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BEARINGS: TECHNOLOGY AND NEEDS

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BEARINGS: TECHNOLOGY AND NEEDS

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ABSTRACT

A brief status report on bearing technology and present and near-term future problems that warrant research support is presented. For rolling element bearings a material with improved fracture toughness, life data in the low Λ region, a comprehensive failure theory verified by life data and incorporated into dynamic analyses, and an improved corrosion resistant alloy are perceived as important needs. For hydrodynamic bearings better definition of cavitation boundaries and pressure distributions for squeeze film dampers, and geometry optimization for minimum power loss in turbulent film bearings are needed. For gas film bearings, foil bearing geometries that form more nearly optimum film shapes for maximum load capacity, and more effective surface protective coatings for high temperature operation are needed.

LIST OF SYMBOLS

C	radial clearance, m (in)
D	diameter, m (in)
E	modulus of elasticity, N/m^2 (psi)
E'	modified modulus of elasticity, $\dfrac{2}{\dfrac{(1-\nu_1^2)}{E_1}+\dfrac{(1-\nu_2^2)}{E_2}}$
F	normal applied load, N (lb)
G	dimensionless material parameter, E'/P_s
H_{min}	dimensionless minimum film thickness, h/R_x
h	film thickness, m (in.)
k	ellipticity parameter, a/b
L	length, m (in)
R	radius m (in)
r	radius of curvature, m (in.)
R_x, R_y	effective radius of curvature, m (in.) $\dfrac{1}{R_x}=\dfrac{1}{r_{1x}}+\dfrac{1}{r_{2x}}$; $\dfrac{1}{R_y}=\dfrac{1}{r_{1y}}+\dfrac{1}{r_{2y}}$
U	dimensionless speed parameter, $(u\mu_a)/(E'R_x)$
u	surface velocity in x direction, m/sec (in./sec)
W_p	dimensionless load parameter, $F/(E'R_x^2)$
Λ	lubricant film parameter, h/σ
μ_a	dynamic viscosity, N sec/m^2 (lb sec/in.^{2})
ν	Poisson's ratio
σ	composite surface roughness, micrometers (μin.)
σ_1, σ_2	surface roughness of bodies 1 and 2, micrometers (μin.)

Subscripts:

1, 2	bodies 1 and 2
x, y	coordinate directions
b	bearing
p	preload

INTRODUCTION

In these days of intensive research efforts in a multiplicity of high technology areas which result in a steady stream of products unknown only a few years ago, one would think that anything as prosaic as bearings, which have been with us since the dawn of civilization, would long since have been relegated to R&D dormancy. That is not the case, however, because we find a continued high level of research and development activity in bearings and other mechanical components. A continued level of substantial support for bearing related R&D activity is a result of the recognized key role that bearings play in rotating machinery. Thus we find that advances in rotating machinery technology have necessitated parallel advances in bearing technology.

Large aircraft gas turbine engines, and transmissions and gear boxes for rotorcraft and turboprops have, for example, been a principal driver for improvements in rolling element bearings. The general demand for greater power density in rotating machinery has led to ever increasing speed demands on both rolling element and fluid film bearings. High speeds have intensified rotor dynamics problems, forcing greater attention on the stability aspects of fluid film bearings and rotor-bearing systems. They have brought about an increasingly wide use of non-circular, whirl resistant journal bearings and of viscous dampers configured as non-rotating journal bearings for vibration attenuation.

High sliding speeds in fluid film bearings can also result in super laminar flow accompanied by sharply increased levels of frictional power loss. In large power generating equipment bearing power loss may reach several thousand kilowatts so bearing power loss is of major concern, especially in today's atmosphere of energy conservation. Accordingly, the development of a better understanding of super laminar flow and of bearing designs which minimize power loss without degrading load capacity should have high priority.

The development of gas film bearings continues with a steadily broadening scope of applications. The obvious benefits are a simpler lubrication system, less mechanical complexity and higher operating temperature capability because of the elimination of organic lubricants. Research has concentrated on compliant types of bearings, many incorporating flexible foils, because of their relatively forgiving nature. Wear resistant coatings are also under intensive development for completely self-acting bearings.

It is not the objective of this paper to present an in-depth treatise on the state of the bearing art. Such an attempt would result in a compendium of encyclopedic proportions. Rather, the objective is to present highlights illustrating where we are and what problems are most urgently in need of solution. A few key references, some of which contain extensive bibliographies, are presented.

ROLLING ELEMENT BEARINGS

The principal driver has seen the need for bearings with higher speed capability and improved endurance in aircraft propulsion systems, most notably large gas turbine engines, and transmissions and speed reducers for rotorcraft and turboprops. Aircraft turbine engine designers want bearings with 3 million DN and even higher speed capability coupled with extended endurance life. Gear box designers want bearings with 2^+ million DN, and high combined thrust and radial load carrying capability, again with extended endurance life and high reliability.

Liquid Lubrication - Parker (1) presents a thorough review of rolling element bearing lubrication. During the mid 1960's it became apparent that jet lubrication of large bore mainshaft ball and roller bearings would not be adequate for DN values beyond 2.3 to 2.5 million. As speeds increase centrifugal effects make it increasingly difficult for oil to penetrate to the critical interior surfaces of the bearing when directed from jets at the sides of the bearing. Brown (2) first presented the concept of underrace lubrication, figure 1. Underrace lubrication allows oil to be metered at controlled flow rates directly into the bearing with an option to use part of the flow exclusively for cooling by bypassing the bearing. Underrace lubrication offers several advantages for extreme speed applications. Churning losses can be kept to a reasonable level by feeding directly into the bearing only enough oil to satisfy the internal lubrication and cooling requirements. By-passed oil flow serves exclusively as a coolant. With underrace lubrication, centrifugal effects assist in pumping oil into the critical areas requiring lubrication. In (3) jet lubrication was found to be ineffective at 2.5 million DN in experiments with 120 mm bore ball bearings, figure 2. Underrace lubrication is effective, in contrast, at DN values to 3 million (4).

Underrace lubrication has also been applied to small bore (35-mm) ball bearings operating at DN values to 2.5 million (5), and to highly loaded tapered roller bearings operating at DN values to 2.4 million (6). It has proven to be particularly effective in solving lubrication problems associated with the cone rib flange in high-speed tapered roller bearings.

It can reasonably be said that the concept of underrace lubrication, in its various forms, constitutes a major advance in the lubrication and cooling of very high speed bearings. Lubrication and the control of frictional heat generation and bearing temperature no longer constitute a problem or limiting factor in high speed bearings.

Elastohydrodynamics - Beginning with the work of Grubin (7), and followed by both analytical and experimental work of numerous other investigators, most notably Dowson and Higginson (8), Archard and Cowking (9), and, most recently, Hamrock and Dowson (10, 11), the calculation of elastohydrodynamic film thickness has become an integral part of rolling bearing analysis and design. An easily used equation for minimum film thickness in either point and line contacts (10) is:

$$H_{min} = 3.63\, U^{0.68}\, G^{0.49}\, W_p^{-0.073}(1-e^{-0.68K})$$

H_{min} is a dimensionless minimum film thickness and K is a function of the contact ellipticity.

$$K = 1.03\left(\frac{Ry}{Rx}\right)^{0.64}$$

The influence of elastohydrodynamic films on bearing performance is determined by the now widely accepted EHD film parameter, Λ, which is the ratio of film thickness, h, to composite surface roughness, σ, where

$$\sigma = \sqrt{\sigma_1^{\,2} + \sigma_2^{\,2}}$$

σ_1 and σ_2 are the RMS roughnesses of the two surfaces in contact. Tallian, et.al., (12, 13) first investigated the relationship between Λ and bearing performance. As shown in figure 3, there is a high probability that surface distress in the form of superficial glazing, smearing and micropitting will occur at $\Lambda \leq 1$, figure 4. The frequency of asperity contact in this Λ region is high. At $.9 \leq \Lambda \leq 1.5$ surface distress may occur. If surface distress does occur it will probably result in early bearing failure due to wear out and/or surface initiated fatigue. At $\Lambda > 3$ very little asperity contact occurs, and the surfaces are, for practical purposes, completely separated. In this region the bearing can be expected to run to its full endurance life expectancy.

In the Λ range between 1 and 3, where many bearings operate, successful operation depends on additional factors such as the boundary lubricating characteristics of the lubricant with the bearing materials involved, bearing kinematics such as spin-roll ratios, temperature and surface texture (other than rms surface finish).

It is in the low Λ region where the departure of bearing life from that predicted by classical fatigue theory becomes the greatest. Tallian and his coworkers have recognized the need for a much broader theory of bearing failure, based on several competing modes of failure (13, 14, 15). The theory allows for material, lubrication and surface topography effects and goes considerably beyond the consideration of Λ alone. This is especially needed in the low Λ range where the scatter in the little available life data is greatest, figure 5, (13) Bearing endurance data are needed in the Λ range up to 2 with controlled values of asperity slopes and traction in order to demonstrate the validity of the theory.

