Present Technology of Rolling-Element Bearings

Richard J. Parker*

The specification of rolling-element bearings has progressed significantly beyond the selection of ball or roller bearings from a manufacturer's catalog. Applications are becoming more commonplace where novel materials, unique lubrication techniques, and sophisticated computer analysis are required to satisfy difficult environmental and operating conditions. Primary motivation for advancements in the state of technology of rolling-element bearings has come from aerospace requirements. In particular, the aircraft gas-turbine engine has provided the impetus and driving force for quantum leaps in ball and roller bearing technology.

Dramatic improvements in high-speed capabilities of ball and roller bearings can be directly attributed to the under-race lubrication technique first described by Brown (ref. 1) in 1970. Its use for mainshaft bearing lubrication of present turbojet engines is commonplace. The successful application of this technique to tapered-roller bearings will encourage their use in higher speed applications.

Improvements in rolling-element fatigue life have been particularly dramatic with AISI M-50, which has evolved to a premium quality steel, incorporating improved processing, melting, and heat-treating techniques (ref. 2). This premium quality steel is now specified by major turbojet engine manufacturers for critical main shaft bearing applications.

The evolution of computerized rolling-element bearing analysis has significantly improved the reliability of design calculations, particularly for high-speed applications. The predictions of rolling-element bearing performance now can include thermal effects, ball or roller dynamics, time transient effects, and variations in lubricant modelling (ref. 3).

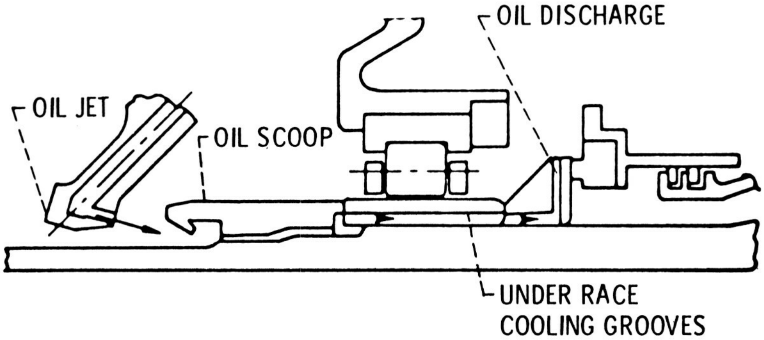
These advancements, among others, signify the progress in rolling-element-bearing technology resulting from efforts to meet advancing aerospace requirements. It is the purpose of this review paper to identify the present technology of rolling-element bearings, the barriers and limits which currently exist, and some of the future requirements that will demand further advancements.

High-Speed Bearing Lubrication

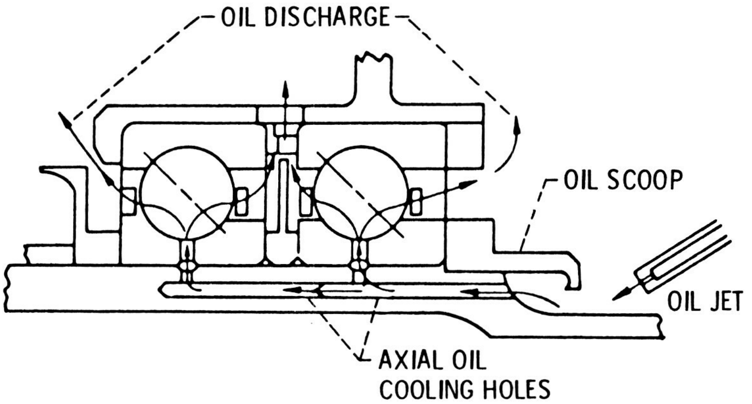
During the mid-1960's, as speeds of the main shaft of turbojet engines were pushed upwards, a more effective and efficient means of lubricating rolling-element bearings was developed. Conventional jet lubrication failed to adequately cool and lubricate the inner-race contact as the lubricant was thrown centrifically outward. Increased flow rates only added to heat generation from churning the oil. Brown (ref. 1) described an "under-race oiling system" used in a turbofan engine for both ball and cylindrical roller bearings. Figure 1 (from ref. 1) shows the technique used to direct the lubricant under and centrifically out through holes in the inner race to cool and lubricate the bearing. Some lubricant may pass completely through under the bearing for cooling only (fig. 1(a)). Although not shown in figure 1, some radial holes may be used to supply lubricant to the cage riding lands.

This lubricating technique has been thoroughly tested for large-bore ball and roller bearings up to 3 million DN. (DN is a speed parameter equal to the bore of the bearing in millimeters multiplied by the speed in rpm.) Results of these tests have been published by Holmes (ref. 4) with 125-mm (4.9212-in.) bore ball bearings, Signer, et al. (ref. 5), with 120-mm (4.7244-in.) bore ball bearings, Brown, et al. (ref. 6), with 124-mm (4.8819-in.) bore cylindrical roller bearings, and Schuller (ref. 7) with 118-mm (4.6457-in.) bore cylindrical roller bearings. An example of the effectiveness of under-race lubrication and cooling is shown in figure 2 from reference 8. Under-race lubricated ball bearings ran significantly cooler than identical bearings run with jet lubrication. Beyond 16 700 rpm

*NASA Lewis Research Center.



(a) CYLINDRICAL ROLLER BEARING.



(b) BALL THRUST BEARING.

