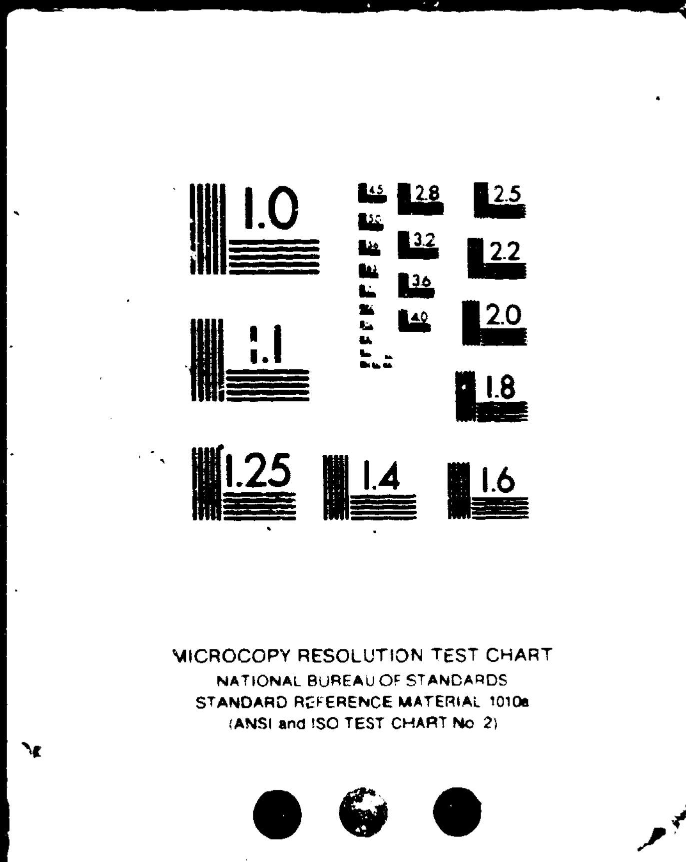
1 OF 1

N85 34409 UNCLAS



NASA Technical Memorandum 87107

Design and Lubrication of High-Speed Rolling-Element Bearings

(NASA-TM-87107) DESIGN AND LUBRICATION OF HIGH-SPEED ROLLING-ELEMENT BEARINGS (NASA) 19 p HC A02/MF A01 CSCL 13I

N85-34409

Unclas
G3/37 22231

Erwin V. Zaretsky
Lewis Research Center
Cleveland, Ohio

Prepared for the
Original Equipment Manufacturing Design Conference
Philadelphia, Pennsylvania, September 9-11, 1985

NASA

DESIGN AND LUBRICATION OF HIGH-SPEED ROLLING-ELEMENT BEARINGS

by

Erwin V. Zaretsky
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

E-2670

ABSTRACT

The speed capability of rolling-element bearings has increased from speeds of less than two million DN to speeds of three million DN. The life and reliability of these bearings have also increased where they are equal to, or greater than, those of bearings with limited speed capability. However, high-speed bearings are not "off-the-shelf" bearings which can be readily ordered from a manufacturer's catalog. Design parameters must be carefully chosen and optimized based upon sophisticated bearing computer programs. Material and lubricant selection must be integrated into the bearing design. Bearing thermal management must be implemented through proper lubrication and cooling. Parameters which can be used to design, specify, and lubricate high-speed bearings are presented and discussed.

INTRODUCTION

In the early years of the aircraft turbojet engine, engine life and reliability were generally limited by rolling-element bearing technology. Bearing life in a jet engine environment was limited to approximately 300 hr MTBR (mean time between removal). Research and development activities by the major U.S. engine manufacturers, bearing companies and government laboratories over the past three decades have resulted in extending bearing lives in engine operation to approximately 30 000 hr MTBR. At the same time bearing speed capability has also been vastly increased (fig. 1). Bearing speeds of 3 million DN and lives 100 times catalog rating have been demonstrated for specially designed rolling-element bearings using material, lubrication and manufacturing technologies which are now commercially available.

Bearings used in commercial turbomachinery operating at higher speeds demand the same design and lubrication considerations as those found in advanced airbreathing aircraft engines. Besides the effect of higher stress at the contact of the rolling element and the outer race due to centrifugal force at higher speeds, heat generation and power loss within the bearings are major problems. Thermal analysis of the internal geometry of the bearing with regard to fits and clearances becomes important. Shaft and housing tolerances on the fit and expansion of the bearing rings are a consideration. Cooling of the bearing through lubrication must be efficiently performed. Material and lubricant type must be selected to both assume compatibility with each other and with the higher temperatures found in high-speed bearings. Current lubricants are limited to bearing temperatures of 425 °F and materials to temperatures above 600 °F.