A second lubrication related factor for which better predictive methods are required is starvation. The influence of lubricant availability in front of the contact (in terms of inlet distance) on the ability of the rolling system to develop an elastohydrodynamic film has been mathematically described by Hamrock (11) and lubricant starvation has been observed experimentally by Wedeven, et.al., (16) and Wolveridge, et.al., (17) in simple ball and cylinder contacts. The problem of observing and predicting starvation in full scale bearings operating at high speeds is considerably more complex than developing mathematical models and conducting simple experiments.

Chiu (18) developed a film replenishment model which incorporates a speed-viscosity parameter and a starvation parameter dependent on oil-air surface tension and rolling element spacing. Experimental data are needed to verify Chiu's model and to assist in developing more effective predictive techniques. The importance of starvation in ball and roller bearings is not well understood. It appears to be significant in gyro bearings and may very well influence the operation of other high speed bearings. Pemberton and Cameron (19) conducted experiments with a cylindrical roller bearing operating at approximately 0.2 million DN and had difficulty producing starvation. The method of lubricating a bearing as well as the operating DN of the bearing may be critical to the onset of starvation, and that hasn't been studied.

Computerized Analysis and Design Methods - The availability of accurate elastohydrodynamic film thickness and traction predictive methods has made possible the development of both quasi-static and dynamic ball and roller bearing analyses which can accurately predict bearing performance. The evolution of large high speed computers has made the use of complex computer programs practical as design and performance prediction tools.

The first widely used ball bearing analysis was conceived by Jones (20, 21). Coulomb friction was assumed to exist in the ball-race contacts. Jones' analysis predicted life quite well but had shortcomings in predicting skidding and slip. Harris (22) first incorporated elastohydrodynamics into a ball bearing analysis. Harris used Archard and Cowking's (9) film thickness equation and an exponential viscosity pressure relationship. As shown in figure 6, Harris' analysis predicted cage slip better than did Jones' earlier race control theory.

A revised version of Harris' computer program called SHABERTH was developed incorporating experimental traction data. Further revisions by Coe (23) resulted in a program which predicted bearing temperatures reasonably well (figure 7).

The development of analytical tools for predicting the performance and life of cylindrical roller bearings parallels that for ball bearings. Harris (24) first introduced elastohydrodynamics into a roller bearing analysis. Since then increasingly reliable programs have evolved through the introduction of more precise traction relationships. Poplawski (25) and Rumbarger (26) generated programs which predicted roller bearing skidding with increasing accuracy, figure 8.

Quasi-static analyses have also been developed for tapered and spherical roller bearings. These programs are being continually updated with more efficient subroutines, especially thermal subroutines which can produce reliable estimates of bearing temperatures.

The quasi-static analyses are suitable only for describing steady-state bearing operation since they tacitly assume that an equilibrium of forces exists at all times. Evidences of transient behavior and instabilities have been observed in gyro bearings, and unsteady state behavior has probably been responsible for numerous high speed bearing failures. These have provided the stimuli for full scale dynamic analyses, such as Gupta (27-30). Even with a high speed digital computer, however, a full dynamic analysis is costly and time consuming, as such a program can only be used very selectively. Quasi-static programs have proven to be very effective engineering design tools. Dynamic analyses may prove of real value in the diagnosis of bearing failures, many of which follow in the wake of the onset of unsteady state dynamic conditions.

Bearing life prediction is presently based on the Lundberg and Palmgren theory with several life multiplying factors applied as suggested in (31). These factors include corrections for materials and lubrication. The lubrication factor is developed from experimental data relating bearing life and Λ . An "average" curve such as that of figure 9 is recommended for use, but we have already seen the wide scatter in fatigue lives at low Λ 's (figure 5) and, as stated earlier, we lack a good understanding of the factors that control bearing life in the low Λ region of operation. At high values of Λ the lubrication factor recommended in (31) is probably valid although even there it may be overly simplistic, based on Tallian's work on competing failure modes (13, 14, 15) and the dramatic advances that have been made in the quality of bearing materials. It may be argued with some validity that the advent of double vacuum melted steels and superior processing techniques has greatly diminished the incidence of classical subsurface fatigue, upon which the Lundberg-Palmgren theory is based. Advanced life-prediction methods should, therefore, recognize the many failure modes that are possible, and incorporate physical and chemical factors in addition to surface texture effects. These should then be incorporated into computer programs for more realistic life prediction.

Materials - A broad scope, in-depth discussion of the state of the art of rolling element bearing materials is given in (33). As stated above, dramatic improvements in bearing endurance life have resulted from modifications in melting and processing techniques applied to existing iron base bearing alloys (as opposed to the development of new alloys). Bearing endurance life improvement realized through the evolution of AISI M-50 bearing steel, for example, is shown in figure 10 (from 33). The extended bearing endurance life resulting from improved alloys has made bearings more susceptible to other failure modes (such as surface initiated fatigue). This, together with an increasing awareness of elastohydrodynamic and other lubrication effects, has made the development of a more comprehensive life (or failure) prediction method even more relevant.

The dramatic improvement in iron base bearing alloys has also made it possible to achieve adequate endurance life at extreme speeds, despite the high cycling rate and the detrimental centrifugal effects (34). With the aforementioned developments in underrace lubrication, computerized design, and now adequate material endurance, one might be tempted to think that all aspects of the 3 million DN bearing problem have been solved. Such is not the case, however, as reported in (34) because fatigue spalling at 3 million DN was followed by crack propagation through the rotating inner ring. This quickly precipitated catastrophic failure (figure 11). At moderate bearing speeds, spalling is a benign type of failure, but at extreme speeds the hoop stress in the rotating ring coupled with inadequate fracture toughness of the ring material combine to make the failure catastrophic. A number of research efforts are presently directed toward obtaining bearing materials with improved fracture toughness. Some of the more promising approaches are discussed in (33). These are critically needed efforts, because fracture toughness presently constitutes a barrier problem preventing rolling element bearing operation at DN values beyond 2.4 million, figure 12.

Bearing ring fracture is also a problem in anti-skid roller bearings (35). These bearings frequently utilize non-circular raceways designed to produce preload or "roller pinching" at points within the bearing. Inner ring flexing can occur in these bearings resulting in fatigue cracks which propagate radially outward from the initiation site at the bore. Similar problems occur in hollow rollers when that approach is used to produce bearing preload. It is clear from this discussion that bearing materials with improved fracture toughness are required to solve several critical bearing problems. This is a major challenge for the bearing materials researcher.

Other areas of bearing materials research which are particularly relevant to present and near future needs and problems are corrosion resistant ferrous alloys, powder-metallurgy produced materials, and ceramics. A large number of conventional oil-lubricated bearings are rendered unserviceable each year because of corrosion, Table 1, (36). Therefore, there is a genuine need for a cost competitive, corrosion resistant alloy with endurance characteristics equivalent to vacuum processed M-50 and AISI 52100. One promising alloy under development is AMS 5749, a martensitic stainless steel. AMS 5749 has shown fatigue characteristics which compare favorably with those of M-50 in bench fatigue tests (37). Endurance tests of full scale bearings should be run to properly assess the potential of AMS-5749.

Powder-metallurgy-produced materials have been under investigation as potential bearing materials for a number of years. Among the P/M alloys under investigation are M-50, T-15 and CRB-7 (an alloy similar to AMS-5749) (38, 39). The potential for producing a fine-grained material with fine, evenly dispersed carbides is recognized and provides the incentive for this research. A viable P/M material might also lead to lower cost bearings through the reduction of waste in processing bearing parts. Results so far have been promising but somewhat erratic. An integral part of the problem is quality control in both making and processing the powders. Continued research is needed to establish good quality control so that the ultimate potential of P/M materials can be assessed.

Ceramics have also been the subject of much research as potential bearing materials since the 1950's. A successful ceramic material would find application in high temperature bearings, and bearings for corrosive and non-lubricating environments. A summary of ceramic bearing materials research is given in (33). Much of the early work proved disappointing; quality control was erratic and the endurance characteristics were generally poor. Recent emphasis has shifted to silicon nitride, which has shown considerable promise in both hybrid (silicon nitride rolling elements coupled with ferrous alloy races) and in all ceramic bearings, (40). It has demonstrated endurance characteristics comparable to those of a good quality forged bearing alloy (41). Its low density, excellent high temperature properties and excellent corrosion resistance insure that it will someday find use in bearings requiring a material with one or more of these properties. Research on silicon nitride should continue with strong emphasis on manufacturing quality control.

Solid Film Lubrication - Operation of rolling element bearings at extreme temperatures, either very high or low, or in vacuum usually precludes the use of conventional oil or grease lubricants. Solid film lubrication can often be used in these unusual environments, and in systems where the life requirements are short and simplicity of mechanical design is of premium importance.

The use of solid film lubrication generally limits bearing life to a small fraction of its full endurance life potential with oil lubrication. Solid lubricants are most often used in the form of bonded films or as transfer films. Gas entrained loose powders of molybdenum disulfide, lead monoxide and graphite have been evaluated in experiments, but to the writer's knowledge have not found use in actual applications (42).

