RSIC-932

ROLLING-CONTACT BEARING REFERENCE SUMMARY

by

William A. Glaeser
David B. Cox
Keith F. Dufrane
Jerrold W. Kannel

Contract No. DAAH01-67-C-1921
Battelle Memorial Institute
Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201

September 1969

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FOREWORD

The purpose of this reference summary is to provide the designer and the test engineer with the current technology of rolling-contact bearings as it pertains to aerospace applications, with the exception of gyroscope spin-axis bearings. The information provided in the summary was obtained from selected technical references including government reports, military specifications, text books, technical journals, and technical papers. References through 30 June 1969 are covered. This information was subjected to critical review and interpretation as well as configuration and performance limitations for use in selecting bearings and their lubricants. Useful references have been chosen and listed for provision of detailed information on specific aspects of rolling-contact bearing technology.

The Rolling Contact Reference Summary was prepared by Battelle Memorial Institute's Columbus Laboratories in response to a request originated by the joint Manned Spacecraft Center/Marshall Space Flight Center committee for the Investigation of Saturn/Apollo Ball and Roller Bearings.*

The report contains a brief review of the basic principles of rolling-contact bearings (geometry, kinematics, and life-rating methods) and a description of the various classes of rolling bearings. Materials and lubricants used in conventional rolling-contact bearings are described. Lubricants for aerospace applications are also described. Proved methods for bearing lubrication are summarized and an explanation of the recent use of elastohydrodynamic lubrication theory to design and selection of high-speed ball bearings is presented.

Aerospace bearings have special design requirements not covered in conventional rolling-contact bearing practices. These requirements are related to operating conditions including high speed, high temperature, hard vacuum, cryogenic temperatures, and oscillating motion. The special design requirements of these operating conditions are covered in three separate sections.

Failure diagnosis is an important aspect in rolling-contact bearing engineering. The various failure modes are described in a separate section together with description, photographs, and diagrams showing how specific surface features provide clues to the causes of bearing failure.

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* NASA Technical Report No. 53844, March 14, 1969.

Successful operation of rolling-contact bearings depends on correct choice and application of lubricants. By far the most persistent and difficult-to-solve bearing-failure problems can be traced to inadequate lubrication or lubricant failure. The bearing and its lubricant are an interacting system. Therefore, sections dealing essentially with the subject of lubrication have been included to familiarize the design and test engineer with lubricants and methods of using them. These sections include information on characteristics of the major classes of aerospace lubricant materials, the general effects of aerospace environments on lubricants, recent advances in optimizing bearing-lubricant system design through elastohydrodynamic calculations, special systems for liquid lubrication of high-speed bearings, and finally, actual test and flight experience with specific lubricants for various types of bearings in various kinds of environments. These sections will enable the design engineer to select the best candidate lubricants for a specific bearing and application environment (or else to determine that his application requires a lubricant outside the current state of the art).

For detailed reference sources on rolling-element bearings, the following are recommended:

H. T. Morton, Anti-Friction Bearings, Second Edition, Ann Arbor, Michigan, 1965.

T. A. Harris, Rolling Bearing Analysis, John Wiley & Sons, 1966.

J. J. O'Connor and John Boyd, Standard Handbook of Lubrication Engineering, McGraw-Hill, 1968.

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ILLUSTRATIONS (Continued)

ILLUSTRATIONS (Concluded)

Section I. PRINCIPLES OF ROLLING-ELEMENT BEARINGS

1. General Characteristics

Rolling-element bearings, as a class, include ball bearings, roller bearings, and needle bearings. This class of bearings is generally used where low friction is desired, especially at low speeds (start-up) and with heavy loads. They provide precise positioning of a rotating part, and they provide combined thrust and radial load support in one bearing.

Rolling-element bearings have a number of moving parts, limiting the designer to a finite number of combinations of choice available in standard sizes. Plain bearings should be considered where friction is not critical and compactness and low cost are desirable. Rolling-element bearings have a finite life in operation. Failure generally occurs by pitting and spalling of rolling-contact surfaces, a form of surface fatigue. This type of failure produces cumulative damage which progresses with accelerating rate from the formation of the first surface spall to complete destruction of the contact surfaces and, in extreme cases, fracture.

Fatigue life is sensitive to bearing load. Experience has shown that doubling the load will reduce bearing life by as much as eight times. Fatigue life is a function of the cube power of load. Thus, any factor which influences the maximum load on rolling elements will be a major influence on bearing life. Variation in life by as much as a factor of 10 is not unusual in rolling-contact bearing practice. Bearing life is measured in terms of number of cycles of rotation or oscillation. If the bearing speed is constant, bearing life can be given in terms of hours. Increases in speed will reduce bearing life, while operating the bearing at speeds lower than its design speed will increase its life.

Although rolling-contact bearings are expected to fail by fatigue long before they wear out, there are circumstances in which wear can be the principal failure mode. In high-temperature bearings and cryogenic bearings (where lubrication is meager), wear occurs at the sliding contacts in the bearing. Retainers will wear and break; balls or rollers can wear to uneven shapes or too small a size. Examples of various failure modes are given in Section IX.

2. Bearing Ratings

Rolling-element bearings are rated by manufacturers on the basis of fatigue life. Catalog ratings are based on Antifriction Bearing Manufacturers' Association (AFBMA) and American Standards Association formulas or on bearing tests and experience. The load capacity formula used is

$$C = F_e \ (i \cos \alpha)^{0.7} Z^{2/3} D^{1.8},$$

where

C = dynamic capacity (lb)

F_e = material and design factor

i = number of rows of balls in any one bearing

α = contact angle

Z = number of balls

D = ball diameter.

A similar formula with slightly different coefficients is used for roller bearings. Statistical analysis of bearing fatigue test data has made it possible to assign probabilities to projected bearing lives. That is, for a given number of standard bearings selected at random from a manufacturer, a load may be selected at which one can say with 99 percent confidence that 90 percent of the bearings will survive for a given life. This is the L_{10} life, used frequently in design of critical machine elements. The average life (L_{50}) is about five times the L_{10} life. Aerospace requirements demand even higher reliability for components.

Harris and Tallian [1] have proposed a method for modifying load capacity or bearing life for reliabilities higher than 90 percent by applying a correction factor to the L_{10} life or capacity. For instance, if the L_{10} life (90 percent reliability) of an angular-contact bearing supporting 5000 pounds is 440 hours at a continuous speed of 10,000 feet per minute, the projected fatigue life for 99 percent reliability by Harris' method must be reduced to 100 hours, or by about 75 percent. The method is empirical, based on statistical analysis of a larger number of bearing failures. As the reliability for predicting bearing life for a given load is increased, those bearings that fail early or prematurely make up the failure population. Early failures are due principally to manufacturing defects, and improvement in bearing quality or inspection methods can reduce the numbers of premature failures. In fact, Harris and Tallian indicate that their studies show a leveling off of bearing life or load capacity reduction above 90 percent reliability. That is, there is a limiting load or life at which 100 percent reliability should be possible.

High reliability in predicting rolling-element bearing life is possible only when bearings are operated under ideal conditions. Some bearing manufacturers base their catalog ratings on bearing fatigue tests in which large numbers of bearings are run under standardized conditions of load, speed, and lubrication. As was pointed out earlier, any factor which alters the anticipated load on a bearing will have a significant influence on the bearing performance life. Such factors include:

Misalignment
Shaft bending, bearing race deflection
Internal clearance, contact angle

High-speed inertial effects
Impact
Type of loading: radial, thrust, or combination

Thermal expansion or distortion
Dirt
Surface finish.

Most of the above factors alter the distribution of load among the rolling elements. Three factors, dirt or debris between rolling elements, surface finish, and impact cause periodic sharp increases in contact stress and early initiation of surface cracking. Internal clearance influences are most prevalent in bearings subjected to radial loading. The load distribution among the rolling elements shown in Figure 1 [2] becomes more concentrated as bearing internal diametral clearance increases. Misalignment and shaft deflection is especially deleterious to roller bearings because it induces roller edge loading.

Bearing catalogs usually contain dynamic load ratings for one speed level and nomographs or correction charts for converting to other speeds and various combinations of loading. The catalog rating systems vary from one manufacturer to another. Most rate on the basis of L_{10} lives, but some rate on the basis of average lives. Generally, the catalogs state the basis of ratings. Load ratings for the same size and type of bearing vary from manufacturer to manufacturer. Differences are not more than a factor of 2, which is not large for rolling-contact bearing variability in performance. Nomographs for general classes of bearings together with basic load rating tables is given by O'Connor and Boyd [3]. A sample of a basic load rating table is shown as Table I [3].

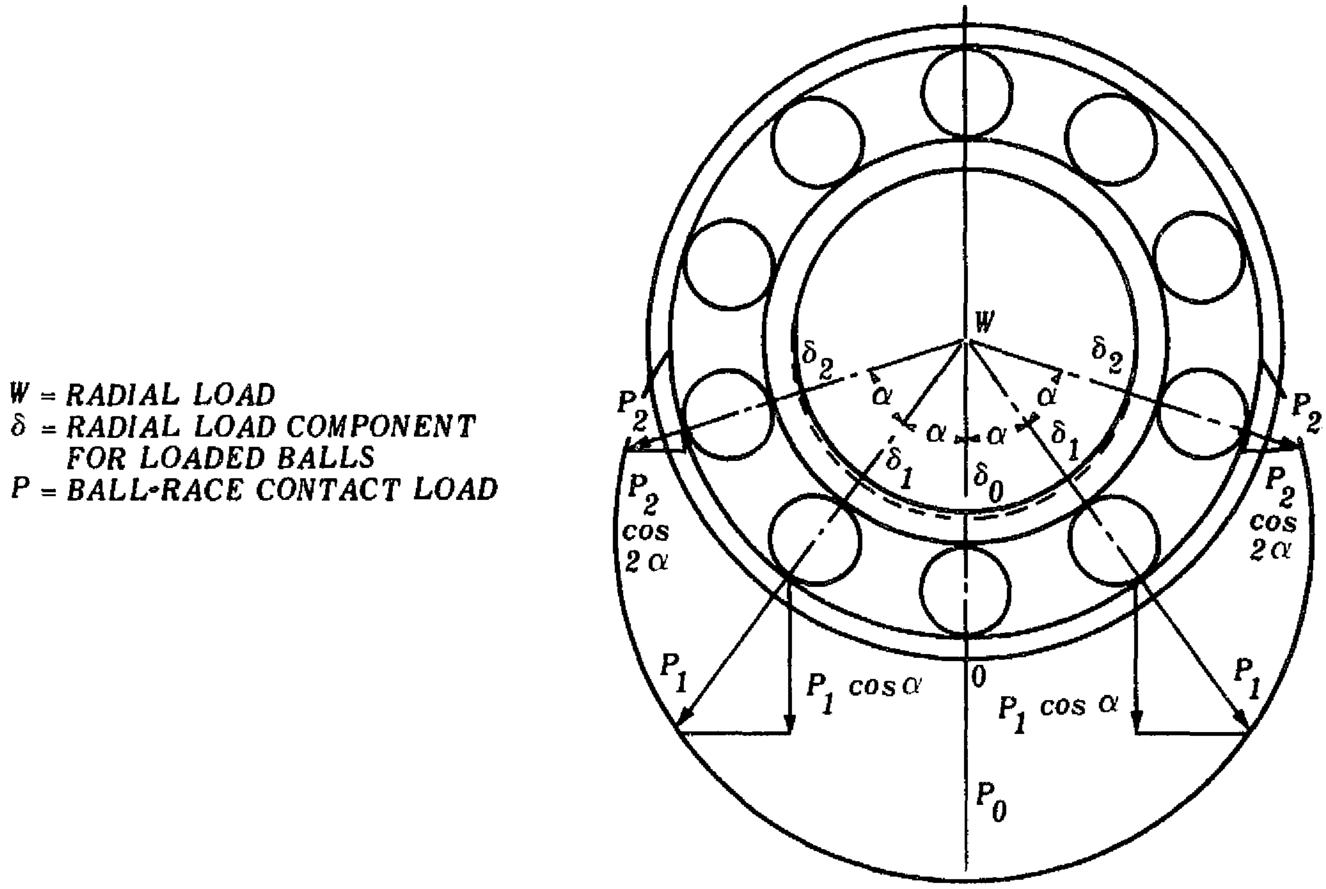


FIGURE 1. RADIAL LOAD DIVIDED AMONG THE LOWER BALLS TRIGONOMETRICALLY

3. Bearing Materials

One important factor which influences bearing fatigue life significantly is the size and distribution of nonmetallic inclusions in the bearing steel. Long stringers located close to the load bearing surfaces are especially deleterious. Inclusion content of bearing steel is just as important to the designer as are the factors listed above. However, the designer has less control over this factor except to insist that his bearings be made from steel of high cleanliness. Bearing materials are discussed further in Section III.

4. Bearing Geometry

The shapes and relative sizes of rolling-element bearings have been established and standardized to provide optimum performance and precision. The range of variations possible in size and geometry within the available standard bearings is described in Section II.

TABLE I. SAMPLE LOAD RATING TABLE FOR DEEP-GROOVE BALL BEARINGS

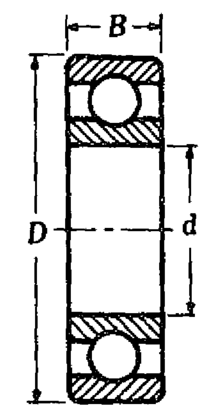
Dimensions and Basic Load Ratings for 200 Series, Radial, Deep Groove Ball Bearings

$$\frac{C}{P} = \frac{\text{basic load rating (from table below)}}{\text{equivalent load (from formula below)}}$$

$$P = XVF_r + YF_a$$

where

X = radial factor given below
V = rotation factor = 1.0 for inner ring / = 1.2 for outer ring } rotating in relation to the load
Y = thrust factor given below
F_r = radial load, calculated
F_a = thrust load, calculated



When $\frac{F_a}{VF_r}$ { is smaller than or equal to e use $X = 1$ and $Y = 0$; is greater than e use $X = 0.56$ and Y from table below

where e is a reference value given in the table below.

$\frac{F_a}{C_0}$ =	0.014	0.028	0.056	0.084	0.11	0.17	0.28	0.42	0.56
e =	0.19	0.22	0.26	0.28	0.30	0.34	0.38	0.42	0.44
Y =	2.30	1.99	1.71	1.55	1.45	1.31	1.15	1.04	1.00

C_0 = basic static load rating from table below.

$$P_0 = X_0F_r + Y_0F_a$$

where $X_0 = 0.6$, $Y_0 = 0.5$.

TABLE I. SAMPLE LOAD RATING TABLE FOR DEEP-GROOVE BALL BEARINGS (Continued)

Bearing No.	d		D		B		Balls		Basic Static Load Rating C_0 (lb)	Basic Load Rating C (lb)
	mm	in.	mm	in.	mm	in.	No.	Diameter (in.)		
200	10	0.3937	30	1.1811	9	0.3543	7	$3/16$	400	805
201	12	0.4724	32	1.2598	10	0.3937	7	$15/64$	685	1,180
202	15	0.5906	35	1.3780	11	0.4331	8	$15/64$	790	1,320
203	17	0.6693	40	1.5748	12	0.4724	8	$17/64$	1,000	1,650
204	20	0.7874	47	1.8504	14	0.5512	8	$5/16$	1,390	2,210
205	25	0.9843	52	2.0472	15	0.5906	9	$5/16$	1,560	2,420
206	30	1.1811	62	2.4409	16	0.6299	9	$3/8$	2,250	3,360
207	35	1.3780	72	2.8346	17	0.6693	9	$7/16$	3,070	4,440
208	40	1.5748	80	3.1496	18	0.7087	9	$15/32$	3,520	5,040
209	45	1.7717	85	3.3465	19	0.7480	9	$1/2$	4,010	5,660
210	50	1.9685	90	3.5433	20	0.7874	10	$1/2$	4,450	6,070
211	55	2.1654	100	3.9370	21	0.8268	10	$9/16$	5,630	7,500
212	60	2.3622	110	4.3307	22	0.8661	10	$5/8$	6,950	9,070
213	65	2.5591	120	4.7244	23	0.9055	10	$21/32$	7,670	9,900
214	70	2.7559	125	4.9213	24	0.9449	10	$11/16$	8,410	10,800
215	75	2.9528	130	5.1181	25	0.9843	11	$11/16$	9,250	11,400

TABLE I. SAMPLE LOAD RATING TABLE FOR DEEP-GROOVE BALL BEARINGS (Concluded)

Bearing No.	d mm	d in.	D mm	D in.	B mm	B in.	Balls No.	Balls Diameter (in.)	Basic Static Load Rating C_0 (lb)	Basic Load Rating C (lb)
216	80	3.1496	140	5.5118	26	1.0236	10	$^3/_4$	10,000	12,600
217	85	3.3465	150	5.9055	28	1.1024	11	$^{25}/_{32}$	12,000	14,400
218	90	3.5433	160	6.2992	30	1.1811	10	$^7/_8$	13,600	16,600
219	95	3.7402	170	6.6929	32	1.2598	10	$^{15}/_{16}$	15,600	18,800
220	100	3.9370	180	7.0866	34	1.3386	10	1	17,800	21,100
221	105	4.1339	190	7.4803	36	1.4173	10	$1^1/_{16}$	20,100	23,000
222	110	4.3307	200	7.8740	38	1.4961	10	$1^1/_8$	22,500	24,900
224	120	4.7244	215	8.4646	40	1.5748	9	$1^3/_{16}$	22,600	25,100
226	130	5.1181	230	9.0551	40	1.5748	9	$1^1/_4$	25,000	26,900
228	140	5.5118	250	9.8425	42	1.6535	10	$1^1/_4$	27,800	28,800
230	150	5.9055	270	10.6299	45	1.7717	11	$1^1/_4$	30,600	30,400
232	160	6.2992	290	11.4173	48	1.8898	12	$1^1/_4$	33,400	32,000
234	170	6.6929	310	12.2047	52	2.0472	12	$1^3/_8$	40,400	36,700
236	180	7.0866	320	12.5984	52	2.0472	11	$1^1/_2$	44,100	37,500
238	190	7.4803	340	13.3858	55	2.1654	11	$1^5/_8$	51,700	44,100

5. Heat Transfer

Rolling-element bearings will fail because of thermal effects. Loss of internal clearance by more rapid thermal expansion of the inner race or rolling elements will cause freeze up or fracture of the bearing. Excessive frictional heating of the retainer will cause damage to the retainer and, sometimes, the retainer will break. Undesirable thermal effects in rolling-element bearings generally come from two sources: (1) externally generated heat conducted down the shaft into the bearing or carried by convection by the lubricant and (2) internally generated heat from friction of rolling elements or shearing of the lubricant. Thermal problems are not anticipated in conventional rolling-element bearing applications. The problem is most often encountered in high-speed bearings, high-temperature environments, cryogenic bearings, meager lubrication, or in bearings employing self-contained lubrication techniques. Recognizing the conditions which can produce thermally induced bearing distress is an important aspect of bearing design.

Any temperature gradient which requires heat to flow from the shaft through the bearing and into the housing is potentially harmful to the bearing. This is because a rolling-element bearing is a thermal barrier to radial heat flow. When the bearing is in motion, the heat conduction path is through rolling-element contacts which are minute in area, and there is interposed between the contact surfaces only a thin film of heat-transfer medium (lubricant). Without sufficient convective cooling by lubricant flow, a radial thermal gradient is developed across the bearing. If the thermal gradient is sufficiently steep, the relative expansion of bearing components will close down the internal clearance and increase rolling-element contact stress. This is particularly deleterious in preloaded angular-contact bearings.

An undesirable thermal gradient resulting from external heat sources can be recognized readily and allowances can be made by provision of sufficient internal clearance or convective cooling. Problems from internally generated (frictional) heat are not as obvious. In angular-contact bearings for instance, high speeds cause an increase in the contact area (Section II). This condition produces an increase in heat generation, and if convection cooling is insufficient, a degenerative process occurs; the results include wear, lubricant decomposition, metal transfer, and early spalling. The analysis of heat generation and flow in rolling-element bearings is a complex process requiring the use of digital computers. Harris [4] has suggested a simplified method for thermal analysis of a ball bearing; his method is reviewed in Section VI. A rough estimate can be made of heat generation in a bearing by assuming a coefficient of friction (0.001 is a conservative value) and using the following relation:

$$Q_f = 4 \, n \, W \, d_m \times 10^{-5}$$

where

Q_f = heat generation (Btu/hr)

n = rpm

W = load (lb)

d_m = pitch diameter (in.).

Assuming that most of the heat removal takes place by convection, one can estimate the minimum amount of lubricant flow required to balance the internal heat generation.

Section II. BEARING DESIGN AND SELECTION

Rolling-element bearings are available from manufacturers in a variety of types, sizes, and materials. The designer has a number of choices not only in type (ball or roller) and size, but also in geometry (contact angle, retainer shape, conformity, etc.). This section describes the range of commercial bearing types and sizes and their applications.

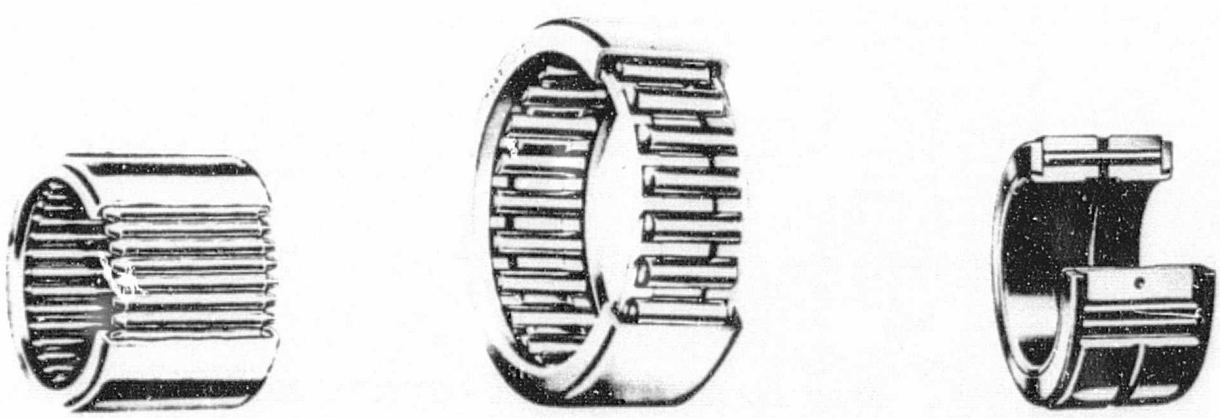
1. Bearing Types

There are three general types of rolling-element bearings: the ball bearing, the roller bearing, and the needle bearing. Within these three general classes, there are a number of variations in design. Standard types of rolling-element bearings are shown in Figures 2, 3, and 4. For details of bearing geometry, the manufacturers' catalogs should be consulted. Manufacturers' catalogs also contain a number of variations in the general bearing types shown in the figures.

The basic applications for the various types of rolling-element bearings shown are summarized in Table II.

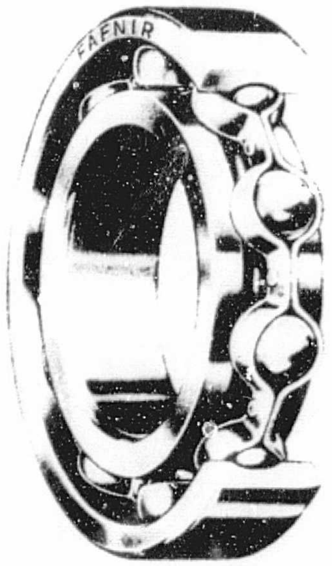
2. Bearing Specification

The designer, in selecting a rolling-element bearing, has a number of choices or decisions to make:

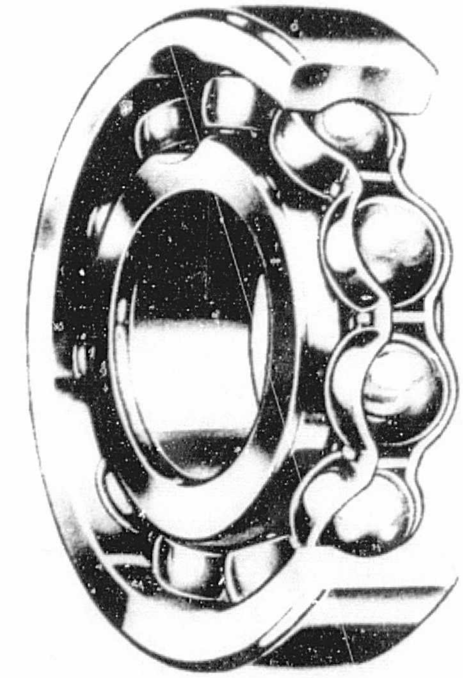


DRAWN CUP NEEDLE *AIRCRAFT NEEDLE*

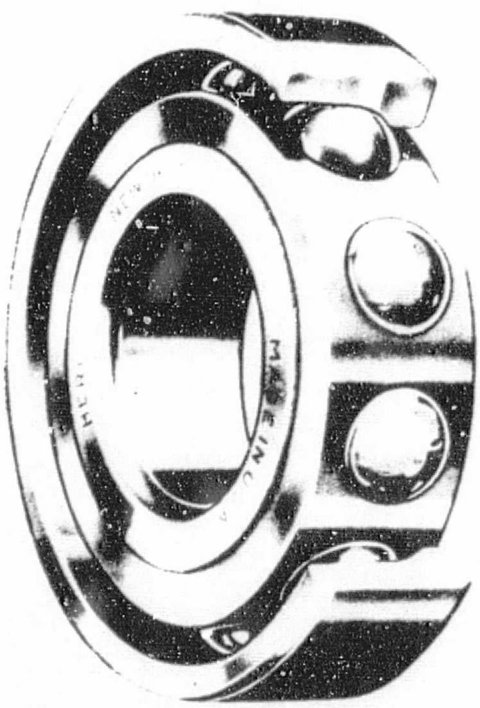
FIGURE 2. NEEDLE ROLLER BEARINGS



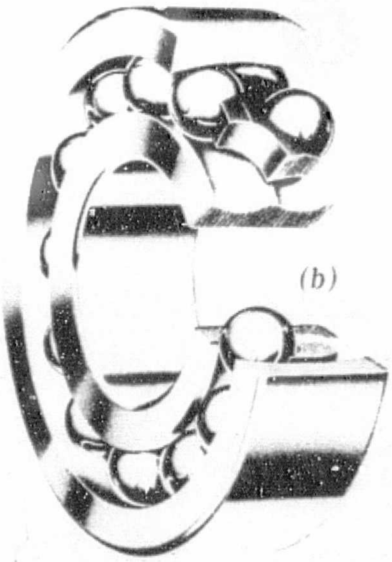
DEEP GROOVE
(CONRAD)



DEEP GROOVE WITH FILLING SLOT



ANGULAR-CONTACT
(SEPARABLE)

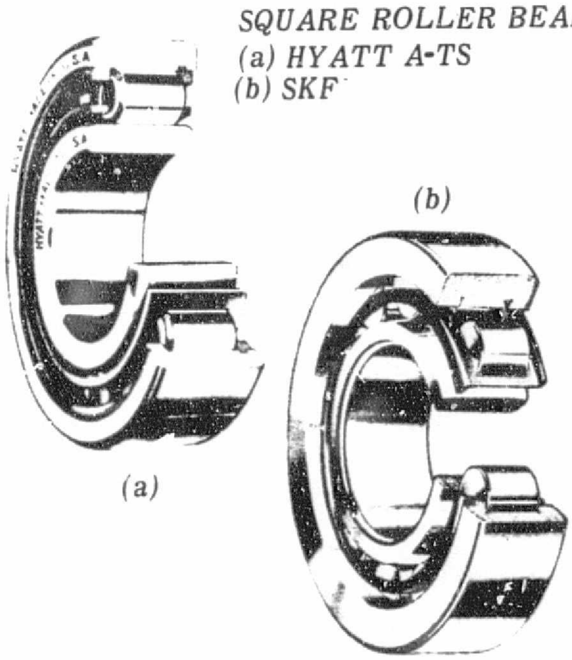


INTERNAL SELF-ALIGNING
BALL BEARING
(a) SKF
(b) NORMA HOFFMAN

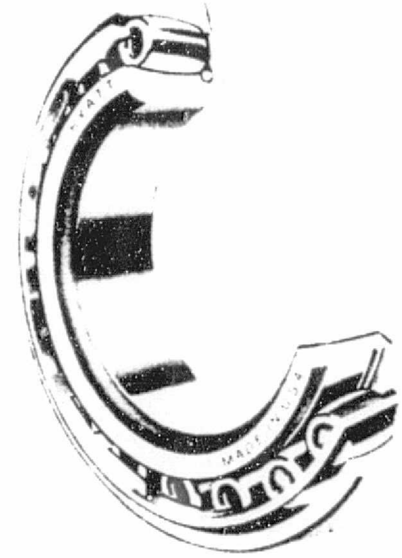
SELF-ALIGNING

FIGURE 3. BALL BEARINGS

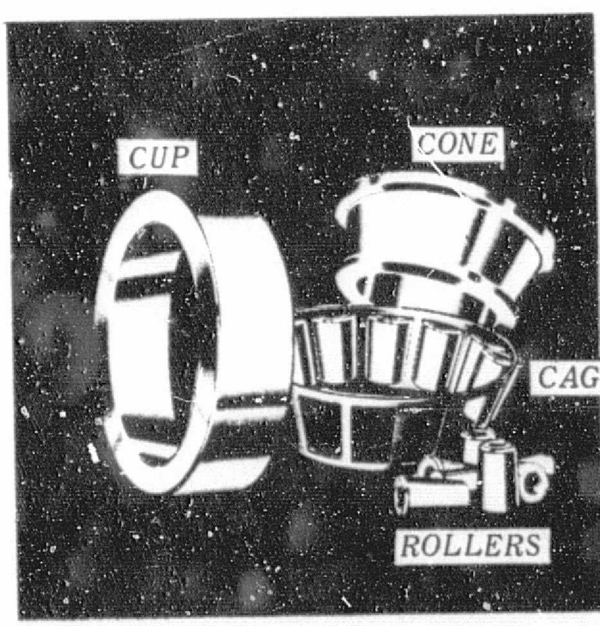
SQUARE ROLLER BEARING
(a) HYATT A-TS
(b) SKF



CYLINDRICAL ROLLER

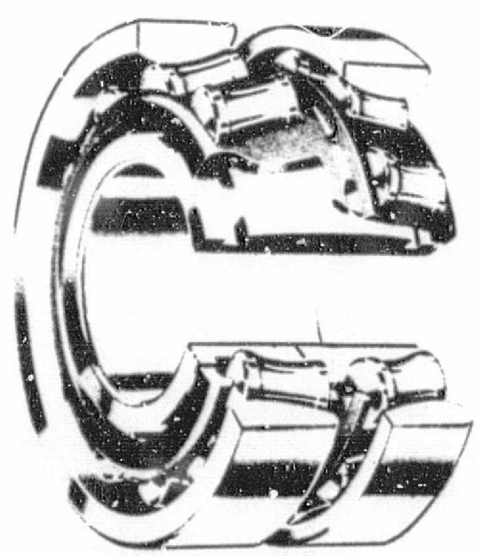


SPHERICAL ROLLER



TAPERED ROLLER

CONCAVEX ROLLER BEARING
(SHAFER)



CONCAVE ROLLER

FIGURE 4. ROLLER BEARINGS

TABLE II. ROLLING-ELEMENT BEARING APPLICATIONS

Bearing Type	Radial Loads	Thrust Loads	Combination Loads	Speed	Remarks
Deep groove ball	Moderate	Moderate	Moderate	High	Can have self-contained lubricant with integral seals
Deep groove ball	Heavy	Moderate	Moderate	Medium	Thrust load causes balls to roll over filling slot
Angular contact	Moderate	One direction, heavy	Radial load should not be greater than thrust	Very high	Usually used as matched pairs, preloaded
Self-aligning ball	Light	Very light	Light	Medium	----
Cylindrical roller	Heavy	---	---	High	Will stand more shock load than ball bearing
Spherical roller	Heavy	Moderate	Moderate	Medium	Used as oscillating bearings
Tapered roller	Very heavy	Heavy	Heavy	Medium	Popular as aircraft wheel bearings
Concave roller	Heavy	Light	Moderate	Medium	Has self-aligning capability
Drawn cup needle	Light	---	---	Medium	Provides thinnest rolling-contact bearing
Aircraft needle	Heavy	---	---	Medium	Used in helicopter rotor hubs

a) Bearing type (ball, roller, needle, angular-contact ball, etc.)

b) Bearing size

bore diameter
bearing weight (determines outside diameter, width, rolling-element size)

c) Retainer type

separable metal (allows diversity in bearing selection)
nonseparable (usually requires a separable bearing design)
no retainer, full complement of rolling elements (provides maximum load capacity for a given bearing size)

d) Bearing geometry

internal clearance
conformity between rolling element and race
contact angle

e) Bearing material

race and rolling-element material (Steel cleanliness must be considered for critical applications.)
retainer material (Self-lubricating aspects must be considered for certain applications like cryogenic bearings, space-vacuum bearings, etc.)

f) Bearing dimension tolerances

g) Lubricant

The above factors give the designer a large number of combinations to choose from. However, when he considers the significance of each factor in his design requirements, the choice becomes easier and the performance will be more effective. The seven factors are discussed in more detail in Table II and Sections III, IV, and V.