In designing a bearing for these higher speeds a balance must be maintained among attaining the required life, incurring surface damage by skidding of the rolling elements, inner-ring fracture due to resulting hoop stresses and fretting problems due to improper fits at operating conditions. Shaft deflections from shaft loading and rotor dynamics can exceed the bearing design limitations. When a tolerance of 100 millionth (0.0001) of an inch is critical to bearing operation and performance these deflections become an important design consideration.

The design of a high-speed rolling-element bearing is a rather sophisticated process. Contrary to the implication of many engineering design texts, it is not possible to select a high-speed bearing from a manufacturer's catalog considering only the external dimensions of the bearing and the applied load and speed. These bearings must be designed for the high-speed application and conversely, the application must be designed for the bearings. Sophisticated bearing computer analysis must be employed for this purpose. It becomes the objective of this paper to review the design and lubrication methods currently used in high-speed, rolling-element bearing design and lubrication as well as material and lubricant selection. While there may be other design methods and analysis employed for this purpose, the methods presented have been established in laboratory experiments and field service to achieve the desired performance results.

COMPUTER ANALYSIS

There are several comprehensive computer programs that are capable of predicting rolling-element bearing operating and performance characteristics. These programs generally accept input data: bearing internal geometry (such as sizes, clearance, and contact angles), bearing material and lubricant properties, and bearing operating conditions (load, speed, and ambient temperature). The programs then solve several sets of equations that characterize rolling-element bearings. The output produced typically consists of rolling-element loads and Hertz stresses, operating contact angles, component speeds, heat generation, local temperatures, bearing fatigue life, and power loss. The critical assumptions currently necessary in the use of these programs are the form of the lubricant traction model and the lubricant volume percent (the assumed volume percent of the bearing cavity occupied by the lubricant). Two of these programs titled "Shaberth" and "Cybean" are available from COSMIC, 112 Barrow Hall, University of Georgia, Athens, GA 30602, or SKF Industries, Inc., 1100 First Ave., King of Prussia, PA 19406.

The Shaberth program simulates the thermal-mechanical performance of a flexible shaft supported by as many as five rolling-element bearings comprising combinations of ball, cylindrical and tapered roller bearings. Cybean analyses a single cylindrical roller bearing. Both programs are capable of calculating the thermal and kinematic performance of high-speed bearings, and Cybean includes a roller skew prediction for misaligned conditions. Figure 2 illustrates the modal system for thermal routines in Cybean. The correlation between predicted and experimental results with these two programs is very good. However, the proper use of these programs requires an engineer knowledgeable in bearing technology.

MATERIAL SELECTION

A commonly accepted minimum hardness for rolling-element bearing components at operating temperature is Rockwell C58. At hardnesses below this value brinelling of the bearing races can occur at stress levels normally experienced. Bearing life is also a function of material hardness. Since hardness decreases with temperature, conventional bearing materials, such as AISI 52100 can be used only to temperatures of about 250 °F. The M-series steels such as AISI M-50 can retain a minimum hardness of Rockwell C58 to approximately 700 °F. Because of its life potential and hardness retention, the AISI M-50 material is the choice material for high-speed ball and roller bearings. The nominal room temperature material hardness of the rolling elements and the races should be Rockwell 63. The hardness difference (ΔH) of the rolling elements minus the races should be zero. This material in its through hardened state can be used to speeds of approximately 2.3 million DN. At higher speeds with the occurrence of a spall in the inner race, hoop stresses due to centrifugal force and press fits can cause the inner ring to fracture causing catastrophic failure of the bearing and probable secondary failure of the rotor system. Clearly this failure mode is unacceptable. As a result, bearing speeds using through hardened steels must be limited speeds of less than 2.3 million DN.

A solution to this problem is the use of carburized or surface hardened steels wherein the material core remains in a soft ductile state and fracture resistant. E.N. Bamberger at the General Electric Company, devised a modified AISI M-50 material which can be carburized. This material is designated M-50 Nil. With the M-50 Nil material, bearing speeds of 3 million DN can be achieved without fear of inner-ring fracture.

Retained austenite content of the AISI M-50 material should be kept at 3 percent or less. This should minimize dimensional growth because of the retained austenite transforming into martensite.

Rolling-element bearing reliability and load capability increases significantly when nonmetallic inclusions, entrapped gases, and trace elements are eliminated or reduced. Improvements in steel making processing, namely double melting in a vacuum, can achieve this. Vacuum-induction melted, vacuum-arc remelted (VIM-VAR) AISI M-50 steel achieves the aforesaid results with significant improvement in bearing life and reliability over single-vacuum melted (VIM) steel. VIM-VAR AISI M-50 and VIM-VAR M-50 Nil are commercially available and should be specified for high-speed bearing application.