Polytetrafluoroethylene base materials have found considerable use in cryogenically cooled bearings in high energy rocket engine turbopumps (43). Feasibility studies of very thin (≈ 5000 A°) sputtered films of hard nitrides and carbides as surface protectants in gyro and turbine engine bearings with short-life requirements are presently being made (44). Thin films are probably more suitable for use in Hertzian contacts than are thick films.

The ultimate utility of solid film lubricants in rolling element bearings is probably limited by the nature of their stressed contacts and kinematics, but one of the factors limiting the success achieved to date may lie in the bearing designs used. Bearings designed for use with liquid lubricants have been the vehicles employed in evaluating solid lubricants. Careful studies to optimize surface texture and bearing design have never been done. Such studies would enhance the level of success with solid lubricants.

HYDRODYNAMIC BEARINGS

The analytical and experimental foundation for hydrodynamic bearing design began with the work of Osborne Reynolds and Beauchamp Tower before the turn of the present centruy. In contrast, rolling element bearing analysis does not predate the work of Lundberg and Palmgren in the 1940's. A reasonably good level of technical understanding of hydrodynamic bearings was, therefore, established before a comparable understanding of rolling element bearings was developed. Probably for this reason the level of research activity on hydrodynamic bearings in recent years has been somewhat lower than that on rolling element bearings. There appear to be several rather select areas in which research on hydrodynamic bearings, directly relevant to visible needs and problems, is being carried on. Those include:

1. Non-circular journal bearings for vibration suppression in high speed rotating machinery.

2. Non-rotating journal bearings as squeeze film dampers, for suppression of vibrations in high speed rotors supported on rolling element bearings.

3. Turbulent flow bearings for more efficient design and energy conservation.

As mentioned earlier the trend toward higher speed, higher power density rotating machinery has made mandatory steady advances in both rolling element and fluid film bearing technology. Higher operating speeds have aggravated problems associated with vibrations due to critical speeds, unbalance and stability. Several of the bearing types developed to combat these problems are discussed in (45). These include multilobe (figure 13), pressure dam and tilting pad bearings.

A critical element in the design and selection of anti-whirl bearings for high speed systems is the ability to predict the instability threshold speed, or the onset of subsynchronous, potentially dangerous, vibration frequencies. In investigations reported in (45) bearing coefficients were determined using finite elements and the flexible rotor was modeled using lumped rotor masses. Theoretical predictions compared reasonably well with experiments, as shown in figure 14. More work is needed to establish more realistic boundary conditions for the solution of the relevant differential equations, and to incorporate real geometry effects (runout, out-of-round, minor unbalance) into the theoretical model. Additional experiments would then be required to test the degree of correlation.

The ability to predict the stiffness and damping coefficients of squeeze film dampers has been limited partly by an inability to properly characterize film cavitation. Recent analytical attempts (46, 47) utilizing more sophisticated extensions of the short-bearing theory have produced some interesting film boundaries. Experimental studies (48) indicate that significant pressure variations may occur in cavitated regions, contrary to the usual assumption of constant pressure.

Additional experimental work in which film shapes can be observed over a range of eccentricity ratios, frequencies, supply pressures, end conditions, etc., needs to be done to obtain data for correlation with, and possible feedback into existing analyses. Squeeze film dampers are an important component of high speed bearing-rotor systems. The ability to accurately predict their performance is vital to continued advancements in rotating machinery technology.

The occurrence of turbulent or superlaminar flow in liquid film bearings has been recognized for over three decades (49). Superlaminar flow is accompanied by greater than expected load capacity, power loss and temperature rise together with reduced flow. Interest of late has centered around large bearings in electric power generating machinery. These bearings have very considerable power losses which, together with the escalating cost of energy, has stimulated research on more energy efficient bearings (50). An in-depth discussion of the state of the art of turbulent bearings and of their economic importance is given in (51). Turbulent lubrication theories, although somewhat empirical in nature, are well developed and have been translated into usable design procedures (52, 53, 54).

There remain, however, areas where further research gains could be translated into significant economic savings. From a fundamental point of view the nature of the flow over a wide range of Reynolds' numbers immediately beyond the laminar flow region is not well understood. Vortex flow appears to persist over a wide range of Reynolds' numbers up to the development of fully turbulent flow, if, in fact, the latter ever comes about (51). It is in this "transition" region where discrepanices occur among the various theories. Further analytical and experimental studies are needed. In the practical arena lubricant supply effects and parasitic churning losses need to be better defined and translated into design guides. Much can probably be done, by way of power loss and energy saving, through studies of the portions of the thin film regions in both journal and thrust bearings that don't contribute appreciably to load capacity. Elimination or minimization of non-productive thin film regions could result in reduced levels of power loss.

GAS FILM BEARINGS

Air or, more generally, gas film bearings have evolved into practical entities used in a fairly broad scope of machinery and mechanisms. Gas bearings can be found in gyros, magnetic recording heads, microcircuit pattern generators, metrology instruments, load transportation pallets, textile spinning rings, space power turboalternators and air cycle machines. The pattern of gas bearing development has been interesting. At first rigid geometry bearings were applied; of late, compliant bearings, most of which incorporate foils of various designs, have been finding more widespread use. Compliant gas bearings have greater tolerance to contaminants and shock loads than rigid geometry bearings.

Foil bearings which have been developed to the point of application include tensioned foil bearings, figure 15, (55), cantilevered foil bearings, figure 16, (56), and bump foil bearings, (57). A more recent version of a bump foil journal bearing, being developed by Mechanical Technology, Inc., is shown in figure 17. A journal bearing that utilizes a telescoped foil within a rigid shell (58) and a thrust bearing that utilizes a spirally grooved membrane on a compliant back (59) were successfully evaluated in experiments conducted with a high speed rotor. Schematic diagrams of these bearing concepts are shown in figures 18 and 19.

Development work is continuing on both the cantilevered and bump foil bearing types to achieve geometries closer to the optimum for both load capacity and stability. Continued research on these bearings is fully warranted by their promise as rotor support systems superior to conventional oil lubricated bearings for a variety of high speed turbomachines. Their advantages are obvious - the complete elimination of conventional lubrication systems and operating temperature capability limited only by the properties of the bearing materials rather than those of organic lubricants.

All of these bearings are completely self-acting which means that their surfaces are in contact at startup, shutdown and perhaps momentarily at high speeds. Wear resistant surface coatings are therefore mandatory on both the foils and the journal or the thrust runner. Foil coatings must be very thin and resilient, adherent, possess good wear properties and be stable at temperatures up to the maximum operating temperature. Foil bearing coatings are under development (60); the ultimate success of gas bearings depends heavily on the degree of success achieved in coating development.

SUMMARY

A brief status report on bearing technology, present and near-term future problems that warrant research support has been presented. Some selected research needs are the following:

1. Rolling Element Bearings

   a. Bearing materials with improved fracture toughness to allow bearing operation to at least 3 million DN without catastrophic failures

   b. Ball and roller bearing life data in the low Λ region with carefully controlled surface textures and lubrication factors for correlation with and feedback into a bearing life or failure theory which recognizes competing failure modes.

   c. A comprehensive bearing life or failure theory that goes beyond Lundberg-Palmgren and incorporates competing failure modes.

   d. Bearing experiments at high DN values for assessment of starvation effects.

   e. Incorporation of advanced life prediction methods into bearing dynamic analyses.

   f. An improved corrosion resistant bearing alloy.

   g. Improved thin film surface protective coatings for unlubricated short life bearings.

   h. Optimization of bearing surface texture and geometry for solid film lubrication.

2. Hydrodynamic Bearings

   a. Experiments with non-rotating short journal bear..gs subjected to oscillating loads to establish the boundaries of cavitated films and pressure distributions in both the film and cavitated regions for prediction of squeeze film damper performance.

   b. Analysis and experiments to determine lubricant supply effects and parasitic churning losses in turbulent flow bearings.

   c. Analysis and experiments to determine optimum geometries for minimum power loss in turbulent flow bearings.

3. Gas Film Bearings

   a. Foil bearing geometries that form more nearly optimum film shapes for maximum load capacity.

   b. Surface protective coatings for use in compliant bearings at temperatures to 1200°F.