3. Geometry and Size

The dimensions of rolling-element bearings are generally described in two terms: bore size and weight. For instance, in ball bearings a 30-millimeter bore, light series has smaller balls, more balls, smaller outside diameter (OD), and smaller width than does the heavy series having the same bore. These differences in dimensions are shown in Figure 5 for the

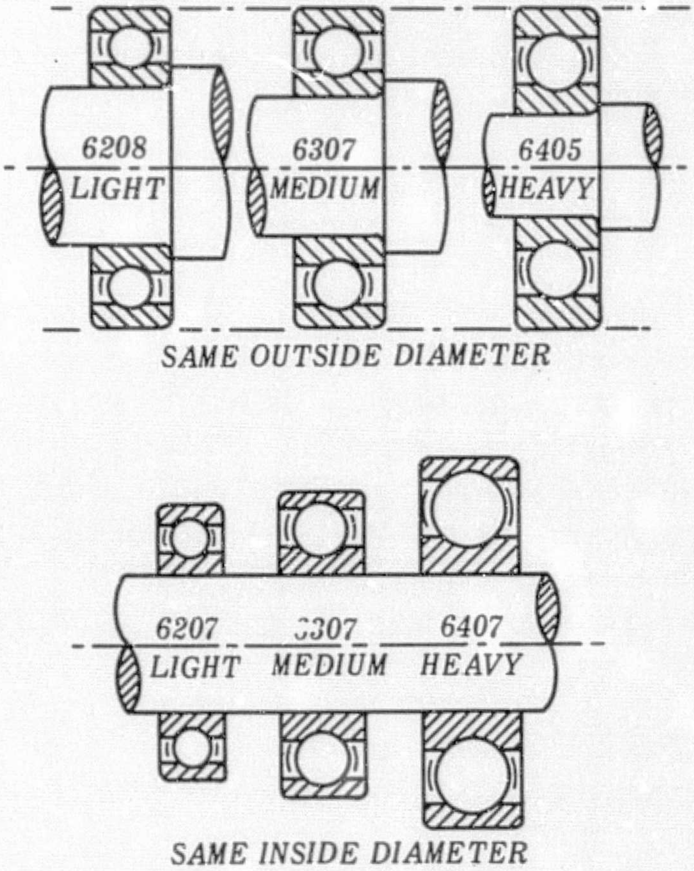


FIGURE 5. BEARING DIMENSIONAL VARIATIONS FOR CONSTANT BORE AND OUTSIDE DIAMETERS

light, medium, and heavy series. Bearing manufacturers usually have an extra-small, extra-light, and extra-large series, also.

4. Geometry

In ball bearings there are three geometrical factors which are important to the designer and which can be manipulated to optimize the geometry for maximum load capacity and minimum heat generation. These factors are conformity, contact angle, and internal clearance.

a. Conformity

Conformity is a term which defines how close the ball curvature fits the race curvature. It can be defined as

$$\sigma = \frac{\left(R-R_B\right) \times 100}{R_B} ,$$

where

R = race radius

R_B = ball radius

σ = conformity (%).

Conformity is also expressed as (R/ball diameter) × 100. Often, the bearing will have two conformity values because the radii of the outer and inner races will differ.

b. Contact Angle

Morton defines contact angle as "that angle formed by a line passing through the points of contact of a ball with the raceways and a perpendicular to the bearing axis when a thrust load is applied . . ." It locates how far from the middle of the race groove the ball contact is moved under an axial load. The geometry of the contact angle α is shown in Figure 6 [2].

c. Internal Clearance

Internal clearance in a ball bearing is the total radial movement possible between the inner and outer race when the balls are positioned at the bottom of the race grooves. This is known as radial play and is influenced by race groove geometry and precision of manufacture. Axial play can also occur as the result of internal clearance and level of ball-race conformity. Internal clearance and conformity in a radial bearing will determine contact angle under a given axial load. Internal clearance in a radial bearing will influence the radial load distribution among the balls. The larger the clearance, the fewer the balls supporting the radial load.

If the radial clearance and end play are known, the contact angle under static conditions can be determined by geometry:

$$\frac{EP}{RC} = \sqrt{\frac{1 - \cos^2 \alpha}{1 - \cos \alpha}} \text{ or } \frac{\sin \alpha}{1 - \sqrt{1 - \sin^2 \alpha}} ,$$

A = INNER RACE CLEARANCE
B = OUTER RACE CLEARANCE
α = CONTACT ANGLE
R_B = BALL RADIUS
R_I = INNER RACE RADIUS
R_o = OUTER RACE RADIUS
EP = END PLAY
RC = RADIAL CLEARANCE

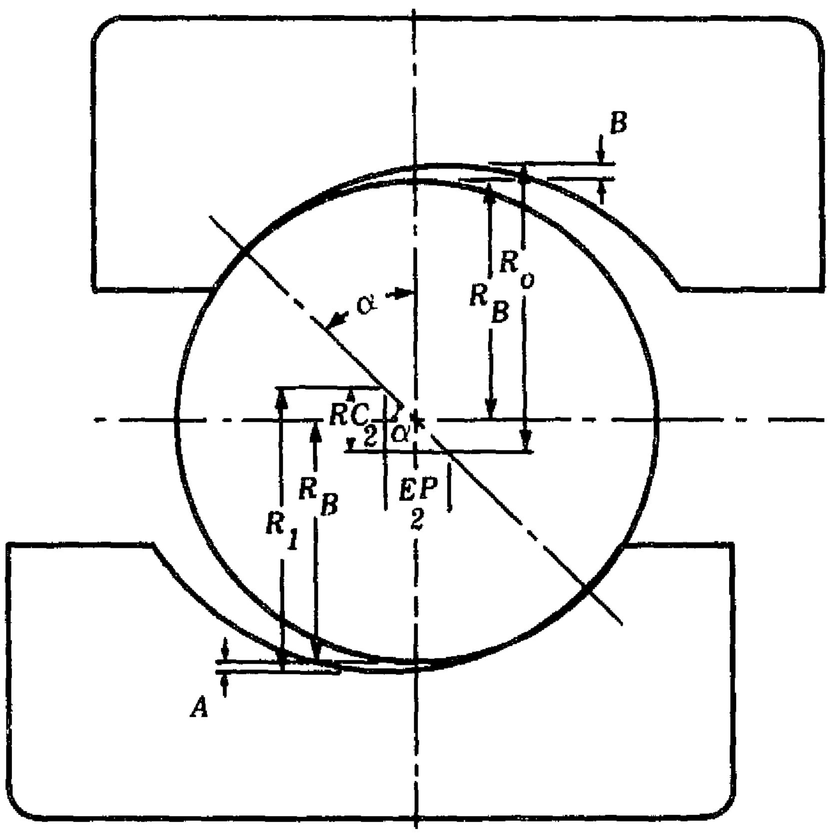


FIGURE 6. CONTACT ANGLE, END PLAY, AND RADIAL CLEARANCE FOR THRUST- OR AXIAL-LOAD CONDITIONS

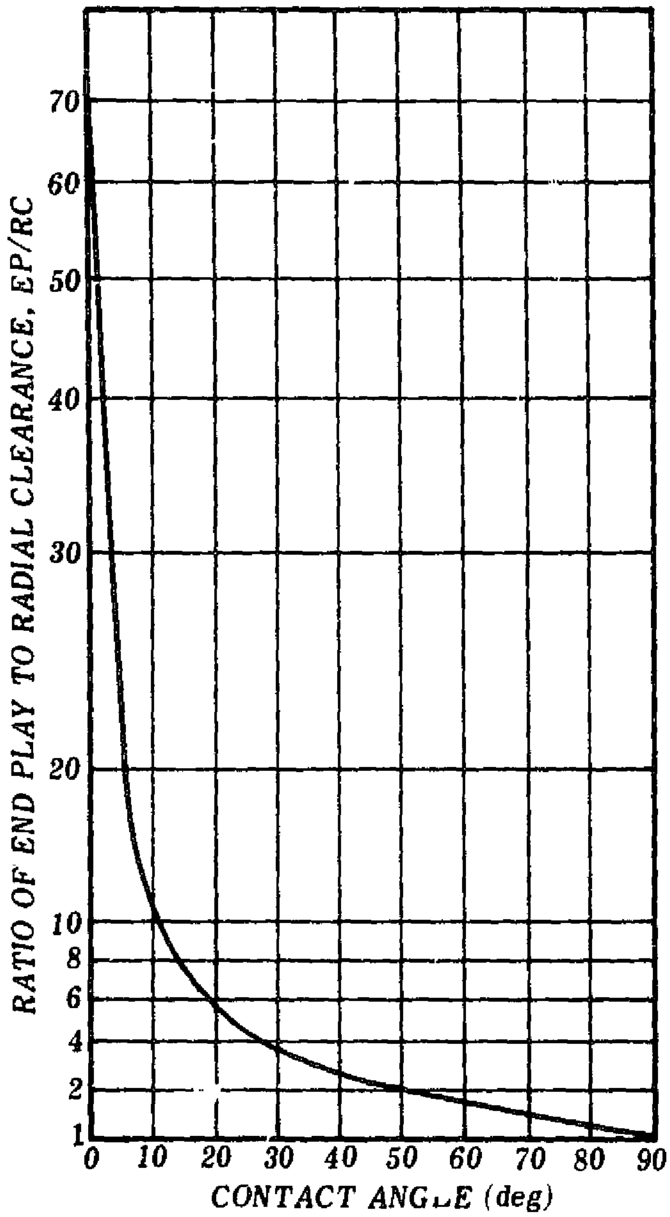


FIGURE 7. RATIOS OF END PLAY TO RADIAL CLEARANCE FOR EACH CONTACT ANGLE

where EP and RC (Figure 6) are end play and radial clearance. The ratio EP/RC as a function of contact angle is shown in Figure 7 [2].

The contact angle of an assembled angular-contact bearing can be estimated by rotating the bearing under thrust load and counting the number of revolutions the cage makes for a given number of inner-race revolutions. The geometry of the contact condition during rolling results in the following relation:

$$\cos \alpha = \frac{E}{d} \frac{(1 - 2\,Nr)}{Ni}$$

where

α = contact angle

E = pitch diameter

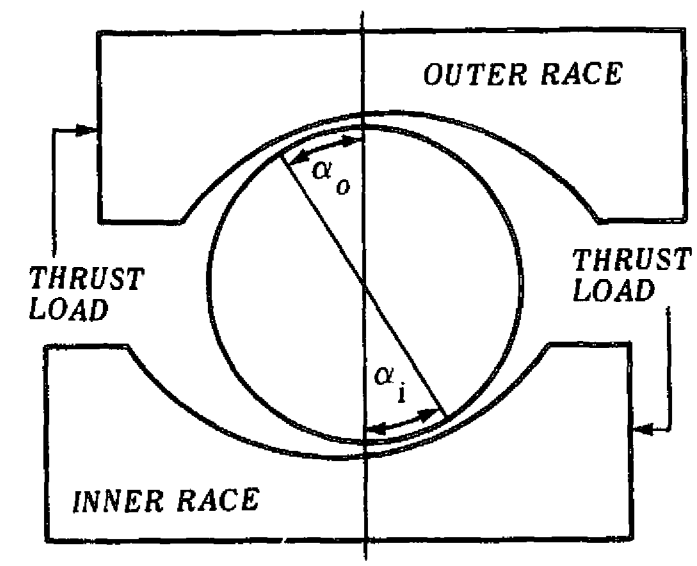
d = ball diameter

Nr = number of revolutions of retainer

Ni = number of revolutions of inner race.

5. Effect of Thrust Load

When an axial load is applied to a deep-groove ball bearing or an angular-contact bearing, the ball-race contact moves up the sides of the race groove (Figure 8) [5]. The bearing then becomes a two-point contact bearing (the ball contacts the outer race at one point and inner race at one point). This represents the condition of the ball-race contact when the bearing is stationary under an axial load, or in a preload condition. The contact angles α_o and α_i



INNER AND OUTER RACE CONTACT ANGLES α_o AND α_i EQUAL AT 0 rpm

FIGURE 8. FREE-BODY DIAGRAM OF A BALL OPERATING UNDER THRUST LOAD AT ZERO SPEED

are the same for the outer and inner races. The balls then move around the race at some fraction of the race rpm. Changing the contact angle changes the rate of travel of the ball around the race. Ball speed can vary around the pitch circle, which occurs to some extent in every ball bearing. In some cases when the variation becomes large, it can lead to erratic bearing friction (periodic torque peaks) and even retainer breakage. In high-speed bearings that operate with radial loading combined with small thrust loads, ball speed variations are prevalent and have been known to cause rapid bearing failure. Barish has reviewed this effect in detail [6]. For slow-speed bearings, small errors in alignment or adjustment can result in "lock-up," preventing rotation when a combined radial and thrust load is applied.

As the bearing speed is increased under thrust load, the condition shown in Figure 9 [5] develops, with the outer-race contact angle α_o decreasing and

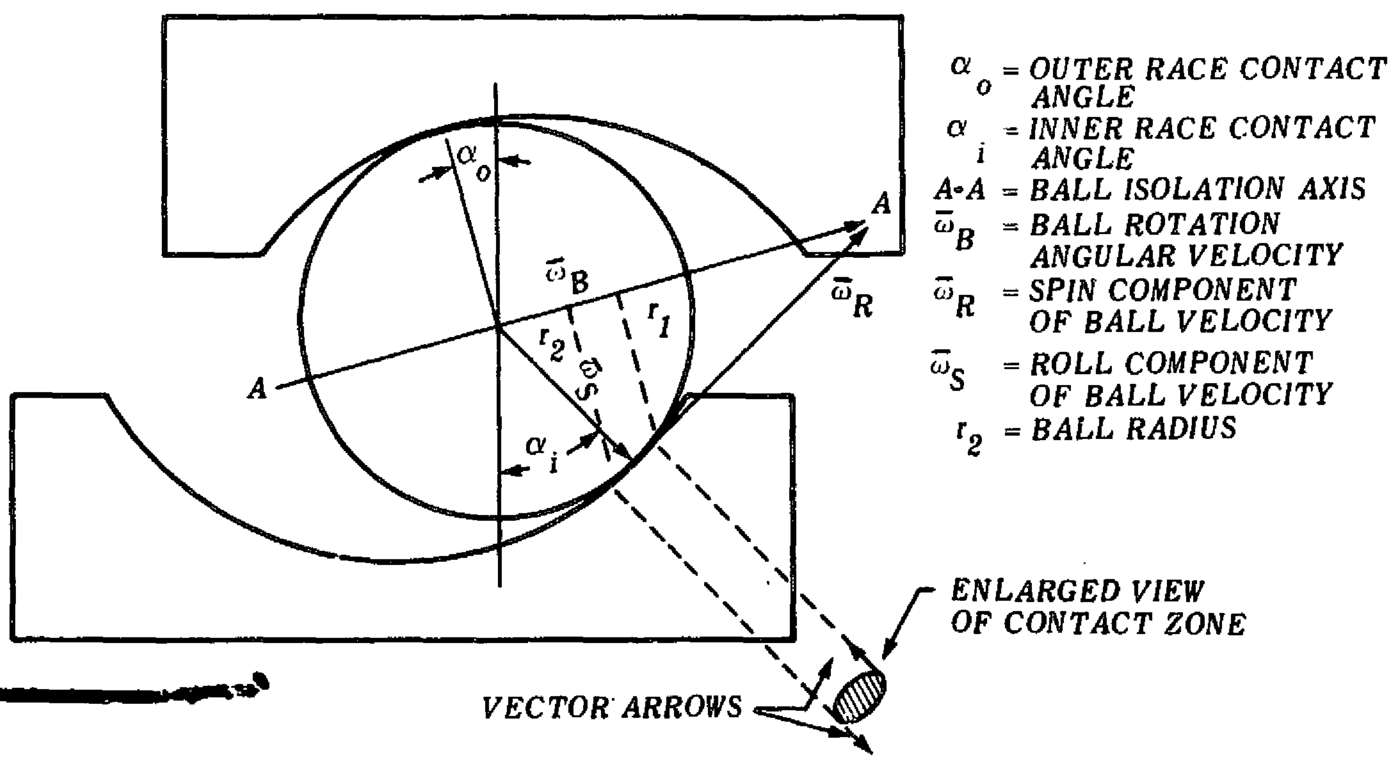


FIGURE 9. SECTION VIEW SHOWING EFFECT OF CENTRIFUGAL FORCE ON BALL CONTACT ANGLE

the inner-race contact angle α_i increasing. The centrifugal force of the ball tends to increase this difference with increasing speed. The condition shown in Figure 9 represents that found in a high-speed bearing.

The ball in Figure 9 has several possible axes of rotation. It can rotate about A-A with velocity vector $\overline{\omega}_B$ as shown. It can also rotate about the axis of the cone formed by the inner race contact with a velocity vector $\overline{\omega}_R$. The vector $\overline{\omega}_B$ can be resolved into the two vectors $\overline{\omega}_R$ and $\overline{\omega}_S$. Since $r_2\overline{\omega}_B$ is larger than $r_1\overline{\omega}_B$, it is evident that the surface velocity of one edge of the contact zone is slower than is the velocity at the other edge; this tends to spin the ball, giving rise to the so-called ball-spin vector. High-speed motion photography of ball movement in high-speed bearings indicates that slip occurs in the contact zone so that a combined rolling $\overline{\omega}_R$ and sliding describes the ball-race condition under axial load. The greater the spin/roll ratio, the greater is the frictional heat generation and the lower the bearing fatigue life for a given contact stress. This condition can be ameliorated by adding radial load, or by using a bearing with a smaller contact angle.

Under combined radial and thrust load at high speed, the condition referred to earlier as variable ball velocity develops. Figure 10 [6] shows the geometry of the condition. At high speed and pure radial load, the ball leaves contact with the inner race on the unloaded side. When thrust is added, the inner race moves axially until it contacts the unloaded ball with a high contact angle. The higher contact angle increases the ball speed; hence, the balls on the "top" of the bearing move faster in the pitch circle than do those on the bottom. This causes excessive ball loading against cage pockets.

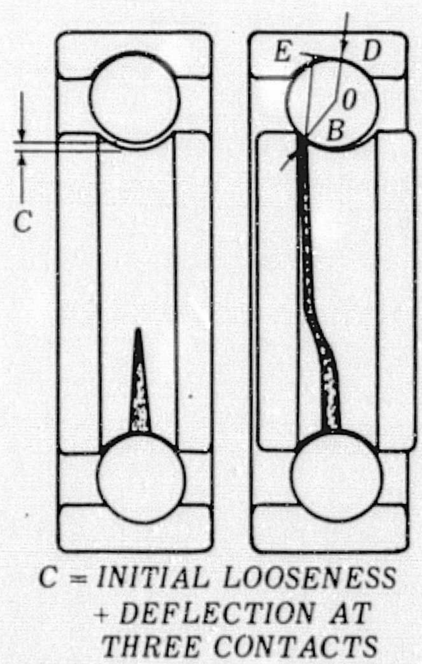


FIGURE 10. BALL PATH PATTERNS AT HIGH SPEEDS

6. Identification Codes

The numbers stamped on the sides of bearing races are an identification code which gives size, type, tolerances, lubrication, and special design features. These numbers are found in the bearing manufacturers' catalogs. Two identification codes exist:

a) Antifriction Bearing Manufacturers' Association System: Bearing bore size in millimeters (or sixteenths of an inch) is listed as the first number followed by the type, in letters, and width and OD code numbers. For instance, 25 BC 02 is a 25-millimeter bore, single-row ball bearing that is 52 millimeters OD by 15 millimeters wide with any type of cage.

b) American Standards Association System: Bearing width and OD indicated as narrow, light, medium, heavy, numbered in hundred; 100 is narrow, 200 is light, etc. The last two numbers indicate bore size. Bore size in millimeters is determined by multiplying all bore numbers 4 and above by 5. A prefix number to the series number indicates bearing type. For instance 7205 is a single-row, angular-contact bearing, light series, 25-millimeter bore.

Other code letters and numbers are used by manufacturers to indicate special features and designs. These are usually explained in the bearing manufacturers' catalogs. Further details on standard identification codes have been given by Morton [2].

7. Precision

Ball and roller bearings are made to a number of degrees of precision. Precision involves race and rolling-element surface finish and dimensional tolerances. The degree of precision in rolling-contact bearings has been standardized and listed in ABEC numbers. The better the precision, the higher the number. The numbers start with ABEC 1 and go through ABEC 3, 5, 7, and 9. These numbers indicate tolerance limits for metric- and inch-dimensional radial ball and roller bearings. The numbers do not pertain to tapered roller bearings, needle bearings, and special types of roller bearings. Tolerance limits are given in tabular form by Morton [2].

Section III. ROLLING-CONTACT BEARING MATERIALS

1. General Requirements

Materials for the races and balls of rolling-element bearings must fulfill three requirements: They must be of sufficient hardness at operating temperatures to support the load and resist brinelling, they must show adequate corrosion resistance, and they must have adequate dimensional stability. The hardness level required for the maximum operating temperature is related to the load on the bearing and, therefore, cannot be exactly specified. However, even very lightly loaded bearings generally require a minimum hardness of 55 Rockwell C (at temperature) for the races and rolling elements. A hardness of at least 58 Rockwell C is desirable for normally loaded bearings [7]. Operation of bearings with components of lower hardness results in premature failures through brinelling or fatigue.

Except for a limited number of alloys in narrow temperature ranges, increases in temperature result in reductions in hardness. The reductions are usually reversible if no microstructural alterations are produced, but they are permanent when the microstructure is altered. The softening of hardened steel, for instance, is reversible until the tempering temperature is reached, at which point further temperature increases result in permanent softening. Therefore, knowledge of the maximum temperature to which a bearing will be exposed is important in selecting the proper material for providing and maintaining adequate hardness.

The selection of bearing materials to resist corrosion and oxidation depends upon the environment and the temperature. Oxidation-corrosion resistance at elevated temperature must be considered as well as hot hardness, and these two requirements are often contradictory. Materials are more readily available for bearings operating at normal or low temperatures in corrosive atmospheres, but compromises must usually be struck between physical properties, corrosion resistance, and service life required.

Dimensional stability is a material's inherent resistance to change dimensions in service. This property, which is especially important for precision bearings and those cycled through large temperature ranges, is required for maintaining the original bearing dimensions, clearances, and preloads. Dimensional stability is attained by producing stable microstructural phases and controlling residual stresses through heat treating. Of primary importance is tempering the steel at temperatures above those to which the bearing will be exposed in operation.

2. Materials for Conventional Bearings

a. Balls and Races

After years of experience, SAE 52100 steel appears to be the most satisfactory material for rolling-element bearings in noncorrosive atmospheres at temperatures to 300°F [2]. Since most normal bearing applications fall under these operating conditions, SAE 52100 steel is the most common rolling-element bearing material.

SAE 52100 steel is a high-carbon, chromium steel that attains a hardness of greater than 60 Rockwell C by formation of a martensitic structure during heat treatment. Fatigue life has been extended by vacuum-melting practices to improve cleanliness. Dimensional stability is attained by cycles of cold quenching and tempering to transform and temper residual austenite remaining after quenching. The material must be tempered at temperatures at least as high as those over which the bearing material is expected to be used. Since heat treatment for maximum dimensional stability is not routinely performed, such treatment should be specified for precision and thermally cycled bearings.

Compositional modifications of SAE 52100 steel are available to increase the hardenability of bearings with cross sections exceeding one-half inch [2]. Four such modified alloys, which essentially contain more manganese, silicon, or molybdenum, are listed in ASTM Standard A 485-63. These, along with the standard grade, are listed in Table III.

b. Roller-Bearing Steels

Carburized grades of case-hardened steels were initially used for roller bearings to prevent breaking caused by roller skewing by providing case hardness and core toughness. Improvements in precision of manufacturing nearly eliminated the breakage problems, but case-hardened steels are still used for reasons of economics where unit loads permit. The accepted steels used in roller bearings are listed in Table IV.

3. Materials for Special Bearing Applications

The selection of materials for unusual applications normally involves a compromise between performance and the desired property to fulfill the special requirements. No universal material comparable to 52100 steel for normal applications is available for unusual applications. Particular applications must be considered individually to select the material with the desired properties.

TABLE III. CHEMICAL COMPOSITION OF STANDARD AND MODIFIED SAE 52100 STEEL

Section Thickness (in.)	Under 1/2	1/2 - 3/4	Over 3/4	Over 3/4	Over 3/4
Grade	Standard	1	2	3	4
Composition (%)					
Carbon	0.98-1.10	0.90-1.05	0.85-1.00	0.95-1.10	0.95-1.10
Manganese	0.24-0.45	0.95-1.25	1.40-1.70	0.65-0.90	1.05-1.35
Silicon	0.20-0.35	0.45-0.75	0.50-0.80	0.20-0.35	0.20-0.35
Chromium	0.90-1.15	0.90-1.20	1.40-1.80	1.10-1.50	1.10-1.50
Nickel, max	---	0.25	0.25	0.25	0.25
Copper, max	---	0.35	0.35	0.35	0.35
Molybdenum	---	0.06 max	0.06 max	0.20-0.30	0.45-0.60

TABLE IV. STEELS USED FOR ROLLER BEARINGS

	Metric	Light Section		Heavy Section		Very Heavy Section	
SAE Number	E52100	8620	4720	4620	4320	4820	E4340
Carburizes	No	Yes	Yes	Yes	Yes	Yes	Yes
Carbon	0.98-1.10	0.18-0.23	0.17-0.22	0.17-0.22	0.17-0.22	0.18-0.23	0.38-0.43
Manganese	0.25-0.45	0.70-0.90	0.50-0.70	0.45-0.65	0.45-0.65	0.50-0.70	0.65-0.85
Phosphorus	0.025 max	0.035 max	0.035 max	0.035 max	0.035	0.035	0.025
Sulphur	0.025 max	0.040 max	0.040 max	0.040 max	0.040 max	0.040 max	0.025
Silicon	0.25-0.35	0.20-0.35	0.20-0.35	0.20-0.35	0.20-0.35	0.20-0.35	0.20-0.35
Chromium	1.30-1.60	0.40-0.60	0.90-1.20	---	0.40-0.60	---	0.70-0.90
Nickel	Trace	0.40-0.70	0.35-0.55	1.65-2.00	1.65-2.00	3.25-3.75	1.65-2.00
Molybdenum	Trace	0.15-0.25	0.20-0.30	0.20-0.30	0.20-0.30	0.20-0.30	0.20-0.30

a. <u>Elevated Temperatures</u>

Increases in operating temperature require materials with increased hot hardness. The hardness (as a function of temperature) for high-temperature bearing materials is shown in Figure 11 [7]. Each type of material has a useful temperature range with sufficient hardness. If the minimum useful hardness is taken as Rockwell C 55, the chart in Figure 12 shows the useful range for each [7]. Since the material, fabrication, and machining costs rise sharply in the same order as does maximum useful temperature, the lowest temperature material that will fulfill the requirements should be specified. On this basis, the recommended materials for various elevated temperature ranges are presented in Table V.

The designer of bearings for elevated temperatures must consider the effect on fatigue life of temperature increases. The performance varies depending on the temperature, material, and lubricant, but is always less than that for normal temperatures. The effect of hardness on load capacity is critical, as shown by the relationship in Figure 13 [7]. For example, at 400°F the relative capacities are 95 percent for M-50, 86 percent for material for high temperature, and 65 percent for SAE 52100 steels (Table V). The capacity of bearings operating at elevated temperature must be derated to an extent depending on the operating temperature and the hardness-temperature characteristics of the material.

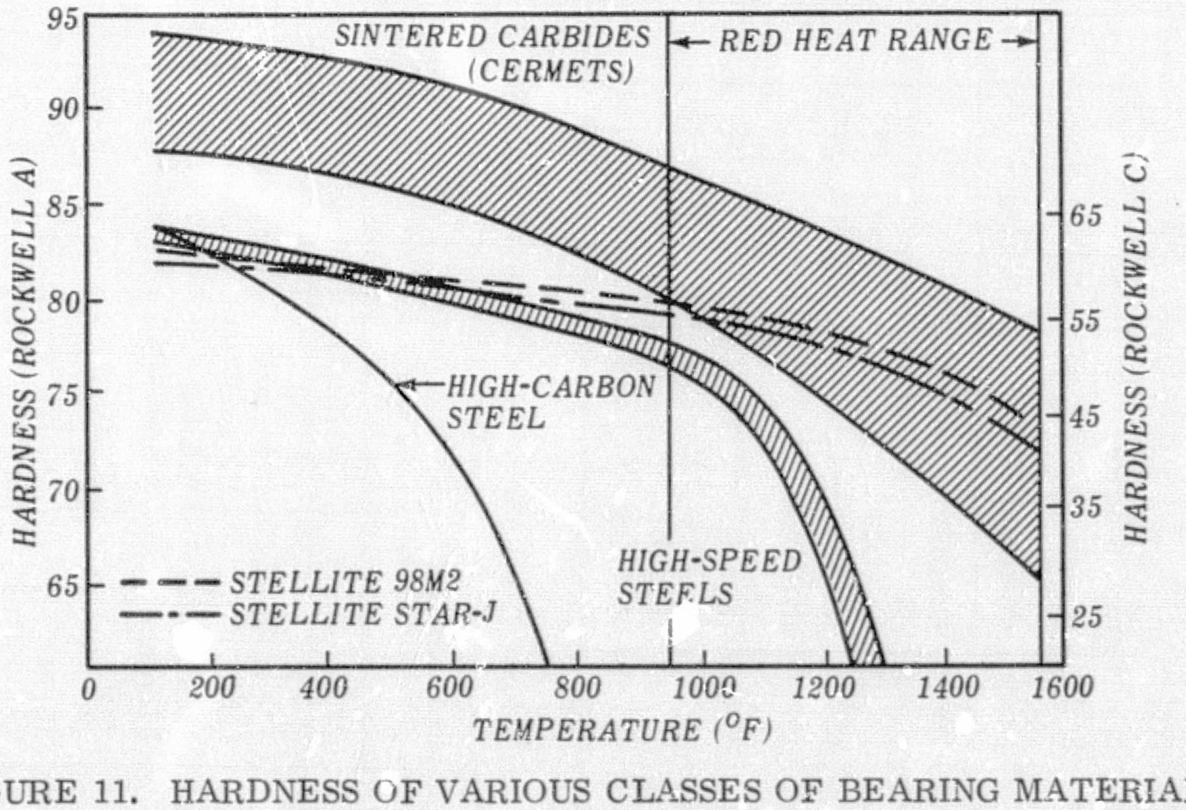


FIGURE 11. HARDNESS OF VARIOUS CLASSES OF BEARING MATERIALS AS A FUNCTION OF TEMPERATURE

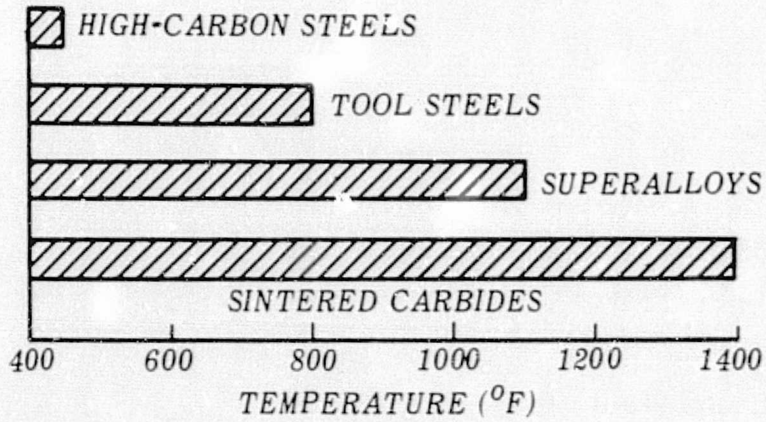


FIGURE 12. MAXIMUM USEFUL TEMPERATURES FOR HIGH-TEMPERATURE BEARING MATERIALS

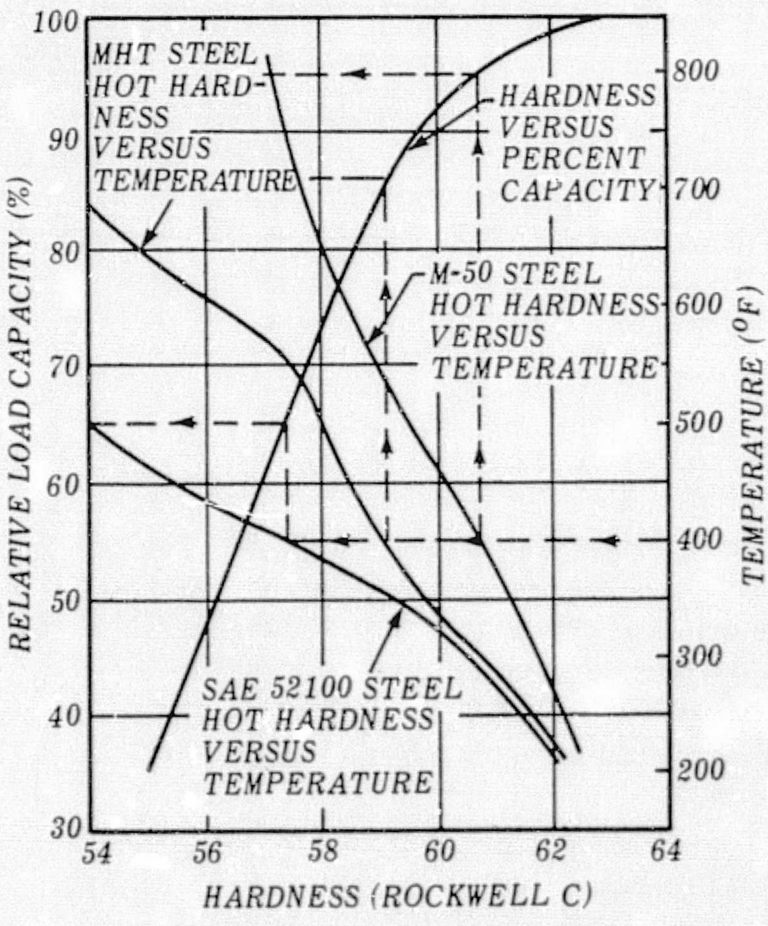


FIGURE 13. RELATIONSHIP BETWEEN TEMPERATURE, HARDNESS, AND LOAD-CARRYING CAPACITY

b. Corrosive Environments

Selection of a material for use in a corrosive environment depends on the severity of the corrosive media. In general, materials exhibiting the desired bearing properties do not possess freedom from corrosive attack. Every application requires a compromise between physical properties, corrosion resistance, and useful life.