For angular-contact ball bearings, the bearing races should be forged to assure parallel grain flow in the ball-race contact. Research has shown that improvement in life of rolling-element bearings occurs with parallel grain flow. For cylindrical roller bearings and deep groove ball bearings, the preferential grain flow already exists without the need for forging.

In recent years the use of ceramic rolling elements has been proposed for high-speed bearing applications because of their lighter weight resulting in lower centrifugal force at the outer-race contact. What has not been considered is that the modulus of elasticity (Young's modulus) and/or Poisson's ratio of these materials are higher than steel such that, even considering the reduction in centrifugal force, the contact stresses at the inner and outer races are greater than with steel rolling elements. This would result in lower lives than with an all steel bearing. Unless there are conditions where the environment is noncompatible with steel, ceramic materials are not recommended for high-speed bearing application.

BEARING DESIGN

Angular-Contact Ball Bearings - Angular-contact ball bearings can be designed and operated for speeds of 3 million DN with acceptable life and reliability. Based upon experience, a nominal contact angle of 24° and race comformities of 54 and 52 percent at the inner and outer races, respectively, can be used as starting parameters in the design. These parameters can be varied in the design processes to optimize fatigue life and minimize power loss and temperature differences between the inner and outer races. The nominal contact angle β will change with load and speed as shown in fig. 3. As speed is increased, the contact angle at the outer race will decrease while the contact angle at the inner race will increase. The bearing designer or user must determine whether at full operating speed and load the ball will ride on the shoulder of the inner race. Should this occur, bearing life can be significantly shortened. The differences in temperature between the inner and outer races will affect the internal bearing clearances and, thus, bearing performance and life. Should the inner-race temperature increase at a greater rate than that of the outer race, the bearing may lock up.

For high-speed applications with bore sizes greater than 75 mm, the cage design should be a one-piece inner-land riding type. Experience has indicated that the cage should be made out of an iron base alloy (AMS 6415) heat treated to a Rockwell C hardness range of 28 to 35 and having a 0.005-cm (0.002 in) maximum thickness of silver plate (AMS 2410). The cage should be balanced within 3 gm-cm (0.042 oz-in). For bore sizes less than 75 mm, single-land, outer-race riding cages may be the preferred design.

The tolerance grade of the bearings should be ABEC-5 or better. However, the ABEC specification does not consider surface finish or waviness. Waviness of the raceway should be 100 μin or less. Surface finishes of the races should be 2 μin or better and 1 μin for the balls. With honing, surface finishes of 1 μin can be obtained for the raceways. However, not all precision bearing manufacturers can attain this finish.

Roller Bearings - High-speed roller bearings present perhaps a greater design challenge than ball bearings. These bearings can be run to speeds of 3 million DN. For these bearings at high speed, surface distress caused by roller skidding or skewing is the cause of failure. Skidding occurs when the radial load on the bearing is inadequate to develop enough tractive force between the raceways and roller. With insufficient tractive force, true rolling cannot be maintained, and roller sliding or skidding results. Where there is a sufficient lubricant film separating the surfaces no damage to the bearing would be anticipated. However, where the film is marginal, damage can occur.

Skewing of a roller is erratic gyration on a axis not parallel to the axis of the bearing. Skewing results in loading of the roller ends producing excessive end wear, wear of the race flanges or guide channels, and wear of the cage or separator pockets. Should this condition continue for any length of time, cage failure will occur or the roller will wear sufficiently to turn 90° in the direction of rolling and lock up the bearing. Hence, the bearing must be designed to prevent skewing and minimize skidding.

The primary method of eliminating skidding is by increasing the contact stress on the individual rollers. This can be done by reducing both the size and number of rollers in a bearing. The amount of reduction will be limited by the desired design life of the application.

Another method for increasing the contact stress is shown in fig. 4. In most roller bearings, the maximum number of rollers under load is 20 percent as illustrated in fig. 4(a). Deflecting the bearing outer race by adding a radial preload at two points 90° to the applied load the number of rollers under load can be increased to approximately 60 percent. This is illustrated in fig. 4(b). The design and manufacture of an elliptical outer raceway can accomplish this result.

There have been suggestions to use preloaded hollow rollers to produce a load on all rollers within the bearing. While at first blush this suggestion has merit, experiments have shown that these hollow rollers can fail by flexture fatigue resulting in catastrophic bearing failure.

For most roller bearings, the roller of the bearing is crowned to prevent edge loading of the roller ends and thus premature failure. Typical features of a crowned roller for a high-speed bearing are shown in fig. 5. The length of the roller should not exceed its diameter. The corner radius runout with the diameter of the roller should not exceed 0.001 in total. Concentricity of the crown profile to the diameter should be within 25 millionth of an inch. Roller ends should be square within 0.0001 in. Surface finishes on the rollers are usually 2 to 5 μin and on the raceway 4 to 6 μin.