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Part III- Ball Bearing Analysis
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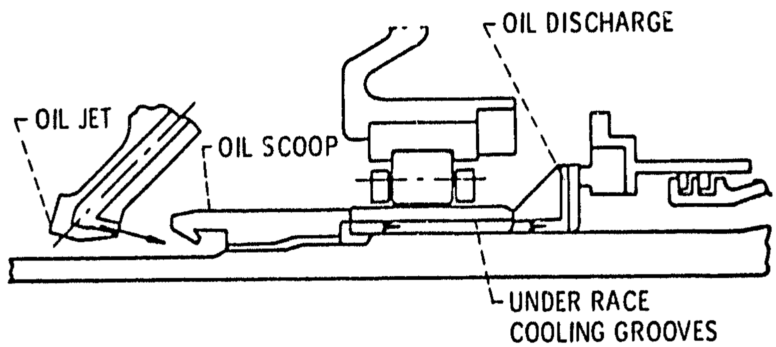
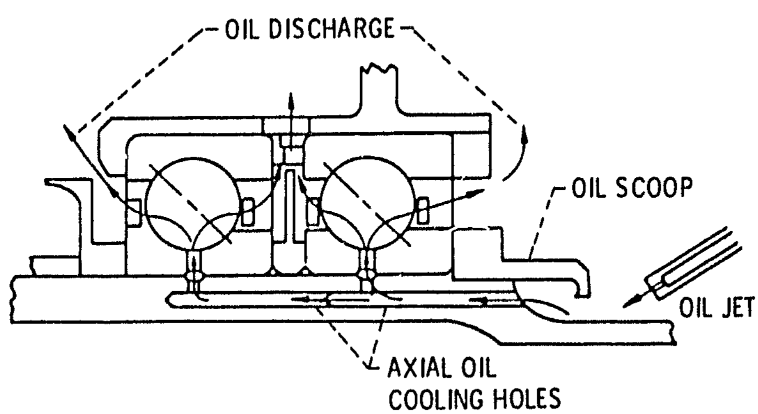
TABLE I. - AIRCRAFT BEARING REJECTIONS BY CAUSE (FROM (36))

CATEGORY	PERCENT OF TOTAL 1969	1971	1977
DIMENSIONAL DISCREPANCIES	15	10	18
CORROSION/PITTING	32	30	29
IMPROPER INSTALLATION, DAMAGED DURING REMOVAL OR HANDLING	7	5	6
WEAR, EXCESSIVE INTERNAL CLEARANCES	--	15	8
FATIGUE, SURFACE OR SUBSURFACE INITIATED	2	3	1
CAGE WEAR	2	3	--
INDENTATIONS/CONTAMINANTS (NICKS, SCRATCHES, DENTS)	14	19	20
OTHER - CHANGE IN DIRECTIVES, SPEC'S, TIME COMPLIANCE, ETC.	28	15	18



(a) CYLINDRICAL ROLLER BEARING.



(b) BALL THRUST BEARING.

Figure 1. - Underrace oiling system for main shaft bearings on turbofan engine. (From ref. 2.)

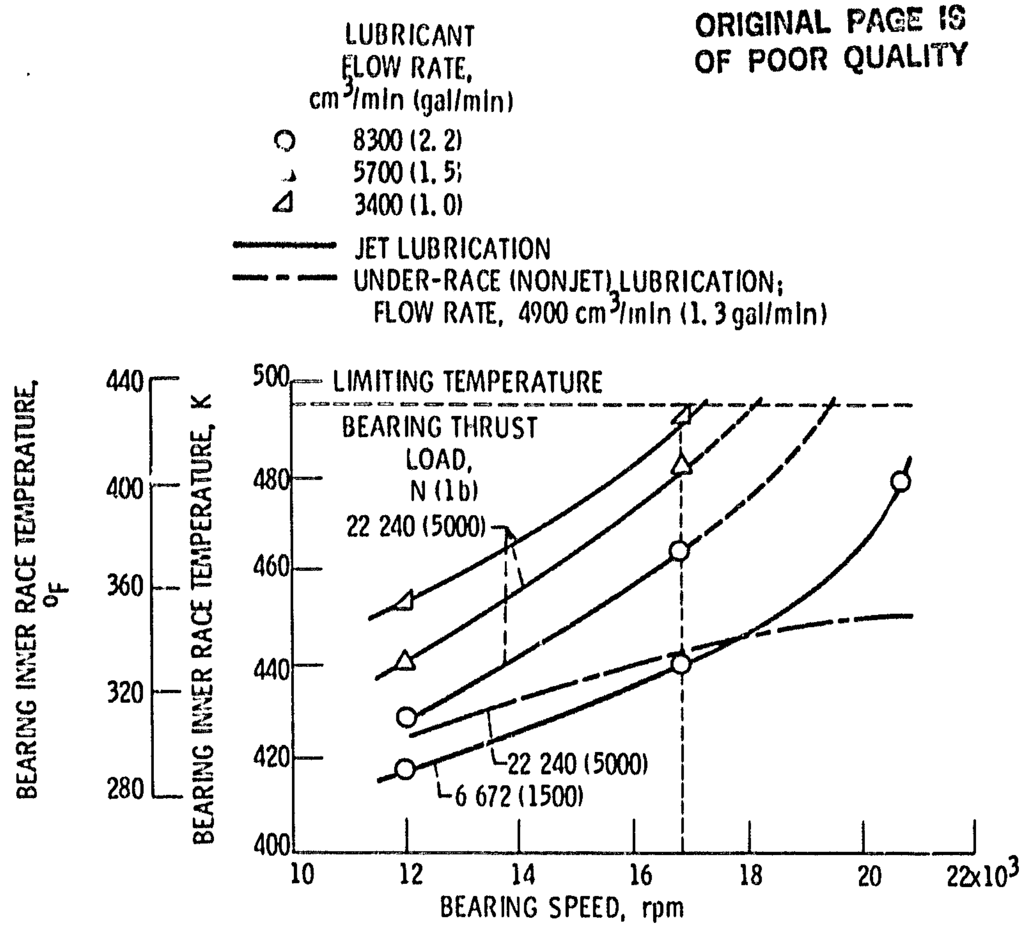


Figure 2. - Effectiveness of under-race lubrication with 120 mm-bore angular contact ball bearings. (Oil-in temperature, 3?4 K (250° F). (From ref. 3.)