Standard or modified 440 C stainless steel (Table V) has been used widely with success in corrosive environments, from liquid hydrogen temperatures (-423°F) to 800°F [2]. It resists oxidizing atmospheres and most mildly corrosive media except chlorides such as are present in sea water. The hardness (and associated load capacity) of 440 C is excellent, and consequently it is widely used where its corrosion resistance is adequate.

Bearings for highly corrosive media, such as heavy water, sea water, sulfuric acid, and chemical pumps, require better corrosion resistance than is provided by 440 C. Precipitation-hardening stainless steels, such as 17-4 PH,

TABLE V. RECOMMENDED MATERIALS FOR HIGH-TEMPERATURE BEARINGS

Temperature (°F)	Material
Room to 350	SAE 52100 [3]
350 to 600	Modified 440 C stainless steel [7] M-50 tool steel [8]
600 to 800	Modified 440 C stainless steel [7] M-1 tool steel [8]
800 to 1100	Haynes Stellite Star J [7] Haynes Stellite 98 M-2
1100 to 1600	Sintered carbides [7] (tungsten carbide, titanium carbide, and small percentages of chromium and columbium carbide with a nickel or cobalt binder)

17-7 PH, AM-350, and AM-355 possess intermediate hardnesses and increased corrosion resistance. Types 302, 304, and 316 provide further increases in corrosion resistance, but have even lower hardnesses. Since all of the austenitic stainless steels show hardness values well below that of 440 C, the bearing load capacity is sharply reduced. These materials should be used only when the environment is extremely corrosive to the harder alloys.

c. Retainers

In addition to selection of the proper material for the races and the balls or rollers, careful consideration must be given to the retainer material. Besides resisting corrosion in the environment to which the bearing is exposed, the retainer must have adequate strength and be compatible in sliding contact with the rolling elements and races. The retainer is also designed to provide lubrication for the bearing in applications where other means of lubrication cannot be provided.

In conventional rolling-element bearings, both metallic and nonmetallic retainers have been used successfully [9]. With normal temperatures, metallic retainers of stamped low-carbon steel and machined iron-silicon-bronze or a leaded brass are used. Nonmetallic retainers are made of cotton-cloth-phenolic laminates which are impregnated with lubricant. These latter materials are suitable for continuous use to temperatures of 275°F. All of these materials have reasonably good properties in sliding contact with the bearing elements. Silver-plated bronze has been used in applications where marginal lubrication exists in part of the operating cycle.

As temperature increases, the strength requirements of the retainer limit the use of copper-base alloys to 600°F, or about the maximum use temperature for liquid lubricants [10]. Nickel-base alloys have been most successful for temperatures above 600°F. H monel and S monel have been used for retainers at operating temperatures of 1000°F and above [9]. Table VI lists the chemical compositions of the three most widely used retainer materials [10].

For applications where conventional lubrication is not possible, such as in vacuum, cryogenic, or extreme-temperature environments, the retainer is designed to be self-lubricating (to provide lubrication both for itself and for the rolling elements). The retainers are made entirely of a composite self-lubricating material, they are coated with a solid lubricant, or they are made with solid lubricant inserts in a metallic retainer. Plastics are commonly used for cryogenic or vacuum applications at low temperatures. Listed in Table VII are some typical nonmetallics that have been used successfully [2]. Design of the bearing must allow for the much lower coefficient of thermal expansion of the plastic materials to assure adequate retainer-race clearance if large temperature ranges are encountered, such as in cryogenic bearings.

TABLE VI. COMPOSITIONS OF COMMON RETAINER MATERIALS

Material	Cu	Ni	Si	Fe	Zn	Mn
Iron-silicon-bronze	Min 90.00	--	3.25	1.50	2.75	Max 1.00
H monel	31	63	3.00	2.00	--	0.75
S monel	30	63	4.00	2.00	--	0.75

Lubrication can be provided for limited periods by solid lubricant coatings on the retainer [11]. While the coatings provide adequate lubricant, they are not self-replenishing, and the bearing life is limited by the life of the coating. Attempts have been made to provide solid-lubricant reservoirs by filling pockets machined in the critical sliding areas with solid lubricant. The two retainer designs in Figure 14 result in extended bearing life at 350° and 750°F, which is a function of the size of the reservoirs [11]. Similar designs are also applicable to very high-temperature bearings, but the service life of bearings at temperatures above 1000°F has been very limited.

TABLE VII. NONMETALLICS USED FOR RETAINERS FOR LIGHTLY LOADED ROLLING ELEMENTS

Material	Durometer Hardness	Temperature Range (°F)		Coefficient of Expansion (in./in./°F × 10^{-5})	Modulus of Elasticity (psi)	Specific Gravity	Acid Attack
Nylon	D 78	-100	230	5.5	200,000	1.14	Inert
Teflon	D 50-65	-423	450	7-10	84,300	2.1-2.3	Inert
Phenolic (laminated)	RM 105		300	2.0	850,000	1.35	Inert
Viton		-100					
Armalon		-300					
Rulon		-300					
Graphitic carbon		-300					

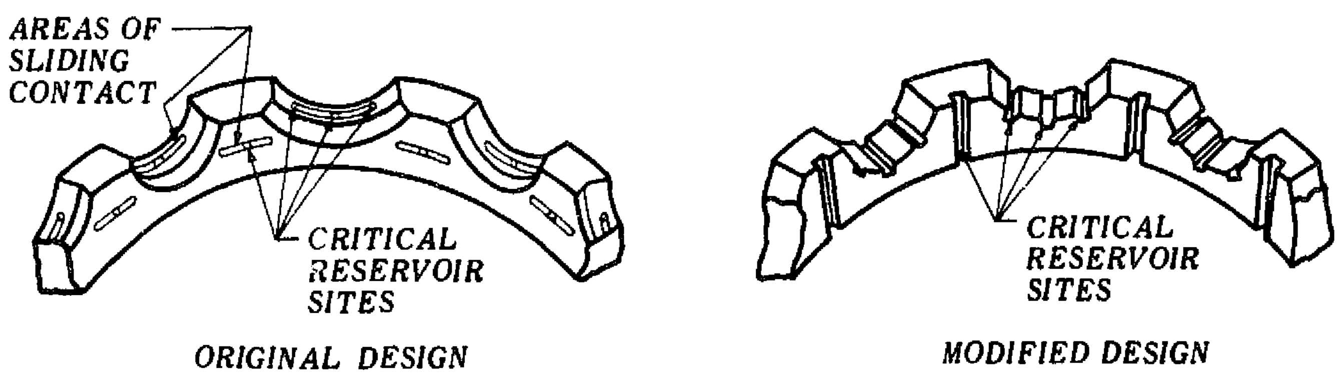


FIGURE 14. SECTION OF MACHINED BEARING RETAINERS WITH SOLID LUBRICANT SLOTS

Section IV. LUBRICANTS

1. Description of Classes

a. Liquids

Liquid lubricants are usually used for rolling-element bearings when it is possible to supply the lubricant conveniently from a reservoir at a rate able to sustain good bearing performance. A liquid lubricant, in addition to doing its principal job of preventing or minimizing wear in the bearing, may also remove heat and contamination from the bearing. Currently, the selection of lubricants for rolling-element bearings is largely empirical and based on extrapolation from past experience. Simple selection by viscosity level, as is possible for full hydrodynamic bearings, is usually not sufficient for rolling-element bearings which operate in the elastohydrodynamic regime.* In addition, the environmental demands of aerospace applications frequently lie outside the capacity of lubricants developed for industrial use. Conventional petroleum lubricating oils are not automatically ruled out for aerospace use, but synthetic oils or specially refined petroleum products must often replace the conventional oils. The following discussion lists the major classes of special and synthetic oils together with some comments on their properties and general limitations. Choice of a specific oil for a specific need will depend on these considerations and others to be discussed later.

(1) Petroleum Oils — Conventional Refining. Petroleum oils prepared by conventional refining usually are restricted to a temperature range from about -20° to 300°F. Not all oils will span the entire range. Napthenic oils will often reach the lower limit naturally, but paraffinic oils usually have a pour point (solidification temperature) of 0°F or even higher due to crystallization of wax components. The upper temperature limit can be due to oxidation, volatility, or both. Of course, in a nonoxidizing environment, this limitation is removed. Also, since petroleum oils are always a mixture of molecules of different sizes, evaporation rate for any given temperature and pressure will usually diminish with time, and viscosity will correspondingly rise with time.

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* The significance of elastohydrodynamics is discussed in Section V.

(2) Petroleum Oils — Super-Refining. The term super-refining has been given to processes in which conventional petroleum oils are deep dewaxed and otherwise treated to produce oils with a temperature range of use from -65° to roughly 500°F. The lower limit is set by high viscosity, and the upper limit is a function of environmental conditions. Some super-refined oils have been tested as jet-engine lubricants at bearing temperatures of over 400°F, but the sump temperature for the bulk of the oil must be kept much lower to achieve reasonable life for the oil. These oils are thermally stable to about 700°F, but they are limited depending on conditions by oxidation stability, volatility, or catalytic deposits formation. Super-refined oils are much more expensive than are conventionally refined oils, and no large market has developed for them. Accordingly, they are in limited supply. Esso and Kendall Refining are sources of supply.

(3) Synthetic Esters. The principal use for synthetic esters today is in jet-engine oils. Nearly all military and commercial jet aircraft in the free world use ester-base engine oils. These oils have good boundary lubricating properties (comparable to petroleum oils), and they have a wide temperature range of use. Depending on the chemical composition of the ester, and consequently its temperature range, esters can be used from -65° up to about 450°F maximum. The high-temperature limit is based on use of antioxidant additives and limited exposure of the ester to that temperature while circulating from sump to bearing and back again. Different viscosity levels of esters may be used depending on whether the engine is a jet thrust device or a turboprop. Higher viscosity is usually considered important to lubricate the reduction gears in a turboprop engine in addition to the main shaft bearings. The higher viscosity at high temperatures for turboprop oils is usually achieved at the expense of some low-temperature fluidity.

The synthetic esters based on organic acids are usually subclassed into diesters, triesters, and polyesters. The practical significance of this classification is that the polyesters are usually somewhat more viscous at any given temperature, but it is possible to select representatives of each subclass which overlap each other in properties for most of their temperature range. U. S. military jet-engine oil specifications are based on performance rather than on composition.

Ester lubricants, either military-specification or otherwise, are available from many major oil companies, some chemical companies, and some specialty lubricants companies.

(4) Silicate Esters. In contrast with the esters of organic acids described above, silicate esters are based on silicic acid. The silicate esters also differ functionally, generally being poorer lubricants in sliding friction. Silicate esters are generally used only as low-temperature hydraulic oils. Oronite Chemical Company is the source of supply.

(5) Polyphenyl Ethers. This class of fluids covers a fairly wide range of viscosity, depending on whether the basic molecule contains four, five, or six phenyl rings. These fluids have outstanding thermal and oxidation stability, and are usable in air to 700°F in some circumstances. However, they do have appreciable volatility at that temperature. Otherwise, they suffer from a rather steep viscosity-temperature slope compared to most mineral oils, esters, or silicones, and they become very viscous below 100°F. Whether that viscosity is excessive depends on the application, but the limitation has prevented this class of lubricant from being used as jet-engine oils, where low-temperature fluidity is necessary for engine starting.

The polyphenyl ethers are outstanding for resistance to ionizing radiation, being equalled only by a few high-phenyl silicones among the lubricants.

The polyphenyl ethers have somewhat poorer boundary lubrication properties when compared with esters and mineral oils, although the reasons for this behavior and the circumstances in which it occurs are not fully understood.

Sources of supply are Monsanto Company and Dow Chemical Company.

(6) Silicones. The silicones, as a class, are polymers of alkyl or aryl siloxanes. Those silicones in which the alkyl groups are all methyl (dimethyl silicones) are obtainable in a very wide range of viscosities since the viscosity can be controlled by degree of polymerization of the basic building block. The silicones are also distinguished by the widest temperature range of use available among lubricant fluids. Dimethyl and some phenylmethyl silicones can span a range from -100° to >500°F. Substitution of some of the methyl groups in silicones by phenyl groups has produced a series of fluids with properties different from those exhibited by the dimethyl silicones. Generally speaking, the substitution of phenyl groups increases the oxidation resistance, but at the expense of less favorable viscosity-temperature characteristics. The very highly phenylated silicones have excellent oxidation and thermal stability but are also somewhat more volatile than are comparable dimethyl silicones, so the benefit may not be usable, depending on the environment.

Dimethyl and low-phenyl silicones are more susceptible to damage by ionizing radiation than are petroleum oils. However, radiation resistance increases with increasing phenyl-group content until the highest phenyl silicones are comparable to the polyphenyl ethers.

Silicones containing no chlorine or fluorine have rather poor lubricating ability for many metal bearing couples in sliding friction. These oils may be very satisfactory in lightly loaded rolling-element bearings, but the imposition of load with subsequent sliding will usually sharply reduce life of bearings made from steel alloys.

(7) Halogenated Silicones. In the last ten years, chlorophenyl and trifluoropropyl silicones have become available. These fluids overcome much or all of the poor boundary lubrication ability of methyl and phenyl silicones. Better lubricating ability in sliding situations has been achieved at some small sacrifice in high and low temperature limits of use. Halogenated silicones are also slightly denser than are their nonhalogen counterparts.

Silicones are available from Dow Corning, General Electric, and to a limited extent, Union Carbide.

(8) Perfluorinated Fluids. Various attempts have been made to synthesize lubricating fluids which have the chemical inertness of fluorocarbon solids, e.g., polytetrafluoroethylene, but which at the same time have a useful liquid range. Recently, a group of perfluoroalkyl ethers with varying molecular weight (and hence viscosities) has been made available commercially. These fluids have a lower limit of about 0° to -40°F, depending on viscosity grade. The highest temperature range of use is about 600°F for thermal and chemical stability, again depending on circumstances. Some metals catalyze decomposition above 600°F, and the lower-molecular-weight species become excessively volatile at 600°F. Some compromise must be made between low- and high-temperature limits for any single fluid.

The perfluoroalkyl ethers have very good boundary lubrication properties for all types of bearing materials within their temperature range. They are also much denser than most other fluids (specific gravity ≈ 2 at room temperature).

Sources of supply are DuPont, Petroleum Chemical Division, and Bray Oil Company.

b. Greases

Lubricating greases are usually used for rolling-element bearings when it is impossible or unnecessary to supply lubricant continuously from a reservoir. Greases are manufactured by thickening a fluid to semi-solid consistency with a minor proportion of a solid, colloidally divided thickener. There are many successful grease thickeners, including various soaps, high-temperature organic materials (e.g., some dyes, pigments, arylureas), as well as inorganic materials (e.g., modified clays, carbon black, and others). The choice for a given application will often depend on the temperature range expected. A grease is in effect its own reservoir. In rolling-element bearings which rotate at ordinary or high speeds, the grease usually should function by being pushed out of the path of the rolling elements and their retainer after the first few revolutions of the bearing. Thereafter, the grease should maintain its position surrounding these elements and leak or "bleed" fluid at a slow rate to supply the needs of the bearing. The thickener need not have any lubricating ability of its own, so long as it is not actually abrasive. However, it is also possible to select grease thickeners which do have some function as solid lubricants, if necessary.

Greases can be manufactured from any of the lubricating fluids described in the preceding section. Therefore, the lubricating properties of the greases will basically depend on the choice of fluid. However, the thickener should be stable under the environmental stresses of heat, mechanical shear, radiation, or whatever. Also, it should allow the fluid to bleed at a rate neither too slow nor too fast.

Greases can act to some extent as shields to prevent access to the interior of the bearing of dirt or other outside material. However, if shielding is very important, the bearing should have suitable seals or shields to provide the first line of defense.

Greases cannot dissipate heat as well as can circulating fluids. Also, they are always more viscous than the base fluids themselves. Therefore, the mechanical consistency of the grease should be chosen to permit it to support its own weight when pushed out of the path of the rolling elements and retainer. If it does not support its own weight, it will slump back into that path and be churned with consequent higher power loss, heat generation, and damage to the grease itself. It is possible for a bearing to be destroyed by overheating due to an excessive amount of grease or the use of a poorly chosen grease. It is particularly important not to overpack a grease-lubricated bearing for this reason.

Greases are characterized for consistency by ASTM penetration* numbers, rather than by viscosity. Rolling-element bearings usually require a grease with a penetration of between 200 and 300. However, the penetration measurement is performed at 77°F, and tells nothing about what consistency a grease will have at other temperatures. Also, greases are non-Newtonian materials whose apparent viscosity under shear depends at least on the shear rate, and often on the past shear history of the material.

Aerospace greases may be obtained from several of the major oil companies and some specialty suppliers. The synthetic fluid and high-temperature specialty greases are usually highly individual in character and must be selected carefully. Air Force-Navy Aeronautical (ANA) Bulletin No. 275f is a useful guide to military specification lubricants, including greases.

c. Solid Lubricants

Solid lubricants are used instead of liquid lubricants or greases only when there is some specific reason why the latter are inconvenient or impossible to use. A primary reason might be use at very high temperatures (beyond the capacity of known fluids), or a requirement that no organic vapors be emitted from the lubricated bearing. There are many different solid lubricant materials available for use, and there are also several different ways of using them. Materials most extensively tested or used in various forms are:

Graphite
Molybdenum disulfide and related dichalcogenides, e.g., WSe_2
Fluorinated polymers

Polyimide resins
Lead oxide (PbO)
Calcium fluoride-barium fluoride eutectic
Various metals (gold, silver, tin, indium).

These solids are usually applied to the bearing before assembly, but some work has been done on continuous supply of solid lubricants as powders carried by a gas stream. When the lubricants are applied before the bearing is assembled, they may be applied as films burnished into the bearing or painted

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* Penetration in millimeters of a standard-sized cone under prescribed conditions.

and baked on with auxiliary binders. Otherwise, they may be packed in pockets or reservoirs in the bearing retainer, or they may form the retainer itself. The variety of ways of using solid lubricants is greater than those for either liquid or grease lubricants.

The principal limitations to the use of solid lubricants is the fact that they usually cannot be resupplied. Once a film of solid lubricant has been worn through, rapid failure may occur. There have been many problems associated with durability and reliability of baked films, particularly when they are used in rolling-element bearings. The use of retainer reservoirs or self-lubricating retainers was stimulated by early failures with bonded films.

Although the total number of solid lubricants investigated is very large, the most commonly used ones are molybdenum disulfide (often in conjunction with a minor proportion of graphite) and polytetrafluorethylene (PTFE). PTFE itself is seldom used as a "pure" material but is best employed in rolling-contact bearings in composite form with other lubricating solids or reinforcing solids which give it more dimensional stability and strength.

2. Environmental Effects on Lubricants

The principal environmental stresses on aerospace bearings are likely to be extremes of temperature, low ambient pressure, ionizing radiation, or some combination of these. The following discussion states some generalizations about how specific lubricants respond to these stresses and then cites some examples of experience in selection and development of successful lubricants.

a. Ionizing Radiation

Lubricants are likely to receive large accumulated dosages of ionizing radiation only in prolonged orbit in space, or in connection with nuclear-powered rockets or aircraft. One estimate suggests that the amount of annual radiation on the surface of a vehicle in polar orbit at a 2300-mile altitude is 2×10^9 ergs per gram (carbon). This is equivalent to 2×10^7 rads, an amount which is beginning to be significant in terms of damage to organic lubricants. However, any lubricant would be shielded to some extent by the metal of the bearing in which it is being used, as well as by any surrounding hardware. Such shielding would cut the actual radiation absorbed by the lubricant quite drastically, since much of that radiation is charged particles such as electrons and protons and is therefore quickly absorbed by the surface. Therefore, ambient radiation in space is not often likely to be a hazard to lubricants. If the application is known to carry a radiation hazard, however, then the order of resistance of lubricants to radiation should be considered (Table VIII).

TABLE VIII. RADIATION TOLERANCE OF LUBRICANTS

Lubricant Material	Rads Absorbed Dose, Serious Damage
Dimethyl silicones	10^7
Perfluorinated fluids	10^7
Synthetic esters	10^8
Petroleum oils	5×10^8
High-phenyl silicones	$2\text{-}3 \times 10^9$
Polyphenyl ethers	3×10^9
Inorganic solids	$>10^{10}$

Greases made from the various organic fluids may be more sensitive to radiation if the thickener is damaged, but several thickeners having radiation resistance to match the fluids are available [12].

b. <u>High Temperature</u>

Lubricants for high temperatures can be divided roughly into two groups. Liquids and greases, at least those of organic origin, are restricted to use below temperatures of about 700°F. Above that temperature, only liquid metals on salts and solids are available. At present, solids known to be useful are restricted to about 1500°F. In either group, the practical temperature limit for any given lubricant is affected by the nature of the ambient atmosphere. If oxygen is present, the lubricant may be restricted to a service temperature of 100°F (or more) less than the limit set by its thermal stability. If the bearing must operate in vacuum, the volatility of the lubricant may limit service temperature to a much lower level than would be possible at atmospheric pressure. However, the volatility consideration is also affected by whether the lubricant is supplied from a circulating system or applied only once, as with greases.

Table IX gives some approximate upper limits for the organic lubricants in an inert environment or an oxidizing atmosphere.

TABLE IX. HIGH-TEMPERATURE LIMITS FOR FLUID LUBRICANTS

	Atmosphere (°F)	
	Inert	Oxidizing
Conventionally refined mineral oils	400-500	250
Super-refined mineral oils	500-600	450
Synthetic esters	500	350-400
Polyphenyl ethers	700	500
Chlorosilicones	650	500
Fluorosilicones	550	400
Silicones	650	400-600
Perfluoroalkyl ethers	700	550-600

In general, the maximum use temperature recommended in Table IX must be reduced by about 100°F if the lubricant is employed in the form of a grease. The reason for this is that grease is usually applied only once, and at high temperature the failure mode for the grease is likely to be volatility. If too much of the liquid phase of the grease evaporates, the grease may become excessively stiff or may dry out and cease to "bleed" the rest of its oil.

In addition, the recommended temperatures must be lowered if the life of the lubricant needs to be more than a few-hundred hours.

The most common solid lubricants are molybdenum disulfide and graphite. These materials have high thermal stability. In an inert atmosphere the former is stable to about 2000°F and graphite is stable to 6000°F. However, oxidation limits the extended use of molybdenum disulfide to about 650°F, and of graphite to about 800°F. Graphite has further limitations in that it loses some of its effectiveness as a lubricant under conditions where its adsorbed water vapor and gases are removed by low pressures or high temperatures.

Other solid lubricants such as other metal sulfides and selenides, lead oxide, barium fluoride, etc., each have a particular temperature limit and in some cases they also have a minimum temperature for proper operation. That is, they may show excessive friction at lower temperatures and only perform well in a limited range. Because there has been limited success with use of solid lubricants in rolling-element bearings, further observations on temperature will be limited to the section on examples of promising performance.

c. Low Temperature

Liquid lubricants or greases based on liquids are largely limited to a minimum temperature of -100°F. There are liquids whose freezing point or pour point is below -100°F, of course, but in general these materials are likely to be too volatile for use at higher temperatures. Table X shows approximate minimum use temperatures for various common lubricant fluids.

TABLE X. LOW-TEMPERATURE LIMITS FOR FLUID LUBRICANTS

	Minimum Use Temperature (°F)
Conventionally refined mineral oils	
Naphthenic	-25
Paraffinic	0 to 20
Super-refined mineral oils	-65
Synthetic esters	
Mil-L-7808	-65
Mil-L-23699	-40
Silicones	-100 to -65
Polyphenyl ethers	50-100
Perfluoropropyl ethers	-40

These temperatures are approximate because excessive viscosity is usually the limiting property. The actual minimum use temperature for any specific fluid and situation will depend on the torque. The drive must have sufficient power to overcome the bearing losses. Greases are more viscous than are the liquids on which they are based, so the torque may be higher for grease-lubricated bearings than for liquid-lubricated ones. However, the same general temperature limits should be used for greases as for liquids. Properly selected greases for rolling-element bearings should channel after the first few revolutions. Thereafter, the viscous drag in the bearing should primarily be that of the liquid because the bulk of the grease has been removed from the path of rotation. The only situation in which torque for a grease would be much larger than that for the oil on which the grease is based is when the bearing has been freshly packed and not rotated until the bearing reached the low temperature. Military-specification greases such as Mil-G-23827, Mil-G-25013, and others must pass a low-temperature torque test in which a freshly packed 204

Conrad-type bearing is soaked at the test temperature and then turned. Break-away and running torque are measured.

d. Vacuum

At the outset of the space program, most space hardware designers assumed that conventional oils and greases would be completely unsuitable because of excessive evaporation rates. They further assumed that solid lubricants would be necessary. A considerable amount of test work and experience since then has shown that this assumption is far too pessimistic. Depending on the circumstances, grease lubrication may be preferred to solids because grease-lubricated bearings are often longer-lasting and more reliable [13-15].

True, oils and the liquid constituents of greases do evaporate in vacuum; however, it is possible to select several types of oils and greases which were developed for high-temperature use, and thereby provide oils with minimum volatility for a given viscosity [12]. Further, it is possible to minimize natural evaporation rates by providing shielding around the lubricant. Through appropriate combination of design and lubricant selection, it has been possible to run grease-lubricated bearings in a vacuum of about 10^{-9} torr for well over two years [15]. In other studies in which test life was not so prolonged, grease-lubricated bearing performance has been relatively quite successful [16].

It is worth noting that even solids, which are ordinarily thought of as nonvolatile, may have appreciable evaporation rates in high vacuum [17]. As with greases, this means that some thought must be given to the way in which solids are to be used as lubricants. However, for missions of moderate length, solid lubricants have already been proved to be useful.

For any new machinery design problem involving lubrication in high vacuum, the designer should examine all the possibilities of speed, load, shielding, mission length, etc. before selecting the bearing-lubricant system. "Conventional" grease lubrication may be the most reliable approach possible. Resort to solid-lubricated systems may not be necessary. However, if a solid lubricant is necessary, then the choice of lubricant and lubrication technique must be made very carefully to achieve desired reliability. This choice necessarily involves special design of bearings specifically for use with solid lubricants.

e. Contact with Rocket Propellants

Most organic lubricants are not compatible with rocket fuels. Particularly bad are liquid oxygen and nitrogen tetroxide; these strong oxidizing agents can cause explosions through contact with most petroleum oils, synthetic fluids, and greases. One approach has been to use rocket fuels themselves as lubricants [18]. Another approach has been to look for materials resistant to reaction with rocket fuels. Some test work done with the perfluoroalkyl ether synthetic fluids and greases based on these fluids using a fluorocarbon polymer as the thickener has shown that these materials are resistant to reaction [19].

3. Combinations of Environmental Variables

The preceding description of effects of environment on lubricants has discussed them one at a time. In practice, of course, some combination of variables exists. Fortunately, the extremes of environment are not likely to be combined all at once. Space applications usually will not include extreme high temperature except during reentry or in close proximity to a rocket motor or a nuclear reactor. There is no vacuum associated with jet-engine bearing lubrication. In many cases, the "ordinary" variables of bearing selection and lubrication may be more important than are the special variables associated with space use. Speed, load, and power requirement must not be forgotten merely because the bearing will operate in vacuum or be exposed to radiation.

Section V. BEARING LUBRICATION

Although lubricants are known to be necessary to insure reasonable performance of a rolling-contact bearing, the actual purpose of the lubricant is often obscure to the design engineer. Generally, it is accepted that a lubricant is used to minimize friction and wear (to permit the bearing to run longer and quieter) and for cooling; however, the mechanisms by which the lubricant performs these functions have usually gone unnoticed. The special demands of many space applications have forced designers to be more concerned with the function of lubricants, because in some cases those special demands have revealed shortcomings in the lubrication methods of more traditional industrial practice.

For all but the relatively high-speed bearing applications, the selection of lubricants and lubrication procedure is based mostly on performance experience, and on the special demands of the environment (high temperature, low temperature, vacuum, oxidizing conditions, salt spray, etc.).

There is no well established set of engineering principles comparable to the hydrodynamic journal bearing design charts based on fluid flow equations from which lubrication can be determined for most rolling-contact bearing applications. Therefore, most critical applications require proof testing before the lubrication design is put to use. Sound principles have been developed through experience for specific bearing lubrication situations (e.g., grease lubrication) and these are summarized in this section.

For high-speed bearings, there are techniques (recently developed) for selection of lubricants and bearing geometry as a system. These techniques are based on the concept of failure by contact fatigue, and the known ability of rolling elements to develop fluid-film support. The fluid-film support maintained between balls and races (or rollers and races) is hydrodynamic, and since the recognition of this lubrication mechanism in the late 1950's, the lubrication process has been known as "elastohydrodynamics."

The elastohydrodynamic lubrication design charts (Figures 15 through 18) can be used to select lubricant viscosity and bearing size for a given set of conditions of speed, load, and surface roughness. A rough approximation of the lubrication condition for a given set of parameters can be obtained from Figure 15 [20], where unfavorable and favorable operating zones are shown for various pitch velocities and lubricant viscosities. From Figure 15, therefore, one can estimate whether the bearing speed is high enough to provide elastohydrodynamic lubrication. If such is the case, then Figures 16 [20], 17 [21], and 18 [21] may be used to optimize bearing and lubricant selection.