Referring to fig. 6, a relatively large crown drop is necessary to accommodate a large misalignment and prevent skewing. This misalignment is caused by improper assembly or shaft bending under load. However, increasing crown drop beyond 0.007 in will not greatly affect skewing.

Unbalance of the roller will cause roller-gyration skewing. Where the roller is not dynamically balanced then roller-gyration skewing may be caused by (a) excessive corner radius runout with the roller diameter; (b) a crown profile skewed to axis of the roller and (c) lack of squareness of the ends of the roller to the diameter.

Cage design considerations should be similar to those of the ball bearings. Clearance of the roller in the cage pocket should be approximately 5 percent of the roller diameter to permit the roller to move within the pocket without excessive wear. Speed variations cause this roller movement.

Channel-to-roller clearances must be small to maintain good guidance of the roller. Total clearance should be approximately 0.0010 to 0.0015 in. This clearance is sufficient to allow for lubrication of the roller end - flange contact and provides for adequate guidance.

Frederi k T. Schuller of the NASA Lewis Research Center has developed a three piece inner-race roller bearing (fig. 7). This concept allows the flanges and races to be manufactured from different materials where required and to be machined with super finishes on the raceway and the roller-flange contacts on the order of 1 to 2 μin RMS. The concept eliminates stress concentrations at lubricant holes in the raceway surfaces at higher speeds found in conventional designs. It also allows for modified configurations of the flange and roller (fig. 8). This bearing design has been successfully run to speeds of 3 million DN. While the bearing is not commercially available, it can be specially ordered from precision bearing manufacturers.

Tapered-Roller Bearing - Tapered-roller bearings have been restricted to lower speed applications than have been ball and cylindrical roller bearings. The speed limitation is primarily due to the cone-rib/roller-end contact which requires very careful lubrication and cooling consideration at higher speeds. The speed of tapered roller bearings is limited to approximately 0.5 million DN (a cone-rib tangential velocity of approximately 700 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.

A computer optimized design of a tapered roller bearing capable of speeds to 2.4 million DN is given in table 1. Tapered roller bearings must use case carburized steels. For higher speed applications materials such as CBS-1000 M and CBS 600 should be considered. To date, M-50 Nil has not been used for tapered roller bearings.

There is a caveat for using tapered-roller bearings at higher speeds. Should there be lubricant starvation to the bearing, the bearing will fail in an extremely short time period. This time is usually less than the time to shut down the system without secondary damage. The problem is currently being worked on. However, no state-of-the-art solution to a commercial user is available.

LUBRICATION METHODS

Jet Lubrication - Where speeds are too high for grease or simple splash lubrication, jet lubrication is used to both lubricate and control bearing temperatures by removing generated heat. In jet lubrication the placement and number of nozzles, jet velocity, lubricant flow rates, and removal of lubricant from the bearing and immediate vicinity are all very important for satisfactory operation.

The placement of jets should take advantage of any natural pumping ability of the bearings. This is illustrated in fig. 9 for a ball bearing with relieved races and for a tapered-roller bearing. Centrifugal forces aid in moving the oil through the bearing to cool and lubricate the elements.

Directing jets in the radial gaps between the races and the cage is beneficial. The design of the cage and the lubrication of its surfaces sliding on the shoulders of the races greatly affects the high-speed performance. The cage has been typically the first element to fail in a high-speed bearing with improper lubrication.

It has been shown that with proper bearing cage design, nozzle placement, jet velocities, and adequate scavenging of the lubricant, jet lubrication can be successfully used for small-bore ball bearings at speeds to 3 million DN. For large ball bearings, speeds to 2.5 million DN are attainable. For large tapered-roller bearings, jet lubrication was successfully demonstrated to 1.8 million DN, although a high lubricant flow rate of 0.0151 m^3/min (4 gpm) and a relatively low oil-inlet temperature of 170 °F were required.

Under-Race Lubrication - A more effective and efficient means of lubricating rolling-element bearings is under-race lubrication. Conventional jet lubrication fails to adequately cool and lubricate the inner-race contact as the lubricant is thrown centrifugally outward. Unfortunately, increased flow rates add to heat generated from oil churning. An under-race lubrication system used in turbofan engines for ball and cylindrical roller bearings is shown in fig. 10. Lubricant is directed under the inner race and centrifugally forced out through a plurality of holes in the race to cool and lubricate the bearing. Some lubricant may pass completely under the bearing for cooling only as shown in fig. 10(a). Lubricant supply holes are usually provided for the cage-land and the roller-flange contacts (fig. 11).