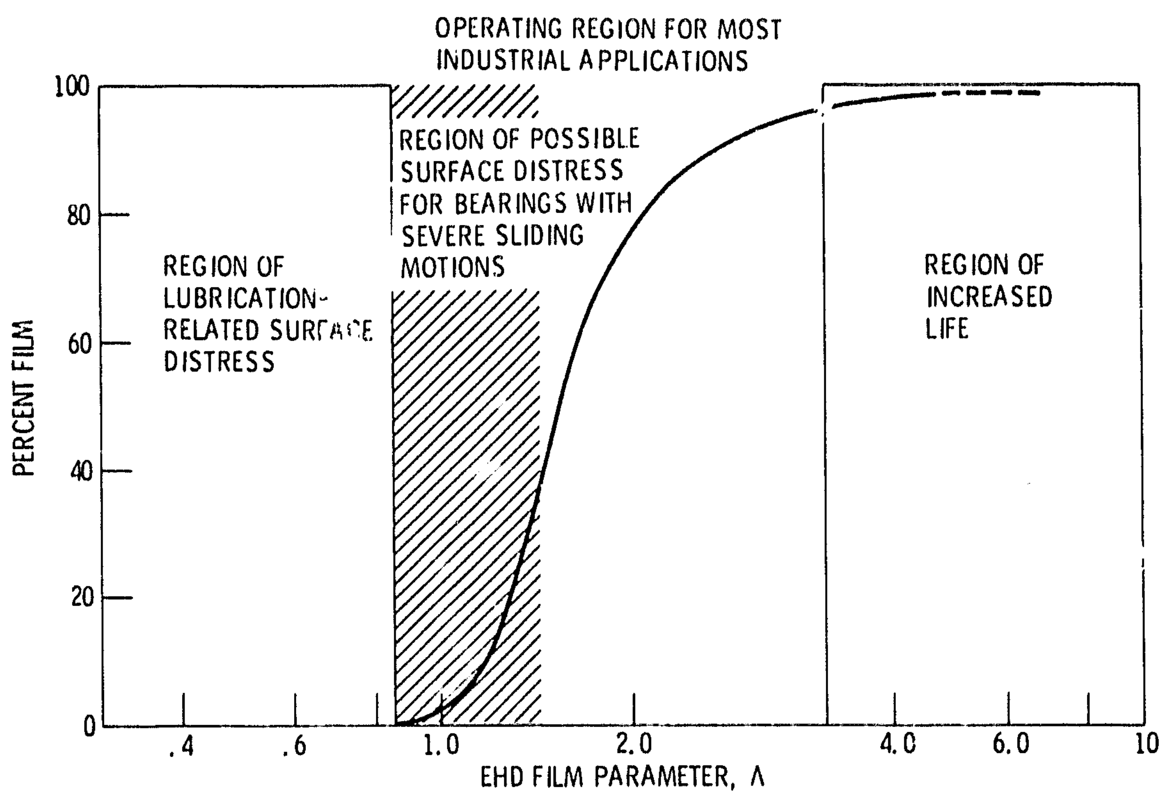
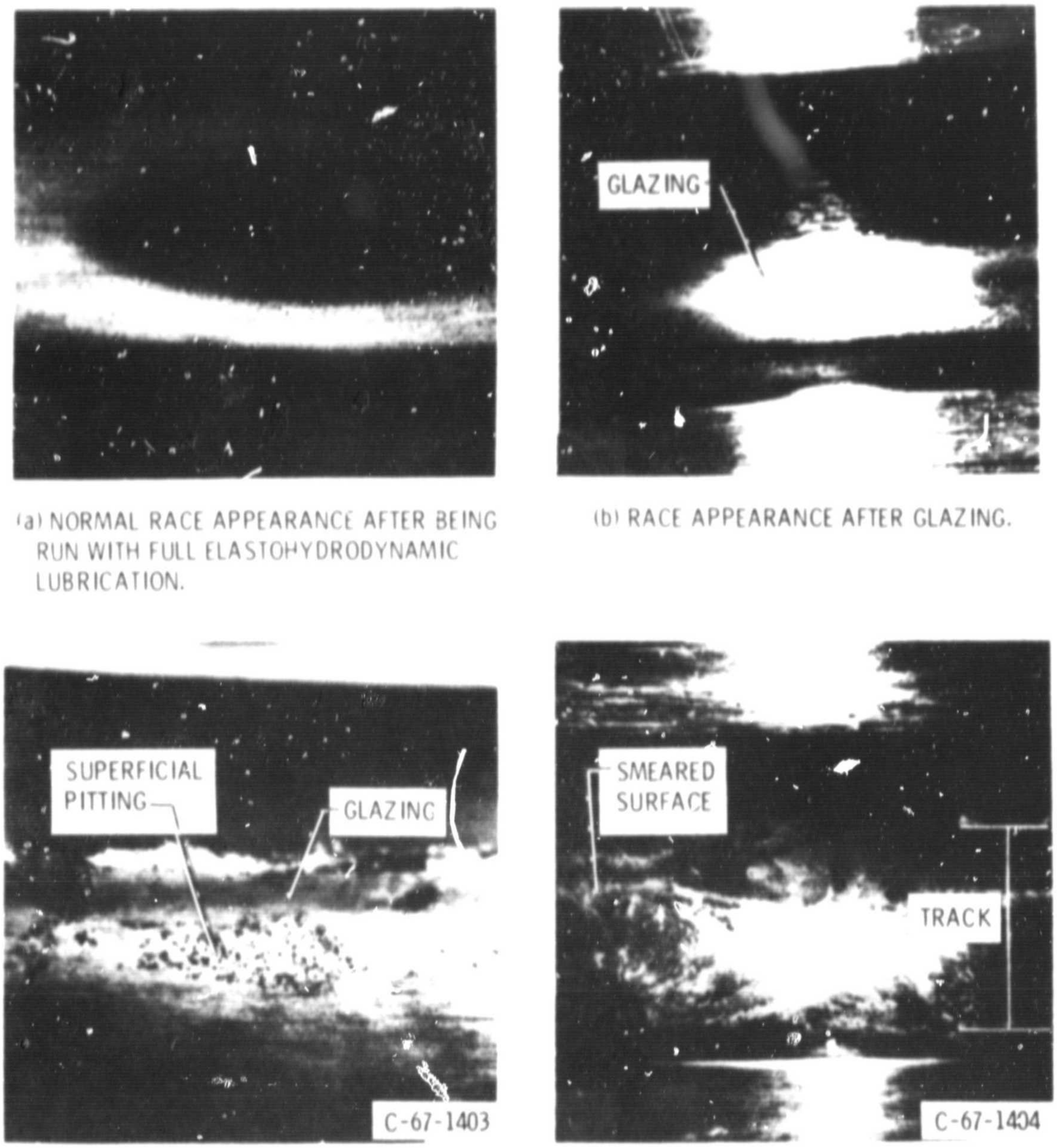


Figure 3. - Percent film as a function of EHD film parameter. (From ref. 13.)



(a) NORMAL RACE APPEARANCE AFTER BEING RUN WITH FULL ELASTOHYDRODYNAMIC LUBRICATION.

(b) RACE APPEARANCE AFTER GLAZING.

(c) RACE APPEARANCE AFTER GLAZING AND SUPERFICIAL PITTING.

(d) RACE APPEARANCE AFTER SMEARING.

Figure 4. - Effect of EHD lubrication on surface damage to bearing raceways. (From [33].)

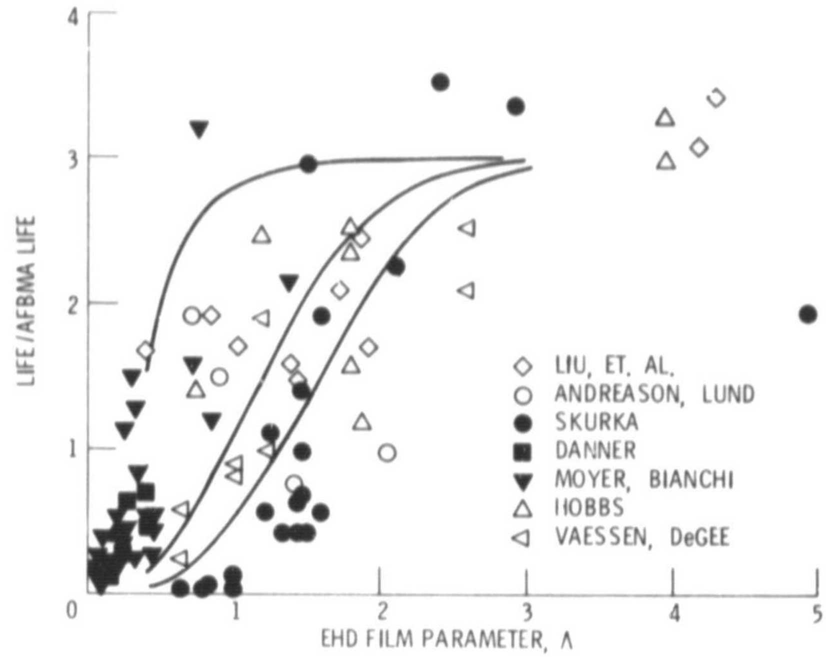


Figure 5. - Effect of Λ-ratio on life. (From ref. 15.)

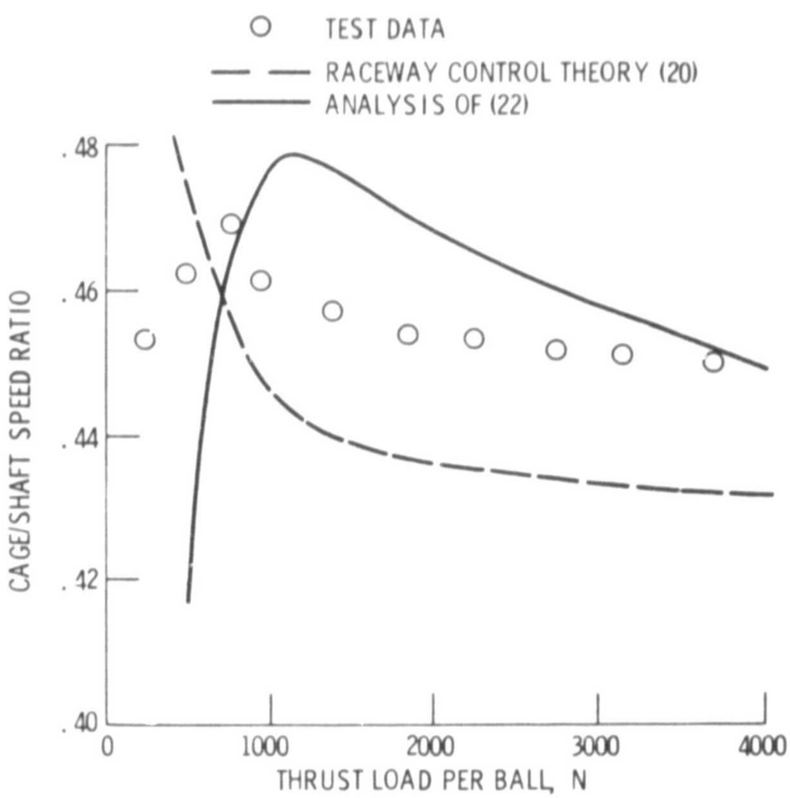


Figure 6. - Cage/shaft speed ratio versus thrust load per ball. (From ref. 22.)