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

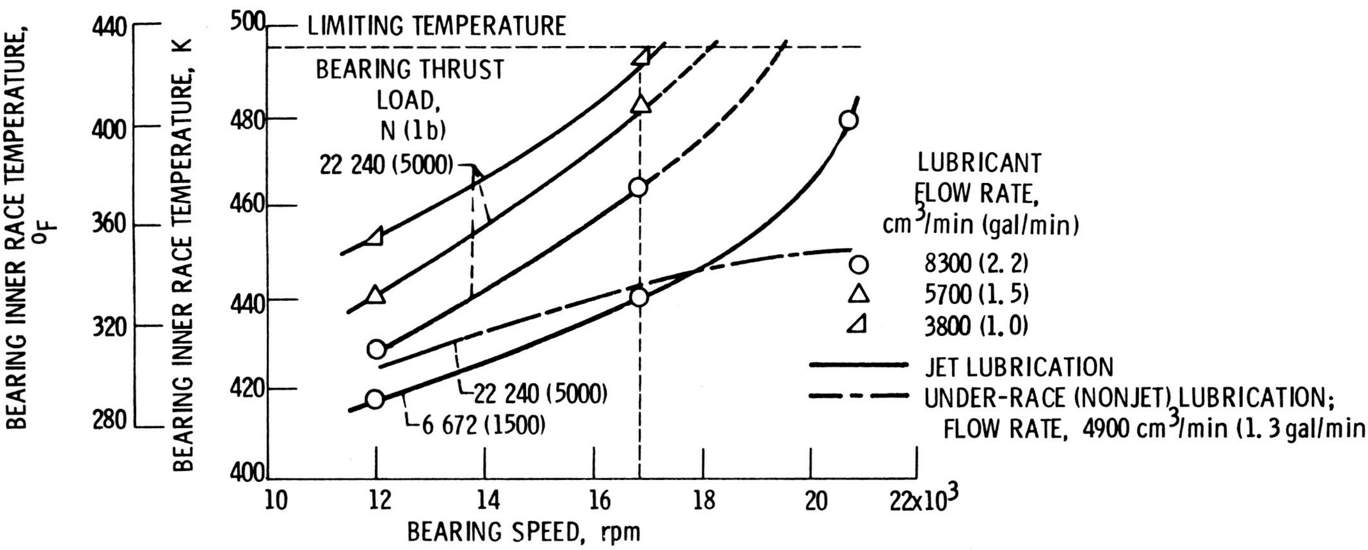


Figure 2. - Effectiveness of under-race lubrication with 120-mm-bore angular-contact ball bearings. Oil-in temperature, 394 K (250° F).(From ref. 8.)

(2 million DN) the bearing temperature with under-race lubrication increased only nominally, while that with jet lubrication increased at an accelerated rate. Only at reduced load could the jet lubricated bearings be run at 2.5 million DN. Under-race lubrication was successfully used under a variety of load conditions up to 3 million DN (refs. 5 and 8).

Applying under-race lubrication to small (<40 mm) bore bearings is more difficult because of limited space available for grooves and radial holes and for the means to get the lubricant under the race. For a given DN value centrifugal effects are more severe with small bearings since centrifugal forces vary with DN^2. Heat generated per unit of surface area is also much higher, and heat removal is more difficult in smaller bearings.

Although operation up to 3 million DN can be successfully achieved with small-bore bearings with jet lubrication, some advantages may be attained if under-race lubrication can be used. Schuller (ref. 9) has shown significantly cooler inner-race temperatures with 35-mm (1.3780-in.) bore ball bearings with under-race lubrication. As shown in figure 3 (taken from ref. 9) the effect is greater at higher speeds up to 72 000 rpm (2.5 million DN).

Tapered roller bearings have been restricted to lower speed applications than ball and cylindrical roller bearings. The speed of tapered-roller bearings is limited to that which results in a DN value of approximately 0.5 million DN (a cone-rib tangential velocity of approximately 36 m/sec (7000 ft/min)), unless special attention is given to lubricating and designing this cone-rib/roller-end contact. At higher speeds centrifugal effects starve this critical contact of lubricant.

In the late 1960's the technique of under-race lubrication, that is, lubrication and cooling of the critical cone-rib/roller-end contact, was applied to tapered-roller bearings. As described in reference 10, 88.9-mm (3.5-in.) bore tapered-roller bearings were run under combined radial and thrust loads to 1.42 million DN with cone-rib lubrication (the term used to denote under-race lubrication in tapered-roller bearings).

A comparison of cone-rib lubrication and jet lubrication was reported in reference 11 for 120.65-mm (4.75-in.) bore tapered-roller bearings under combined radial and thrust loads. These bearings were of standard catalog design except for the large end of the roller, which was made spherical for a more favorable contact with the cone-rib. Those bearings that used cone-rib

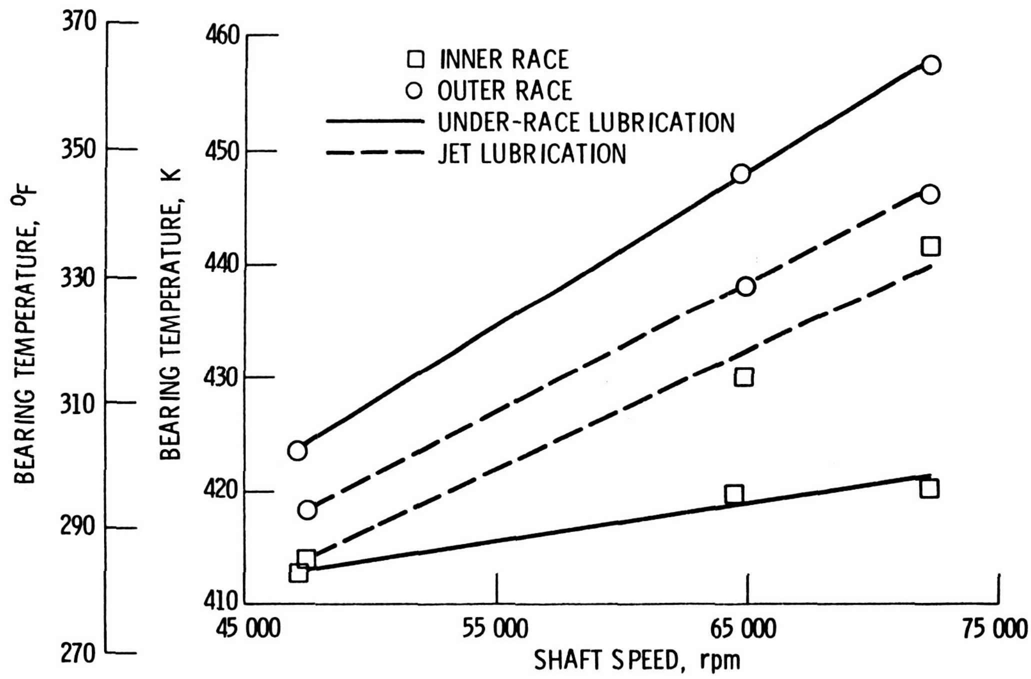


Figure 3. - Effect of under-race lubrication with 35-mm-bore angular-contact ball bearings. Total oil flow rate, 1318 cm^3/min (0.348 gal/min); oil-in temperature, 394 K (250° F). (From ref. 9.)

lubrication also had holes drilled through from a manifold in the cone bore to the undercut at the large end of the cone (fig. 4). The results of reference 11 show very significant advantage of cone-rib lubrication as seen in figure 5. At 15 000 rpm (1.8 million DN) the bearing with cone-rib lubrication

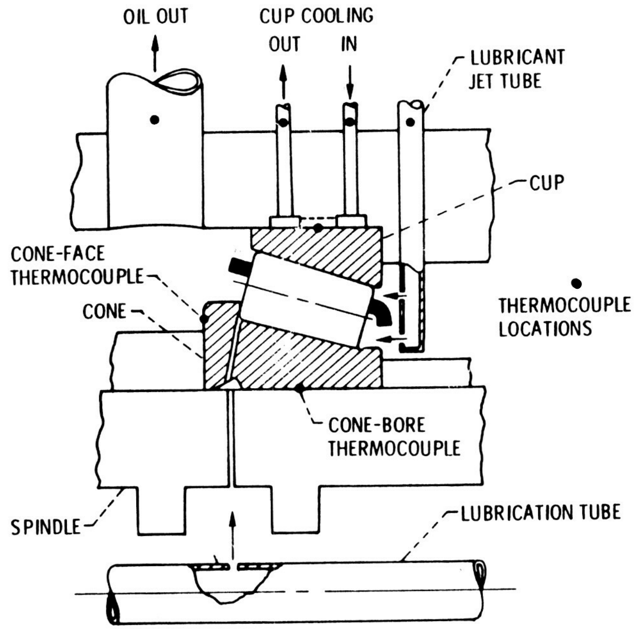


Figure 4. - Tapered-roller bearing with cone-rib and jet lubrication. (From ref. 11.)