Proper "Bearing Thermal Management" is a requirement for successful high-speed bearing operation. This can be further achieved by outer-race cooling with under-race lubrication (fig. 12). By control of lubricant flow to the outer- and inner-races internal clearances of the bearing are maintained and controlled over a range of loads and operating speeds to 3 million DN (fig. 12(a)). For the tapered-roller bearing (fig. 12(b)), by supplying lubricant to the cone-rib/cage contact as well as the cone-rib/roller-end, contact speeds of 2.4 million DN can be reached.

LUBRICANT SELECTION

The criteria for a liquid lubricant to function in a rolling-element bearing are that (a) it be thermally and oxidatively stable at the maximum bearing operating temperature, and (b) it form an elastohydrodynamic (EHD) film between the rolling surfaces. The EHD film, which is generally dependent on lubricant base stock and viscosity, is 5 to 100 μin thick at high temperatures. When a sufficiently thick EHD film is present, rolling-element bearings will not usually fail from surface distress. Instead, they fail from rolling-element fatigue which usually manifests itself, in the early stages, as a shallow spall with a diameter about the same as the contact width.

A requirement for long-term high-temperature bearing operation is that the EHD film thickness, h, divided by the composite surface roughness, $(\sigma_1^2 + \sigma_2^2)^{1/2}$, equal 1-1/2 or greater, where σ_1 and σ_2 are the surface finishes of the raceway and rolling elements, respectively. The EHD film thickness is a function of several lubricant and bearing operating variables. However, as a general rule, the minimum viscosity required of a lubricant is 1 cSt at operating temperature. This same research indicated that the ester based lubricants (table 2) meeting the MIL-L-23699 specification could provide the necessary lubrication requirements to 425 °F in an air environment. While other base stock lubricants could give satisfactory operation to 600 °F, they were precluded from further consideration because of their cost and/or commercial availability. Further, at temperatures above approximately 450 °F a low oxygen environment would be required to minimize lubricant oxidation for most lubricant types.

CONCLUSION

The speed capability of rolling-element bearings has increased from speeds of less than 2 million DN to speeds of 3 million DN. The life and reliability of these bearings have also increased where they are equal to, or greater than, those of bearings with limited speed capability. However, high-speed bearings are not "off-the-shelf" bearings which can be readily ordered from a manufacturer's catalog. Design parameters must be carefully chosen and optimized based upon sophisticated bearing computer programs. Material and lubricant selection must be integrated into the bearing design. Bearing thermal management must be implemented through proper lubrication and cooling.

BIBLIOGRAPHY

Bamberger, E.N., Averbach, B.E., and Pearson, P.K., Improved Fracture Toughness Bearings, AFWAL-TR-84-2103, Jan. 1985.

Bamberger, E.N., Zaretsky, E.V., and Signer, H., Endurance and Failure Characteristic of Main-Shaft Jet Engine Bearing at $3x10^6$ DN, J. Lubr. Technol., Vol. 98, No. 4, Oct. 1976, pp. 580-585.

Bamberger, E.N. et al., Life Adjustment Factors for Ball and Roller Bearings - An Engineering Design Guide, ASME, N.Y., 1971.

Fischer, G.K., ed., Advanced Power Transmission Technology, NASA CP-2210, 1983, pp. 193-269.

Greby, D.F., What Turbine Technology is Teaching us About High-Speed Roller Bearings, Mach. Des., Vol. 42, No. 11, Apr. 20, 1970, pp. 229-234.

Hamrock, B.J., and Anderson, W.J., Rolling-Element Bearings, NASA RP-1105, 1983.

Harris, T.A., Rolling Bearing Analysis, 2nd ed., John Wiley & Sons, N.Y., 1984.

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.

Schuller, F.T., Operating Characteristics of a Three-Piece-Inner-Ring Large-Bore Roller Bearing to Speeds of 3 Million DN, NASA TP-2355, 1984.

Zaretsky, E.V., Schuller, F.T., and Coe. H.H., Lubrication and Performance of High-Speed Rolling-Element Bearings, ASLE Preprint 85-AM-3F-2, May 1985.

TABLE 1. - COMPUTER OMTIMIZED DESIGN FOR HIGH-SPEED TAPERED-ROLLER BEARING

Dimension	
Cup half angle	15°53'
Roller half angle	1°35'
Roller large end diameter, mm (in)	18.29 (0.720)
Number of rollers	23
Total roller length, mm (in)	34.18 (1.3456)
Pitch diameter, mm (in)	155.1 (6.105)
Bearing outside diameter, mm (in)	190.5 (7.500)
Roller crown radius, mm (in)	25.4×10^3 (1000)
Roller spherical end radius, percent of apex length	80