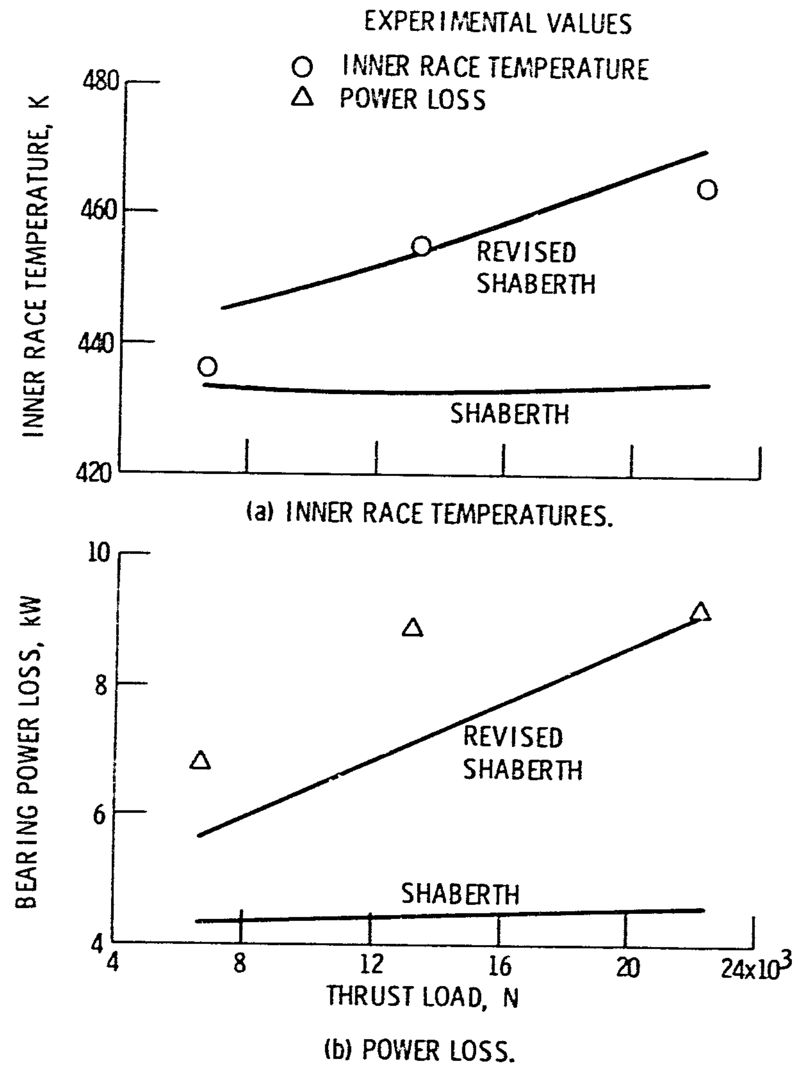


Figure 7. - Comparison of measured and calculated values of bearing operating characteristics as functions of thrust load using two versions of SHABERTH. Shaft speed, 16 700 rpm; lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 2 percent. (From ref. 23.)

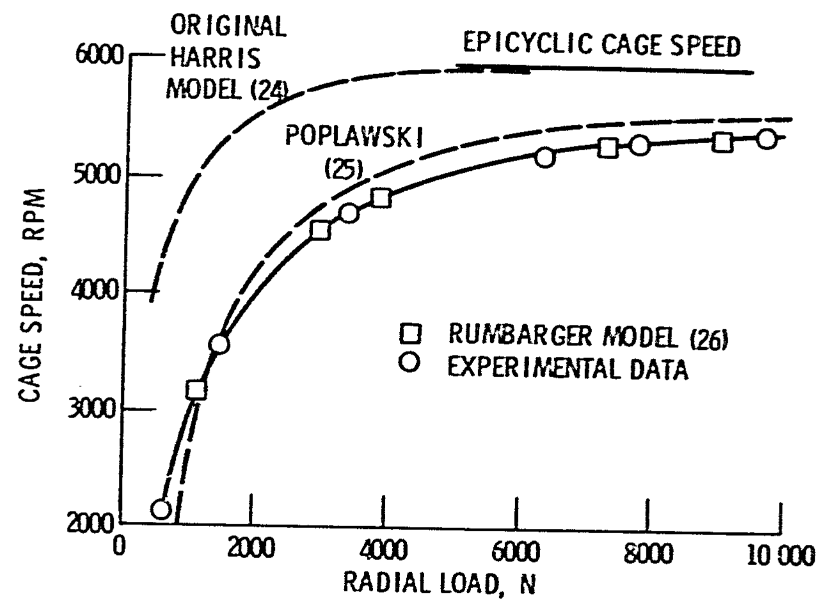


Figure 8. - Roller bearing cage speed correlation.

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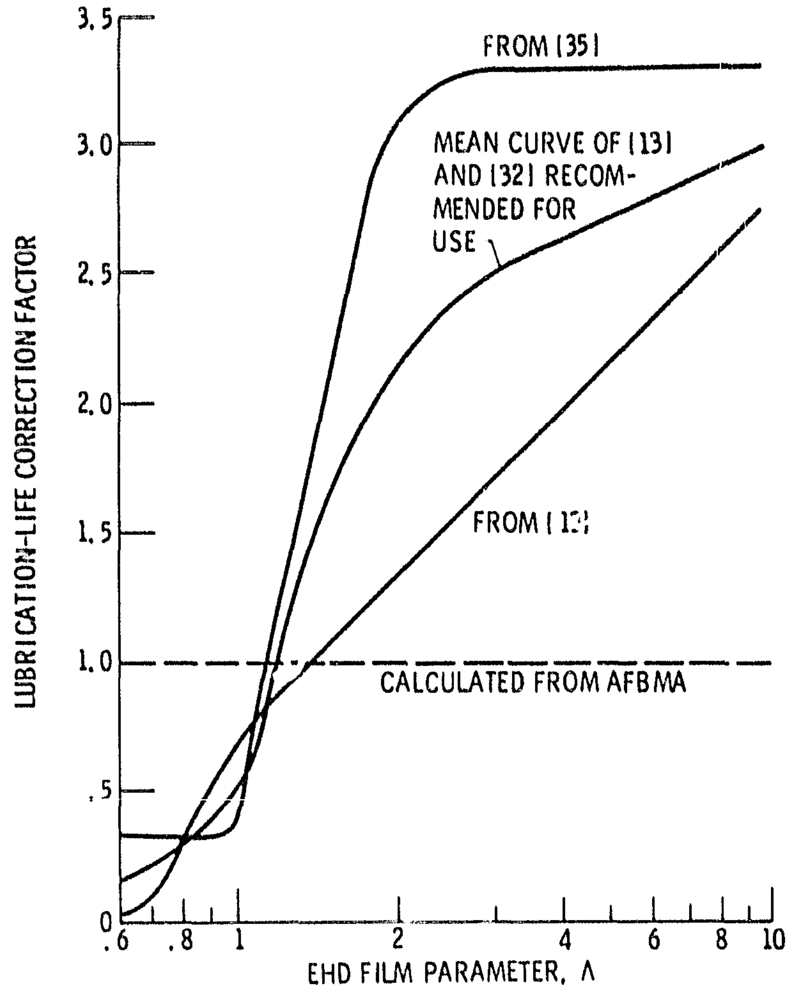

Figure 9. - Lubrication-life correction factor as a function of lambda. (From ref. 31.)

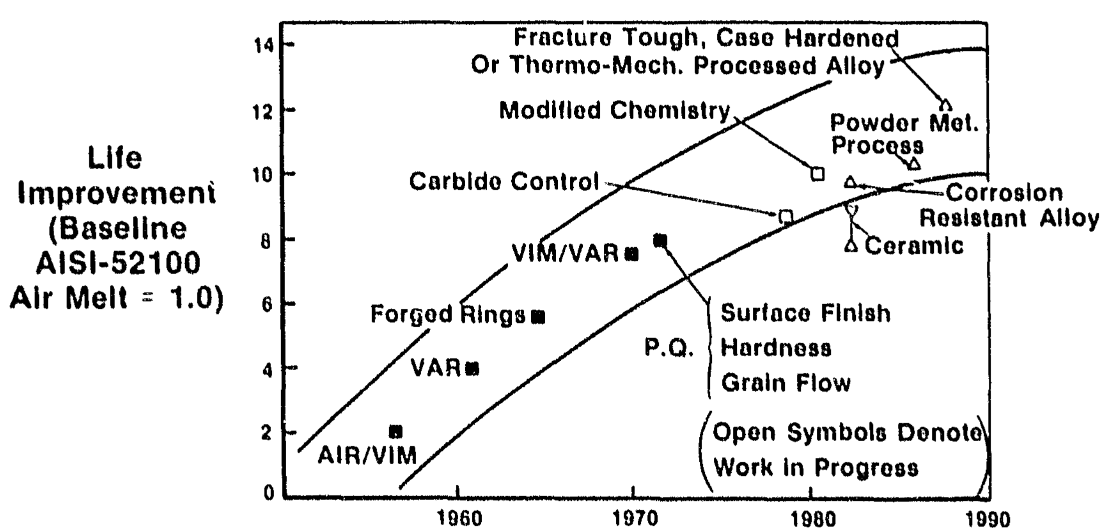

Figure 10. - Evolution of A.I.S.I M-50 bearing steel. (From ref. 33.)