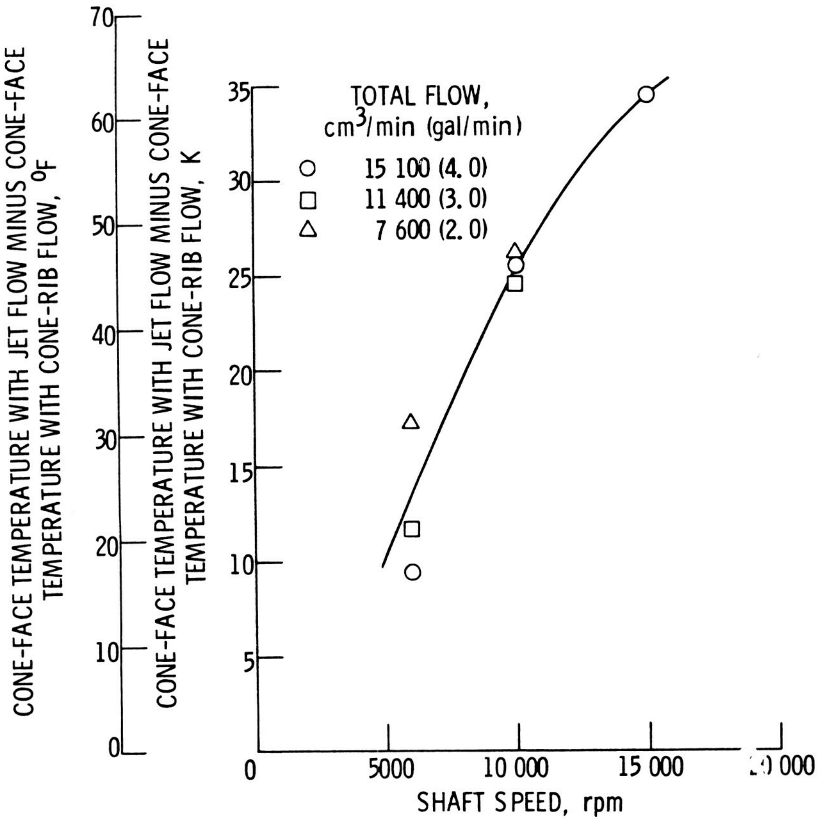


Figure 5. - Advantage of cone-rib lubrication over jet lubrication with tapered-roller bearings. Oil-in temperature, 350 K (170° F). (From ref. 11.)

had a cone-face temperature 34 K (62° F) lower than one with jet lubrication. Furthermore, reference 11 shows that the tapered-roller bearing would operate with cone-rib lubrication at 15 000 rpm with less than half the flow rate required for jet lubrication at that speed.

Further work has shown successful operation with large-bore tapered-roller bearings at even higher speeds. Orvos (ref. 12) reported on long-term operation of 107.95-mm (4.25-in.) bore tapered-roller bearings under pure thrust load to 3 million DN with a combination of cone-rib lubrication and jet lubrication. Optimized design 120.65-mm (4.75-in.) bore tapered-roller bearings were run under combined radial and thrust load with under-race lubrication to both large end (cone-rib) and small end to speeds up to 2.4 million DN (ref. 13).

Under-race lubrication has been shown to very successfully reduce inner-race temperatures. However, at the same time, outer-race temperatures either remain high (ref. 11) or are higher than those with jet lubrication (fig. 3 (from ref. 9)). Outer-race cooling can be used to reduce the outer-race temperature to levels at or near the inner-race temperature. This would further add to the speed capability of under-race lubricated bearings and avoid large differentials in bearing temperature that could cause excessive internal clearance.

The effect of outer-race cooling (or cup cooling in the case of tapered-roller bearings (fig. 4)) is shown in table I (from ref. 13). The cup outer-surface temperature is decreased to the cone bore temperature with cup cooling. With the 35-mm (1.3780-in.) bore ball bearings of reference 9, outer-race cooling significantly decreased outer-race temperatures as shown in figure 6.

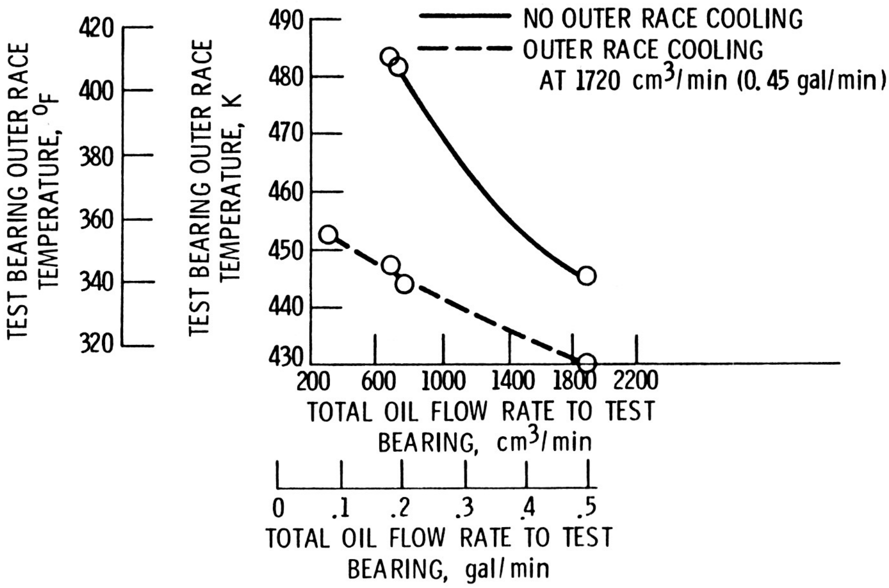


Figure 6. - Effect of outer-race cooling on outer-race temperature of 35-mm-bore ball bearings at 72 300 rpm. Oil-in temperature, 394 K (250° F). (From ref. 9.)

TABLE I. - EFFECT OF CUP COOLING ON TAPERED-ROLLER BEARING TEMPERATURES

[Shaft speed, 18 500 rpm; oil-in temperature, 364 K (195°F); total flow rate without cup cooling, 0.0114 m^3/min (3.0 gal/min); from ref. 13.]