TABLE 2. - PROPERTIES OF TETRAESTER LUBRICANT

Additives	Antiwear, oxidation inhibitor, antifoam
Kinematic viscosity, cS, at:	
311 K (100 °F)	28.5
372 K (210 °F)	5.22
477 K (400 °F)	1.31
Flash point, K (°F)	533 (500)
Autoignition temperature, K (°F)	694 (800)
Pour point, K (°F)	214 (-75)
Volatility (6.5 h at 477 K (400 °F)), wt %	3.2
Specific heat at 477 K (400 °F), J/(kg)(K)(Btu/(lb)(°F))	2340 (0.54)
Thermal conductivity at 477 K (400 °F), J/(m)(s)(K) (Btu/(h)(ft)(°F)	0.13 (0.075)
Specific gravity at 477 K (400 °F)	0.850

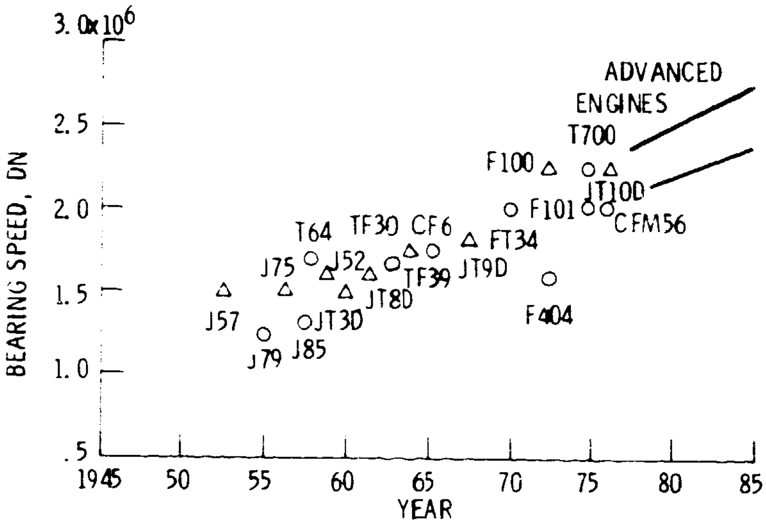


Figure 1. - Increased engine speeds required higher bearing speeds.

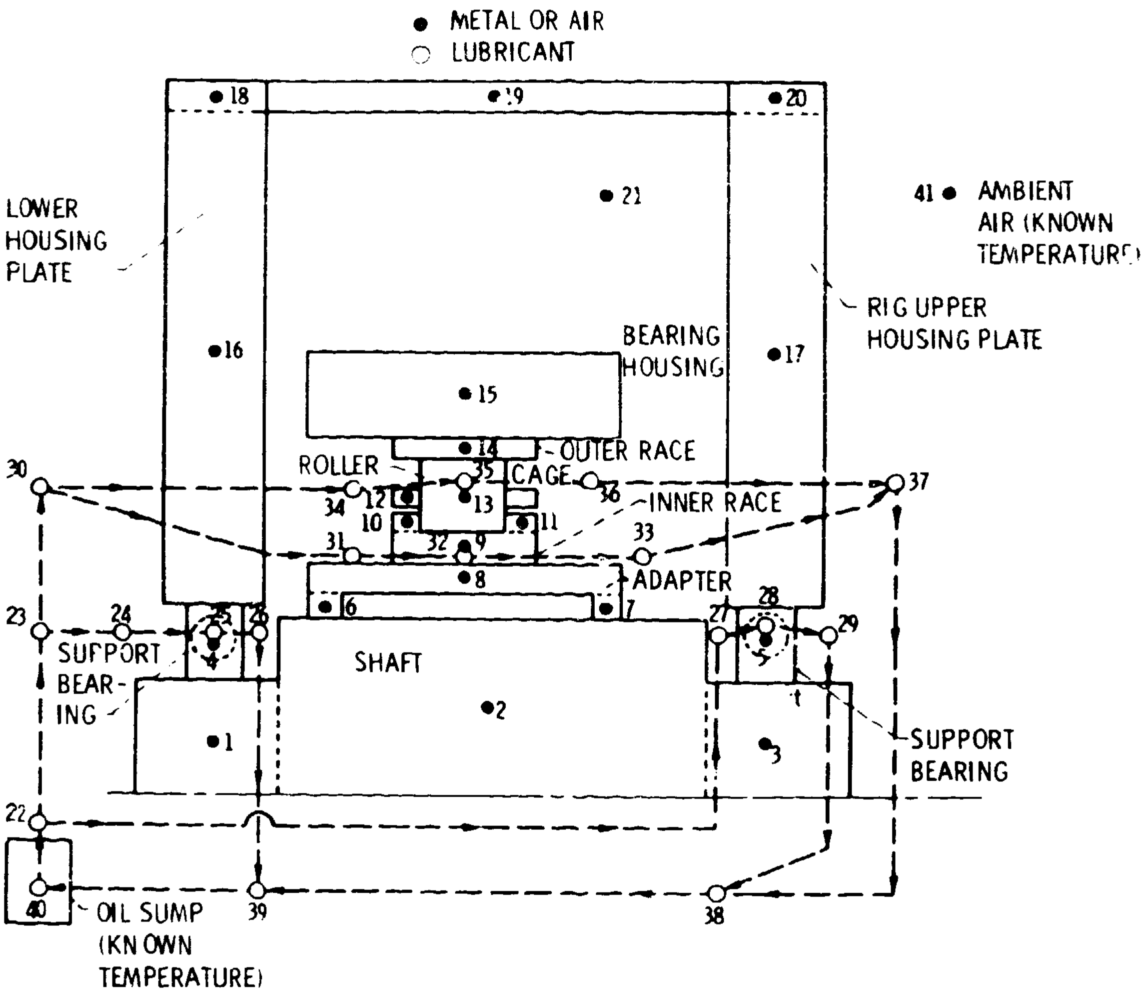
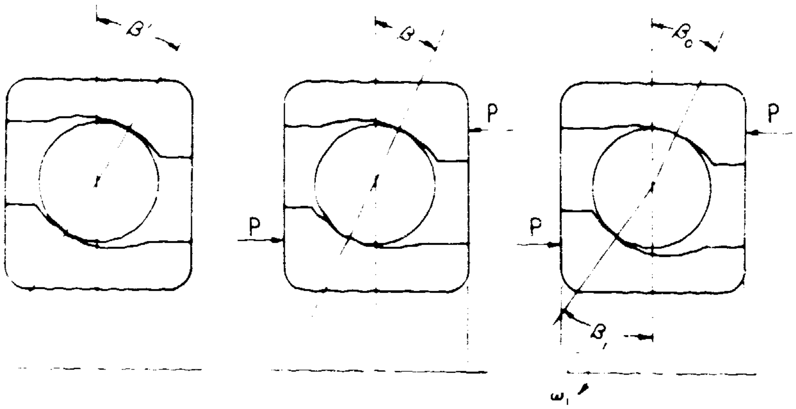