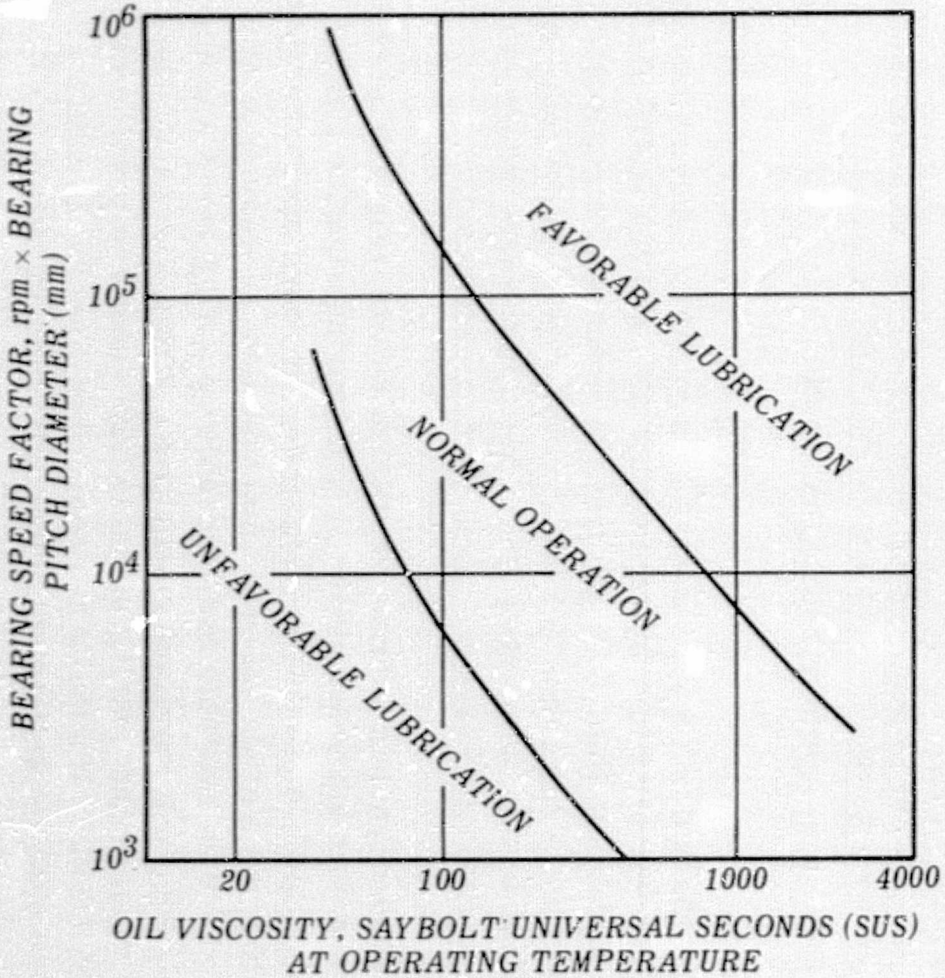


FIGURE 15. ELASTOHYDRODYNAMIC LUBRICATION PREDICTION CHART FOR BALL BEARINGS

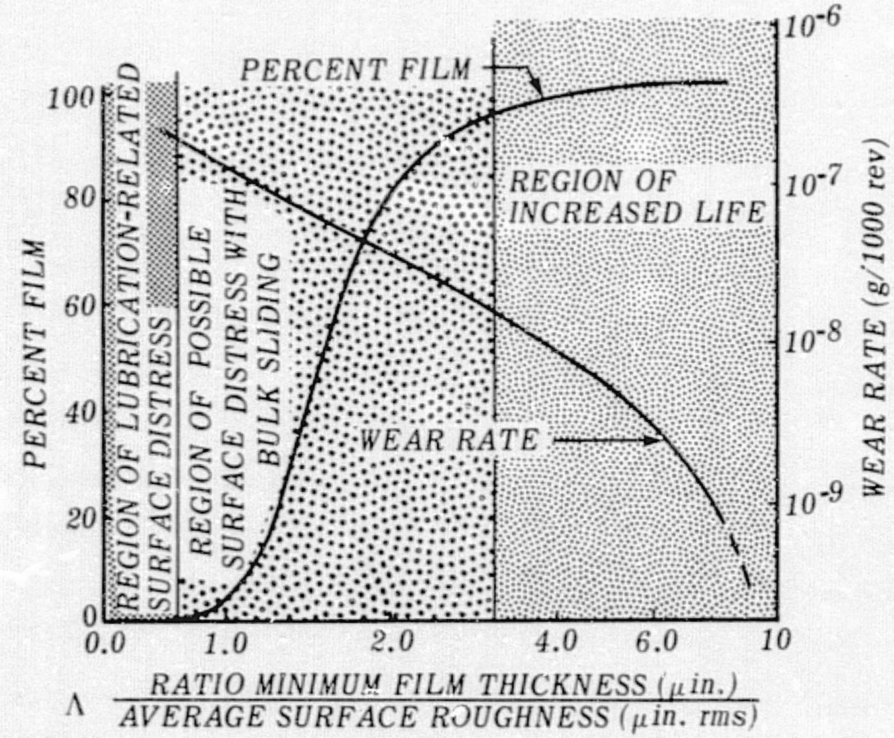


FIGURE 16. COMPOSITE PLOT OF EHD LUBRICANT FILM THICKNESS AND MICROWEAR

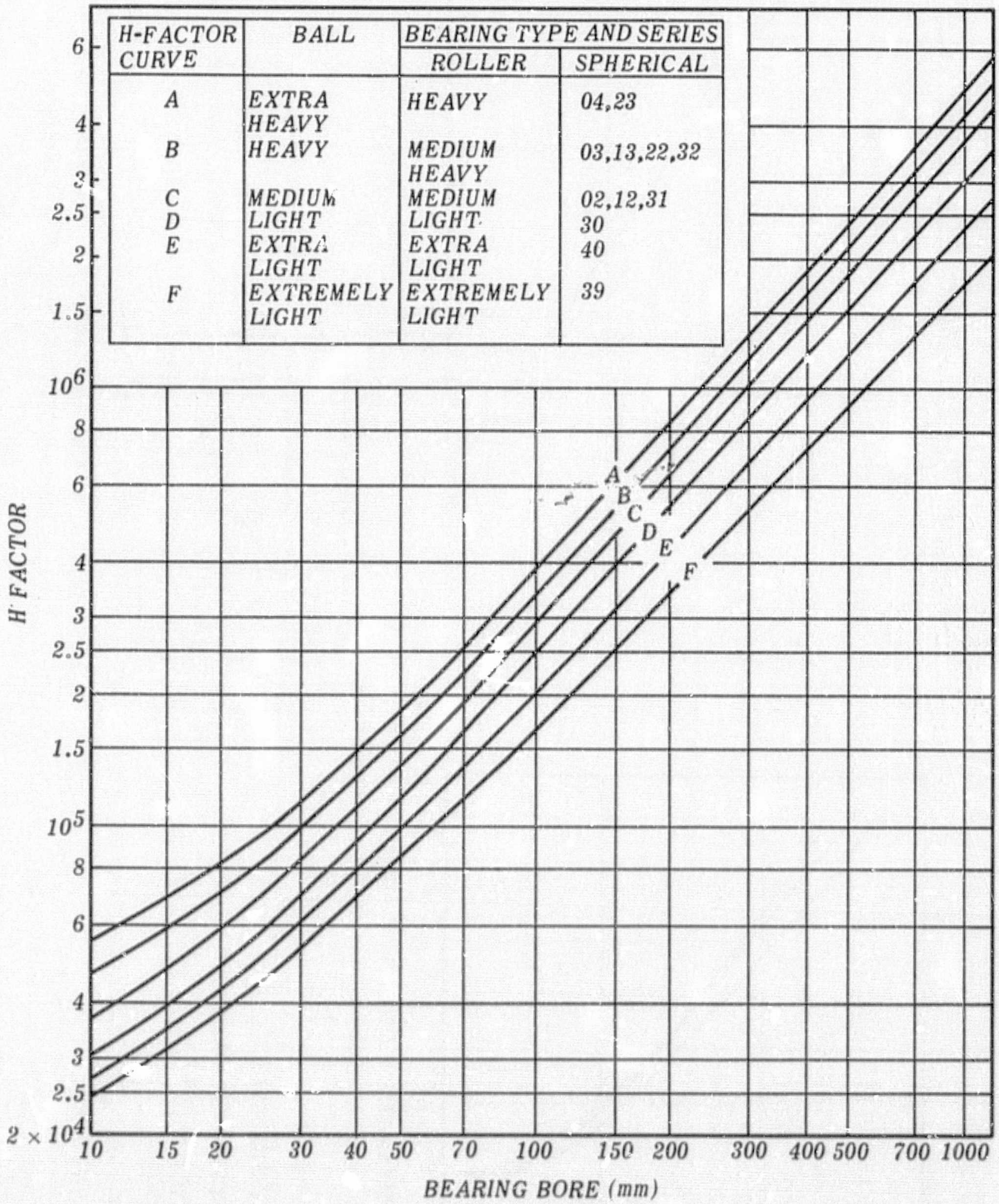
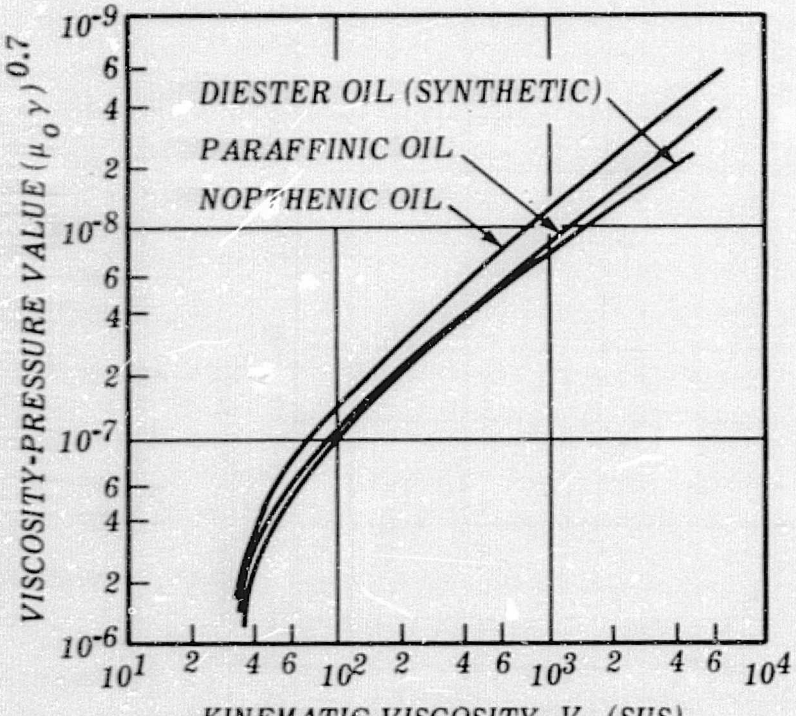
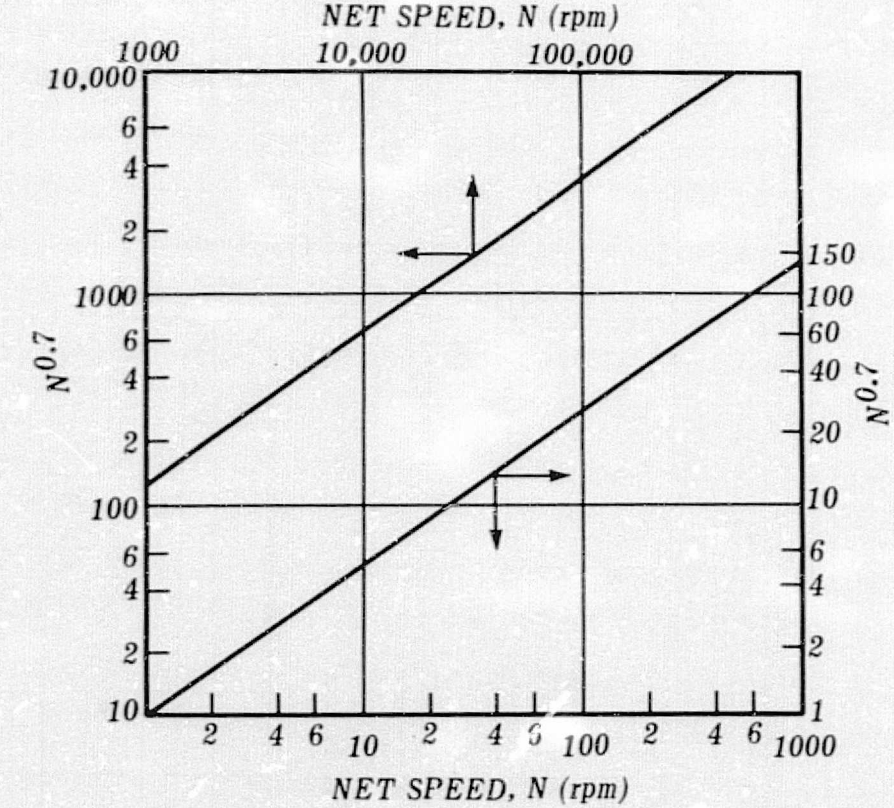


FIGURE 17. H-FACTOR AS A FUNCTION OF BEAR BORE



(a)

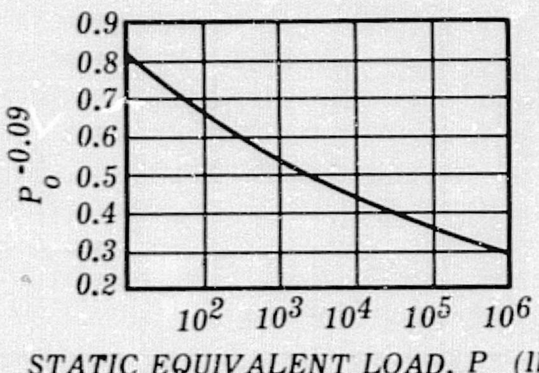


(b)

(a) VISCOSITY-PRESSURE VALUE $(\mu_o \gamma)^{0.7}$ AS A FUNCTION OF KINEMATIC VISCOSITY.

(b) SPEED FACTOR $(N^{0.7})$ AS A FUNCTION OF NET SPEED.

(c) LOAD FACTOR $P_o^{-0.09}$ AS A FUNCTION OF EQUIVALENT LOAD.



(c)

FIGURE 18. LOAD FACTOR, SPEED, AND VISCOSITY FACTOR TO BE USED IN OBTAINING LUBRICATION PARAMETER

The design charts are based on the film thickness factor Λ given by

$$\Lambda = H\left(\mu_o \gamma\right)^{0.7} N^{0.7} P_o^{0.09}.$$

H can be obtained from Figure 17; $\left(\mu_o \gamma\right)^{0.7}$ can be obtained from Figure 18 (a); speed factor $N^{0.7}$ can be obtained from Figure 18 (b); and load factor $P_o^{0.09*}$ can be obtained from Figure 18 (c). The film-thickness factor Λ then is used with Figure 16 to estimate the percent film support between the load supporting rolling elements. As can be seen from Figure 16, it is best to try to maintain Λ well above a value of 2.0. (A numerical example will be provided here.)

*Some doubt exists about the validity of this expression for load factor [22]. For very high bearing loads (producing stresses in excess of 250,000 psi), the accuracy of these design charts diminishes.

Section VI. HIGH-SPEED, HIGH-TEMPERATURE BEARINGS AND LUBRICATION SYSTEMS FOR HIGH-SPEED BEARINGS

Perhaps the most consistent limitations in bearing technology are those associated with speed and, often concomitantly, with temperature. For example, the power-to-weight ratio of turbine engines is limited by the speed the turbine can rotate. Therefore, efforts are continuously being made to increase the speed capabilities of bearings. Current interest in bearing speed capabilities is in DN values (bearing bore diameter in millimeters times shaft speed in rpm) of the order of 2.5 to 3×10^6 DN or higher. For high-speed situations (DN $> 1.5 \times 10^6$), bearing performance limitations due to fatigue or any gradual degradation of operation are approaching only secondary importance. For these cases, failures are more likely to be of a rapid and catastrophic nature and will presumably be caused by overheating and a loss of clearance between bearing components, or by some form of instability of the ball separator. Experience in using high-speed bearings is demonstrated by the curves of Figure 19 [23].

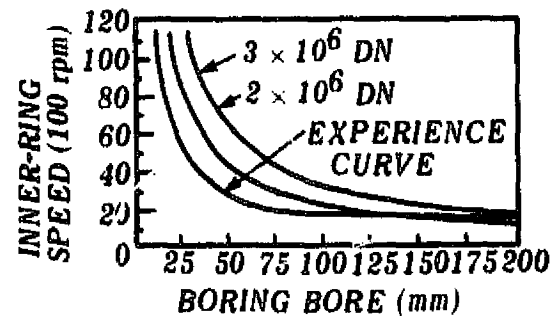


FIGURE 19. EXPERIENCE DATA SPEED-SIZE SPECTRUM FOR BALL BEARINGS

The design of high-speed bearing systems is currently an empirical science and special experiments are required to develop each new spindle concept. There are efforts under way to develop the state of the art of theoretical bearing dynamics to the stages where it can be applied to specific designs such as presented by Walters, et al,* but their efforts are still in the early stages of development. However, researchers have attempted to develop empirical laws from their studies. These few rules and the reported experience in high-speed bearing design can be useful in selecting lubrication systems and bearing designs for future aerospace applications.

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*AN INVESTIGATION OF THE BEHAVIOR OF ANGULAR CONTACT BALL BEARING UNDER DYNAMIC LOAD by C. Walters, et al., Final Report on Contract NAS [illegible]-21255, George C. Marshall Space Flight Center, to be released in 196[illegible].

1. Lubricant Mist Systems

One type of lubrication system often used for high-speed bearings is an air-oil mist system as shown in Figure 20 [24]. The mist concept is particularly applicable for high-speed bearings where some wastage of oil is not objectionable and a source of clean, dry air is available. The air-oil mixture is continuously fed into the bearing to provide cooling of the bearing and to inhibit contaminant particles from entering the bearing. By selection of lubricant-air mixture and supply rates, lubricant churning effects can be kept to a minimum and bearing speeds in excess of 600,000 DN can be obtained. An empirical relationship [25, 26] used for determining the flow rate and hence lubricant waste is given in Figure 21(a). The data for the figure were obtained on a 75-millimeter bore bearing operating at speeds to 13,000 rpm. The required minimum oil flow as a function of temperature is given in Figure 21(b).

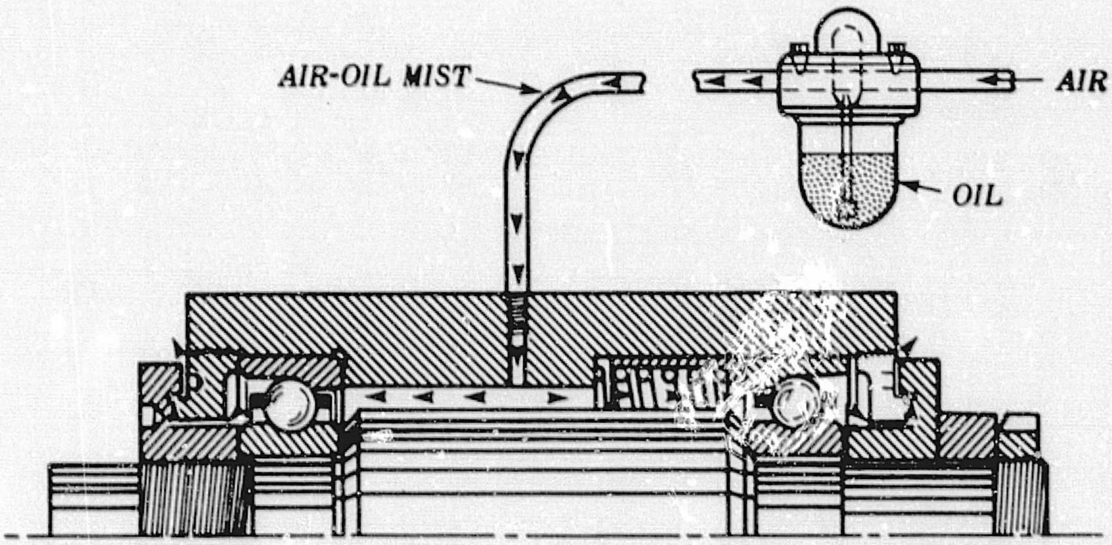


FIGURE 20. AIR-OIL MIST OR OIL-FOG LUBRICATING SYSTEM

2. Oil-Jet Lubrication

A commonly used concept for lubricating high-speed bearings is to employ a series of lubricant jets in the vicinity of the bearing as shown in Figure 22 [24]. The jets of oil are used to cool the inner ring of the bearing. One method is to focus the jet at the retainer-ring interface so as to draw the heat away from the bearing race without allowing the bulk of the lubricant to enter the bearing contact zone. Another concept is to allow for the lubricant to enter the bearing through small holes in the shaft and to be transmitted through

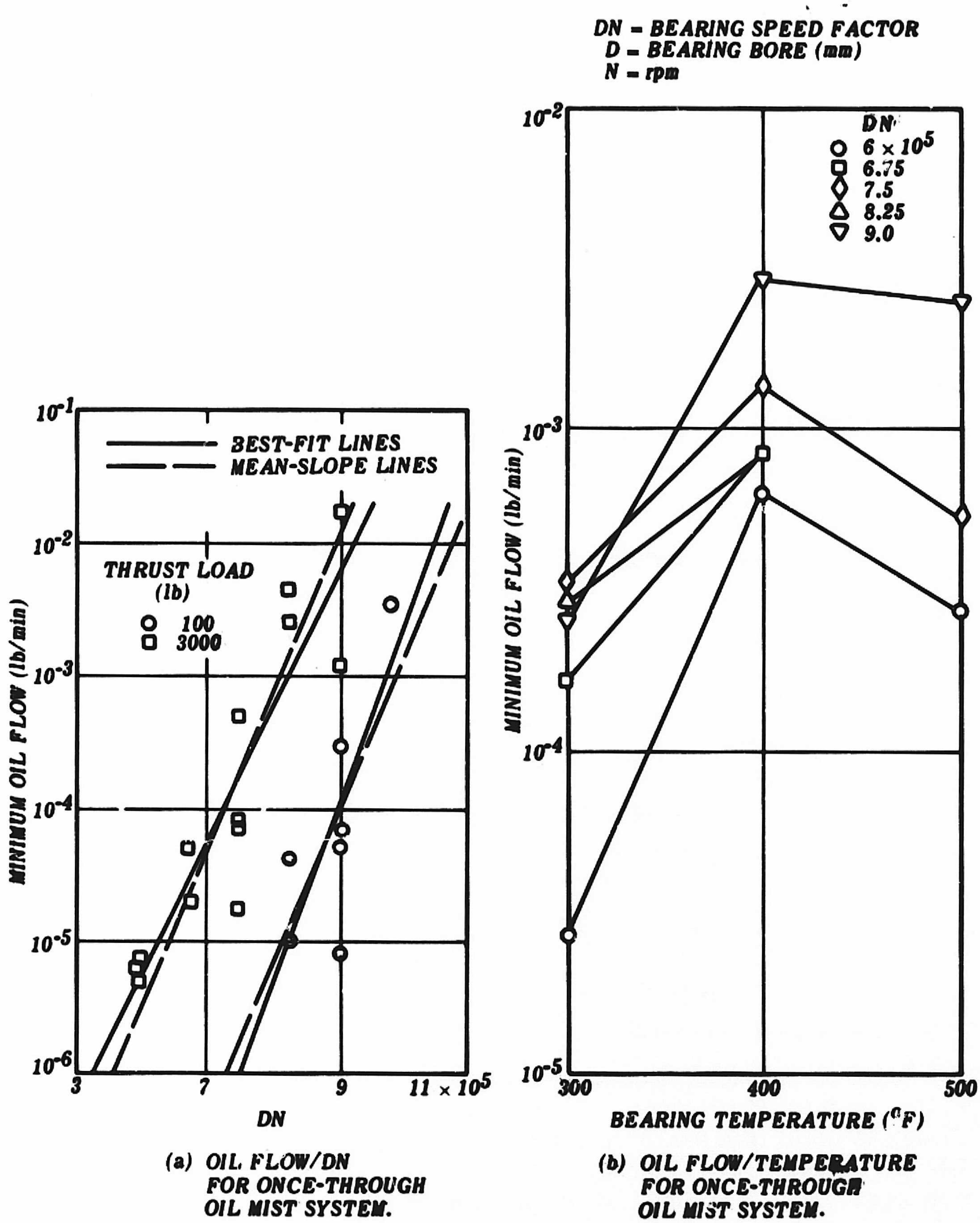


FIGURE 21. LUBRICANT FLOW REQUIRED FOR MIST LUBRICATION SYSTEMS

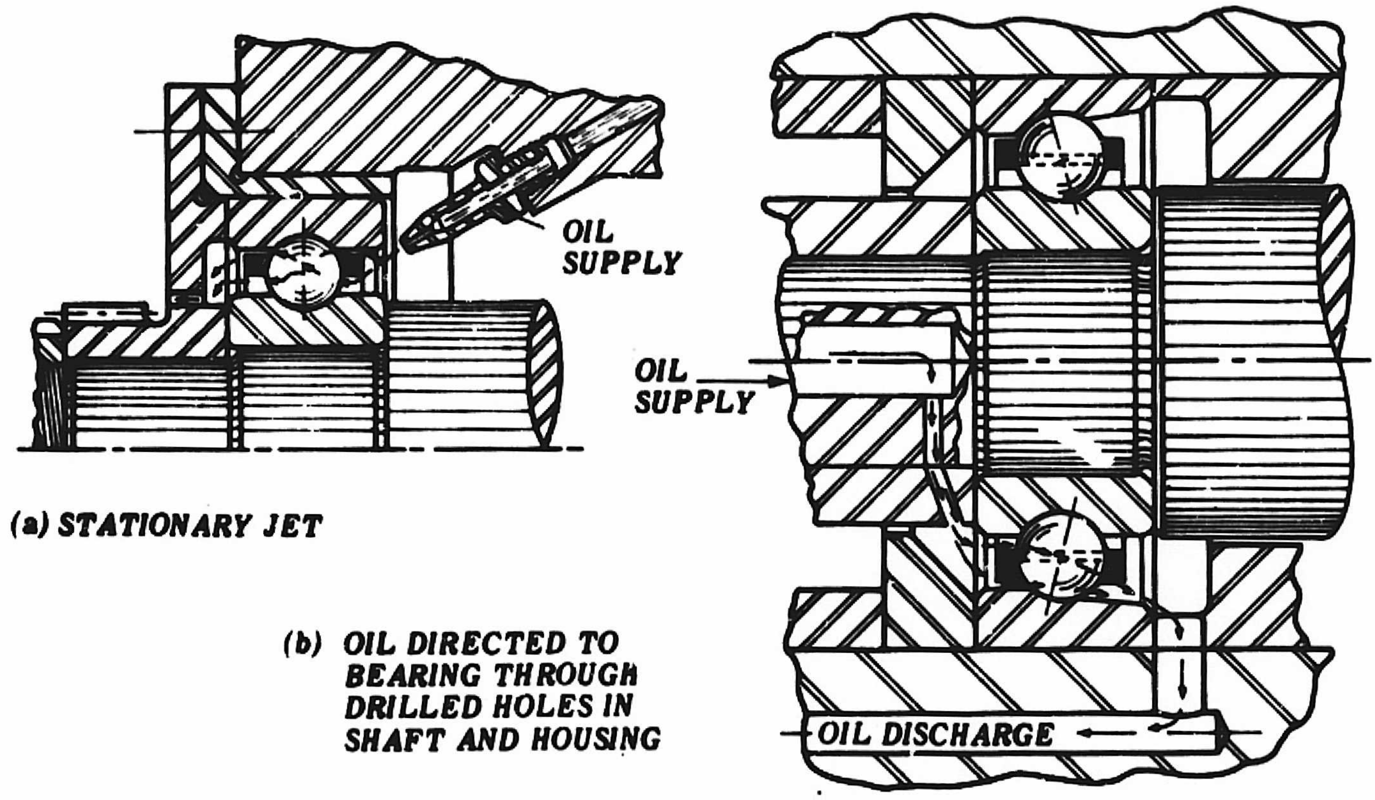


FIGURE 22. OIL-JET SYSTEM

the bearings. For very high-speed bearings multiple jets on both sides of the bearing are an effective way of removing heat. The basic problem with jet lubrication is association with heating due to churning of the lubricant.

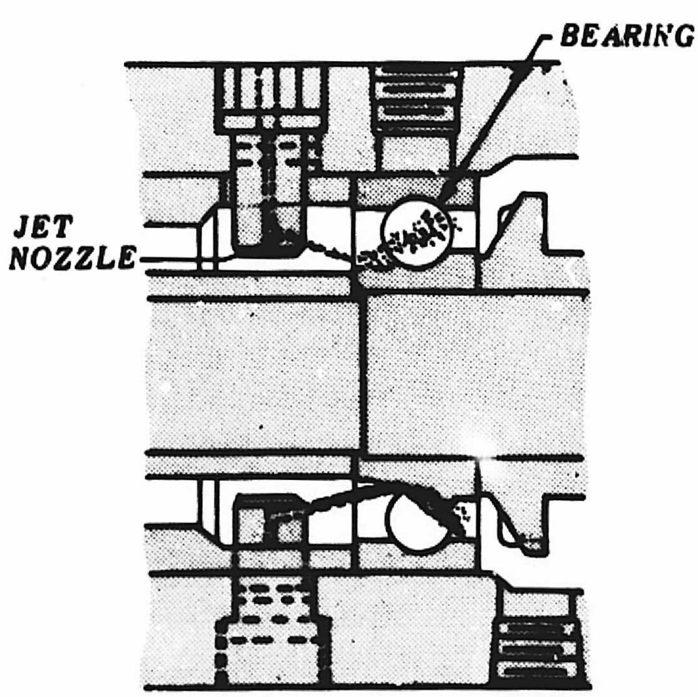
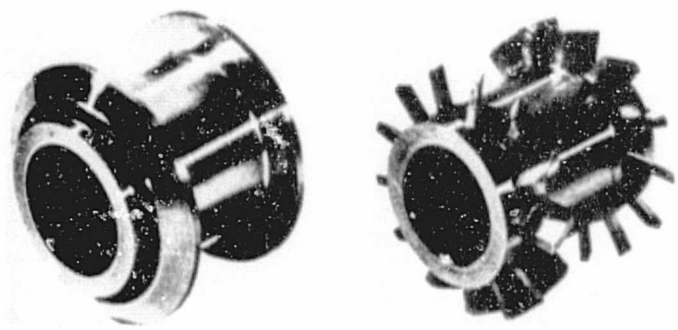
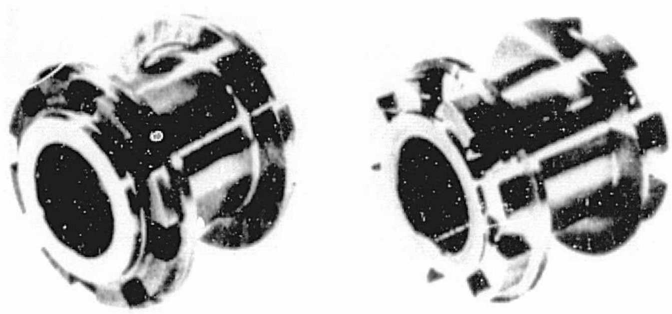


FIGURE 23. OIL-JET POSITION

Matt and Giannotti [27] recently developed an apparatus using a 20-millimeter bearing to operate at speeds to 90,000 feet per minute (1.8×10^6 DN). They used single jets located on the loaded side of each bearing at the inner ring as shown in Figure 23. Special slingers shown in Figure 24 were used to remove the oil from the bearing contact zone. The effect of slinger design on bearing temperature is given in Figure 25. Their experiments were apparently conducted with varying quantities of lubricant flow of the order of $1/2$ gram per minute. They found lubricant flow rate to be of only secondary

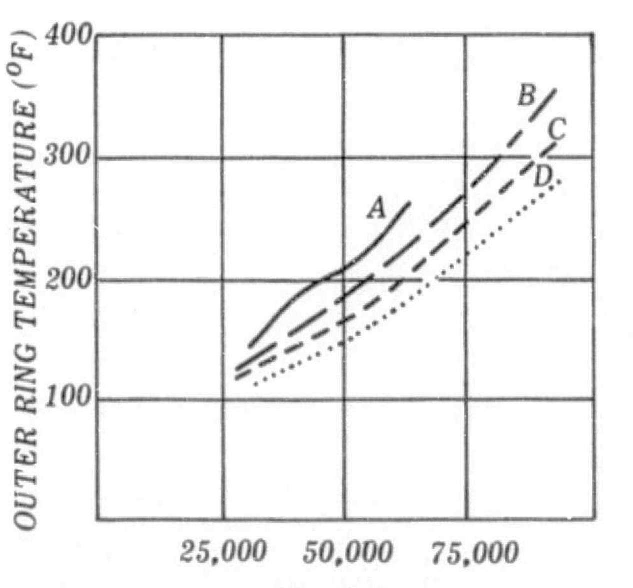


(a) STANDARD SLINGER *(b) FAN TAIL SLINGER*



(c) JET-OIL SLINGER *(d) JET-OIL SLINGER W/REDUCED SHOULDERS*

FIGURE 24. STANDARD SLINGER, FAN TAIL SLINGER, JET-OIL SLINGER, JET-OIL SLINGER WITH REDUCED SHOULDERS



A STANDARD SLINGER
B JET-OIL SLINGER
C JET-OIL W/REDUCED SHOULDERS
D FAN TAIL SLINGER

FIGURE 25. THE EFFECT OF SLINGERS

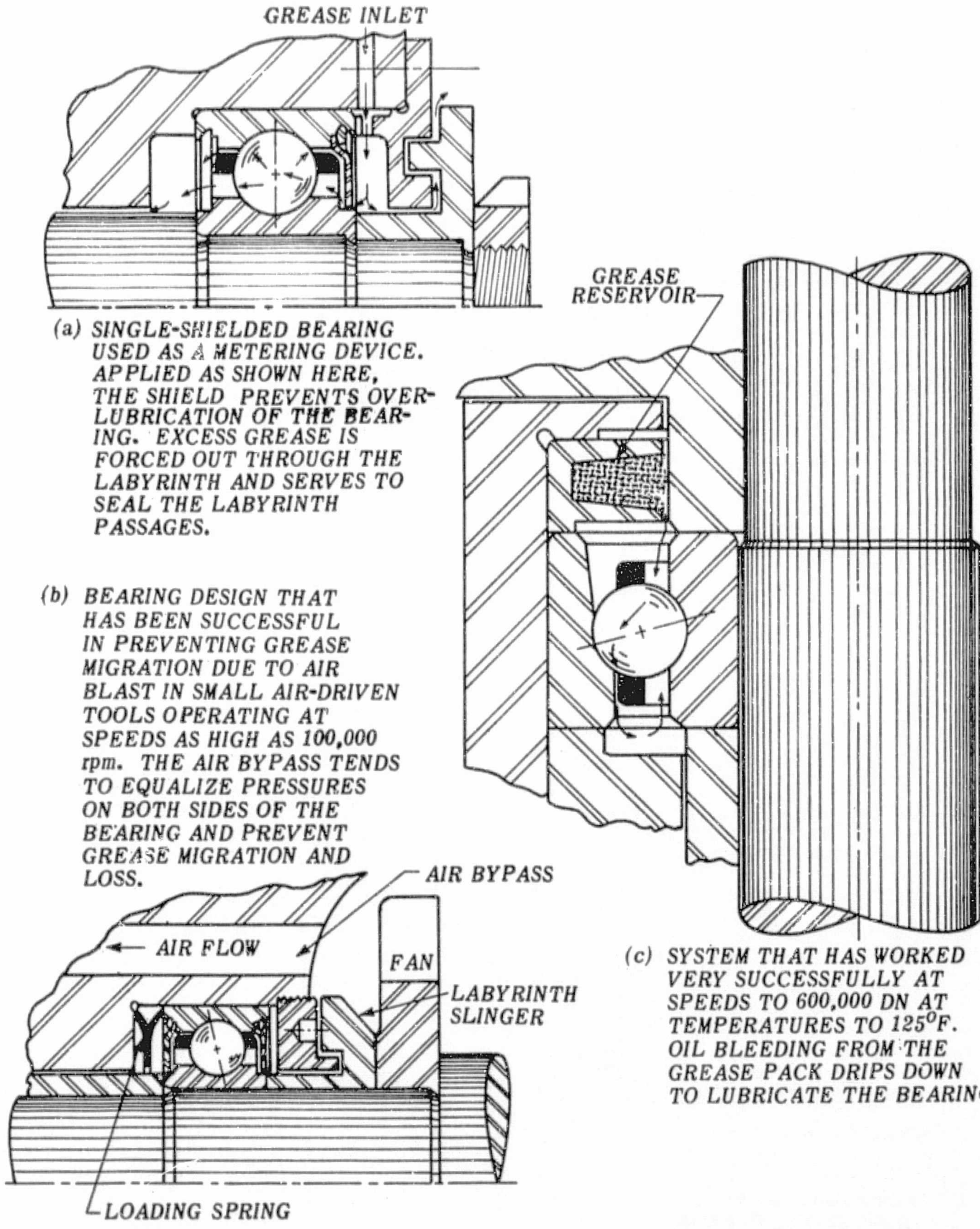
importance. The most important feature was the slinger system for ejecting the lubricant away from the bearing.

3. Grease Lubrication

Since greases are commonly used for lubrication of moderate-speed bearings it is natural to evaluate their potential for lubrication of high-speed bearings. It would be expected that a single grease packing would be unsatisfactory due to the lack of heat transfer capabilities. Nonetheless, studies with bearings operating at speeds as high as 10^6 DN have been made using greases as lubricants.

Rounds [28] reported good results at values of 500, 000 DN provided the temperature of the bearing was below 200°F. Some of Rounds' designs for high-speed grease lubrication are given in Figure 26.