Cup cooling flow rate, m^3/min (gal/min)	Temperature, K (°F)				
	Cone face	Cone bore	Cup outer surface	Oil-out	Cup cooling oil-out
0	389 (240)	423 (302)	438 (329)	426 (307)	---
0.0038 (1.0)	391 (245)	424 (303)	424 (304)	426 (308)	386 (235)

Under-race lubrication has been well developed for larger bore bearings and is currently being used with many aircraft turbine-engine mainshaft bearings. Because of the added difficulty of applying it, the use of under-race lubrication with small-bore bearings has been minimal, but the benefits have been demonstrated. It appears that the application of tapered roller bearings at higher speeds using cone-rib lubrication is imminent, but the experience to date has been primarily in laboratory test rigs.

The use of under-race lubrication in all the previous work referenced includes the use of holes through the rotating inner race. It must be recognized that these holes weaken the inner-race structure and could contribute to the possibility of inner-race fracture at extremely high speeds. This subject of fracture of inner races is discussed in a later section of this paper. It is apparent, however, that the fracture problem in the inner races exists even without the lubrication holes.

Rolling-Element Fatigue Life

The life of a rolling-element bearing is, for design purposes, generally considered the life to fatigue spalling of the raceways or rolling elements. The assumption is made that the bearing is properly maintained, not abused, and properly lubricated with oil or grease. The life, in stress cycles, to fatigue spalling is heavily dependent on the load on the bearing, the quality of the material, the size of the bearing, and, to a lesser extent, the lubricant.

Major advances have been made in the past two decades in the quality of bearing materials. Improved quality includes improved processing and cleanliness and greater control on material chemistry and heat treatment. In particular, AISI M–50, the bearing material used almost exclusively by aircraft gas-turbine-engine manufacturers in the United States, has seen development that has resulted in significant life improvements (ref. 2). The largest improvements are related to improved vacuum melting techniques. Specifically, the vacuum arc remelting (VAR) processing technique (ref. 2) produces a very homogeneous material with reduced nonmetallic inclusions, entrapped gases, and trace elements.

This process, when used in combination with the vacuum induction melting process (VIM), produces an outstandingly clean material (VIM–VAR). VIM–VAR AISI M–50 is currently being specified for virtually all main shaft bearings by major U.S. aircraft turbine engine manufacturers.

An example of the exceptionally long fatigue life that can be attained with VIM–VAR AISI M–50 is presented in reference 14. A group of 120-mm-bore, angular-contact ball bearings was endurance tested at 3 million DN and a thrust load of 22 200-N (5000 lb). The 10-percent fatigue life obtained was over 100 times the predicted AFBMA life. This long life includes lubrication effects, which are beneficial to life at these high speeds, so that the improvement attributed to VIM–VAR AISI M–50 was a factor of 44 (ref. 14).

AISI M–50 is a through-hardened material, heat treated such that the hardness is uniform throughout the section of a bearing component. Surface-hardened materials are frequently used for rolling-element bearings, primarily in the railroad and automotive industries. Most common are the carburized materials such as AISI 9310, 8620, and 4320 which are typically heat treated with hard cases of Rockwell C 58 to 62 and soft, tough cores of Rockwell C 20 to 48, depending on section size.

These common carburized materials have not seen widespread use in bearings in the aerospace industry, primarily because of their relatively low operating temperature capability of about 422 K (300° F) (ref. 15). Several carburized steels have been developed for higher temperature use, primarily though the addition of alloying elements such as Cr and Mo. CBS-600, CBS–1000M, and Vasco X–2 are carburized steels with continuous service capabilities of 505 K (450° F), 589 K (600° F), and 644 K (700° F), respectively. The hardness retention of these steels compared with the common carburized steels and through hardened materials is shown in figure 7 (ref. 16).

While some prototype bearings have been made with these materials, in particular, tapered-roller bearings of VAR CBS–1000M (refs. 12, 13, and 17), bearing fatigue life data are scarce. Accelerated rolling-element fatigue data and gear fatigue data (refs. 18 to 20) indicate that the rolling-element fatigue lives of these advanced carburized steels are comparable with the standard lower temperature carburized steels and with VAR AISI M–50 through-hardened steel. Life data from accelerated tests in the rolling-contact fatigue tester (ref. 18) are shown in figure 8. These advanced carburized steels show good life potential in these accelerated tests, but full-scale bearing life test data are scarce. It is apparent that more development is needed before they see widespread use in critical aircraft bearing applications.

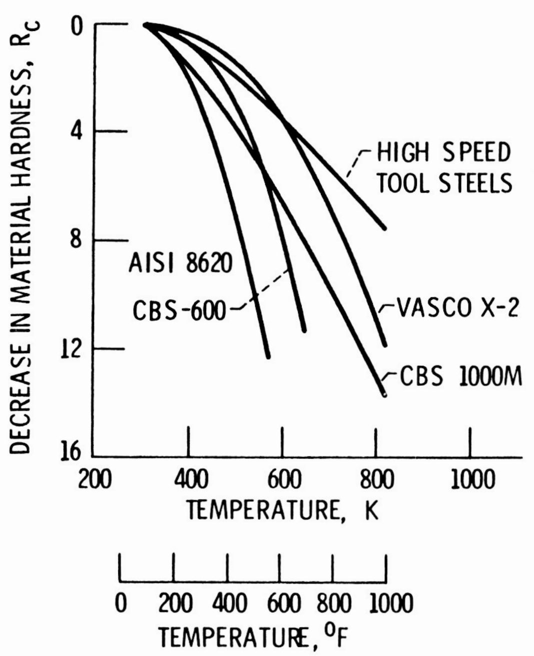


Figure 7. - Hardness retention of advanced carburizing grade steels compared with AISI 8620 steel and through-hardened, high-speed tool steels. (From ref. 16.)

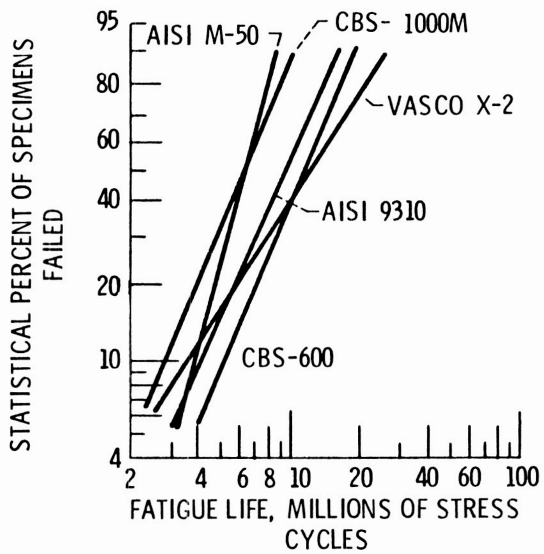


Figure 8. - Rolling-element fatigue life of carburizing grade steels and VAR AISI M-50 steel. (From ref. 18.)

