," ANSI/ABMA-20:1996, The American Bearing Manufacturers Association, Washington, DC.
3. Czyzewski, T. (1975), "Influence of a Tension Stress Field Introduced in the Elastohydrodynamic Contact Zone on Rolling Element Fatigue," Wear, **34**, 2, pp. 201–214.
4. Coe, H.H. and Zaretsky, E.V. (1986), "Effect of Interference Fits on Roller Bearing Fatigue Life", ASLE Transactions, **30**, 2, pp. 131–140.
5. Jones, A.B. (1946), Analysis of Stress and Deflections, Vol. 1, New Departure Div., General Motors Corp.
6. Lundberg, G.; and Palmgren, A. (1947), "Dynamic Capacity of Rolling Bearings," Acta Polytech, Mechanical Engineering Series, **1**, 3, Stockholm, Sweden.
7. Zaretsky, E.V., ed. (1999), STLE Life Factors for Rolling Bearings, STLE SP–34, second edition, Society of Tribologists and Lubrication Engineers, Park Ridge, IL.
8. Zaretsky, E.V., Poplawski, J.V. and Root, L.E. (2007), "Reexamination of Ball-Race Conformity Effects on Ball Bearing Life," Tribology Transactions, **50**, 3, June 2007, pp. 336–349.
9. Juvinall, R.C. (1967), "Engineering Considerations of Stress Strength and Strain," McGraw Hill, NY.
10. Weibull, W. (1939), "A Statistical Theory of the Strength of Materials," Ingeniorsvetenskapsakad, Handl., no. **151**.
11. Weibull, W. (1939), "Phenomenon of Rupture of Solids," Ingeniorsvetenskapsakad Handl., no. 153.
12. Weibull, W. (1951), "A Statistical Distribution Function of Wide Applicability," J. Appl. Mech. Trans. ASME, **18**, 3, pp. 293−297.
13. Poplawski, J.V., Peters, S.M. and Zaretsky, E.V. (2001), "Effect of Roller Profile on Cylindrical Roller Bearing Life Prediction—Part I Comparison of Bearing Life Theories," Tribology Transactions, **44**, 3, pp. 339–350.
14. Palmgren, A. (1945), "Ball and Roller Bearing Engineering," 1st ed., Translation by G. Palmgren and B. Ruley, SKF Industries, Philadelphia, PA.
15. The Timken Company (2004), "Timken Aerospace Design Guide for Precision Metric Ball and Cylindrical Roller Bearings", 3M 04-04-06 Order No. 5912, The Timken Company, http://www.timken.com/en-us/solutions/aerospace/ Documents/aerospace%20product%20design%20guide.pdf, Accessed 10/15/2008.
16. Harris, T.A. (1991), "Rolling Bearing Analysis," Third Edition, John Wiley & Sons, Inc. , ISBN 0-471-51349-0.

17. Poplawski, J.V., Rumbarger, J.H., Peters, S.M., Flower, R., and Galaitis, H. (2002) "Advanced analysis package for high speed multibearing shaft systems: COBRA-AHS," Final report, NASA Contract NAS3–00018.
18. Poplawski, J.V., Peters, S.M. and Zaretsky, E.V. (2001), "Effect of Roller Profile on Cylindrical Roller Bearing Life Prediction—Part II Comparison of Roller Profiles," Tribology Transactions, **44**, 3, pp. 417–427.
19. Lundberg, G.; and Palmgren, A. (1950), "Dynamic Capacity of Roller Bearings," Acta Polytechnica, Mechanical Engineering Series, **2**, 4, Stockholm, Sweden.

REPORT DOCUMENTATION PAGE	*Form Approved* *OMB No. 0704-0188*

1. REPORT DATE *(DD-MM-YYYY)*	**2. REPORT TYPE**	**3. DATES COVERED** *(From - To)*
01-11-2009	Technical Memorandum	

4. TITLE AND SUBTITLE
Interference-Fit Life Factors for Roller Bearings

5a. CONTRACT NUMBER

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
Oswald, Fred, B.; Zaretsky, Erwin, V.; Poplawski, Joseph, V.

5d. PROJECT NUMBER

5e. TASK NUMBER

5f. WORK UNIT NUMBER
WBS 877868.02.07.03.04.01

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
John H. Glenn Research Center at Lewis Field
Cleveland, Ohio 44135-3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E-16468-2

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, DC 20546-0001

10. SPONSORING/MONITOR'S ACRONYM(S)
NASA

11. SPONSORING/MONITORING REPORT NUMBER
NASA/TM-2009-215285

12. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified-Unlimited
Subject Category: 37
Available electronically at http://gltrs.grc.nasa.gov
This publication is available from the NASA Center for AeroSpace Information, 443-757-5802

13. SUPPLEMENTARY NOTES

14. ABSTRACT
The effect of hoop stresses in reducing cylindrical roller bearing fatigue life was determined for various classes of inner-ring interference fit. Calculations were performed for up to 7 fit classes for each of 10 bearing sizes. The hoop stresses were superimposed on the Hertzian principal stresses created by the applied radial load to calculate roller bearing fatigue life. A method was developed through a series of equations to calculate the life reduction for cylindrical roller bearings. All calculated lives are for zero initial internal clearance. Any reduction in bearing clearance due to interference fit would be compensated by increasing the initial (unmounted) clearance. Results are presented as tables and charts of life factors for bearings with light, moderate, and heavy loads and interference fits ranging from extremely light to extremely heavy for bearing accuracy class RBEC-5 (ISO class 5). Interference fits on the inner ring of a cylindrical roller bearing can significantly reduce bearing fatigue life. In general, life factors are smaller (lower life) for bearings running under light load where the unfactored life is highest. The various bearing series within a particular bore size had almost identical interference-fit life factors for a particular fit. The tightest fit at the high end of the tolerance band produces a life factor of approximately 0.40 for an inner-race maximum Hertz stress of 1200 MPa (175 ksi) and a life factor of 0.60 for an inner-race maximum Hertz stress of 2200 MPa (320 ksi). Interference fits also impact the maximum Hertz stress-life relation.

15. SUBJECT TERMS
Rolling element bearings; Roller bearings; Interference fit; Bearing life prediction; Stress

16. SECURITY CLASSIFICATION OF:			**17. LIMITATION OF ABSTRACT**	**18. NUMBER OF PAGES**	**19a. NAME OF RESPONSIBLE PERSON**
a. REPORT	**b. ABSTRACT**	**c. THIS PAGE**			STI Help Desk (email:help@sti.nasa.gov)
U	U	U	UU	26	**19b. TELEPHONE NUMBER** *(include area code)* 443-757-5802

Standard Form 298 (Rev. 8-98)
Prescribed by ANSI Std. Z39-18