Figure 2. - Nodal system used for thermal routines in Cybean.

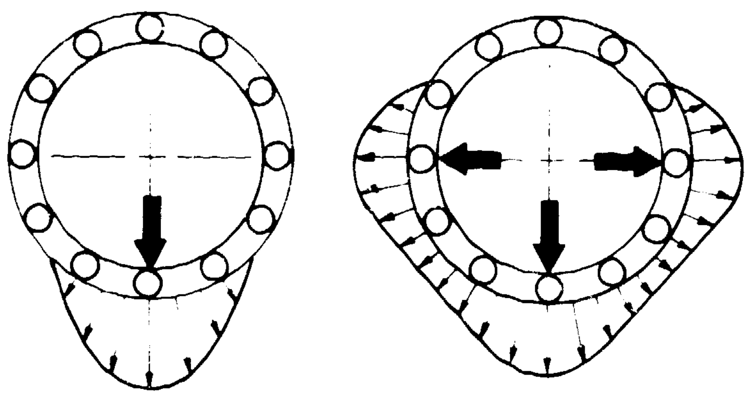


(a) Free-contact angle (no load).

(b) Contact angle when under load.

(c) Contact angles when under load and at high speed.

Figure 3. - Changes in contact angle with load and speed.



(a) Radial load.

(b) Radial load and two-point preload.

Figure 4. - Roller bearing load distribution for simple radial load and with two-point preload.

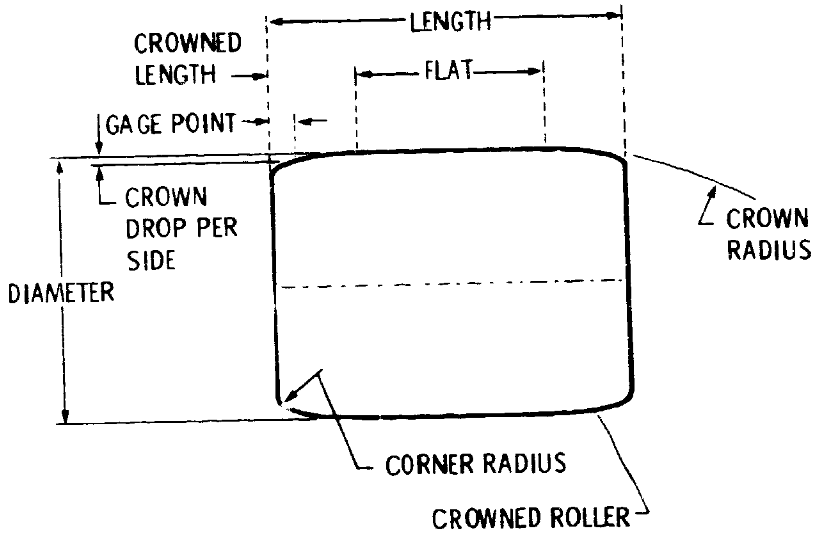


Figure 5. - Crowned roller for high-speed roller bearing application.