Computerized Analysis

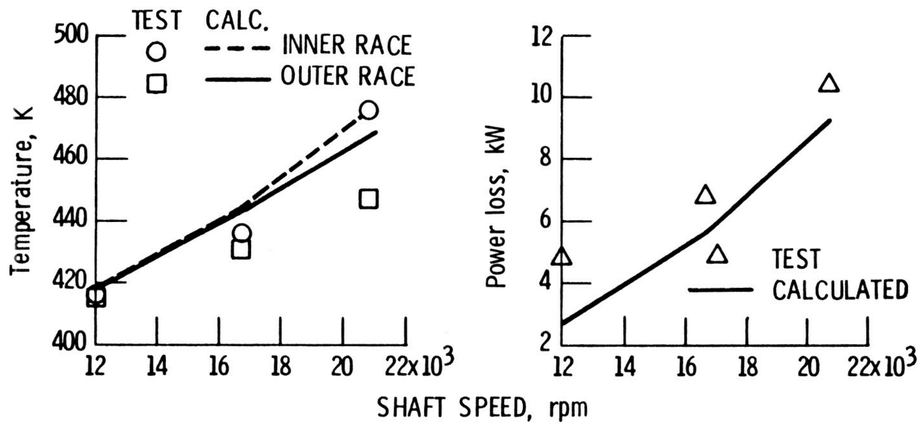
Rolling-element bearing life and performance predictions have been greatly enhanced by the use of some of the newer computer codes now available to design engineers. The ability to closely simulate the performance of a rolling-element bearing also aids in failure analysis of systems where the bearing or external conditions imposed on the bearing are suspect.

Computer analysis has progressed from the early elastic solutions of ball-bearing load distributions considering inertial and centrifugal effects (ref. 21) to a more generalized theory including lubrication and traction effects (ref. 22) and to current programs such as Shaberth (ref. 23) and Cybean (ref. 24), which include thermal analysis and calculation of temperature distributions in the bearing. These latter programs give essentially steady-state solutions (considered quasidynamic) and are useful for the majority of high-speed rolling-element bearing applications.

Computer programs of a fully dynamic nature have also been developed (refs. 25 to 29) in which time transient motions and forces of the rollers or balls and the cage are determined. Computing time with a dynamic program such as that described in reference 27 can become excessive (ref. 30). A direct comparison of a quasidynamic program (ref. 23) and a fully dynamic program (ref. 26) was described in reference 30. The "quasi-dynamic" program is more practical as a bearing design tool where fatigue life, torque, and heat generation are of primary interest. The dynamic program, although consuming large amounts of computer time, appears to be valuable as a diagnostic tool, especially where cage motions are of interest. Shaberth and Cybean, both quasidynamic computer programs, are discussed in more detail in references 3, 31, and 32.

The new computer codes can generate a great amount of output data describing bearing performance at given input conditions. Output data such as load distributions, Hertzian stresses, operating contact angle or skew angle, component speeds, heat generation, local component temperatures, bearing fatigue life, and power loss are typical. But, how well does this output predict actual conditions within an operating bearing? Only a few attempts have been made to compare predicted performance with experimental data. A major problem has been that of obtaining appropriate experimental data with enough detail and accuracy to make a correlation meaningful. Currently, the analyses compute operational characteristics such as component temperatures and roller skew angles which have not yet been experimentally verified.

A comparison of calculated and experimental performance of high-speed ball bearings is presented in reference 33. The analysis provided a good prediction of temperatures and power losses in jet-lubricated 120-mm-bore angular-contact ball bearings (fig. 9). In this work the critical assumptions were the form of the lubricant traction model and the lubricant volume percent (the assumed volume percent of the bearing cavity occupied by the lubricant).



(a) Race temperatures. (b) Bearing power loss.

Figure 9. - Comparison of calculated and experimental ball-bearing temperatures and power loss as functions of shaft speed using Shaberth computer program. Thrust load, 6672 N (1500 lb); lubricant flow rate, 8300 cm^3/min (2.2 gal/min); lubricant volume, 2 percent. (From ref. 33.)

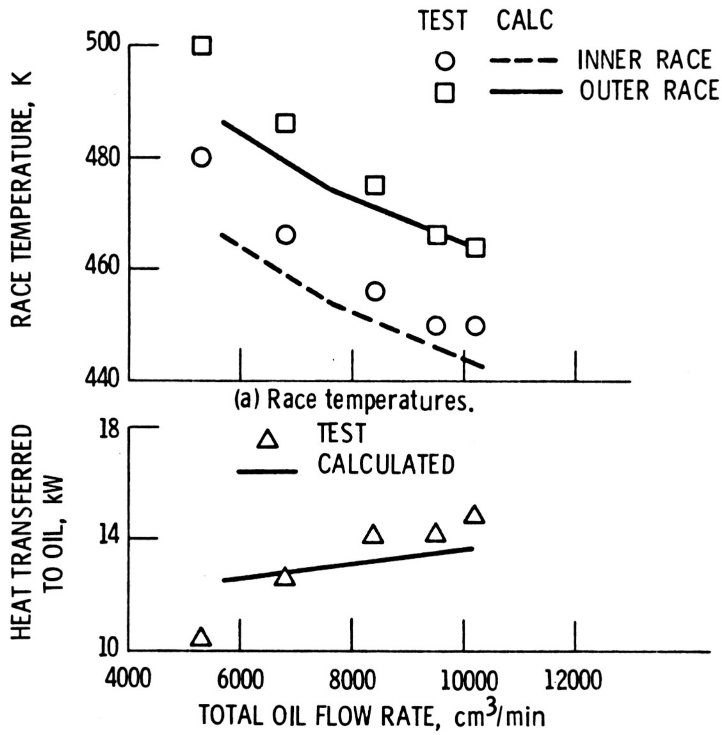
Reference 34 presents a good correlation of predicted and experimental data for 118-mm-bore cylindrical roller bearings at speeds up to 3.0 million DN. Figure 10 shows the good correlation for race temperatures and heat transferred to oil at various flow rates. This work also shows the importance of knowing the operating diametral clearance that exists when mounting fits, temperatures, and rotational speed are considered.

Another correlation of predicted and experimental data for cylindrical roller bearings is shown in reference 35. Experiments were performed with 124.3-mm-bore bearings at speeds up to 3 million DN, and the results were compared with analysis using the computer program described in reference 6. Predictions of both heat rejection to the oil and outer-race temperature were within 10 percent of the experimental values.

The work reported in references 33 and 34 emphasizes the importance of selecting an appropriate volume percent of lubricant in the bearing cavity, since heat generation and bearing temperatures are dependent on this factor. Further work needs to be done to show how lubricant volume percent varies with bearing type and design, lubrication method, lubricant flow rate, and shaft speed. Despite the limitations, these computer programs have proven to be valuable tools in the development of rolling-element bearings for specific difficult applications.