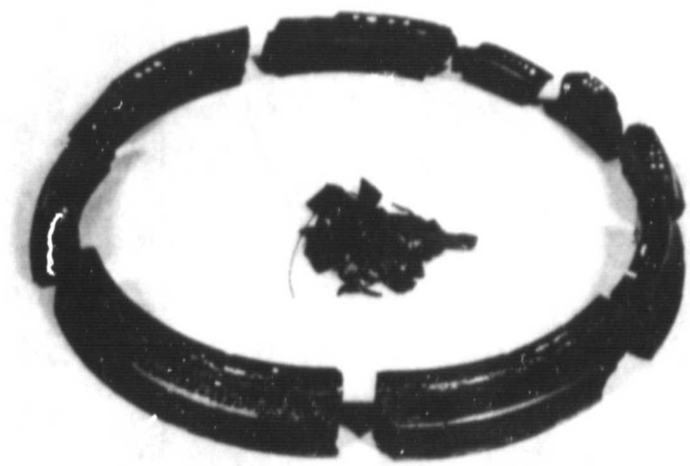

Figure 11. - Fractured bearing inner race initiated by a rolling-element fatigue spall. (From ref. 34.)

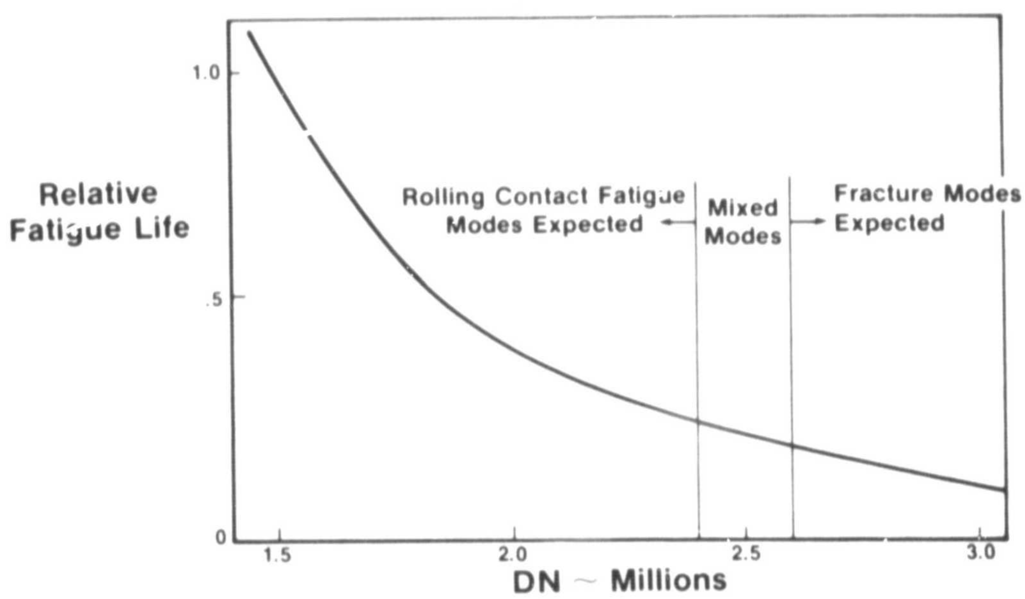

Figure 12. - Mainshaft bearing lives and expected failure mode as a function of speed for a constant bore size. (From ref. 33.)

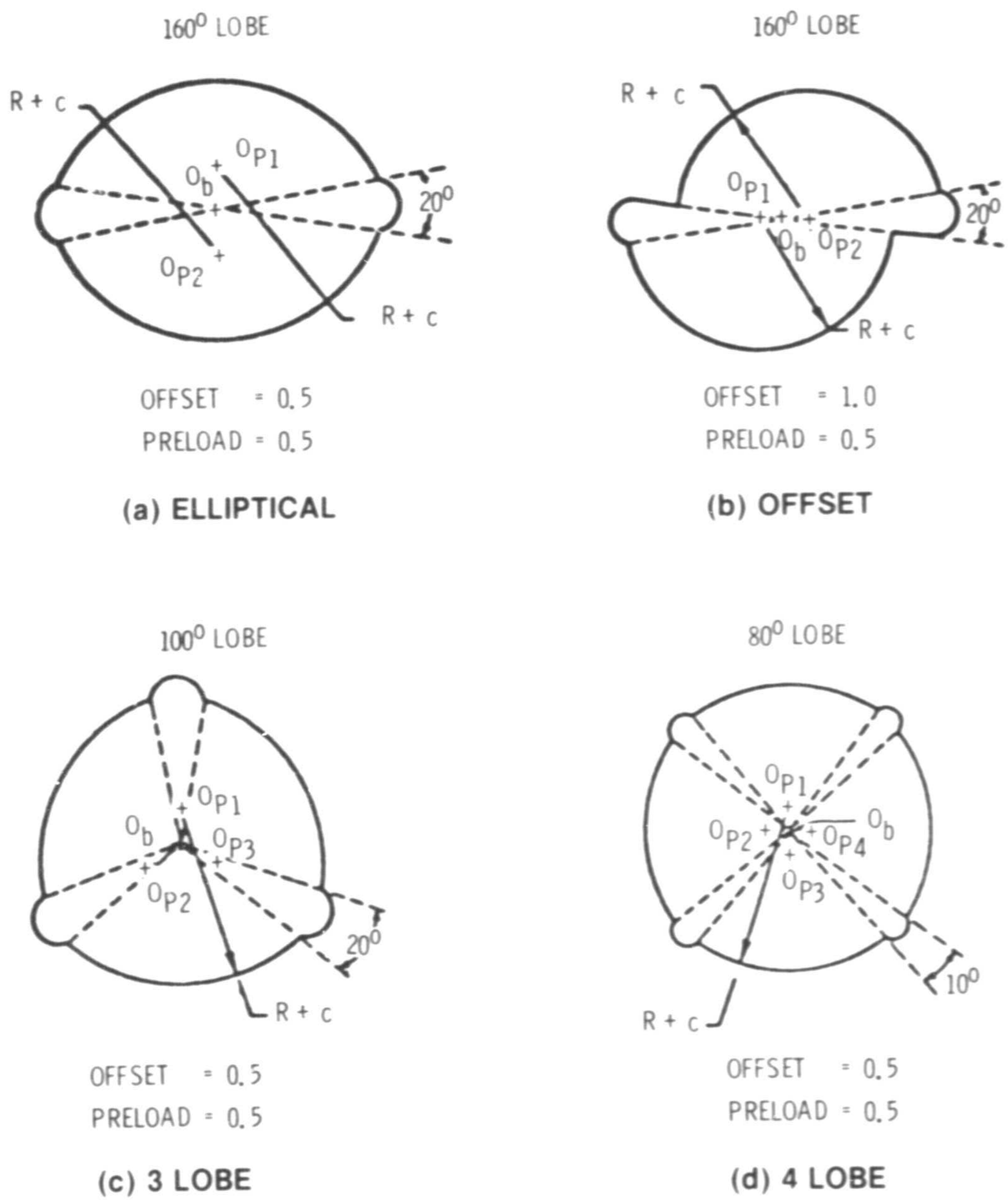

Figure 13. - Multilobe bearing geometry. (From ref. 45.)

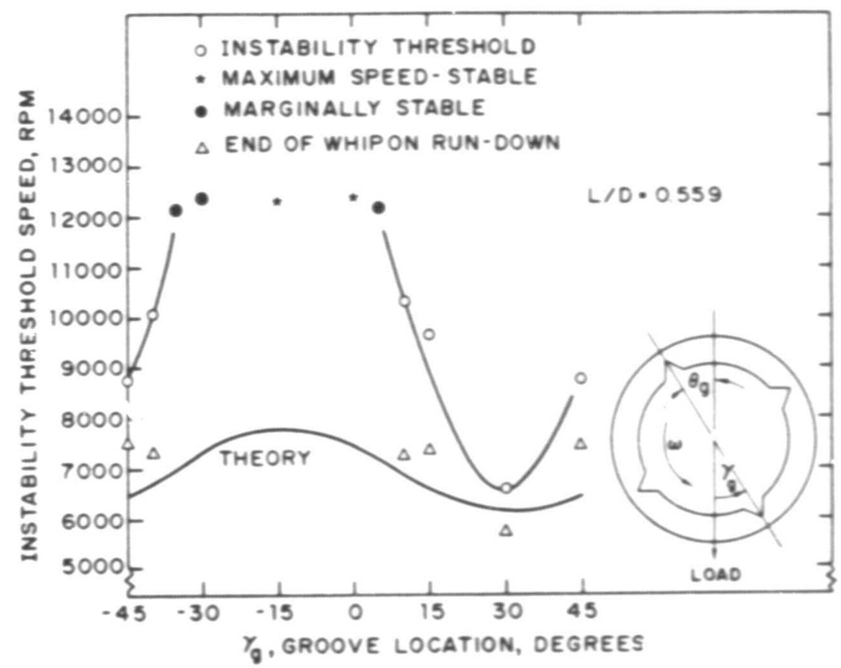

Figure 14. - Summary of correlation of instability threshold speed with γ_g. (From ref. 45.)