Accinelli [29] explored the possibility of using grease-lubricated ball bearings at DN values from 1×10^6 to 2×10^6. For his experiments he developed an apparatus utilizing 20- to 25-millimeter ball bearings rotating at speeds to 100, 000 rpm. Although he obtained 106 hours of performance at 10^6 DN with an 18-pound axial load in one experiment, he observed that greases provided inadequate lubrication at the critical sliding surfaces (ball-to-separation and separator control surfaces).



(a) SINGLE-SHIELDED BEARING USED AS A METERING DEVICE. APPLIED AS SHOWN HERE, THE SHIELD PREVENTS OVER-LUBRICATION OF THE BEARING. EXCESS GREASE IS FORCED OUT THROUGH THE LABYRINTH AND SERVES TO SEAL THE LABYRINTH PASSAGES.

(b) BEARING DESIGN THAT HAS BEEN SUCCESSFUL IN PREVENTING GREASE MIGRATION DUE TO AIR BLAST IN SMALL AIR-DRIVEN TOOLS OPERATING AT SPEEDS AS HIGH AS 100,000 rpm. THE AIR BYPASS TENDS TO EQUALIZE PRESSURES ON BOTH SIDES OF THE BEARING AND PREVENT GREASE MIGRATION AND LOSS.

(c) SYSTEM THAT HAS WORKED VERY SUCCESSFULLY AT SPEEDS TO 600,000 DN AT TEMPERATURES TO 125°F. OIL BLEEDING FROM THE GREASE PACK DRIPS DOWN TO LUBRICATE THE BEARING.

FIGURE 26. POSSIBLE METHODS FOR USING GREASE LUBRICANTS FOR HIGH-SPEED BEARINGS

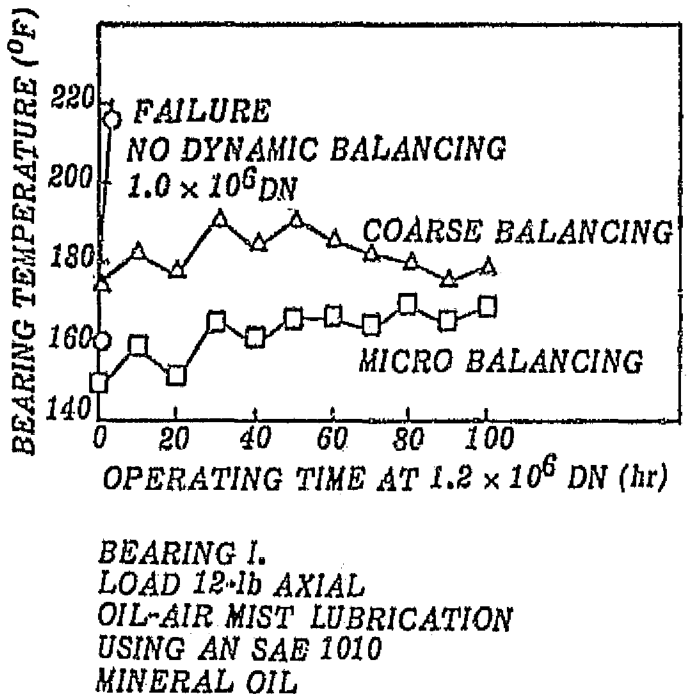


FIGURE 27. EFFECT OF DYNAMIC UNBALANCE ON BEARING OPERATION, OIL-AIR MIST LUBRICATION

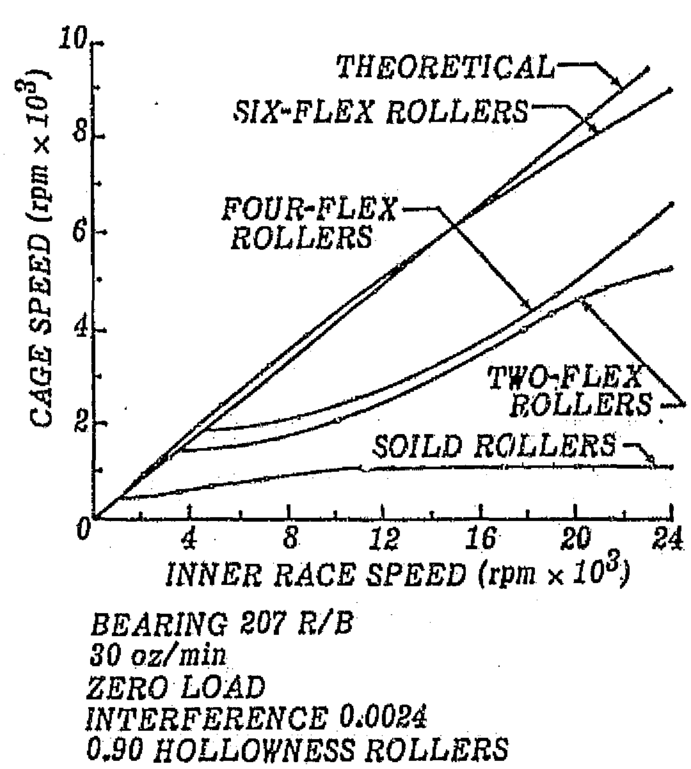


FIGURE 28. SEPARATOR SPEED VERSUS INNER RACE SPEED

4. Retainer and Stability Problems

A fundamental problem with high-speed bearings occurs as a result of instabilities and overheating of internal components. As evidenced by Figure 27, Accinelli [29] found that operation of his apparatus depended heavily on the type of balance for the system. Although the results presented in this figure are not surprising, they do illustrate dramatically the critical importance of balance.

Dynamics problems with high-speed roller bearings include cage slip [30] and ball skidding [31]. The factors controlling these effects depend heavily on the particular bearing and must be evaluated for each design. One method for inhibiting skidding in a roller bearing is by the use of hollow rollers. Harris [31] discusses effects of hollow rollers, and Bradley [32] experimentally verified this theory, as demonstrated by Figure 28. This figure illustrates not only the accuracy of Harris' prediction, but also the amount of skidding occurring in a solid roller bearing.

Kliman [33] has developed an empirical formula for determining what he terms as a limiting speed for a ball bearing,

$$\frac{DN^3 d^3}{\cos^3 \beta} = 31 \times 10^8 ,$$

where

D = bearing pitch diameter (mm)

N = inner race speed (rps)

d = ball diameter (in.)

β = initial contact angle.

Kliman states that for extended bearing life, 60 percent of the speed should be considered maximum. Special retainer designs such as the channelled retainer discussed by Moran [34] can help alleviate overheating of a retainer to permit high-speed bearing operation.

5. Temperature Considerations

One of the fundamental problems with high-speed bearings is associated with the thermal gradients developed within the bearing. Internal heating is generated in a bearing by the slip at contacts between the bearing elements and by the churning of the lubricating oil. The heating problem becomes even more complex when the bearing must withstand internal heat sources. The most common liquid-lubricated, high-temperature bearing problems for the aerospace industry occur in gas-turbine engines.

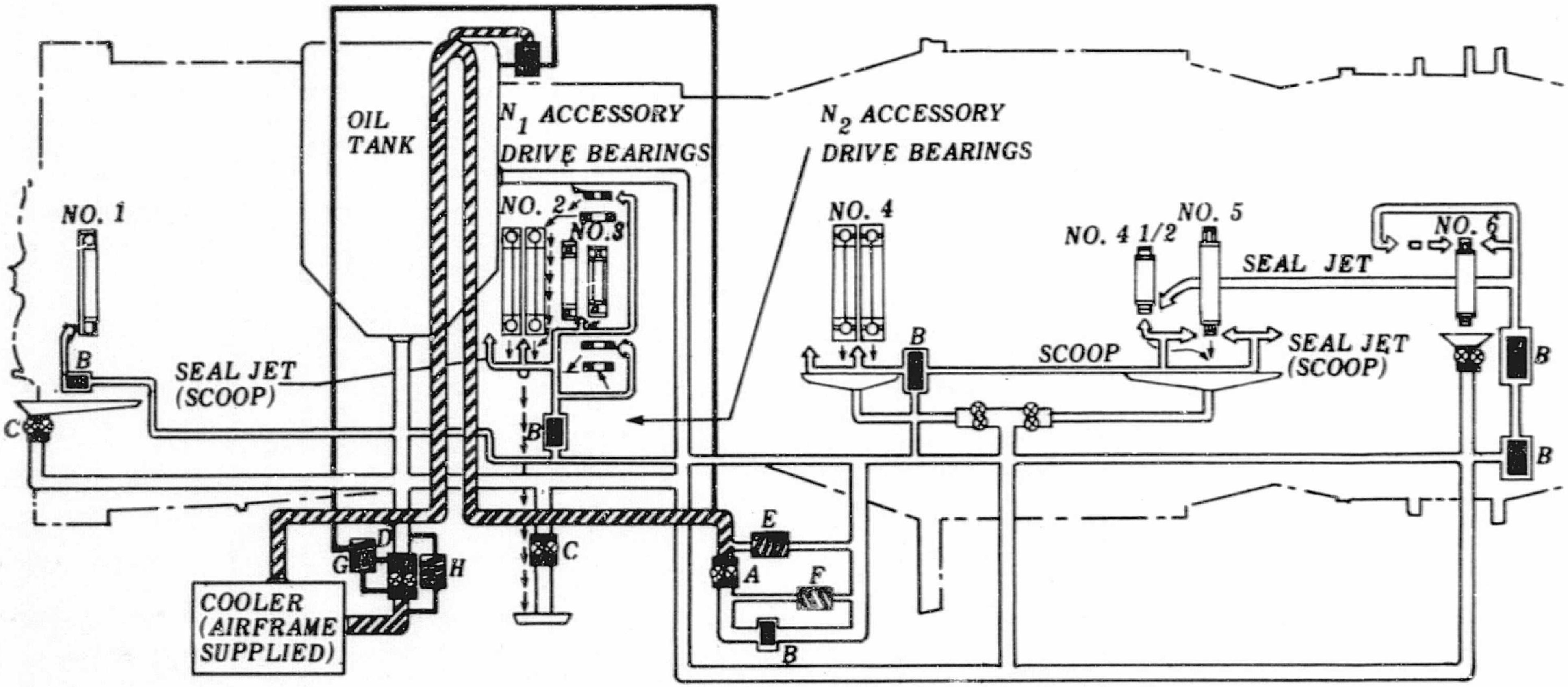
6. Turbine-Engine Lubrication

Although early turbine engines used the oil-air mist lubrication system, modern engines employ the jet or spray concept. The engine oil system for a commercial jet engine is shown in Figure 29 [35]. This drawing illustrates a type of lubrication system required for high-temperature application. The oil flow situation is summarized in Table XI [35].

Figure 30 summarizes bearing fatigue life and retainer life for a 150-millimeter bearing operating at 400°F with two high-temperature lubricants. Recently, full-bearing fatigue tests on 120-millimeter bore bearings were conducted at temperatures to 600°F. One interesting feature of these tests is that operating temperature was obtained by the heat generated within the bearing with a 3-gram per minute oil flow rate. The most satisfactory lubricant evaluated was a synthetic hydrocarbon fluid.

7. Heat-Transfer Analyses

One important aspect of any discussion on lubricants and bearings are the thermal problems arising both external and internal to the bearing. To



A OIL PRESSURE PUMP
B OIL STRAINERS
C SCAVENGE PUMPS
D OIL BOOST PUMP
E MAIN PRESSURE REGULATING VALVE
F MAIN SCREEN BY-PASS
G BOOST PUMP REGULATING VALVE
H BOOST PUMP RELIEF VALVE

AIRFRAME SUPPLIED LINES INDICATED BY SHADING

FIGURE 29. SCHEMATIC DRAWING OF COMMERCIAL JET ENGINE OIL SYSTEM

TABLE XI. TYPICAL TEMPERATURE AND HEAT REJECTION PATTERN FOR AN ADVANCED TURBOFAN TRANSPORT ENGINE*

Position	Bearing Type	Bearing Oil Flow (lb/min)	Bearing Heat Rejection (Btu/min)	Bearing Race Temperatures (°F)		Seal Type	Seal Oil Flow (lb/min)	Seal Heat Rejection (Btu/min)
				Outer	Inner			
1	Ball	7	150	425	365	Face contact	4	130
1-½F						Face contact	3	100
1-½R						Face contact	6	140
2	Ball	20	700	425	365	Face contact	5	190
3	Roller	10	270	525	485	----		
3F						Face contact	8	200
3R						Face contact	8	200
4	Roller	3	50	525	505	Ring contact	6	250
Totals		40	1170				40	1210

Item	Oil Flow (lb/min)	Heat Rejection (Btu/min)
Mainshaft bearings	40	1170
Mainshaft seals	40	1210
Accessory geartrain and gearbox	20	1600
Churning and wall pickup	---	870
Engine totals	100	4850

*Based on advanced engine at cruise power with Type III oil at 325°F oil-in temperature.

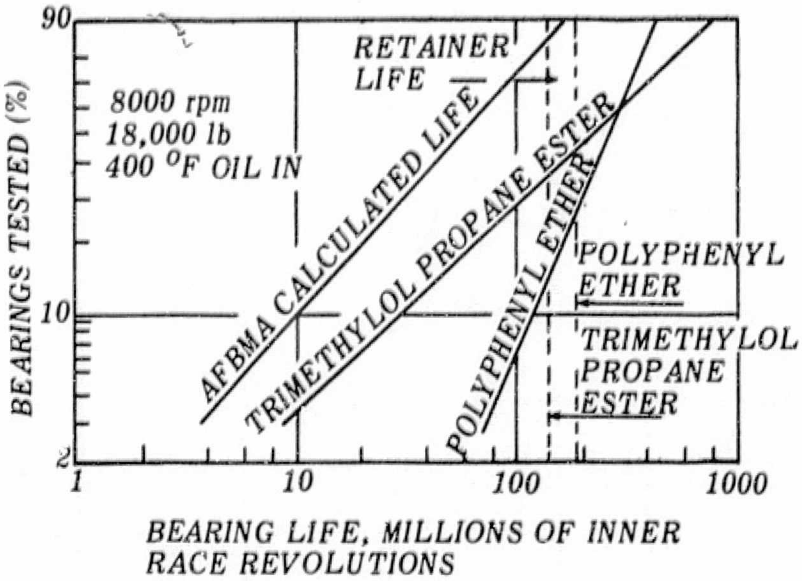
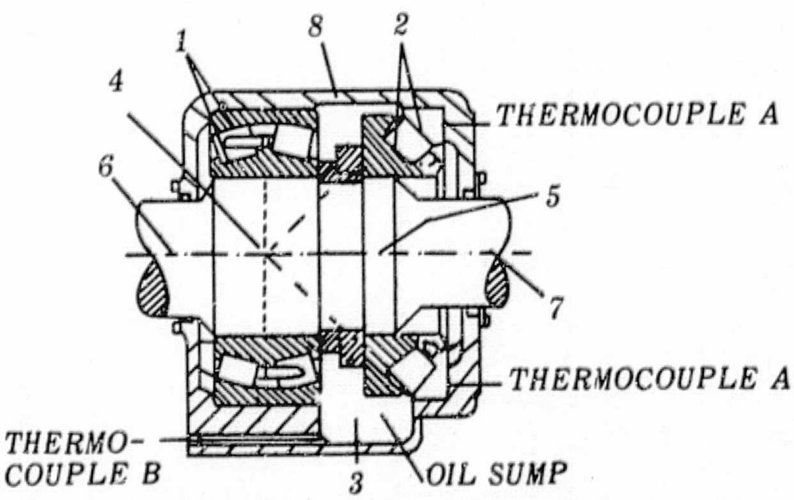


FIGURE 30. BEARING FATIGUE LIFE AND RETAINER LIFE DATA



1 CONDUCTION OF HEAT FROM THE INNER AND OUTER RINGS OF EACH BEARING TO THE SHAFT AND HOUSING, RESPECTIVELY
2 HEAT CONVECTION FROM EACH BEARING TO AN AIR-OIL MIST IN THE HOUSING
3 HEAT CONDUCTION ALONG THE SHAFT
4 HEAT CONVECTION FROM THE ROTATING SHAFT TO MACHINERY COMPARTMENT AIR
5 NATURAL CONVECTION OF HEAT, FROM THE HOUSING SURFACE TO COMPARTMENT AIR
6 RADIATION OF HEAT FROM THE HOUSING TO SURROUNDING STRUCTURES

FIGURE 31. SECTION VIEW OF BEARING HOUSING SHOWING TEMPERATURE NODE AND THERMOCOUPLE LOCATION

obtain realistic estimates of the bearing temperature, a complex heat balance study must be conducted. These analyses, which are unique to the particular bearing system, depend on the heat paths as well as the heat sources in the system. The calculations normally require high-speed computer programs.
Harris [4] presents a technique for the temperature prediction without the need of exhaustive analysis. Harris uses an 8-node system as shown in Figure 31. Analyses of the type discussed by Harris should be straightforward, and based on comparisons with experimental data, they appear to yield realistic results.

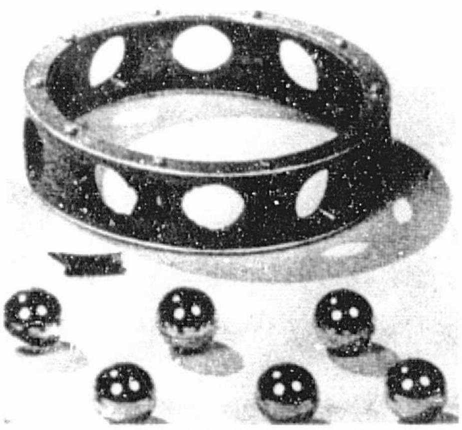
Section VII. CRYOGENIC BEARINGS

Ball and roller bearings are used as mainshaft bearings in pumps for cryogenic fluids. Developments in design and materials during the past 10 years have enabled engineers to successfully use rolling-contact bearings in cryogenic pumps with the bearings totally immersed in the cryogenic fluid or cold gas.

Operation of commercial ball bearings at liquid hydrogen temperatures was attempted in 1947 [36]. Twenty minutes operation at 5000 rpm was considered an achievement at the time. Primary source of failure was excessive wear of steel retainers. Since that time, gradual improvement in cryogenic bearing performance both in life and speed capabilities has come about primarily through the development of self-lubricating retainers. Attention to the effects of bearing geometry has also provided improvements, especially for high-speed applications. The need for self-lubricating retainers became evident when it was established that removal of residual oil films from ball and race surfaces was required before use in a cryogenic environment. The protective oil film (remaining from manufacturing processes and deliberately applied to prevent rusting during storage and shipment) will solidify at cryogenic temperatures and act as an adhesive to cause immobilization of the bearings. As a consequence, thoroughly degreased bearings will develop excessive frictional heating at locations where balls slide in retainer pockets, retainers slide on race guide surfaces, and balls spin against race surfaces. Even in cryogenic fluids, the frictional energy is sufficient to cause charring of phenolic retainers and development of temper colors on steel surfaces. In liquid hydrogen, the native oxide films which provide some protection against adhesive wear are not reformed as they are worn away; therefore, the need for inherent lubrication is vital for this environment. Self-lubricating retainers made of polytetrafluoroethylene (PTFE) or graphite-based materials provides a transfer-type of solid-film lubricant deposition which is quite effective at cryogenic temperatures. Solid lubrication films are transferred from the retainer pockets to the ball or roller surfaces, and from the rolling elements to the surfaces of the races.

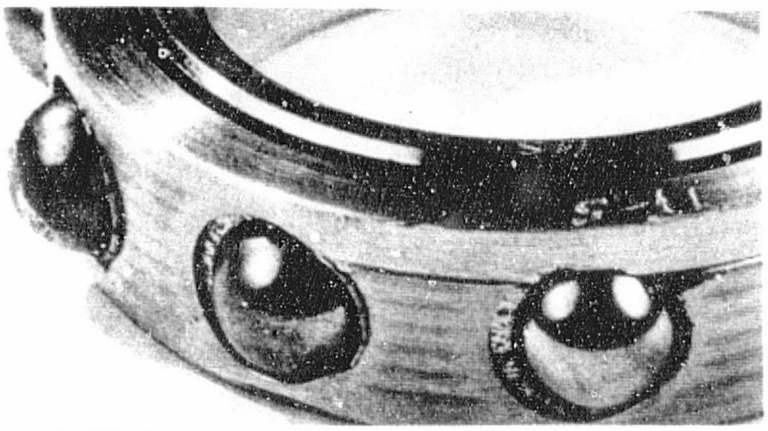
1. Typical Cryogenic Bearing Failures

A large percentage of cryogenic bearing failures have been attributed to defective retainers. Figure 32 shows several examples of retainer-caused bearing failures from high-speed tests in liquid hydrogen made at the



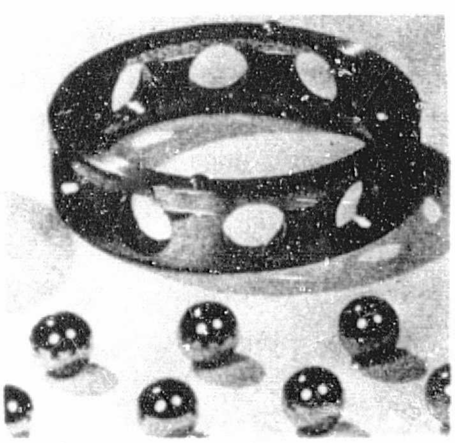
(a) LAMINATED GLASS CLOTH WITH PTFE BINDER (BEARING 16-S). DELAMINATION AFTER 349 MINUTES.

(b) GLASS-FIBER-FILLED PTFE (BEARING 15-S). CRACKS IN BALL POCKET AFTER 464 MINUTES.

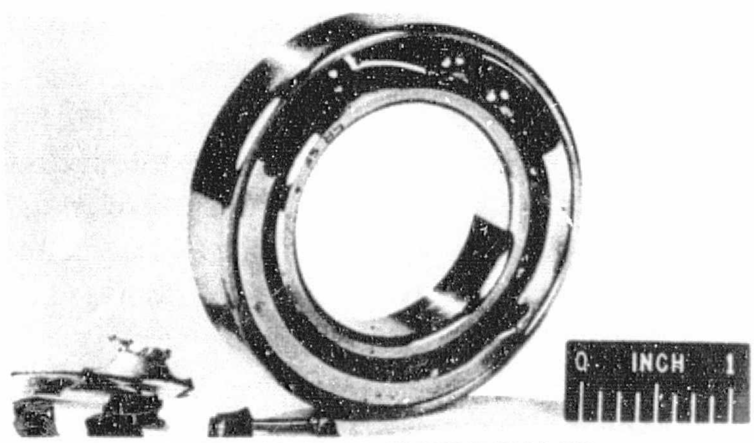


(c) SILVER COMPOSITE (BEARING 11-S). SHROUD RUBBED ON OUTER-RACE LAND AND MOVED RELATIVE TO BODY; BALL WORE INTO SHROUD. BEARING JAMMED AFTER 138 MINUTES.

(d) COPPER COMPOSITE (BEARING 17-S). SHROUD RUBBED ON OUTER-RACE LAND; HIGH WEAR AFTER 213 MINUTES.



(e) SILVER COMPOSITE (BEARING 18-S). SHROUD RUBBED ON OUTER-RACE LAND; BALL POCKET CRACKED AFTER 234 MINUTES.



(f) MOLYBDENUM DISULFIDE-FILLED POLYIMIDE (BEARING 19-S). COMPLETE FAILURE AFTER 22 MINUTES.

FIGURE 32. MECHANICAL DAMAGE OF BEARING RETAINERS

NASA-Lewis Research Center [37]. Although these represent high-speed performance (approaching one million DN), more moderate speeds also produce similar failures.

Figure 32(a) shows delamination of a laminated fabric-PTFE material. Defective material (poor binding between laminate layers) was blamed. Figure 32(b) shows cracking of the retainer at its thinnest cross section. Differential thermal contraction between the plastic and the aluminum shroud at cryogenic temperatures caused tensile fracture. Figure 32(c) shows a problem prevalent in steel-shrouded, solid-lubricated retainers. The shroud is used to provide structural strength. Differential thermal contraction will cause the plastic or graphite material to loosen in the shroud. In this case, the balls came into contact with the stainless steel shroud and, owing to the severe galling tendency of the stainless steel, the bearing jammed. Figure 32(d) and (e) shows a retainer made with too large a clearance between the guiding surfaces of the bearing. Retainer instability at high speeds caused heavy loads, sliding, and excessive wear. Figure 32(f) shows the structural failure of a retainer material with insufficient mechanical strength at temperature.

Failures have also been attributed to overheating from ball-spin-generated frictional heating in angular-contact bearings subjected to heavy thrust loads. (This effect is discussed in Section II.) Rusting of degreased bearings from condensation during pump storage or shutdown can immobilize a shaft. Use of 440 C stainless steel will reduce rusting.

2. Design of Cryogenic Bearings

Cryogenic bearings generally fall into two classifications (Table XII):

a) Moderate speed and load for cooling system or vacuum system pumps

b) Very high-speed, lightly loaded in rocket engine turbopumps.

TABLE XII. CRYOGENIC BEARING APPLICATIONS

Device	Speed (rpm)	Load (lb)	Life
Cooling system pump	1800-3600	50-500 thrust	5000 hr to 1 yr
Rocket engine turbopump	20,000	100	20 min to 1 hr

The former are relatively long-life systems (5000 hours to 1 year) while the latter are short-life systems (10 minutes to a few hours). The high-speed cryogenic bearings require more careful consideration of ball-race geometry than do cooling-system bearings.

a. Moderate Speed

Moderate-speed bearings can be angular contact, Conrad (deep groove) radial, or roller bearings. Selection of bearing type is governed essentially by the considerations given in Section II. The system of a roller bearing at one end of the shaft and angular-contact bearings at the other end provides for shaft expansion and contraction, thrust loads, and radial loads. In some designs, one end of the shaft remains remote from the cryogenic environment and a conventionally lubricated bearing can be used. However, care must be taken that the temperature of the remote bearing does not go below the frost point during shut-down because the upper bearing can jam with frost crystals.

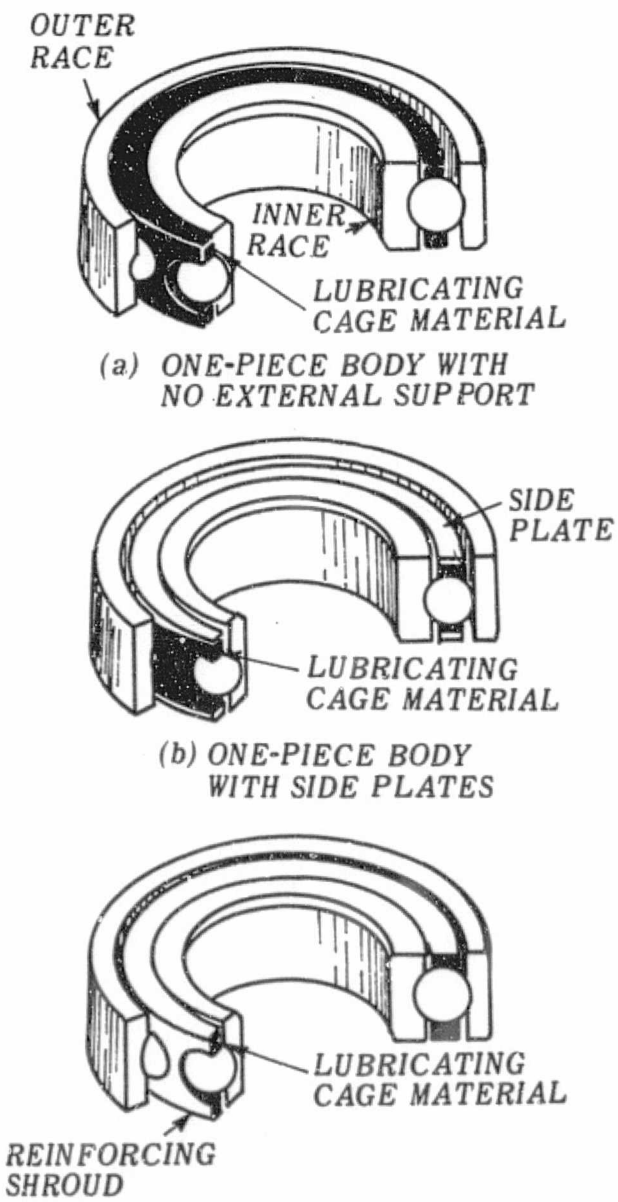
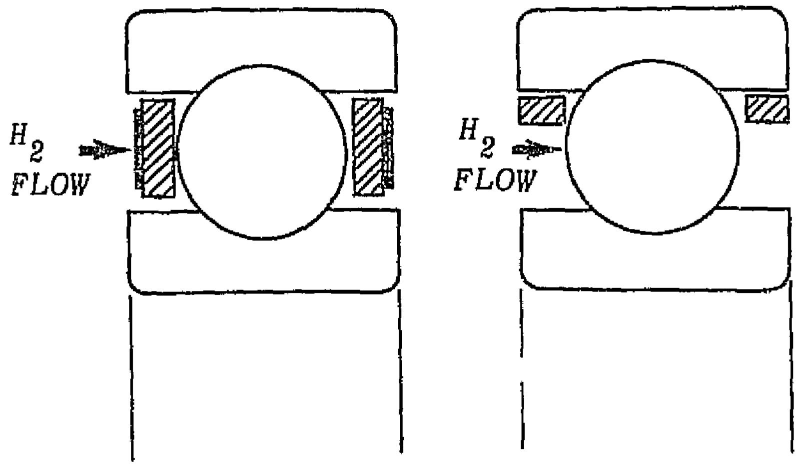


FIGURE 33. INNER-RACE LOCATED CAGE DESIGNS

The only available design parameters (Section II) which need to be considered for moderate-speed cryogenic bearings are internal clearance and retainer design. The internal clearance should be opened somewhat to accommodate differential thermal expansion or contraction of bearing components. (A rolling-contact bearing is a poor heat-transfer system and will not readily stabilize localized hot spots.) Standard bearings are often used with a filled PTFE retainer substituted for the conventional steel retainer. One ball is usually removed from the compliment to provide space for the more bulky plastic retainer. Several retainer materials are in use but most are based on PTFE strengthened with a metal or inorganic filler. Unfilled PTFE is not structurally adequate for retainer applications. Typical retainer designs are shown in Figures 33



(a) CONVENTIONAL DESIGN (b) THIN-LINE DESIGN

FIGURE 34. OUTER-RACE LOCATED CAGE DESIGNS

and 34 [37]. Polytetrafluorethylene has been found effective in a number of cryogenic media including liquid nitrogen, liquid hydrogen [5], and liquid oxygen [38].

Carbon graphite has been used as a retainer material for moderate-speed cryogenic bearings, especially where a radiation environment rules out the use of PTFE. Carbon graphite is impact sensitive and if used may require metal shrouding to provide shock resistance. It is well known that graphite lubricates only in the presence of water vapor or other adsorbed species; therefore, performance at cryogenic temperatures is not as effective as in ambient moist air.

Sizing the retainer requires careful consideration of the relative shrinkage of bearing components during cooling from room temperature to cryogenic temperatures. The retainer should be guided by light contact with the race lands. Usually the retainer is designed to be inner-race riding. Therefore, the retainer should be sized so that upon shrinkage at running temperature the desired clearance is achieved. Table XIII shows the relative shrinkage of various retainer materials when cooled to liquid hydrogen temperature [37].

TABLE XIII. TOTAL CONTRACTION OF PTFE MATERIALS FROM 70° TO -423° F

Material	Total Contraction in Cage Radial Direction (in./in.)	Total Contraction in Cage Width Direction (in./in.)
100 percent PTFE	0.0215	Same as radial direction
15 percent glass fibers, 85 percent PTFE	0.0084	0.0164
25 percent glass fibers, 75 percent PTFE	0.0090	0.0165
65 percent bronze, 35 percent PTFE	0.0140	Same as radial direction

b. High Speed

Angular-contact bearings are generally used in high-speed cryogenic bearings. High-speed dynamics of angular-contact bearings (Section II) reveal that ball centrifugal force produces unequal contact angles at inner and outer race contacts. The difference in contact angle results in a combined rolling-sliding mode for the balls, with resultant frictional heat generation. The higher the speed and the larger the ball, the larger the amount of heat generation. By opening ball-race conformity, reducing ball diameter, and reducing contact angle, ball-spin frictional heating can be minimized. NASA-Lewis personnel have developed a computer program for analysis of an optimization of bearing geometry to reduce frictional heating caused by ball "spin."

3. Ball and Race Materials

Although SAE 52100 steel has been used successfully in cryogenic bearings (carburized steel has also been used in cryogenic roller bearings), SAE 440 C is recommended as more appropriate. The principal reason for using 440 C stainless is to reduce the possibility of rusting. This material is not truly stainless, but is superior in rust resistance when compared with SEA 52100 steel.

The use of 440 C stainless steel introduces a problem of dimensional stability, however. This is of particular importance to carefully sized bearings. Dimensional instability is the result of retained austenite formed during heat treatment. Cryogenic temperatures will cause transformation of retained austenite to untempered martensite with an accompanying increase in volume. This can be sufficient to cause immobilization of the bearing. Sub-cooling the steel to -300°F followed by tempering is recommended as part of the heat-treatment cycle when cryogenic applications are anticipated.

Section VIII. AIRFRAME BEARINGS

Airframe bearings are bearings that operate at less than 100 rpm or with oscillating motion. They are usually heavily loaded and grease lubricated. These bearings are commonly used in control surface linkages, landing-gear retraction hinges, wing privots, etc. Plain bearings are used in similar applications where low friction and wear are not critical design factors. Rolling-contact bearings do not have the high load capacity of plain bearings because the rolling elements tend to indent the race surfaces at a limiting load level. Despite their load limitations, rolling-contact-airframe bearings have been used in many applications covering the temperature range -65° to 600°F. Experimental bearings, using high-temperature alloys, cermets, and solid lubricants have been operated at higher temperatures (to as high as 2000°F) [39].