Required Technology Advancements

One of the major forces pushing the technology of rolling-element bearings is higher shaft speeds and higher DN. This stems mainly from the aircraft turbine engine manufacturers who desire higher performance and efficiency from their engines with shaft speeds approaching 3 million DN (fig. 11 (ref. 2)). As previously discussed, operation of ball and cylindrical roller bearings up to this speed has been successfully accomplished in laboratory tests with proper regard for lubrication and cooling techniques. However, the high speeds bring on other problems for which solutions are not yet in hand but which, in most cases, are currently being studied. Several of the problem areas are discussed in the following paragraphs.



(b) Heat transferred to lubricating oil.

Figure 10. - Comparison of calculated and experimental cylindrical roller bearing data using a diametral clearance of -0.02 mm in the computer program. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant volume, 2 percent. (From ref. 34.)

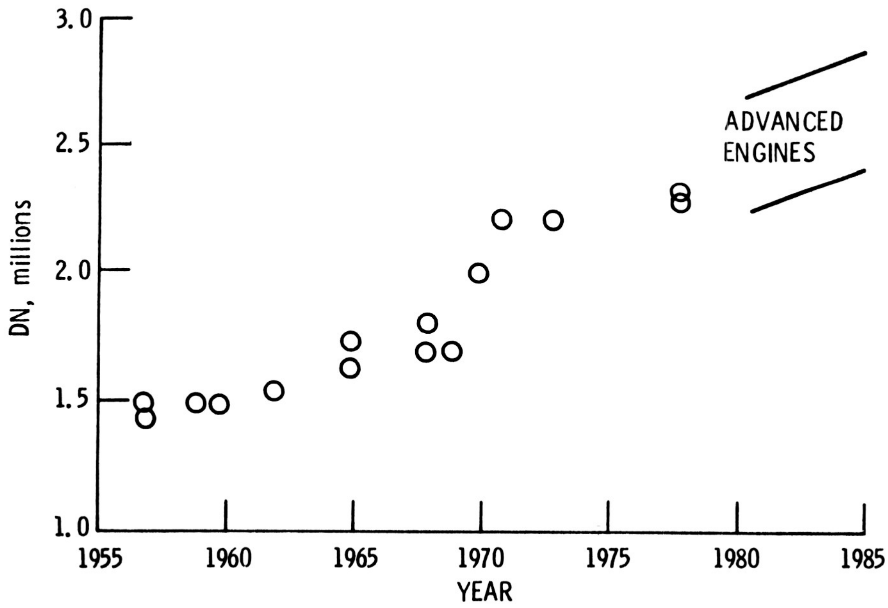


Figure 11. - Trend in aircraft engine main bearing DN. (From ref. 2.)

Rolling-Element Fatigue Life

As bearing speeds increase, bearing fatigue life decreases as shown in figure 12. This decrease is due to the increased rate of stress-cycle accumulation and centrifugal effects on the rolling elements. Therefore, to maintain an acceptable removal and/or overhaul time, increased rolling-element fatigue life in stress cycles or revolutions must be attained. This primarily applies to the thrust ball bearing on the high-speed shaft of a multishaft turbine engine where loads cannot be easily reduced and, in fact, may even be increased in the newer engines with higher thrust-to-weight ratios. Even though premium quality AISI M-50 material has shown great increases in rolling-element fatigue life, further life increases are needed. Modifications of the current alloys or the development of new materials along with improved processing and heat treatment is required to provide the desired fatigue life improvements.

Improved Fracture Toughness

It has been shown in references 14, 17, and 36, and discussed in some detail in reference 2, that bearing races made of through-hardened materials are susceptible to catastrophic fracture when exposed to the high tensile hoop stress often present in high-speed ball and rolling bearing inner races. To prevent this mode of failure, materials with improved fracture toughness must be used when bearing speeds exceed about 2.4 million DN (ref. 2). Case-hardened materials have fracture toughness values (ref. 15) high enough that they should prevent the fracture mode of failure. Further work is needed to assure that a case-hardened material can be developed that will also provide the required case hardness (Rockwell C 60 to 64), high-temperature hardness (up to 589 K (600° F) for main shaft engine application), and rolling-element fatigue life. Currently, programs sponsored by the DOD and the NASA are pursuing improved fracture toughness materials.

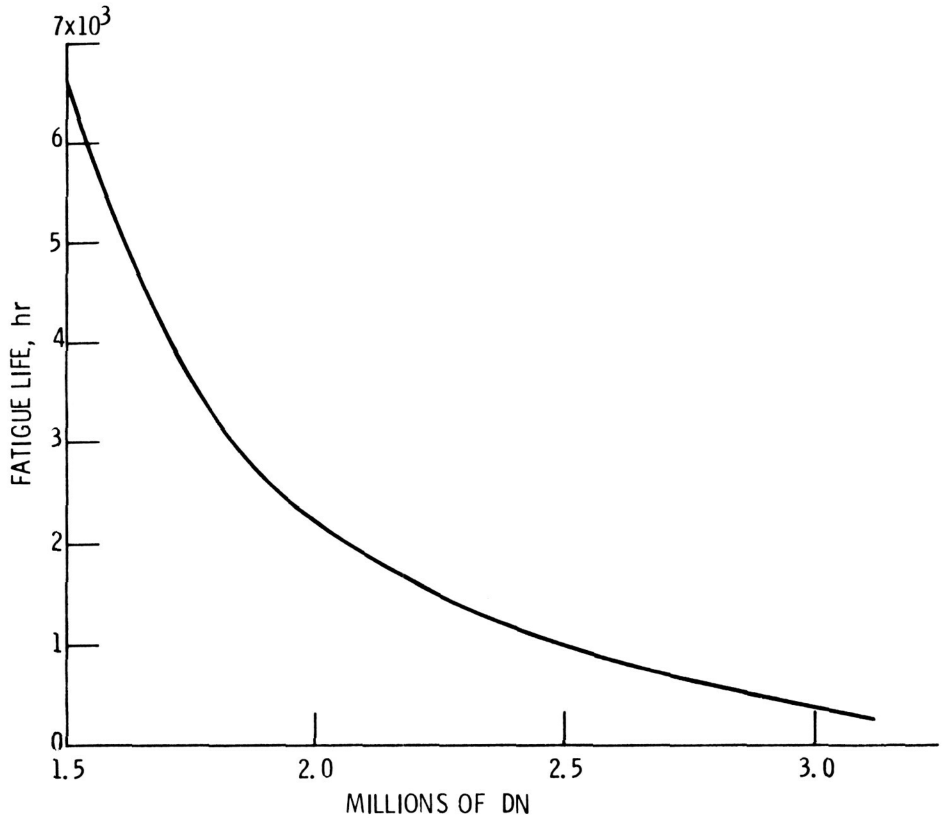


Figure 12. - Typical effect of DN on bearing fatigue life for a given bearing size.