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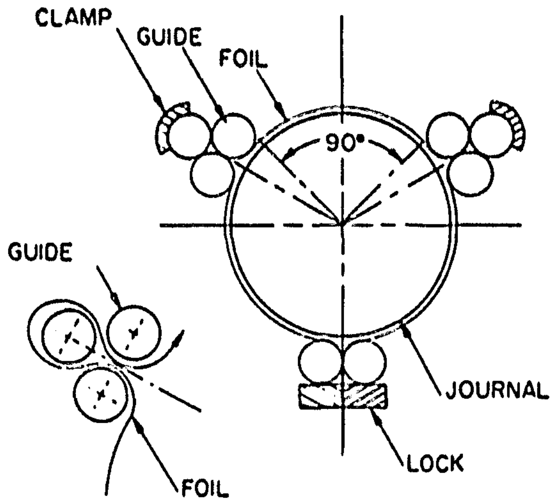

Figure 15. - Schematic diagram of tensioned foil-bearing. (From ref. 55.)

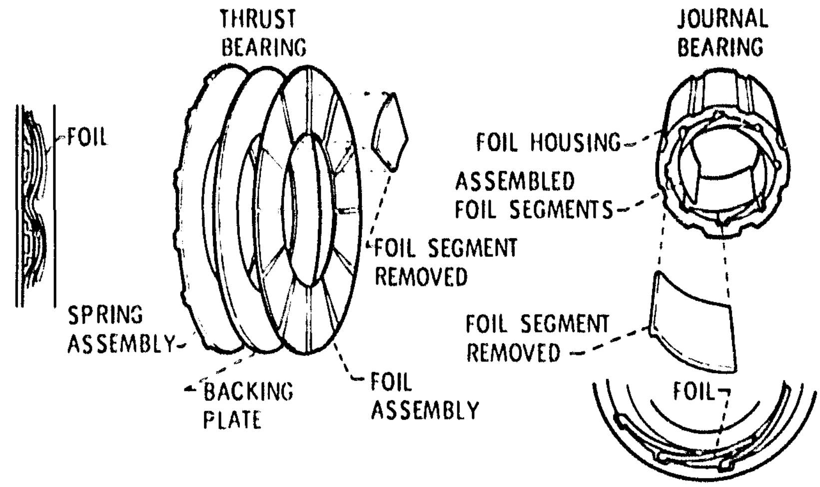

Figure 16. - Cantilevered foil bearings (Garrett Corp.). (From ref. 56).

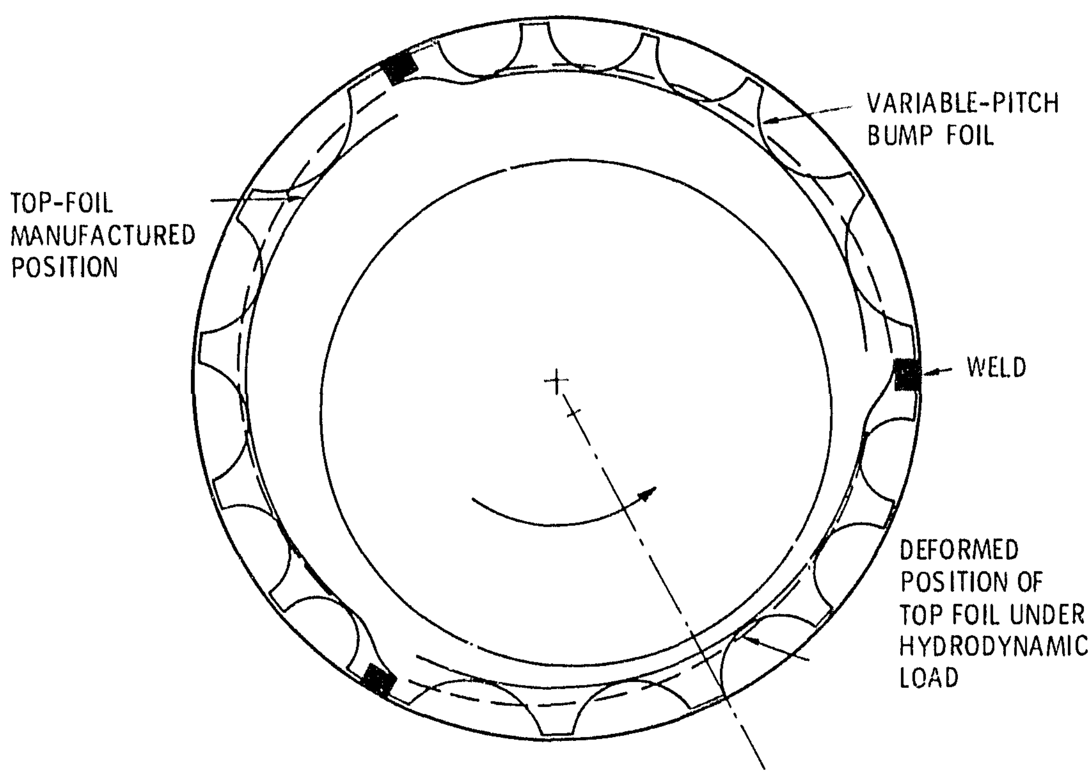

Figure 17. - A three sector bump foil journal bearing with variable pitch bump foils. (Mechanical Technology, Inc.)

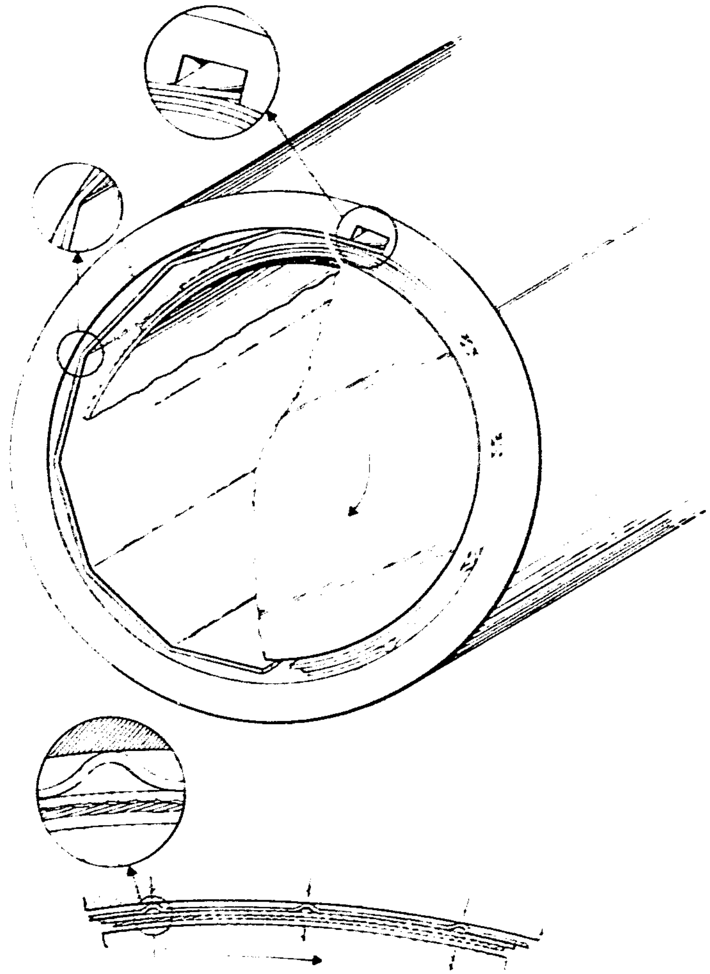

Figure 18. - Schematic drawing of journal foil-bearing (Design B). Structure before and after insertion of journal is depicted. Details illustrate rounded vertices and retaining groove. (From ref. 58.)

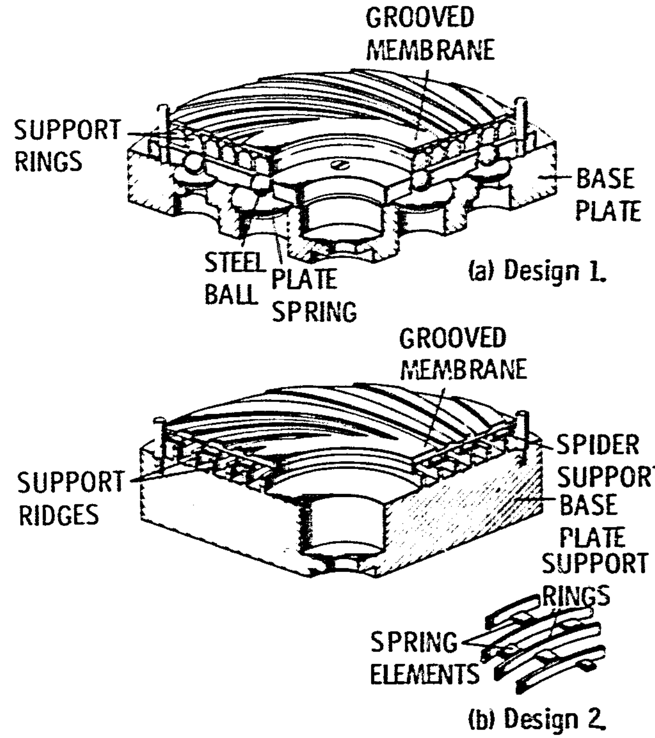

Figure 19. - Isometric drawings of thrust bearings. (From ref. 59.)