1. Load Capacity

Because of the aforementioned tendency for the rolling element to indent or "brinell" the races, airframe rolling-contact bearings are rated based on their "static load capacity." Military Standard (MS) bearings are available specifically for airframe applications and there are available static load limits and load-life data. Typical MS airframe bearing specifications are shown in Figures 35 and 36.

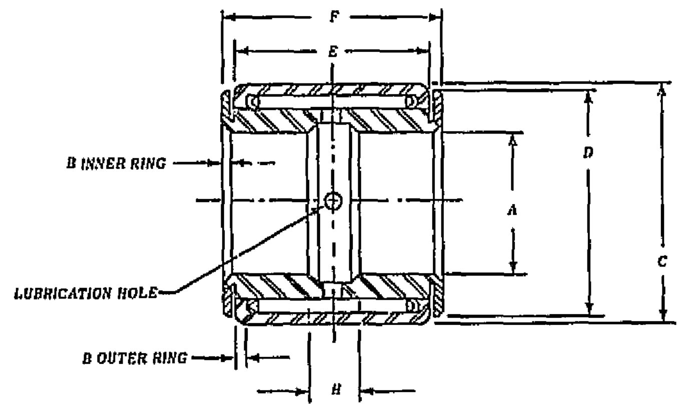
The aircraft static capacity is rated as $3\frac{1}{4}$ to $3\frac{1}{2}$ times the AFBMA basic static capacity for SAE 52100 steel. An approximate formula for determining aircraft static capacity for roller bearings is 11,000 $n\ell d$, where

n = number of rollers

ℓ = effective roller length

d = roller diameter.

F. W. Williams, in an analysis of airframe-bearing load ratings [40], has shown the effect of maximum hertz stress and bearing design on the extent of permanent deformation under static loads. The deformation characteristics are summarized in Figure 37. This figure shows that with a ball in a race with 53 percent conformity (race radius = 53 percent ball diameter) the permanent deformation of a roller on a flat (approximating a cylindrical roller-bearing configuration) is 10 times that of the ball race for the same maximum



DASH NO.	A +.0000 -.0005 BORE INNER RACE	B +.016 -.000 RAD OR 45° MAX BEVEL	(a) C ±.0005	D ±.010	E +.000 -.010	F +.000 -.005	H +.000 -.062	CLAMPING MIN DIA	AIRCRAFT STATIC CAPACITY LB	WEIGHT LB APPROX	TOTAL RADIAL PLAY MAX
-3	.1900					.625		.409	2050	.040	
-4	.2500	.022	.6250	.562	.500	.562	NONE	.500	2670	.025	
-5	.3125		.6875	.625		.625		.502	3000	.050	
-6	.3750		.7500	.688		.812		.625	5820	.060	
-7	.4375		.8125	.750	.750		.188	1.062	6390	.090	.0015
-8	.5000		1.0000	.938		.875		.844	7030	.120	
-10	.6250	.032	1.1250	1.062			.250	.909	8040	.150	
-12	.7500		1.2500	1.188				1.094	13400	.210	
-14	.8750		1.3750	1.312	1.000	1.125		1.219	14900	.240	
-16	1.0000		1.5000	1.438			.375	1.344	16500	.270	
-20	1.2500		1.8750	1.812	1.250	1.375		1.641	24500	.300	.0017

C O P Y

OIL HOLE DATA		
BORE SIZE	NO. OF HOLES INNER RING	NO. OF HOLES OUTER RING
-3 THRU -5	NONE	NONE
-6 THRU -20	2	1

(a) HOUSING DIMENSION FOR BEARING.

MATERIAL: STEEL, MIL-S-7420, MIL-S-6690, MIL-S-7493, QQ-S-624, QQ-S-633, QQ-S-763, FED STD NO. 66 STEEL NO. 50100, 51100, AND 52100.
PLATING: CADMIUM PLATE, QQ-P-416, TYPE I, CLASS 3.
MACHINE FINISH: MIL-STD-10, SEE PROCUREMENT SPECIFICATION.

REMOVE ALL BURRS AND SHARP EDGES.

DIMENSIONS IN INCHES, UNLESS OTHERWISE SPECIFIED, TOLERANCES: DECIMALS ±.005.
DIMENSIONS TO BE MET AFTER PLATING.

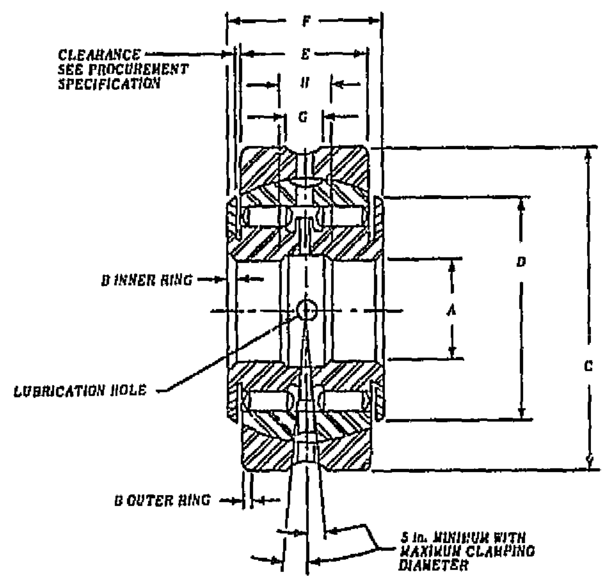
THE AIRCRAFT STATIC BEARING CAPACITY RATINGS AS NOTED REPRESENTS THE DESIGN LIMIT LOAD OR THE HIGHEST LOAD WHICH CAN BE PLACED ON THE BEARING WITHIN AN ALLOWABLE .0001 INCH BRINELL OF THE OUTER RACE. HIGHER LOADS WILL DANGEROUSLY BRINELL THE RACES AND PERMANENTLY DEFORM THE ROLLERS. THE STATIC WORKING LOAD OF THE BEARING SHOULD BE TAKEN AS 1/3 OF THE AIRCRAFT STATIC CAPACITY.

(B) STANDARD REVISED.

FOR DESIGN FEATURE PURPOSES, THIS STANDARD TAKES PRECEDENCE OVER PROCUREMENT DOCUMENTS REFERENCE HEREIN. REFERENCED DOCUMENTS SHALL BE OF THE ISSUE IN EFFECT ON DATE OF INVITATIONS FOR BID.

P.A.	TITLE	MILITARY STANDARD
USAF - AFSC Other Cust Navy - Wep Army - ORD	BEARING, ROLLER, NEEDLE - SINGLE ROW, THIN SHELL, TYPE II, ANTIFRICTION	MS24462

FIGURE 35. A MILITARY AIRFRAME BEARING SPECIFICATION



DASH NO.	A +.0000 -.0007 BORE INNER RACE	B +.015 -.000 RAD OR 45° MAX BEVEL	C		D ±.010	E +.000 -.005	F +.000 -.005	G ±.031	H +.000 -.062	CLAMPING DIA MAX	CLAMPING DIA MIN	AIRCRAFT STATIC CAPACITY LB	WEIGHT LB APPROX	TOTAL RADIAL PLAY MAX	HOUSING SIZE GAGE +.0000 -.0001 LOW	HOUSING SIZE GAGE +.0000 -.0001 HIGH
-6	.3750	.022	1.1875	+.0000 -.0005	.812	.469	.562	.125	.188	.781	.641	6800	.130	.0030	1.1867	1.1872
-7	.4375	.032	1.3125		.875	.531	.625			.844	.703	8800	.174	.0031	1.3110	1.3122
-8	.5000		1.5000		1.031	.656	.750			1.000	.844	13000	.293		1.4991	1.4997
-9	.5625		1.6875		1.093	.781	.875	.156		1.062	.891	17700	.420		1.6860	1.6872
-10	.6250		1.7500		1.156	.906	1.000		.250	1.094	.953	23200	.520	.0032	1.7491	1.7497
-12	.7500		1.8750		1.281	1.000	1.125			1.156	1.078	30000	.630	.0034	1.8741	1.8747
-14	.8750		2.1250	+.0000 -.0005	1.500	1.125	1.250		.375	1.375	1.250	38700	.870	.0037	2.1238	2.1246
-16	1.0000		2.2500		1.625					1.500	1.375	43000	.960	.0041	2.2488	2.2496
-20	1.2500		2.5000		1.906	1.040				1.781	1.625	47100	1.070		2.4988	2.4996
-24	1.5000		2.7500		2.156					2.062	1.875	54900	1.230		2.7488	2.7496
-32	2.0000		3.250	+.0000 -.0008	2.656					2.503	2.375	70600	1.490	.0045	3.2485	3.2496

OIL HOLE DATA			
BORE SIZE	NO. OF HOLES		
	INNER RING	OUTER RING	SPHERICAL HOUSING
-6 THRU -10	2	4	2
-12 THRU -32	4	4	2

MATERIAL: STEEL MIL-S-7420, MIL-S-6690, MIL-S-7493, QQ-S-624, QQ-S-633, QQ-S-763 FED STD NO. 66 STEEL NO. 50100, 51100, AND 52100.
PLATING: CADMIUM PLATE, QQ-P-416, TYPE I, CLASS 2.
MACHINE FINISH: MIL-STD-10. SEE PROCUREMENT SPECIFICATION.

REMOVE ALL BURRS AND SHARP EDGES.

DIMENSIONS IN INCHES. UNLESS OTHERWISE SPECIFIED, TOLERANCES: DECIMALS ±.005.
DIMENSIONS TO BE MET AFTER PLATING.

C
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P
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THE AIRCRAFT STATIC BEARING CAPACITY RATINGS AS NOTED REPRESENTS THE DESIGN LIMIT LOAD OR THE HIGHEST LOAD WHICH CAN BE PLACED ON THE BEARING WITHIN AN ALLOWABLE .0001 INCH BRINELL OF THE INNER RACE. HIGHER LOADS WILL DANGEROUSLY BRINELL THE RACES AND PERMANENTLY DEFORM THE ROLLERS. THE STATIC WORKING LOAD OF THE BEARING SHOULD BE TAKEN AS 2/3 OF THE AIRCRAFT STATIC CAPACITY.

B STANDARD REVISED.

FOR DESIGN FEATURE PURPOSES, THIS STANDARD TAKES PRECEDENCE OVER PROCUREMENT DOCUMENTS REFERENCED HEREIN. REFERENCED DOCUMENTS SHALL BE OF THE ISSUE IN EFFECT ON DATE OF INVITATIONS FOR BID.

P.A.	TITLE	MILITARY STANDARD
USAF - AFSC Other Cust Navy - Wep Army - ORD	BEARING, ROLLER, NEEDLE - DOUBLE ROW, HEAVY DUTY, SELF ALIGNING, TYPE IV, ANTIFRICTION	MS24464

FIGURE 36. A MILITARY AIRFRAME BEARING SPECIFICATION

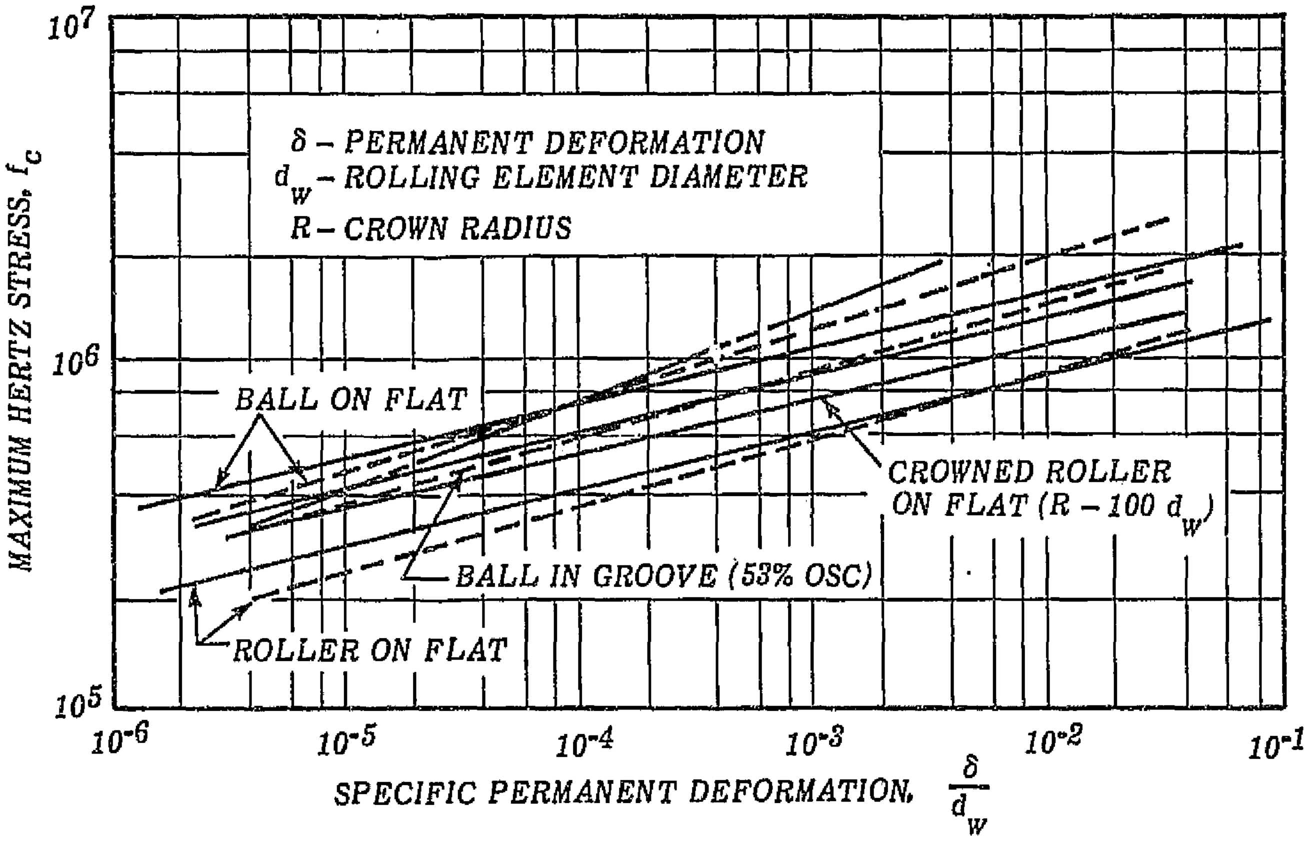

FIGURE 37. MAXIMUM HERTZ STRESS VERSUS SPECIFIC PERMANENT DEFORMATION

hertz stress. Williams points out that this is misleading because the actual load-carrying capacity of the ball in pounds is less than the roller, assuming the same bearing envelope dimensions.

The load limit reflects an arbitrarily chosen depth of permanent indentation on the raceway (usually 0.0001 inch) caused by the load. This does not necessarily mean that deformation of the magnitude described above is detrimental to bearing performance.

2. Dynamic Load Ratings

Airframe bearing dynamic ratings can be considered on the basis of load-life data available for oscillating bearings [41, 42]. Figure 38 shows the difference in bearing failure characteristics between oscillating bearings and continuous-rotating bearings at a given load. The lower slope for the continuous-rotation bearings indicates a larger scatter in bearing lives. Although the median life for continuous-rotating bearings is about six times that for oscillating bearings, the early failures are about the same in equivalent life.

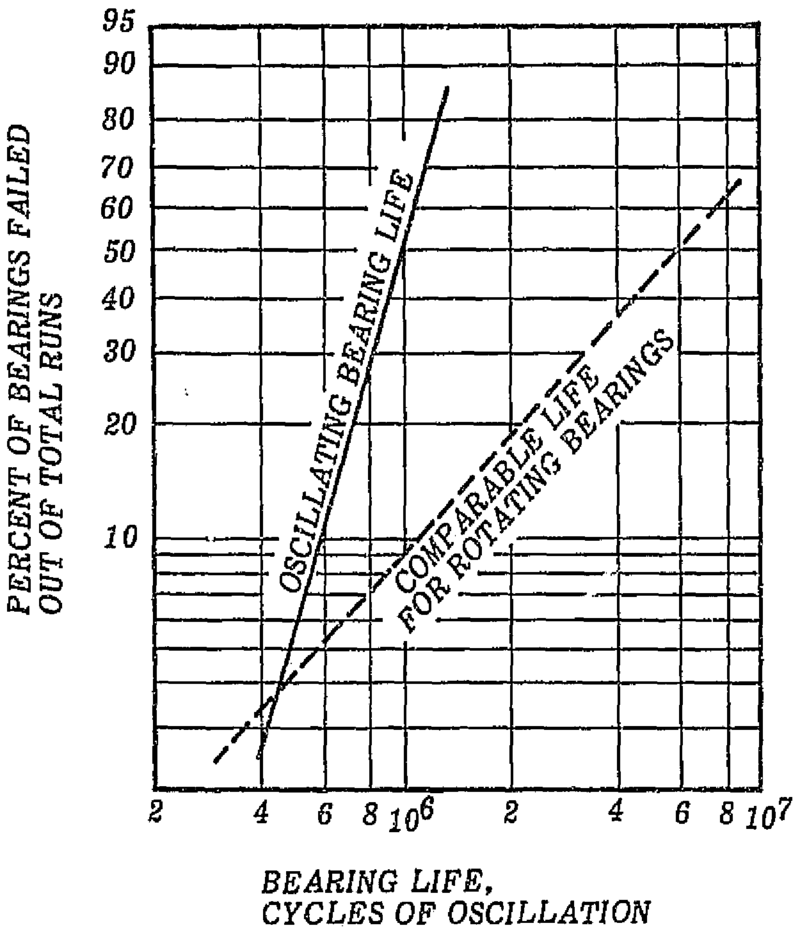

FIGURE 38. PERCENT BEARING FAILURE VERSUS LIFE FOR OSCILLATING BEARINGS

Therefore, for high-reliability applications (less than L_{10} lives), the oscillating bearing life approaches the life values for bearings of the continuous-rotating type.

It has been recommended [43] that the "basic dynamic capacity" of an airframe bearing is "the constant radial load at which 10 percent of the bearings tested will fail through fatigue of the ball or race material within 2000 cycles." The basic dynamic capacity is found for MS bearings in data sheets accompanying the specification sheets.

3. Lubrication

Airframe bearings are generally lubricated with grease. Seals or shields are recommended where relubrication is not anticipated. Since motion is slow and oscillatory, there is no elastohydrodynamic effect and antiwear additives are provided in the lubricant for boundary lubrication conditions. Generally, airframe bearings are packed fuller than are moderate and high-speed continuous-rotation ball bearings. This is because a greater reservoir of lubricant is tolerable in a bearing where friction losses from lubricant shearing by rolling elements is negligible.

4. Bearing Torque

The coefficient of friction for a rolling-contact airframe bearing is relatively low (0.001 to 0.003). Therefore, these bearings contribute a very small amount to the total friction of mechanisms and linkages. However, when the temperature is decreased the grease stiffens and bearing torque increases. Figures 39 and 40 [43] show the starting and running torques for a number of MS-type bearings as a function of temperature. These bearings were lubricated

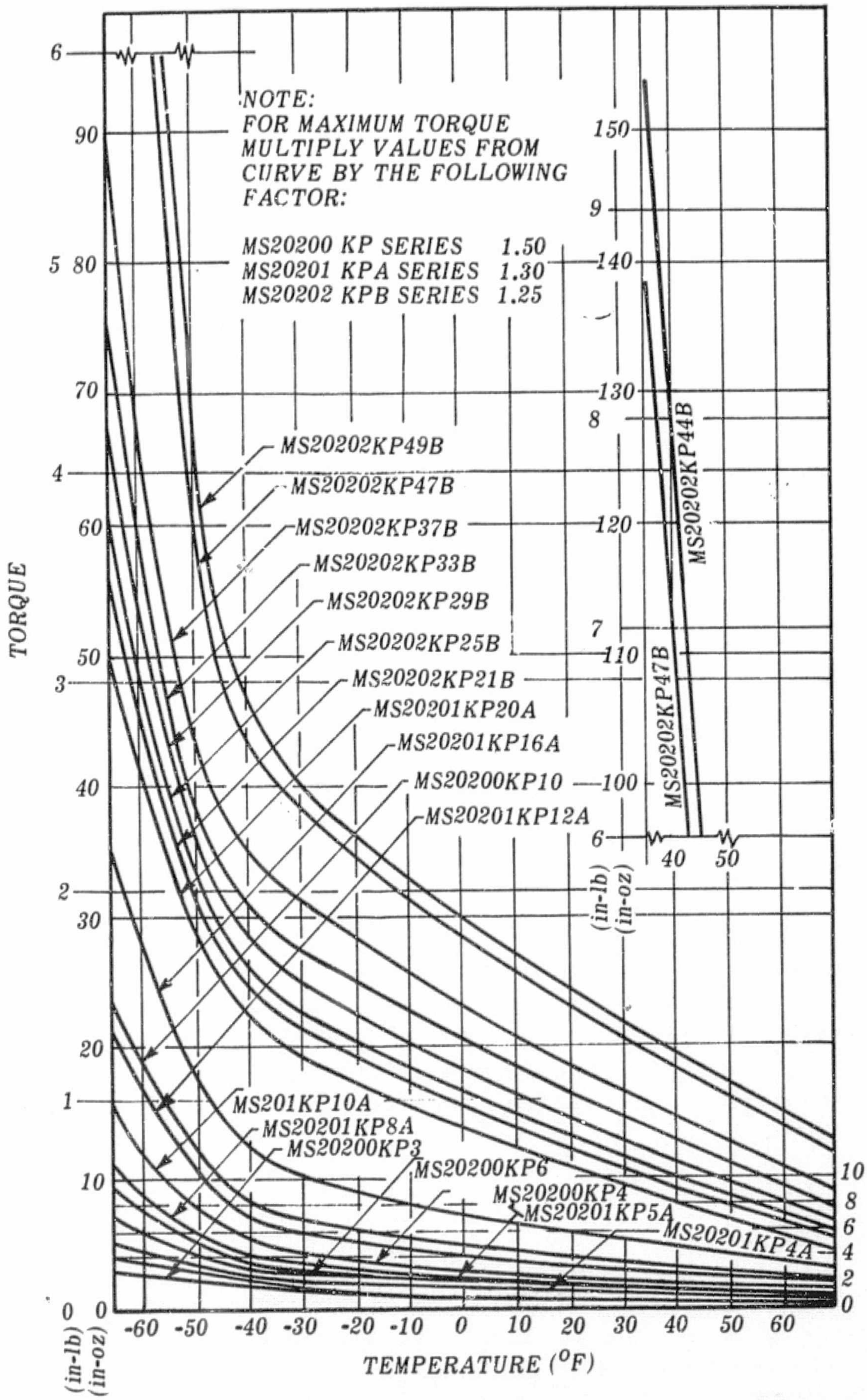

FIGURE 39. BEARING RUNNING TORQUE AS A FUNCTION OF TEMPERATURE FOR A VARIETY OF MS BEARING TYPES

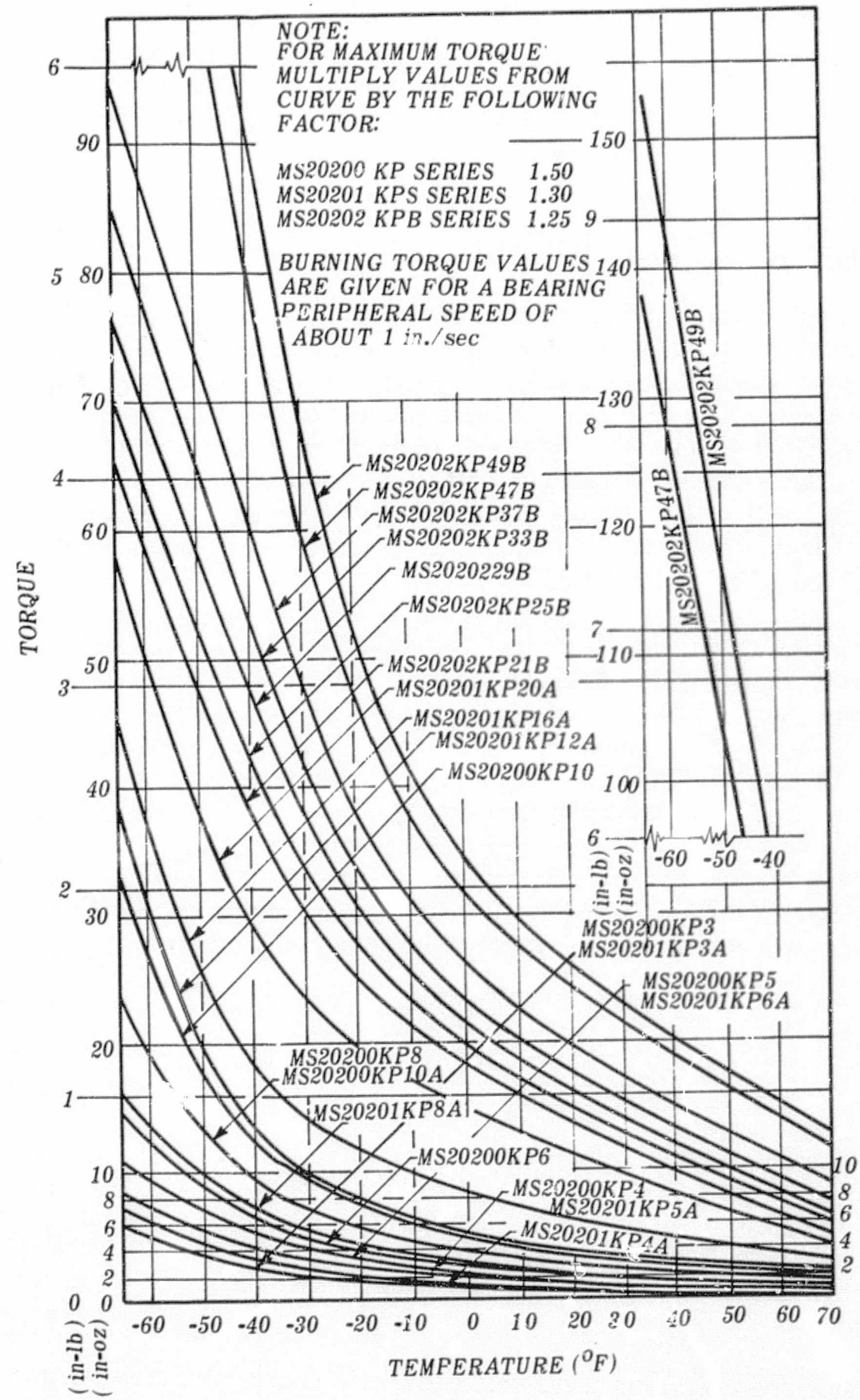

FIGURE 40. BEARING STARTING TORQUE AS A FUNCTION OF TEMPERATURE FOR A VARIETY OF MS BEARING TYPES

with MIL-G-23827 grease. At very heavy loads, or loads that approach the aircraft static load capacity, bearing torque increases as a result of the increase in elastic contact area. Under combined elevated temperature and high load, rolling-contact airframe bearing torque can approach plain bearing levels.

5. High Temperature

Airframe bearings made of SAE 52100 steel can be used at temperatures to about 300°F. Martensitic stainless steel bearings (440 C) can be used at 450°F and tool steel (M-2 tool steel) can be used to 600°F [42]. Other bearing materials have been used at temperatures to 900°F. The life of SAE 52100 tool steel and 440 C stainless steel bearings is reduced significantly, however, at the higher temperatures. Figure 41 shows the effect of temperature on steel bearing life at oscillating conditions. The use of titanium-carbide balls with steel races has enabled operation at temperatures as high as 900°F, as shown in Figure 42 [40].

Elevated temperatures increase the effect of plastic deformation. This is especially important in heavily loaded oscillatory bearings where oscillation amplitude is just sufficient to cause roller-path overlap or less. When oscillation amplitude is very small and no roller-path overlap occurs, the rolling elements gradually indent the race surface (the metal tends to creep), and eventually the indentations are deep enough to cause interference with larger amplitude motion. When the roller paths just overlap, fatigue failure is accelerated and occurs in the overlap regions. An example of this effect is shown in Figure 43 [42]. When oscillation amplitude is large, roller-path overlap no longer has any significant effect on bearing performance.

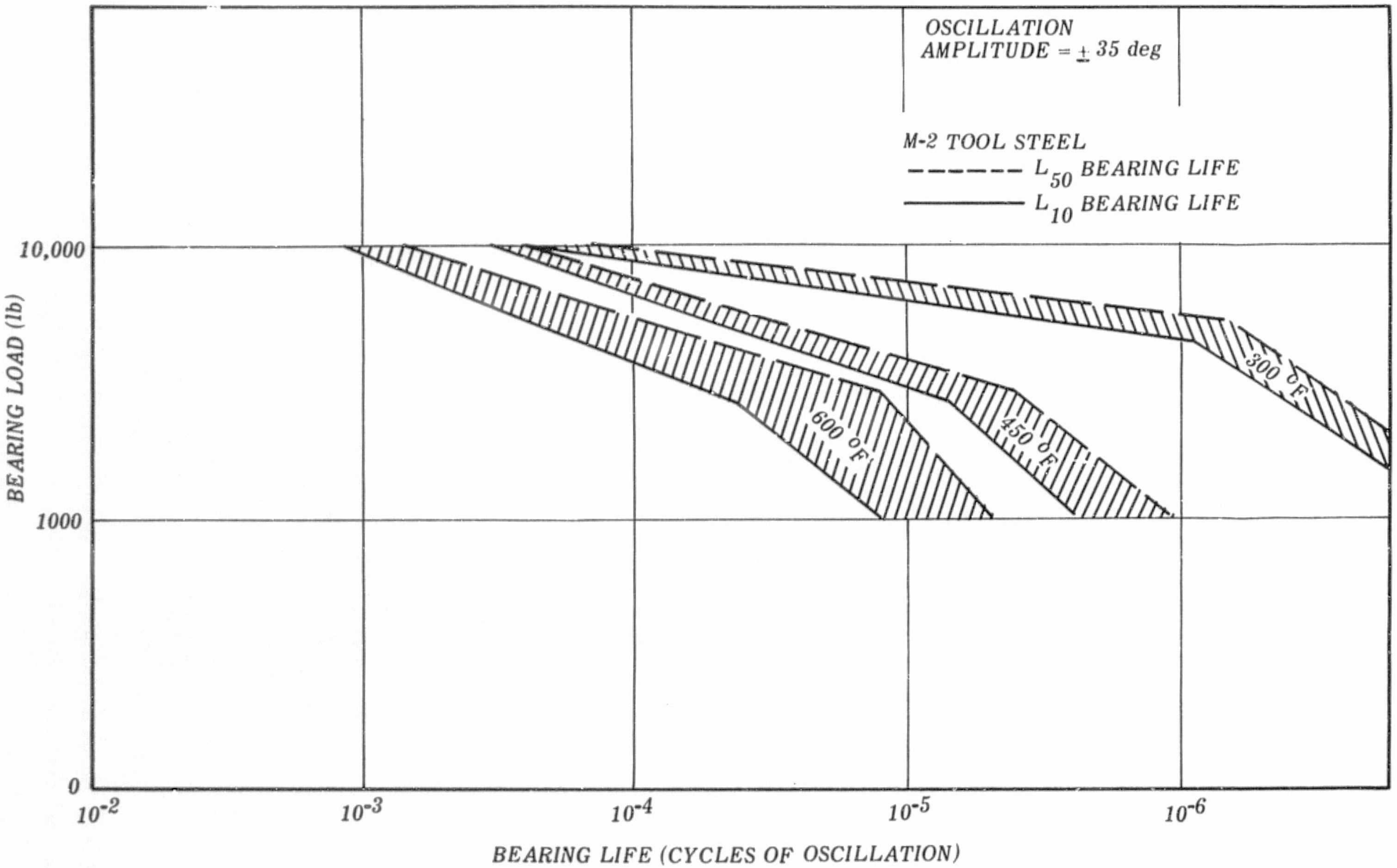

FIGURE 41. LOAD-LIFE CHARACTERISTICS OF BR-6 BEARINGS MADE OF VACUUM-MELTED M-2 TOOL STEEL SHOWING EFFECTS OF TEMPERATURE ON BEARING LIFE

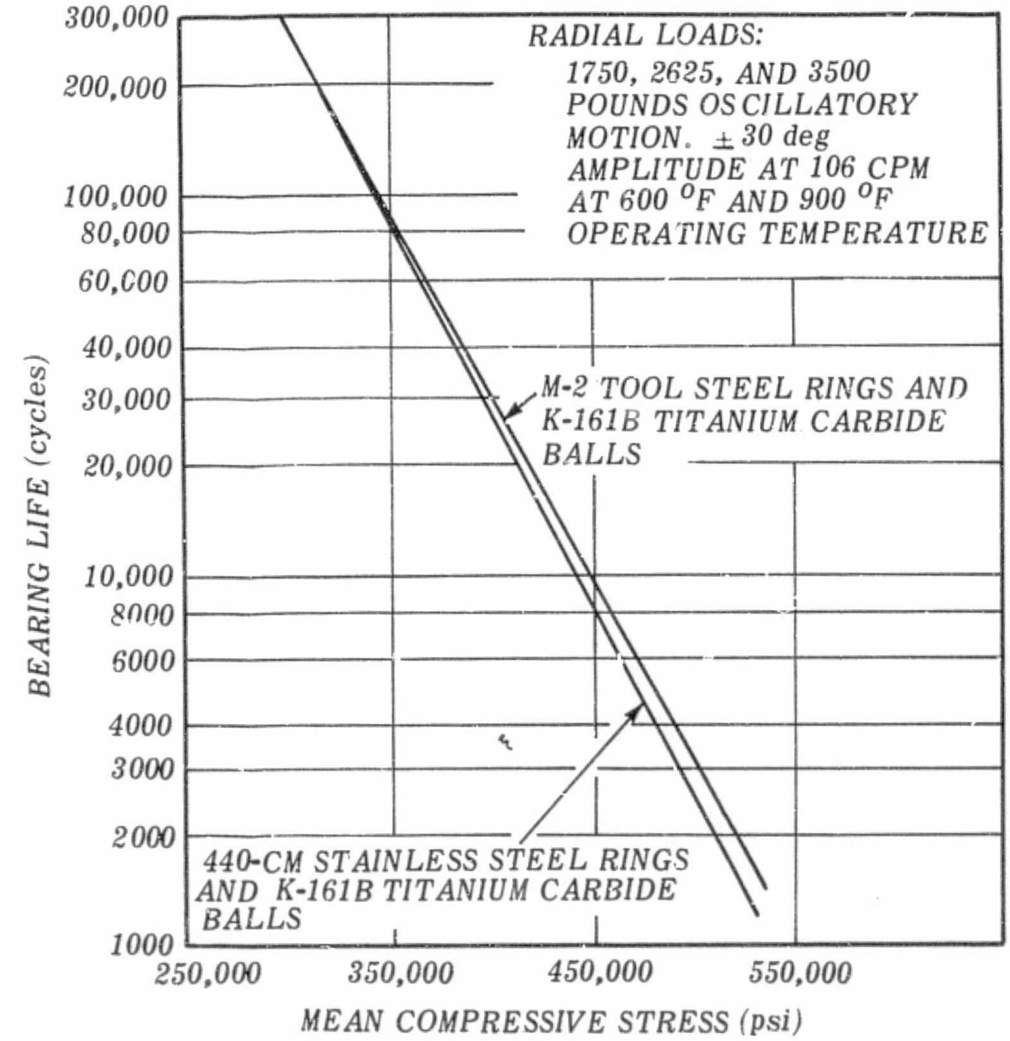

FIGURE 42. EFFECT OF HERTZ STRESS ON PERFORMANCE OF ROLLING-ELEMENT BEARINGS TO 900°F

FIGURE 43. INNER RACES FROM HEAVILY LOADED CONCAVE ROLLER BEARINGS AFTER 1000 CYCLES OF OSCILLATION

Section IX. FAILURE DIAGNOSIS

With proper manufacturing, selection, installation, and maintenance, the incidence of failure in rolling-element bearings is typically very small. Material fatigue becomes the limiting factor in life, and follows a statistical failure distribution. Since there are numerous possible causes of bearing damage that will reduce serviceable life to levels well below the limiting fatigue life, analyzing bearing failures is of great value in establishing the cause of failure. The results of the analysis provide a sound basis for design modifications to extend component reliability.