Survivability

Of major concern in aircraft engine and transmission systems is the prevention of catastrophic failure in the unfortunate event of loss of lubricating oil. A typical requirement is 30 minutes of continued normal operation of a helicopter engine and transmission after loss of lubricating oil supply (ref. 37). This requirement was set primarily for military operations, but it is also applicable to civilian use, such as for off-shore drilling operations. Another military requirement for turbine engines is a capability of continued operation after a 1-minute interruption in the lubricant supply to the bearings.

These requirements are increasingly more difficult to meet as bearing speeds increase above 2 million DN for ball and cylindrical roller bearings. For tapered-roller bearings at speeds where lubrication through the cone to the cone-rib-roller-end contact is required (above about 1 million DN), lubricant interruption causes almost immediate bearing failure. Before tapered-roller bearings can be safely used in high-speed positions in engines or transmissions, a solution to this limitation must be found. Similar limits are applicable to a thrust-carrying cylindrical roller bearing (ref. 38), which also has critical roller-end-rib contacts. Solutions to the survivability problem may take the form of design modifications, supplementary lubrication systems, reduced-friction coatings or materials, or a combination of these.

Corrosion Resistance

The problem of corrosion is, of course, not unique to high-speed bearings as are the previous areas. The problem is most severe in systems with long periods of nonuse. A summary of causes for bearing rejection at a U.S. Navy facility (ref. 39) shows that corrosion accounts for nearly one-third of the bearing rejections from their aircraft systems, including drivelines, wheels, and accessories. Air Force experience (ref. 40) confirms that corrosion is a major cause of rejection at overhaul of aircraft turbine engine bearings.

The corrosion problem is being attacked on three major fronts: corrosion-resistant materials, corrosion-resistant coatings or surface modifications, and corrosion-inhibited lubricants. Materials such as AISI 440C and AMS 5749 are called corrosion resistant or stainless steels, but under some severe conditions in aircraft bearings, they will corrode. They are, however, more corrosion resistant than the common aircraft bearing materials such as AISI M-50, AISI 52100, and the commonly used case-carburized materials. Ceramics such as silicon nitride are truly corrosion resistant in the aircraft environment. Although this material is not yet well developed for widespread aircraft application, it could be applied in some specific cases.

Chromium-ion implantation has shown significant improvement in the corrosion resistance of AISI M-50 (ref. 41). Other surface modifications or coatings may also provide similar benefits but more work is needed in this area. It is probable that a combination of approaches, such as the use of silicon nitride balls or rollers and coated raceways may be a viable solution for some applications that require long periods of nonuse.

Currently, the lubricant provides some measure of corrosion protection in aircraft bearings simply by keeping the components coated and moisture free. Corrosion inhibiting additives for the commonly used MIL-L-23699 lubricant have shown promise in laboratory tests (ref. 42), and confirmation of the degree of improved corrosion resistance in actual aircraft systems is under way.

Concluding Remarks

This paper has reviewed the present state of technology of rolling-element bearings. The most recent advancements related to aircraft engine and transmission systems were emphasized, since it is within this area that major improvements have been realized. Improvements in the speed capabilities of large-bore ball and cylindrical and tapered-roller bearings have been realized with through-the-race lubrication. Spherical-roller bearings and small-bore (<40-mm) ball and cylindrical roller bearings have seen much less development for higher speeds, but efforts are currently under way.

Premium quality AISI M-50 steel has shown greatly improved rolling-element fatigue life over the previously used materials. However, further improvements as well as in fracture toughness and corrosion resistance are needed.

Life and performance predictions of rolling-element bearings are now much better because of the use of recently developed computer programs. The programs are valuable in designing for new, difficult applications and in diagnosing operating systems in which problems or failures have occurred. Further improvements in the analysis are needed, particularly in the definition of the lubricant-air mixture in an operating bearing and in the efficiency of fully dynamic programs which now consume large amounts of computer time.

References

1. Brown, P. F.: Bearings and Dampers for Advanced Jet Engines. SAE Paper 700318, Apr. 1970.
2. Bamberger, E. N.: Materials for Rolling-Element Bearings. Bearing Design—Historical Aspects, Present Technology and Future Problems, W. J. Anderson, ed., The American Society of Mechanical Engineers, 1980, pp. 1-46.
3. Pirvics, J.: Computerized Analysis and Design Methodology for Rolling Element Bearing Load Support Systems. Bearing Design—Historical Aspects, Present Technology, and Future Problems, W. J. Anderson, ed., The American Society of Mechanical Engineers, 1980, pp. 47-85.
4. Holmes, P. W.: Evaluation of Drilled Ball Bearings at DN Values to Three Million. NASA CR-2004 and NASA CR-2005, 1972.
5. Signer, H.; Bamberger, E. N.; and Zaretsky, E. V.: Parametric Study of the Lubrication of Thrust Loaded 120-mm Bore Ball Bearings to 3 Million DN. J. Lubr. Technol., vol. 96, no. 3, July 1974, pp. 515-524.
6. Brown, P. F.; et al.: Mainshaft High Speed Cylindrical Roller Bearings for Gas Turbine Engines. PWA-FR-8615, Pratt / Whitney Aircraft Group, 1977.
7. Schuller, F. T.: Operating Characteristics of a Large-Bore Roller Bearing to Speeds of 3×10^6 DN. NASA TP-1413, 1979.
8. Zaretsky, E. V.; Signer, H.; and Bamberger, E. N.: Operating Limitations of High-Speed Jet-Lubricated Ball Bearings. J. Lubr. Technol., vol. 98, no. 1, Jan. 1976, pp. 32-39.
9. Schuller, F. T.; and Signer, H. R.: Performance of Jet- and Inner-Ring-Lubricated 35-Millimeter-Bore Ball Bearings Operating to 2.5 Million DN. NASA TP-1808, 1981.
10. Lemanski, A. J.; Lenski, J. W., Jr.; and Drago, R. J.: Design Fabrication, Test, and Evaluation of Spiral Bevel Support Bearings (Tapered Roller). Boeing Vertol Co., 1973. (USAAMRDL-TR-73-16, AD-769064.)
11. Parker, R. J.; and Signer, H. R.: Lubrication of High-Speed, Large Bore Tapered-Roller Bearings. J. Lubr. Technol., vol. 100, no. 1, Jan. 1978, pp. 31-38.
12. Orvos, P. S.; and Dressler, G. J.: Tapered Roller Bearing Development for Aircraft Turbine Engines. Timken Co., 1979. (AFAPL-TR-79-2007, AD-A069440.)
13. Parker, R. J.; Pinel, S. I.; and Signer, H. R.: Performance of Computer Optimized Tapered-Roller Bearings to 2.4 Million DN. J. Lubr. Technol., vol. 103, no. 1, Jan. 1981, pp. 13-20.
14. Bamberger, E. N.; Zaretsky, E. V.; and Signer, H.: Endurance and Failure Characteristics of Main-Shaft Jet Engine Bearing at 3×10^6 DN. J. Lubr. Technol., vol. 98, no. 4, Oct. 1976, pp. 580-585.
15. Jatczak, C. F.: Specialty Carburizing Steels for Elevated Temperature Service. Met. Prog., vol. 113, no. 4, Apr. 1978, pp. 70-78.
16. Anderson, N. E.; and Zaretsky, E. V.: Short-Term Hot-Hardness Characteristics of Five Case Hardened Steels. NASA TN D-8031, 1975.
17. Parker, R. J.; Signer, H. R.; and Pinel, S. I.: Endurance Tests with Large-Bore Tapered-Roller Bearings to 2.2 Million DN. ASME Paper 81-Lub-56, Oct. 1981.
18. Nahm, A. H.: Rolling-Element Fatigue of Gear Materials. (R78AEG476, General Electric Co.; NASA Contract NAS3-14302.) NASA CR-135450, 1978.
19. Townsend, D. P.; Parker, R. J.; and Zaretsky, E. V.: Evaluation of CBS 600 Carburized Steel as a Gear Material. NASA TP-1390, 1979.
20. Townsend, D. P.; and Zaretsky, E. V.: Endurance and Failure Characteristics of Modified Vasco X-2, CBS 600, and AISI 9310 Spur Gears. ASME Paper 80-C2/DET-58, Aug. 1980.
21. Jones, A. B.: A General Theory for Elastically Constrained Ball and Radial Roller Bearings Under Arbitrary Load and Speed Conditions. J. Basic Eng., vol. 82, no. 2, June 1960, pp. 309-320.
22. Harris, T. A.: Ball Motion in Thrust-Loaded, Angular Contact Bearings with Coulomb Friction. J. Lubr. Technol., vol. 93, no. 1, Jan. 1971, pp. 32-38.
23. Crecelius, W. J.: Users Manual for Steady State and Transient Thermal Analysis of a Shaft-Bearing System (SHABERTH). SKF-AL77P015, SKF Industries, Inc., 1978. (ARBRL-CR-00386, AD-A064150.)
24. Kleckner, R. J.; Pirvics, J.; and Castelli, V.: High-Speed Cylindrical Rolling Element Bearing Analysis "CYBEAN"—Analytic Formulation. J. Lubr. Technol., vol. 102, no. 3, July 1980, pp. 380-390.
25. Walters, C. T.: The Dynamics of Ball Bearings. J. Lubr. Technol., vol. 93, no. 1, Jan. 1971, pp. 1-10.
26. Gupta, P. K.: Dynamics of Rolling-Element Bearings—Part III, Ball Bearing Analysis. J. Lubr. Technol., vol. 101, no. 3, July 1979, pp. 312-318.
27. Gupta, P. K.: Dynamics of Rolling-Element Bearings—Part I, Cylindrical Roller Bearing Analysis. J. Lubr. Technol., vol. 101, no. 3, July 1979, pp. 293-304.
28. Conry, T. F.: Transient Dynamic Analysis of High-Speed Lightly Loaded Cylindrical Roller Bearings, I—Analysis. NASA CR-3334, 1981.

29. Conry, T. F.; and Goglia, P. R.: Transient Dynamic Analysis of High-Speed Lightly Loaded Cylindrical Roller Bearings, II—Computer Program and Results. NASA CR-3335, 1981.
30. Schulze, D. R.: An Evaluation of the Usefulness of Two Math Models for Predicting Performance of a 100-mm Bore, Angular Contact, High-Speed, Thrust Bearing. AFWAL-TR-80-2007, Air Force Wright Aeronautical Laboratories., Apr. 1980. (AD-A089161.)
31. Gupta, P. K.: A Review of Computerized Simulations of Roller Bearing Performance. Computer-Aided Design of Bearings and Seals, F. E. Kennedy and H. S. Cheng, eds., The American Society of Mechanical Engineers, 1976, pp. 19-29.
32. Sibley, L. B.; and Pirvics, J.: Computer Analysis of Rolling Bearings. Computer-Aided Design of Bearings and Seals, F. E. Kennedy and H. S. Cheng, eds., The American Society of Mechanical Engineers, 1976, pp. 95-115.
33. Coe, H. H.; and Zaretsky, E. V.: Predicted and Exerimental Performance of Jet-Lubricated 120-Millimeter-Bore Ball Bearings Operating at 2.5 Million DN. NASA TP-1196, 1978.
34. Coe, H. H.; and Schuller, F. T.: Comparison of Predicted and Experimental Performance of Large-Bore Roller Bearing Operating to 3.0 Million DN. NASA TP-1599, 1980.
35. Brown, P. F.; Dobek, L. J.; and Tobiasz, E. J.: High-Speed Cylindrical Roller Bearing Development. PWA-FR-12598, Pratt & Whitney Aircraft Group, Aug. 1980. (AFWAL-TR-80-2072, AD-A095357.)
36. Clark, J. C.: Fracture Failure Modes in Lightweight Bearings. J. Aircr., vol. 12, no. 4, Apr. 1974, pp. 383-387.
37. Lenski, J. W.: Test Results Report and Design Technology Development Report—HLH/ATC High-Speed Tapered-Roller Bearing Development Program. T301-10248-1, Boeing Vertol Co., 1974. (USAAMRDL-TR-74-33, AD-786561.)
38. Morrison, F. R.; Pirvics, J.; and Crecelius, W. J.: A Functional Evaluation of a Thrust Carrying Cylindrical Roller Bearing Design. J. Lubr. Tech., vol. 101, no. 2, Apr. 1979, pp. 164-170.
39. Cunningham, J. S., Jr.; and Morgan, M. A.: Review of Aircraft Bearing Rejection Criteria and Causes. Lubr. Eng., vol. 35, no. 8, Aug. 1979, pp. 435-440.
40. Jones, H. F.: Discussion to Review of Aircraft Bearing Rejection Criteria and Causes. Lubr. Eng., vol. 35, no. 8, Aug. 1979, p. 441.
41. Hubler, G. K.; et al.: Application of Ion Implantation for the Improvement of Localized Corrosion Resistance of M-50 Steel Bearings. NRL-MR-4481, Naval Research Laboratory, March 1981. (AD-A097230.)
42. Brown, C.; and Feinberg, F.: Development of Corrosion-Inhibited Lubricants for Gas Turbine Engines and Helicopter Transmissions. Lubr. Eng., vol. 37, no. 3, Mar. 1981, pp. 138-144.