The criteria for bearing failure vary greatly depending on the bearing application. Increased radial clearance in an oscillating airframe bearing, for example, may well go unnoticed, while a slight clearance increase in an instrument bearing may spell complete component failure. Whatever the criteria for failure, including increased vibration, clearance, noise, torque, or temperature, the bearing should be removed for examination at the onset of failure. Even though the bearing is not inoperative, further operation usually leads to self-aggravating increases in damage until total failure results. The advanced phases of the failure often obscure the initiating cause and prevent proper diagnosis. In particularly troublesome applications, removal of bearings for examination before there is any evidence of failure is often helpful in detecting early symptoms.

1. Procedure for Analysis

Several important steps are required for accurate analysis of a bearing failure [44]. While all of the information in each step may not be required for a given bearing failure, it is usually essential for a comprehensive analysis.

Step I. Obtain complete information on the application and operating environment. Included are:

a) Description of application, including method of mounting

b) Type, direction, and magnitude of loading

c) Speed (constant, variable, or intermittent)

d) Lubrication (type of system, lubricant, and filtering)

e) Temperature of environment, lubricant, and bearing

f) Sources of debris

g) Sources of electrical current passing through bearing

h) Methods used for installation.

Step II. Thoroughly examine all surfaces of the bearing components at low (3 to 50×) and high (100 to 500×) magnifications to classify the type of damage. Photographing significant areas is helpful for future reference.

Step III. Select areas for metallographic sectioning for confirming visual observations.

Step IV. Consider assembled information in comparing observed and expected life under the given conditions and determine the probable failure cause.

The visual examination is the most important part of the failure analysis and usually provides the necessary clues for establishing the cause of failure. The types of failures are described in the following sections, and photographs of typical examples are included for comparison with the bearing being analyzed.

2. Types and Examples of Failures

For convenience of analysis, all bearing failures can be divided into two categories, contact fatigue failures and nonfatigue failures. Nonfatigue failures often initiate fatigue failures, but will be considered separately because the real initiating cause of failure is not fatigue. Therefore, when fatigue failure is identified, further analysis is required to establish whether some nonfatigue cause had initiated it.

a. Contact Fatigue Failure

Any structural material that is subjected to repeated or reversed stress cycles of sufficient magnitude will ultimately fail by fatigue. Although steel in structural applications apparently exhibits an endurance limit below which reversing stresses will not cause fatigue failure, for practical bearing applications no such endurance limit exists [1]. Fatigue failure is, therefore, the normal limiting factor in bearing life.

A typical spall, or flaking, resulting from fatigue failure of a bearing raceway is shown in Figure 44 [45]. The spalls are often elliptical in shape

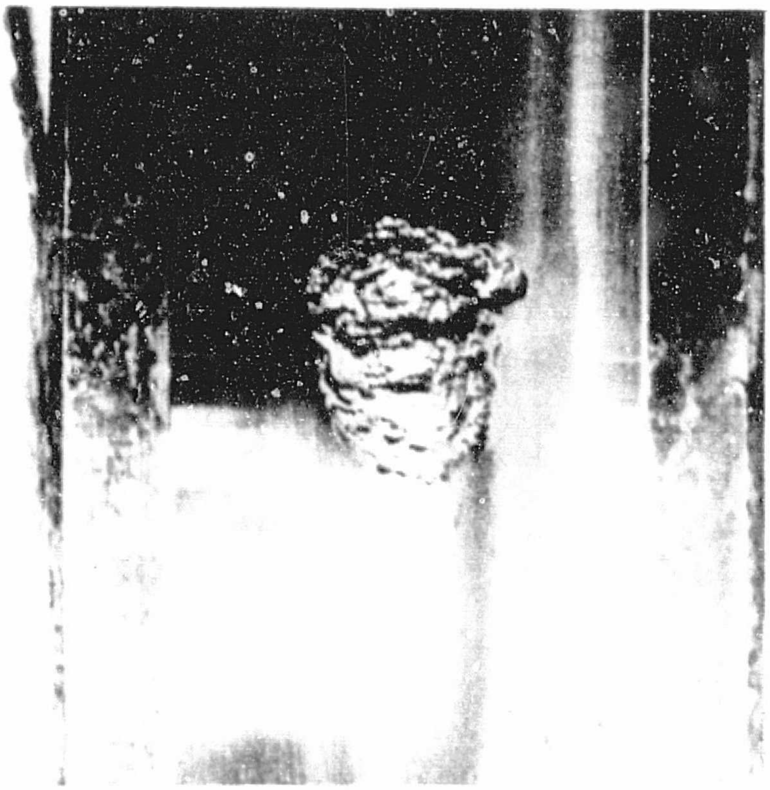

FIGURE 44. TYPICAL BEARING RACEWAY SPALL RESULTING FROM FATIGUE FAILURE

as they first appear on the raceway. Continued bearing operation expands the spall until eventually the entire raceway has the same roughened appearance.

A mathematical treatment of the fatigue life under various loading and operating conditions has been presented by Harris [1] and Tallian [46], and Bisson and Anderson [9] describe the parameters influential in fatigue failures. The most important variables are load, hardness, temperature, and lubricant. Load has a very marked effect; an inverse cubic relationship exists between load (as related to contact stress) and life [9]. The importance of this dramatic relationship will be seen in the analysis of failures resulting from nonfatigue causes. Decreases in component hardness, increases in temperature, and decreases in lubricant viscosity also shorten the fatigue life. Therefore, the extension of fatigue life for a given bearing should begin with a reduction of load (or contact stress). If the load is fixed, a larger bearing with reduced contact stresses on the components should have the desired effect. Additional improvement will also result from appropriate adjustment of the other variables.

b. Nonfatigue Failures

The most dramatic benefits of bearing failure analysis are derived from identifying nonfatigue causes of failure. These are the causes that greatly reduce normal bearing life, and although they often result in fatigue failure of the bearing careful analysis can reveal their identity.

(1) Mechanical Damage. There are numerous ways in which bearings can be damaged before and during mounting, and in service. Damage of this type is very common and is usually easily recognizable.

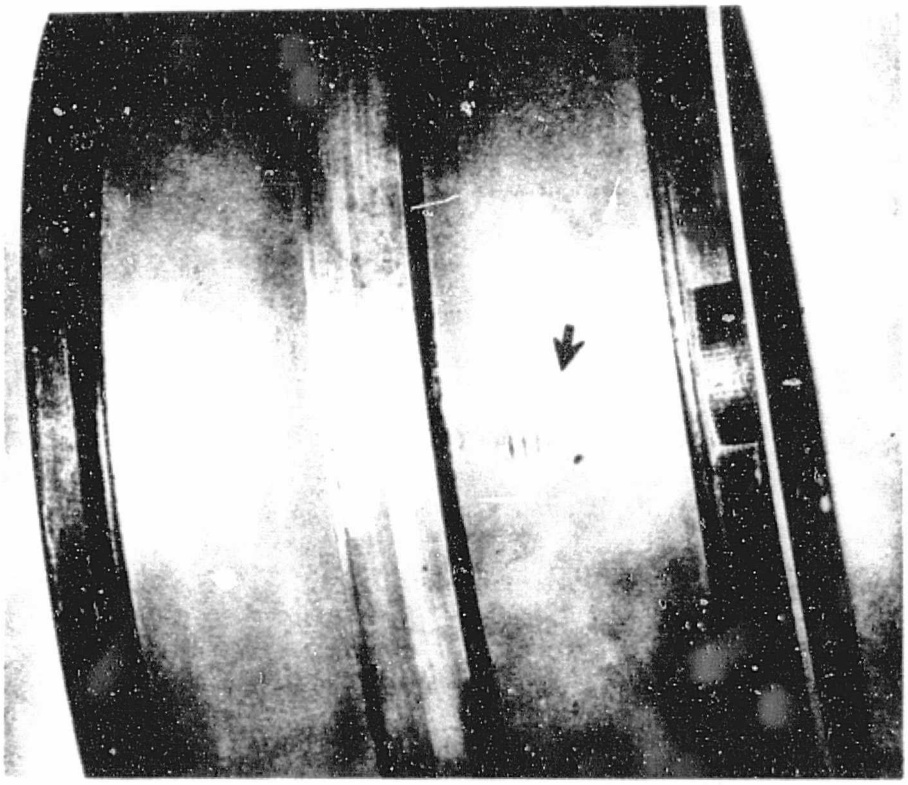

FIGURE 45. INDENTATIONS OF RACEWAY OF SPHERICAL ROLLER BEARING CAUSED BY ABUSE IN ASSEMBLY

FIGURE 46. BROKEN FLANGE ON INNER RACE RESULTING FROM HAMMERING IN MOUNTING

The primary cause of damage before mounting is forcible assembly. Close-clearance bearings and those using filling slots are particularly susceptible. For instance, Figure 45 shows indentations on the raceway of a spherical roller bearing resulting from tilting the rollers and hammering them into position [47]. The indentations cause rough operation and serve as stress risers to initiate fatigue spalling. Therefore, fatigue spalls occurring at the filling slot should be examined carefully for evidence of abusive assembly. Similar scratches and denting are found on both the rollers and raceways of spherical roller bearings when the races are forcibly swiveled during assembly, and on the elements of cylindrical roller bearings when they are maltreated during assembly.

Improper mounting procedures are a further common cause of bearing damage. Improperly applied (or nonuniform) pressure and hammering result in dented or broken raceways. Figure 46 shows a broken flange on an

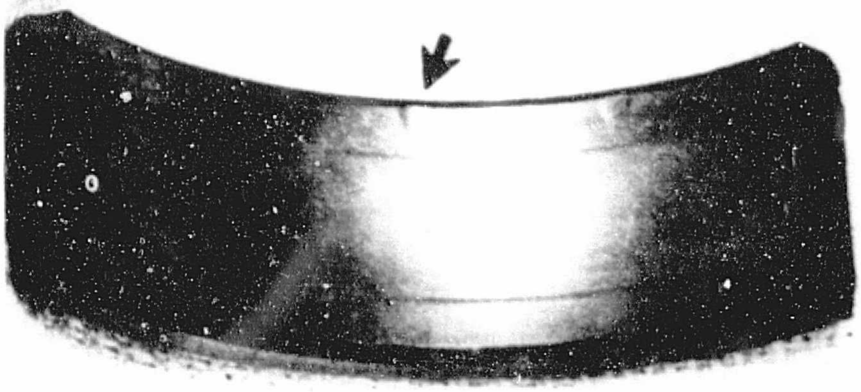

FIGURE 47. DENTS ON EDGE OF RACEWAY OF DOUBLE-ROW, SELF-ALIGNING BALL BEARING CAUSED BY APPLYING FORCE TO OUTER RACE RATHER THAN TO INNER RACE WHEN MOUNTING ON SHAFT

inner race caused by hammering during mounting [47]. Applying force on the outer race to mount the inner race, and vice versa, causes denting by the rolling elements at the edges of the raceways [47]. Figure 47 shows dents at the edge of the outer raceway of a double-row, self-aligning ball bearing resulting from applying force on this race when the inner race was being mounted on the shaft [47]. Similar damage can be imparted to the balls, as shown in Figure 48 [47]. Like the damage caused by improper assembly, these dented areas cause rough operation and serve as sites for fatigue failures.

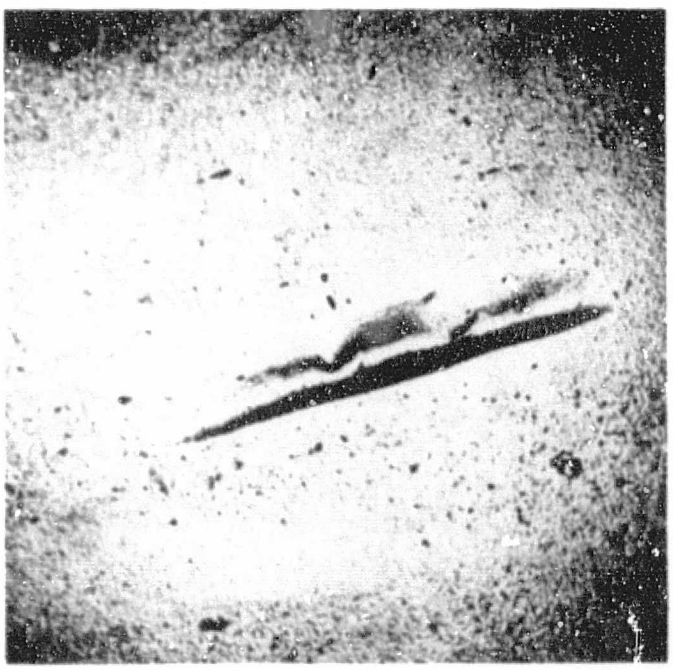

FIGURE 48. INDENTATION ON BALL CAUSED BY PRESSURE AGAINST EDGE OF RACEWAY IN FORCIBLE ASSEMBLY, OR BY IMPACT WITH SHARP OBJECT

Cleanliness during mounting is also very important. Allowing metal chips to enter a bearing by placing it on a dirty floor or bench is a common start of ultimate bearing failure. Such dirt can cause denting, noise, and locking of the rolling elements. The use of a torch to heat a bearing race during assembly is also extremely hazardous. Severe temperature gradients are produced, which can cause cracking or warping. Excessive heating will lower hardness and change dimensions [47]. The characteristic colors of oxide films that appear on bearing components is often a clue that excessive heating has taken place during mounting.

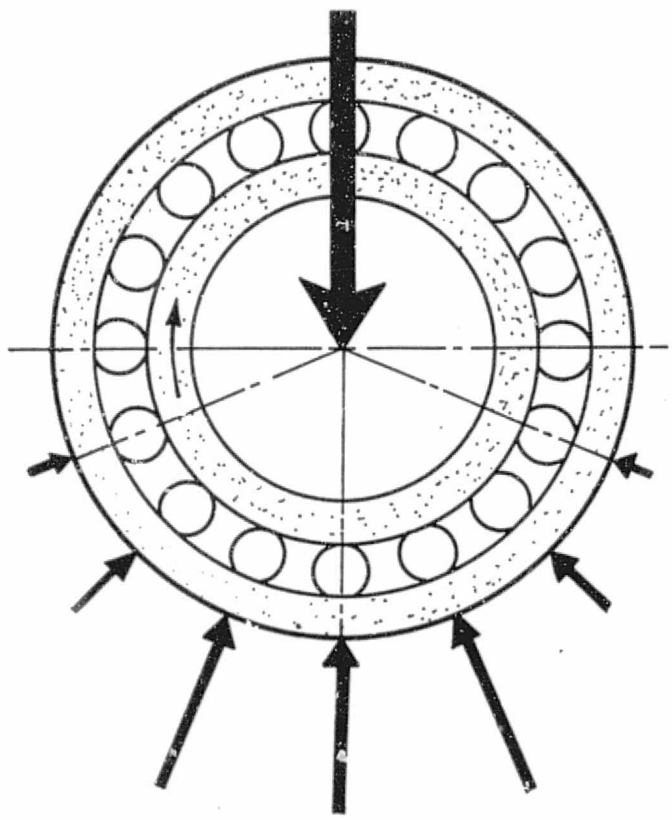

FIGURE 49. HOW THE APPLIED LOAD OF CONSTANT DIRECTION IS DISTRIBUTED AMONG THE ROLLING ELEMENTS OF A BEARING

Since the shaft and housing into which a bearing is mounted provide its alignment, improper geometry can initiate failures from misalignment. An understanding of the characteristic load zones on the raceways of improperly mounted bearings first requires knowledge of the normal patterns. Figure 49 shows how a constantly applied radial load is distributed among the bearing elements [45]. The length of the arrows represents the magnitude of the load. The rotating ring will have a continuous 360-degree load pattern, while the stationary ring will have a load pattern of less than 180 degrees [45]. Figures 50 and 51 [45] illustrate the expected patterns for a radial loading with the outer race fixed and rotating, respectively. The patterns of the load zones are usually visible after the bearing has been in service and result from a difference in reflectivity associated with surface films or very slight wear. The load zone on a deep-groove ball bearing operated with an axial (thrust) load is shifted from that of a pure radial load (Figure 52) [45]. This one-load condition results in a uniform continuous-load pattern on a full 360 degrees of both races. A combination of axial and radial load results in the pattern illustrated in Figure 53 [45], which is somewhere between the patterns resulting from a pure axial and radial load.

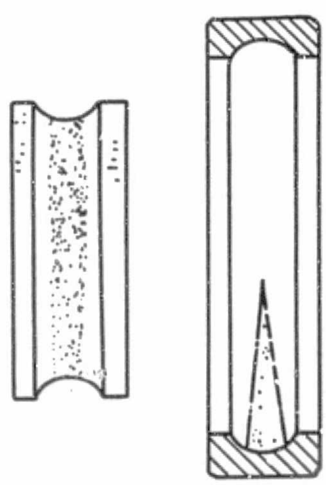

FIGURE 50. NORMAL LOAD ZONE, INNER RACE ROTATING RELATIVE TO LOAD

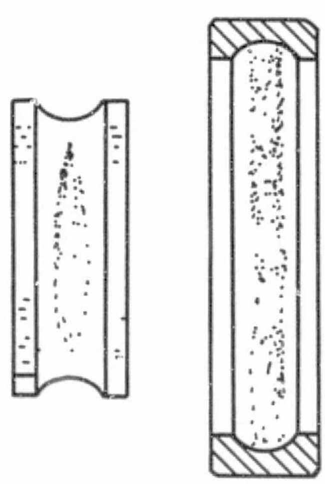

FIGURE 51. NORMAL LOAD ZONE, OUTER RACE ROTATING RELATIVE TO LOAD OR LOAD ROTATING IN PHASE WITH INNER RACE

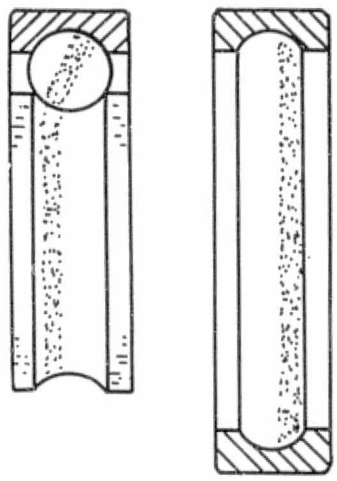

FIGURE 52. NORMAL LOAD ZONE IN A DEEP-GROOVE BALL BEARING WITH AN AXIAL LOAD

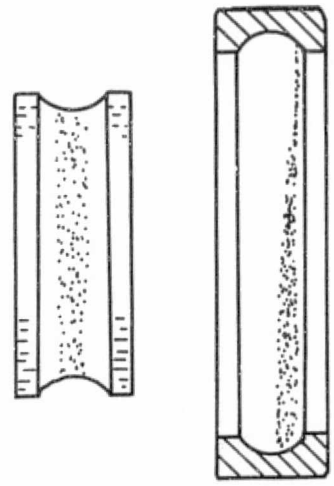

FIGURE 53. NORMAL LOAD ZONE WITH COMBINED AXIAL AND RADIAL LOAD

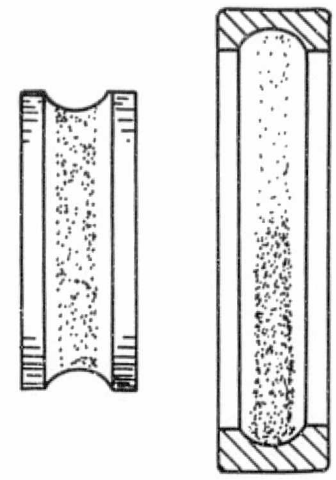

FIGURE 54. LOAD ZONE RESULTING FROM EXCESSIVE INTERFERENCE FIT OF INNER RACE ON SHAFT

Abnormal bearing mounting will cause characteristic changes of the load-zone patterns. An excessive shrink fit of an inner race on its shaft removes the bearing clearance and results in a preloading of the elements. In this case, the load zone will be continuous around both races and will usually be wider and heavier on the portion of the outer race where the applied load is superimposed on the internal preload (Figure 54) [45]. In effect, the bearing is carrying a higher load than that for which it was designed, and early fatigue spalling often results. Excessive interference can also produce hoop stresses that are large enough to cause cracking of the inner race. Figure 55 [47] shows a crack and associated fatigue

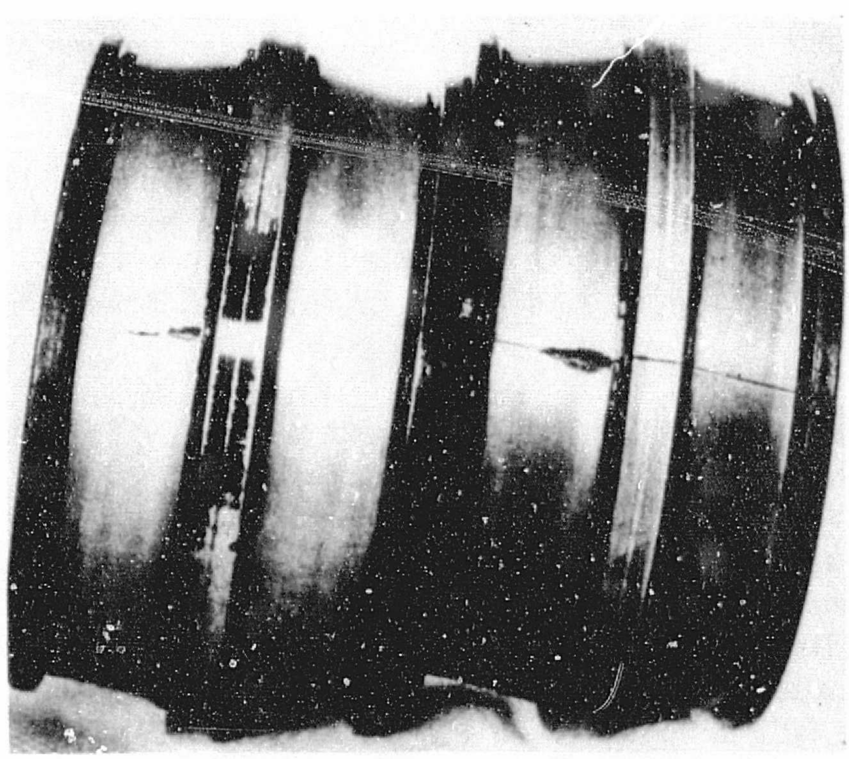

FIGURE 55. CRACKING CAUSED BY EXCESSIVE INTERFERENCE WITH SHAFT

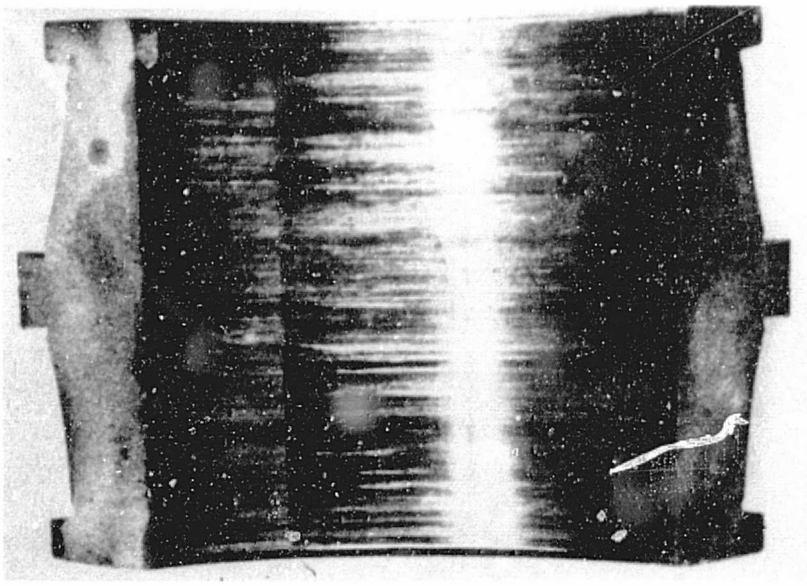

FIGURE 56. WEAR OF INNER BORE OF RACEWAY CAUSED BY INSUFFICIENT INTERFERENCE WITH SHAFT

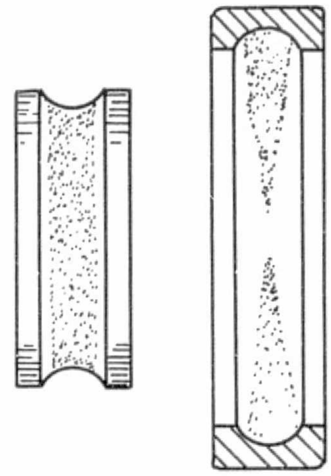

FIGURE 57. LOAD ZONE PATTERN RESULTING FROM OUT-OF-ROUND HOUSING BORE

spall resulting from excessive interference with the shaft. At the other extreme, insufficient interference can permit the race to rotate on the shaft, which results in wear and cracking [47]. In Figure 56, the raceway wore as a result of slipping on the shaft.

The load-zone patterns on bearing raceways also serve as indications of mounting out-of-roundness, misalignment, and shaft-to-housing misalignment. Figure 57 [45] illustrates the load zone pattern resulting from an out-of-round housing that pinched the outer race. In a study of the effect of housing and shaft out-of-roundness on raceway geometry, an originally round raceway was shown to assume closely the geometry of an out-of-round shaft [48]. The out-of-roundness causes high local loading, which can result in premature fatigue spalling.

Misalignment results in readily identifiable load-zone patterns. Figures 58 and 59 [45] illustrate the characteristic patterns when the races of a ball bearing are misaligned. Besides causing local high loads in the races which can cause premature fatigue spalling, misalignment produces excessively high stresses on the retainer. In such cases the retainer may crack or break and result in very early failure. Misaligned races in roller bearings produce high stresses near the ends of the rollers in local portions of the races, which frequently produces premature fatigue spalling. In Figure 60 [47] a trail of fatigue spalls were produced by high end-loading from misalignment. Recognition

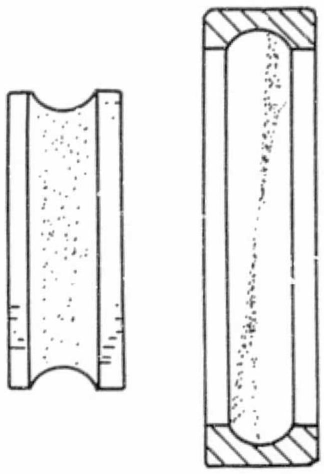

FIGURE 58. LOAD ZONE PRODUCED WHEN OUTER RACE IS MISALIGNED RELATIVE TO THE SHAFT

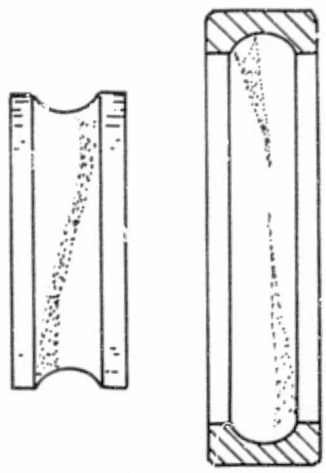

FIGURE 59. LOAD ZONE PRODUCED WHEN INNER RACE IS MISALIGNED RELATIVE TO THE HOUSING

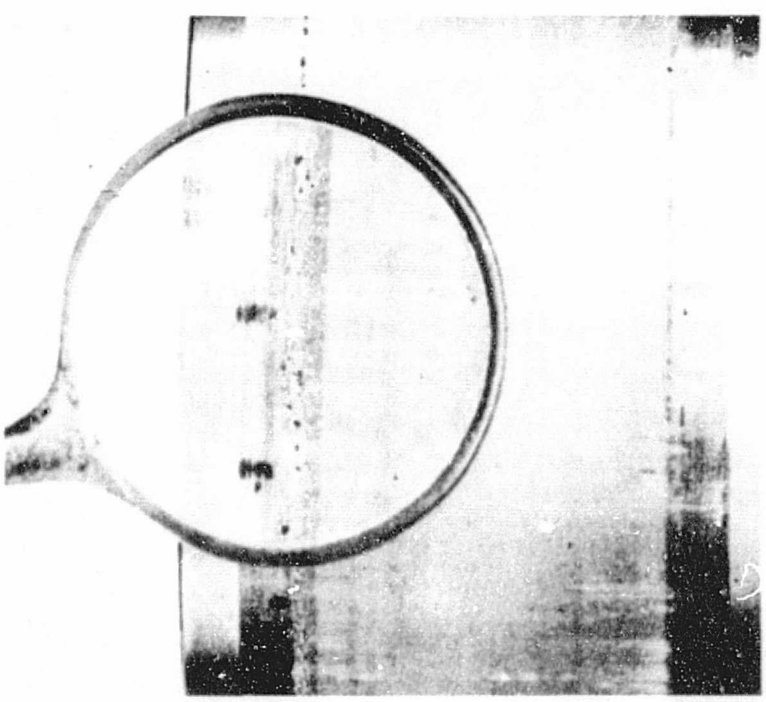

FIGURE 60. FATIGUE SPALLS NEAR END OF INNER RACEWAY OF CYLINDRICAL ROLLER BEARING CAUSED BY EDGE LOADING ON ROLLERS THROUGH MISALIGNMENT

from the load-zone patterns of failures caused by misalignment is usually straightforward.

Other characteristic load-zone patterns result from combinations of misalignment or distortion problems, and from improperly located bearings. For instance, axial loads can be imposed on a radial bearing by clamping its races in improper locations. Therefore, any bearing that exhibits an abnormal load-zone pattern should be carefully studied to determine the cause or combination of causes responsible.

(2) Abrasive Dirt. Rolling element bearings that are properly lubricated and are operating under normal conditions will not exhibit wear even after long periods of service. Therefore, the clearance should not change. If increases in clearance are noted, wear by abrasive dirt is often the cause [45].

The entrance of any dirt that is of sufficient hardness to cut or lap the bearing components will result in wear. The early stages can be detected by the dull or matte finish on the rolling elements and raceways, such

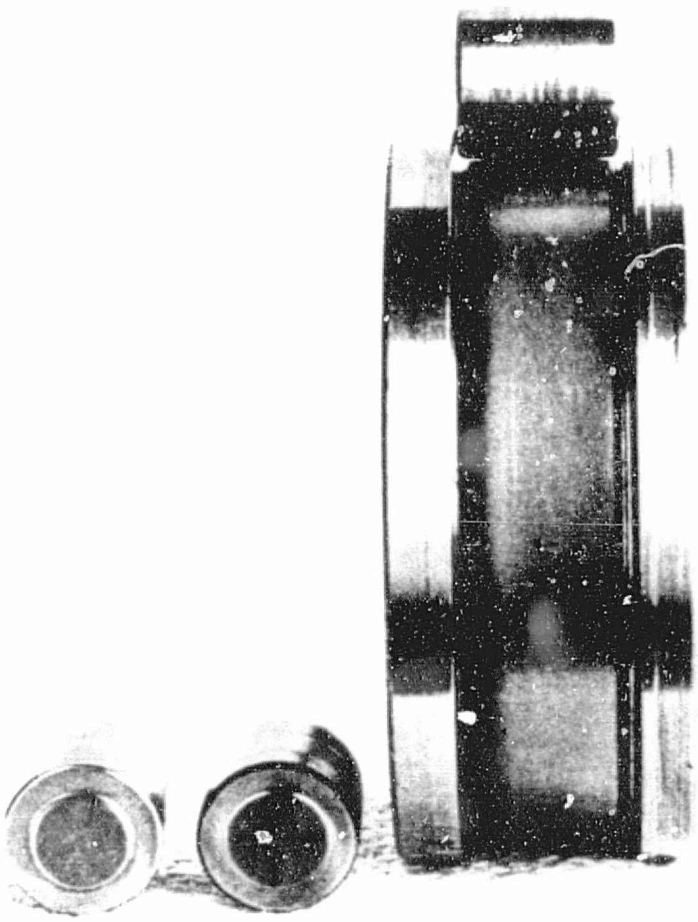

FIGURE 61. DULL OR MATTE FINISH CAUSED BY THE LAPPING ACTION OF ABRASIVE PARTICLES IN THE EARLY STAGES OF ABRASIVE WEAR

as that shown in Figure 61 [44]. Often associated with abrasive wear are irregular dents in the raceways, which are caused by the rolling elements passing over large portions of foreign matter. Such dents can also result from foreign material that is too soft to cause abrasive wear, in which case rough operation or retainer damage occurs without increases in bearing clearance.

As the wear from abrasive particles continues, excessive wear at internal locations of sliding contact is observed. The ends of rollers and the raceway flange of roller bearings, and the retainer pockets and guiding surfaces of all rolling-element bearings, are particularly susceptible to excessive wear. Recognition of abrasive wear is often possible by simple visual examination.

(3) <u>True and False Brinelling</u>. True brinelling is an indentation in a bearing raceway caused by plastic flow when an excessive stress is exerted through the rolling element. The source of the high stress is often shock loading during operation, or hammering before and during assembly. True brinell marks are characterized by their regular but often poorly defined shape and the presence of the original grinding scratches throughout the dented region. Their presence on a raceway can cause rough operation or jamming as the rolling elements pass over the dented region.

False brinell marks result from fretting corrosion that occurs between a rolling element and the raceway when a bearing is subject to vibration or small-amplitude motion while in static contact. The surface may be rough or polished in the mark, but the original grinding scratches are always rubbed out. While false brinell marks can usually be found in any bearing that has been in service for any length of time, only extreme cases of deep marks cause bearing malfunction. They, like true brinell marks, can cause rough operation or bearing jamming.

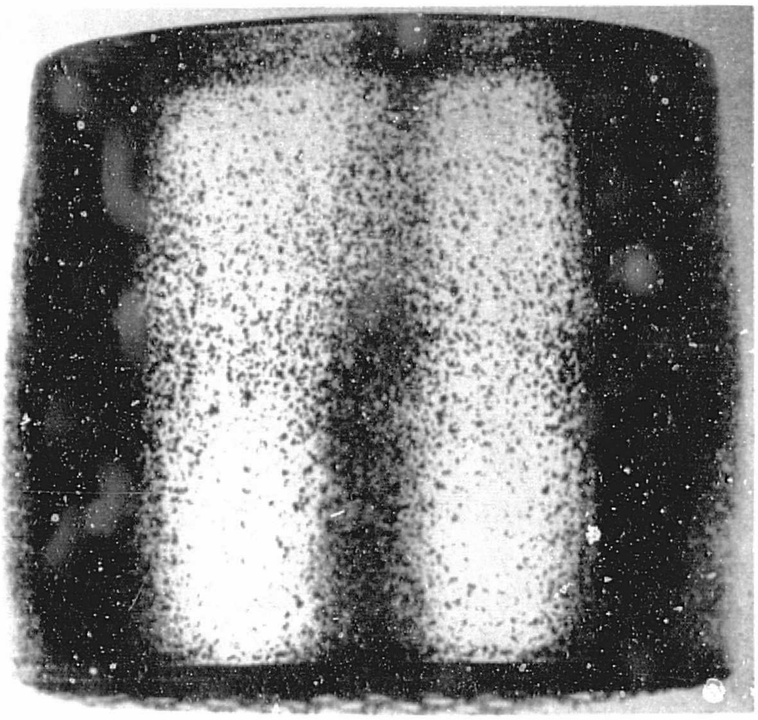

FIGURE 62. CORROSION OF ROLLER SURFACE CAUSED BY FORMATION OF ACIDS IN LUBRICANT WITH MOISTURE PRESENT

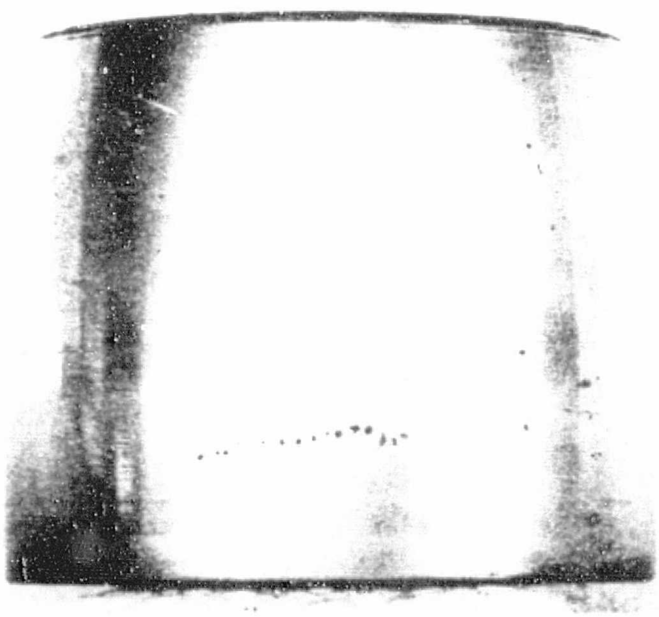

FIGURE 63. ELECTRICAL PITTING ON SURFACE OF A SPHERICAL ROLLER CAUSED BY PASSAGE OF RELATIVELY LARGE CURRENT

(4) Corrosion. With the exception of bearings made of stainless steel or other special materials, the finished surfaces of bearings are highly susceptible to corrosion by water, acid, and other agents [47]. The corrosion can occur after degreasing, before installation, or during service. The entrance of water or other corrosive fluids into the lubricant is often responsible for corrosion during service. The corrosion usually occurs in localized pits, such as on the roller in Figure 62 [47]. While mild corrosion often does not interfere with normal bearing operation, extensive corrosion can result in rough operation, seizure fatigue spalling, and stress-corrosion cracking.

(5) Electrical Current. The passage of electrical current through a bearing produces arcing between the components, which results in pitting of both surfaces. The pitting can take many forms depending upon the amperage of the current and the time of passage. With relatively large currents the pits are separated and large, as on the roller in Figure 63 [47]. With lower currents and in the presence of vibration, "fluting" results (Figure 64) [47]. The initiation of

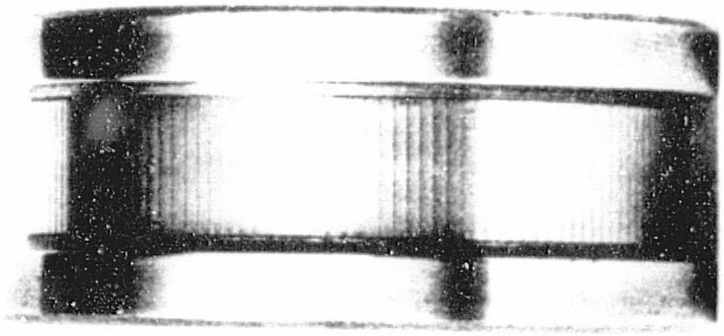

FIGURE 64. FLUTING ON INNER RACEWAY OF CYLINDRICAL ROLLER BEARING CAUSED BY PROLONGED PASSAGE OF SMALL CURRENT IN PRESENCE OF VIBRATION

fluting requires a synchronization of electrical current and vibration, but once started it is probably self-perpetuating [47]. The resulting roughened surfaces cause poor bearing operation and can lead to fatigue spalling.

The identification of electrical pitting can be confirmed by metallographic sectioning. The electrical arc raises the temperature of the surface and causes tempering of the structure, which can be observed metallographically. Higher electrical currents cause melting and other readily identifiable high-temperature effects. Elimination of damage by electrical current requires routing the current around the bearing rather than through it.

(6) Lubrication. Numerous possibilities exist for failure caused by inadequate lubrication. The failures are often severe because they are self-aggravating. Metal smearing and flow, worn or broken retainers, fatigue spalling, and oxide films from high temperatures can be associated with lubricant failure. Lubricant failure can be of several forms. If the bearing is being cooled by the lubricant, an interruption in flow can raise operating temperatures and initiate failure. On the other hand, excessive lubricant can cause bearing heating with the same end result. Incorrect viscosity at operating temperature can result in inadequate elastohydrodynamic film thicknesses, which will accelerate failure by wear and fatigue. While identification of the damage resulting from inadequate lubrication is usually straightforward, applying the correct lubrication technique to a rolling element bearing is complicated by the several operating requirements (Section V).

(7) Materials and Manufacturing Defects. In spite of care in the manufacturing of bearings, they will occasionally fail as a result of shortcomings in the materials and fabrication procedures. Improper hardness, poorly controlled heat treatment procedures, and large impurity inclusions can initiate early bearing failures. Excessive grinding temperatures can cause premature cracking or spalling. Poor bearing geometry, such as out-of-round

rollers and races and improper dimensions, is also a potential trouble source. In most cases, establishment of this type of defect requires detailed metallographic examination or dimensional measurements.

Section X. TRENDS IN ADVANCED LUBRICATION FOR AEROSPACE ROLLING-CONTACT BEARINGS

1. Lubrication

The following subsections are based on reports available either in the open literature or in NASA technical documents and contractor reports. For the most part, the comments given here are paraphrased from the original reports, with an occasional critical comment supplied by the present author. It is apparent that for some types of aerospace applications more than one solution to lubrication is possible. This discussion does not try to evaluate any of the reported approaches in terms of an optimum solution. They have been selected, however, with the intent of providing useful precedent.

It will be obvious that the most complete data have been obtained on ground-based tests. Actual flight experience is presently limited to knowledge of whether the lubricated apparatus worked or failed. It is encouraging to note that very few lubrication failures in space have been found.

a. Lubrication by Liquids

One of the earliest demonstrations that liquid lubricants and greases might be used in vacuum conditions was reported by Freundlich and Hannan [49] in 1961. They examined a series of mineral oils, esters, and silicones by testing them in R-2 bearings in electric motors at 10^{-5} torr and 75°C (167°F). They found that examples of all three types of fluids were capable of giving 1000 hours life with the strong indication that much longer times would have been possible.

The next year, Coit and Sorem [50] published data on lubrication of ball bearings of an electric motor at pressures of about 4×10^{-5} torr and 200°F. Distillate mineral oils were found to be too volatile, but several heavier-molecular-weight oil fractions (known as "bright stocks") gave bearing life in the 100-hour range. Several synthetic oils were tested, such as silicones, polyphenyl ethers, esters, and polyethylene oxides. Generally poor results were obtained with these, except for the polyethylene oxides, which gave good performance despite apparent high volatility in evaporation tests. The results in this paper are most important for providing an early indication that correlations between volatility and bearing performance are not simple. Surface adsorption of thin films of lubricant can prolong lubrication even when the bulk lubricant has evaporated.

Further information on the use of mineral oils is contained in a paper by Murray, Lewis, and Babecki [51]. They tested Type 205 angular-contact ball bearings at 30 rpm, 20°F, and pressures of 10^{-7} torr or lower. They found they could get a bearing life of greater than 1000 hours by using a mineral oil described only as "proprietary low-vapor-pressure oil." After 1200 to 1400 hours, the torque rose suddenly in six different bearings. Torque dropped again if air was admitted, and the results could be repeated. The authors speculated that depletion of water and oxygen traces was the cause of the torque rise. Water and oxygen are considered necessary to reoxidize iron bearing surfaces which were worn off by use. This paper also notes that the oil tested had been used in an unspecified orbiting satellite for more than 8 months with successful results.

A more recent paper by Harris, Read, and Thompson [16] records test experience with several different silicones and a mineral oil in small electric motor bearings at 3000 rpm. Temperatures ranged from about 30° to 56°C (86° to 132°F), and pressures from about 10^{-9} to 10^{-7} torr. The pressures were low at the start, but the vapors produced in the test environment always raised them. Bearing lives of the order of 1000 to 7000 hours were achieved, with some tests still running at the time of reporting. One of the most significant findings in this work was that the bearing design was very important. Full-complement bearings were much shorter-lived than ones with phenolic retainers. Another finding which was not unexpected is that provision for lubricant reservoirs (a sintered nylon sleeve impregnated with oil in contact with the bearings) improved bearing life.

Perhaps the most extensive record of test experience with oil-lubricated bearings in space applications has been compiled over a number of years by Clauss, Silversher, and various co-workers [14, 15, 52]. (This work also includes greases and solids and will be referred to again.) At the time of a report dated April 1968, various test bearings had been run for over 4 years in vacuum with oil and grease lubricants. Some of these tests (limited to 15 months) included radiation to 10^7 rads, in addition to the other environmental conditions. These included light bearing loads, speed of 8000 rpm, temperatures around 175°F, and a vacuum of 10^{-8} torr. The ball bearings used were Conrad-type, R-3 bearings, some with phenolic retainers and some with steel-ribbon retainers.

Many different kinds of lubricating oil were tested in this program. The longest test run was for a fluorosilicone oil which was still running after 48,000 hours (5½ years). However, very long lives were also obtained with a chlorophenylsilicone, a high-phenyl silicone, and at least two specially refined petroleum oils. Curiously, tests on a series of synthetic esters showed much shorter lifetimes than did tests employing silicones and petroleum oils. Lifetimes of up to 2700 hours were obtained for esters, but as a class, they seemed to be distinctly inferior in this test.

Some other conclusions drawn from this program are that there is a correlation between viscosity of the lubricating oil and lifetime, but only a very poor one between vapor pressure of the lubricant and lifetime. The mechanism of lubrication failure with oils usually is loss of the oil from the bearing area, but this loss can be due to creep as well as to evaporation. Also, there is an implication again that an adsorbed film of polar lubricant or some fraction of the lubricant can help maintain lubrication by resisting creep or evaporative loss.

Liquid lubricants for high-temperature applications have been studied more intensively in the last few years than ever before. The Air Force and the Navy use ester-base lubricants for current jet engines. The Navy oil is expected to perform at temperatures to 350°F and good experience has been accumulated. The Navy specification is MIL-L-23699, and MIL-L-7808 is the Air Force specification.

Jet-engine oils for more advanced aircraft such as the SST require temperatures beyond the capacity of current esters. The NASA-Lewis Research Center has been conducting a program to evaluate candidate lubricants for the SST, and a whole series of technical notes and contractor reports has been issued documenting the results of this program [53-57]. The maximum service temperature for partially inert atmospheres seems to be on the order of 600°F; however, in completely inert atmospheres, a synthetic hydrocarbon fluid has been used for at least a few hours at 700°F. The perfluoroalkyl ether fluids mentioned in Section IV seem to be limited to short-term service at 600°F.

Little literature on radiation-resistant lubricants for aerospace use has been published in the last few years. One reference to an evaluation of a combination coolant/lubricant for SNAP-8 mentioned that polyphenyl ethers and ten other fluids were examined [58]. (The original report has not been studied, and the abstract available failed to mention results of the evaluation; however, polyphenyl ethers are most likely to have been the preferred materials.) Quite a few papers on radiation-resistant lubricants published in the early 1960's showed that polyphenyl ethers could resist radiation to over 10^9 rads and temperatures to 500°F, imposed concurrently. Some partially hydrogenated polyphenyls were used as reactor coolants in the Piqua, Ohio, Organic-Cooled and Moderated Reactor, but these were not particularly good lubricants. Also, at reactor radiation flux levels, the formation of radiation degradation products was fairly rapid and the coolant required frequent purification and makeup.

A paper by Butner and Rosenberg [18] has discussed successful short-term lubrication of ball bearings by rocket fuels such as liquid oxygen, liquid hydrogen, nitrogen tetroxide, and RP-1 rocket fuel. They found that bearings made of 440 C stainless steel and having retainers of glass-filled (or supported) PTFE were the most successful of the many material combinations tried. The bearings could be run submerged in the fuel, but lubricant supplied by jets was considered better.

A number of publications have begun to appear summarizing some of the successful experience obtained with various types of space satellites. An article by Delaat, Shelton, and Kimzey [19] on the lubricant requirements for the Apollo missions notes that oils were used only in shock struts of the docking mechanisms and in sealed bearings of tape recorders and cameras. The use of sealed systems, of course, permits the use of liquid lubricants without fear of evaporation in space vacuum. Some other sealed-system applications have been summarized by H. E. Evans of NASA-Goddard Space Flight Center in a presentation at the American Society of Lubrication Engineers in Philadelphia in May 1969. This presentation, and a similar one by S. Drabek of General Electric Space Systems, are to be published in 1969 as part of a symposium by the ASLE [59, 60]. Table XIV summarizes the applications of liquid lubricants noted in those papers.

"Apiezon" products are specially formulated for vacuum usage, but the other lubricants in this series were developed originally for specific lubricating ability. The MIL-L-6085A product is a low-volatility synthetic instrument oil; MIL-L-7808 is a jet-engine oil; and Terresso V-78 is a well known gyroscope lubricant (now no longer available). The success of these oils in the satellite applications is probably due to their being protected from space vacuum.

b. Lubrication by Greases

The paper by Freundlich and Hannan [49] contained data on grease use in vacuum. This paper was an early demonstration that some greases could be used successfully in ball bearings in low-pressure environments.

A 1964 paper by Schwenker, of the Air Force [13], gives a general discussion of military-specification greases and their potential application to

TABLE XIV. SATELLITE LUBRICATION BY LIQUIDS

Satellite	Mechanism	Bearing Type	Lubricant	Successful Life
Nimbus	IR Sensor	Optical prism rotor	MIL-L-7808 (ester	32 months +
		Drive motor	MIL-L-6085A (synthetic)	32 months +
	Gyro		Terresso V-78 (mineral oil)	One failure in 24 months, some more than 6 years
	Momentum flywheels		MIL-L-6085A	32 months
OGO	Wobble drive torque tube		Brayco KK949A (mineral oil)	1 year (+ ?)
	Drive motor and SR-2 speed reducer		MIL-L-6085A	1 year (+ ?)
	Reaction wheel	1-inch deep groove	MIL-L-6085A	1 year (+ ?)
OSO	Azimuth bearing (sail)	No. 3308 and No. R-18	Apiezon C	Not given
	Various small bearings		Apiezon C	Not given
	Tape recorder bearings	R-2	MIL-L-6085A	Not given
Applications Technology Satellite	Antenna bearings	20-mm	Apiezon C	Not given
	Gravity gradient boom drive	"Instrument"	F-50 oil (chlorosilicone)	Not given

space applications. A more recent description of the military specification greases and their uses is contained in Air Force-Navy Aeronautical Bulletin No. 275f, 15 June 1966. This document is an Air Force-Navy Aeronautical Bulletin intended to guide selection of both oils and greases, primarily for aircraft applications.

A useful description of test results on recently developed high-temperature, low-volatility greases is given by Christian and Bunting [61]. They found favorable high-temperature properties, good load-carrying ability, and good resistance to space vacuum in R-4- and 204-type bearings for greases based on a perfluoroalkyl ether fluid. A closely related grease was selected by Delaat, Shelton, and Kimzey [19] for use in the Apollo program because the grease also showed good compatibility with rocket fuel, and resistance to the high-oxygen content in the environment of the Apollo command module. This paper contains useful data on many other greases studied in the same program.

The series of publications from Lockheed authored by Clauss, Silversher, and others also contains a large number of significant test data on grease-lubricated bearings. The most recent report [15] shows that some greases have been running for over 4 years. A general conclusion is that many greases can be expected to last for more than 2 years. Test conditions were similar to those used for the liquids, including vacuums generally in the range of 10^{-7} to 10^{-9} torr, and temperatures to about 225° F. The 2-year life estimate allows for temperatures to 225° F, but higher temperatures definitely would lower lubrication life. The primary cause of failure in all this work was loss of liquid from the grease due to bleeding and evaporation. When too much liquid has been lost, greases become dry and stiff, and either cannot lubricate by themselves or they interfere with ball-retainer motion. The most successful greases in the Lockheed program were based on halogenated silicones (chlorophenyl and trifluoropropyl) and phenylmethyl silicones.

Another recent report of grease test results is in the paper by Harris, Read, and Thompson [16]. Their tests in small electric motor bearings included several greases which were not well identified. However, one of them, G-300, is apparently a chlorophenylsilicone thickened with a lithium soap. Harris et al. found lifetimes of over 7000 hours with the greases in certain types of bearings, and less life in others. This may also be the same grease identified by Williams [62] as a high-temperature airframe-bearing grease capable of providing low torque in oscillating ball bearings.

In cases where high temperature is the principal stress, the recent results of Sliney and Johnson [63] are of interest. Four greases were tested in the slave bearings of a rig designed to test solid lubricants at 2000 to

5000 rpm and at 1500°F. The grease tests themselves were run at 325° to 600°F. Grease life varied from over 700 hours at 325°F to a maximum of 20 hours at 600°F. The most successful greases were based on a fluorocarbon oil which is probably the perfluoroalkyl ether type mentioned earlier. Greases based on polyphenyl ethers performed somewhat less well on both life and bearing torque characteristics.

Some grease experience was related in the papers by Drabek and Evans at the ASLE Aerospace Lubrication Symposium in May 1969 [59, 60]. Bearing failure in a Nimbus II solar array drive was traced to excessive inner-race temperature for a particular bearing. After a redesign, the race temperature was reduced from 300° to 160°F. This raised the bearing-lubricant life from 26 days to over 32 months. The grease in this case was the lithium soap-chlorophenyl silicone grease mentioned above. Various greases were mentioned as being used in the OGO and OSO satellites, but these were presumably used in sealed systems where vacuum was not a factor. The chlorophenyl silicone grease was also used in two places in the Applications Technology Satellite.

c. Lubrication by Solids

As noted in Section IV, solid lubricants can be used in more different ways than can liquids or greases. Research and test programs on solid lubricants have been widely scattered in the literature, and only part of the work has been directly addressed to space applications. Also, the great majority of solid lubricant work has been done on sliding bearings rather than on rolling bearings. All of these considerations make it difficult to break down the pertinent literature into neat categories. Most of the interesting references include more than one kind of lubricant and method of application for the same investigation. Therefore, the following discussion will be organized by reference rather than by type of solid lubricant or means of application.

Vest and Ward of NASA-Goddard Space Flight Center have reported [64] that composite retainers for ball bearings can give good life even when motion is oscillatory or slow-speed rotary. The retainers were composed of PTFE and MoS_2 reinforced with fiberglass. Oscillatory tests were run on R-6 bearings and 100-rpm tests were run on B542 bearings. Loads were light, temperature was presumably room temperature, and vacuum was of the order of 10^{-7} to 10^{-9} torr. Torque increase was used as the indication of failure. Successful operation was demonstrated for times of 10,000 to 12,000 hours.

Harris et al. have reported a variety of different test results with solid lubricants [65-67]. Tests were done at 3000 rpm in electric motor bearings, some of which had retainers and some did not. Better life was obtained generally with bearings having retainers. Table XV shows the lubricants used. Vacuum was of the order of 10^{-9} to 10^{-10} torr. Temperatures

TABLE XV. SOLID FILM LUBRICANTS TESTED

Classification	Lubricant Coating	Retainer
Soft metal films	Lead	Lead-bronze
	Lead	Phenolic
	Lead	---
	Gold	---
	Gallium	---
	Silver	Tool steel
	MoS_2 burnished	---
Lamellar solid and combination films	MoS_2 + lead	---
	MoS_2 + gold	---
	MoS_2 + sodium silicate	---
Plastic film	PTFE	---
	--	Lead-bronze
Self-lubricating	--	PTFE + glass
	--	PTFE + mica
	--	PTFE + glass + MoS_2

were those generated by the motor and bearing and ranged from 50° to 90°C.

In the Harris study, the best performance among the metal-film lubricants was obtained from electrodeposited lead. However, they felt that silver had not been evaluated sufficiently to discard. Use of lead-bronze retainers together with lead films was also promising because it reduced the torque of cageless bearings without lowering life. Thirteen pairs of bearings using lead-films completed from 8000 to 15,000 hours of continuous running without failure. One test run at -40°C ran for 7200 hours before failure.

Molybdenum disulfide coatings gave erratic results in the work by Harris. A sodium-silicate-bonded MoS_2 film did give 7300 hours bearing life in one case, but many other tests were shorter lived due to flaking of the coating and jamming of the bearings. PTFE retainers gave lives no longer than 2600 hours, and increasing axial load shortened that life drastically.

The work at Lockheed by Clauss, Silversher, et al. [15] has shown that a reinforced Teflon (PTFE) retainer used with R-3 bearings could give lifetimes to about 10,000 hours, provided the axial load was light. In general, failure of these bearings was due to wear of the ball pockets or breakage of the retainer. Several other plastic composite types were also tried with rather poor results. This program also included some work with bonded solid-film lubricants, but they generally showed lifetimes of no more than 3 months at best, and test reproducibility was poor.

Murray, Lewis, and Babecki [51] studied the lubrication of Type 205 angular-contact ball bearings in both a constant rotation and oscillatory mode. Test temperature was -25° to 20°F, and the vacuum was 10^{-7} torr or higher. Tests included dry bearings (phenolic retainer), a sodium-silicate-bonded MoS_2-graphite film, a burnished MoS_2 film, an MoS_2 film applied over evaporated metal layers, and a second type of burnished MoS_2 film. Bearing speeds were relatively low. Typical test lives were from a few hundred hours to a maximum of 2000 hours. The silicate bonded MoS_2-graphite film gave the best results of the group. The work also showed that it is necessary to "run-in" the bearings carefully before testing and that failure is usually due to wear of the lubricant film followed by jamming of the bearing by debris.

The use of polyimide resins as a self-lubricating retainer has been demonstrated by Devine and Kroll [68]. The polyimide worked better when modified with a graphite filler. Testing was concentrated on runs at 10,000 rpm in 204 size bearings. High temperatures (to 700°F) were used in air atmosphere. Lifetimes ranged from tens of hours to a few hundred hours. The bearing life depended strongly on some mechanical design details, as well as on the composition of the lubricant. The polyimide appeared to function by transferring a thin film to the ball surfaces. Retainer wear and breakage was a typical source of failure.

2. Application of Elastohydrodynamics to Bearing Design

For the past decade, significant activity in elastohydrodynamic (EHD) lubrication science has been in progress. However, only in the past few years has this science started to emerge from the academic areas into practical application. Now there are some publications on practical EHD worthy of discussion.

a. Experience Reported by SKF Industries

The evidence of practical applications of EHD lubrication is doubtless shrouded by "proprietary rights" exercised by the users and suppliers of bearings. SKF Industries have made notable efforts to demonstrate the validity of designing and selecting bearing systems around the concept of effective lubrication, and to supply techniques for these designs.

b. Studies Reported by Given

A part of the SKF studies were reported by Given at an engineering conference at Dartmouth [20]. To approximate the type of lubrication condition possible in a bearing, Given presented the data in Figure 17 (Section V) of this document. This figure was prepared on the hypothesis that lubricant film thickness is insensitive to loading. The three regimes shown in this figure are indicative of the three lubrication regimes discussed earlier. According to Given, the chart "enables one to rapidly evaluate whether one must take steps to avoid a drastic reduction in bearing fatigue life or if a bonus life might be expected." Apparently the normal life regime is based on a value of Λ between 1.5 and 3.5. Given says further that present technology indicates that bearing life is favorably influenced by higher Λ values.

To obtain a better feel for the influence of the Λ factor on wear rate and film presence, Given presented Figure 17. The wear rate was determined by a radio-tracer technique, and SKF typically uses electrical continuity (asperity clash counting schemes) to measure the extent of the lubricant film. For values of $\Lambda >> 3.5$ it is apparent that the lubricant film is nearly complete.

c. SKF Design Charts

Based on their experience with effects of lubrication regime on bearing life, SKF engineers prepared a series of figures to be used for predicting lubricant effects on bearing performance [21]. These charts are reproduced as Figures 18 and 19 (Section V). These charts are based on classified film-thickness equations and the film factor Λ is given by

$$\Lambda = H\left(\mu_o \lambda\right)^{0.7} N^{0.7} P_o^{0.09}, \tag{4}$$

where P_o is the equivalent static load. Therefore, by using Equation (4) estimation for the lubrication parameter which is more consistent than that obtainable from the more general chart of Figure 17 can be obtained.

To compare the prediction using the SKF charts with those using Equation (2), $P_h = 200,000$ psi and $\lambda = 10^{-4}$/psi. From Equation (3) it can be shown that for $N = 0.3$

$$\frac{h_G}{h_i} = 3.7.$$

This implies that the thickness predicted by Equation (1) (which is nearly the same as used in the SKF studies) yields a thickness which is nearly the same as is the ratio for critical values of Λ. Thus, predictions about conditions where effective separations occur are somewhat consistent.

3. NASA-Lewis Coordinated Studies

NASA-Lewis Research Center is coordinating several research programs to determine usable design criteria for the lubrication of mainshaft gas-turbine engine bearings for advanced aircraft such as the SST system. These research programs, which are being conducted at General Electric [69], SKF [70], and Battelle's Columbus Laboratories (BCL) [71], involve studies dealing with various aspects of the bearing-lubrication process. The major aspects of these programs were discussed in Section VI of this document dealing with high-speed, high-temperature bearings. However, one subject of the coordinated effort of direct interest here concerns the General Electric-BCL studies. General Electric is conducting a series of full-bearing fatigue tests under various loads, speeds, and temperatures. At BCL, studies are being conducted on fundamental aspects of the lubrication process for conditions which overlap those used for the General Electric tests. The purpose of the BCL study is to obtain insight into the role of lubrication on the full-bearing results, and also to obtain methods for extrapolating these results.

The General Electric phase of the program involved conducting a series of fatigue tests with 120-millimeter bore, angular-contact bearings. The majority of the experiments on this program reported by Bamberger [69] were conducted at 600°F with M-50 steel. The following conclusions are pertinent to this document.

a) A synthetic paraffin oil will provide satisfactory bearing operation at 400° to 600°F in a nitrogen atmosphere; temperature in the range studied makes little difference on performance.

b) A polyphenyl ether does provide satisfactory lubrication at 600°F.

c) A perfluoroalkyl ether fluid provides adequate lubrication, although "the adequacy of the lubricant film is not consistent."

At BCL [71], a disk machine was designed to operate under conditions similar to the full-bearing conditions used in the General Electric fatigue tests. A special pair of disks was designed to accurately simulate the ball-race contact conditions, including the ball-spin conditions of the bearing. The disks were finished with a 2- to 3-microinch finish. The disk apparatus is equipped with an X-ray technique for measuring lubricant film thickness and an electrical continuity scheme for measuring film breakdown.

The preliminary experiments indicated that a 120-millimeter bore bearing will operate with a finite film of synthetic paraffin oil lubricant (film about 1 microinch thick) at 600° F for typical bearing loads. More comprehensive experiments for other candidate lubricants which have just recently been completed show results which are consistent with the full-bearing fatigue-test data.

Section XI. CONCLUSIONS

Rolling-contact bearings for aerospace applications require careful consideration of three factors: design, materials, and lubricants. All of these factors play important roles in the reliable operational life of rolling-element bearings. Proper design insures safe stress levels in the rolling-element contacts (bearing life is very sensitive to contact stress level). Bearing material selection must be concerned with providing adequate load support (resistance to penetration and plastic deformation), contact fatigue resistance, corrosion resistance, and capability of producing high surface finish. Since rolling-contact bearings are generally procured as complete units, design and materials selection are somewhat limited to the standard varieties supplied by the manufacturer. The designer does exercise some influence on bearing geometry by specifying contact angle, conformity, and rolling-element size, for instance. It is important that the aerospace designer exercise as much control over the bearing geometry and material selection and processing as is provided him in selection or specification of rolling-contact bearings from bearing manufacturers.

Lubricants and lubrication are extremely important elements in bearing performance and have a significant influence on the reliable operation of rolling-contact bearings. The designer has probably more choices to make in the selection of lubricants and lubricating systems than he has in bearing design and materials selection. There are many lubricants and lubricating systems that can be used for one bearing design, depending on the conditions of operation. In any moderate to high-speed rolling-element bearing design, the performance can be greatly enhanced by selection of a lubricant and lubrication system which insures full film lubrication between rolling elements. This means that during rotation, the bearing load is carried on a fluid film rather than by metal-to-metal contact. The evaluation of the film condition can be made by a semi-empirical method which requires a few, although not extensive, calculations. Experimental evidence of the validity of this concept, although not extensive, is certainly consistent. It is expected that the effectiveness of this approach (elastohydrodynamic analysis) will increase as a more accurate characterization of lubricants is obtained under the conditions of shear, pressure, temperature, and very short transit time through the contact zone. It is possible that it could be applied not only to bearing fatigue but also to design for minimum wear in preloaded bearings.

Lubricants have essentially been the limiting factor in high-temperature bearing applications. As higher-temperature lubricants have developed, bearing operating temperatures have increased as well.

New concepts are being developed in bearing design to provide greater diversity in rolling-contact bearing application. In recent years, for instance, experiments have been carried out on bearings with hollow rollers. The hollow roller concept provides greater shock resistance, ability to comply to unfavorable thermal gradients, and less tendency for misalignment. This concept is still in the developmental stage but holds promise for future bearing applications.

New materials and material processing are being applied to rolling-contact bearing requirements for longer life at heavier loads and higher temperatures. In recent years, M-50 tool steel has become an accepted bearing material for high-temperature use. More recently M-50 steel has demonstrated improved fatigue resistance over AISI 52100 steel. Still further improvement in fatigue resistance has been accomplished by ausforming M-50 tool steel rolling elements. Fatigue resistance of AISI 52100 steel bearings has been improved by the use of special heat treatment techniques which produce residual compressive stresses in the race surfaces.

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