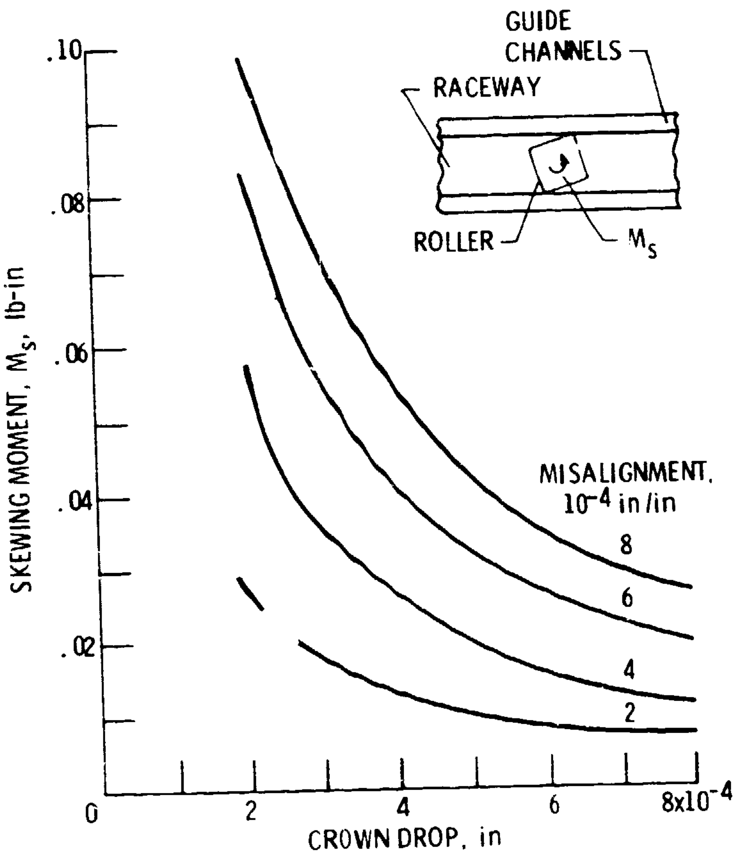


Figure 6. - Effect of crown drop on skewing moment for various amounts of misalignment.

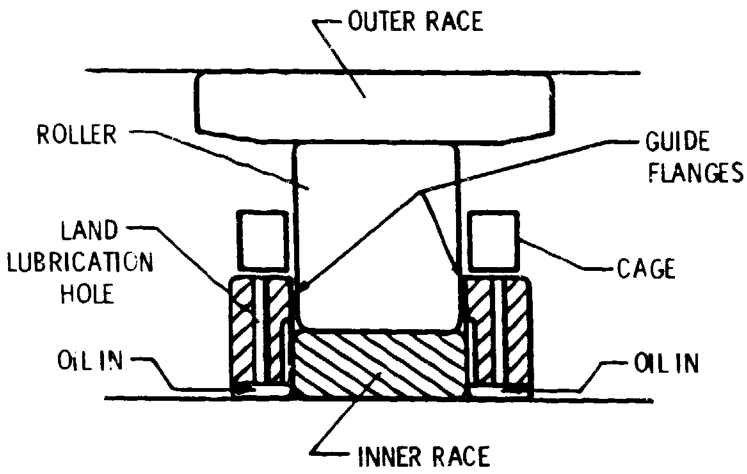
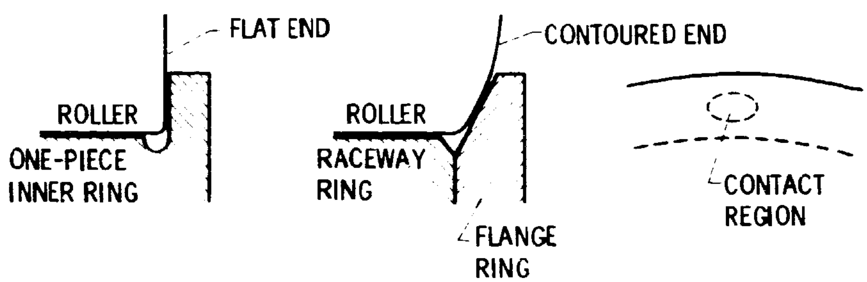


Figure 7. - Three piece inner-race roller bearing.



(a) Conventional, one-piece inner race

(b) Modified configuration, three-piece inner race.

Figure 8. - Roller end-contact geometry of one- and three-piece inner-race bearing.

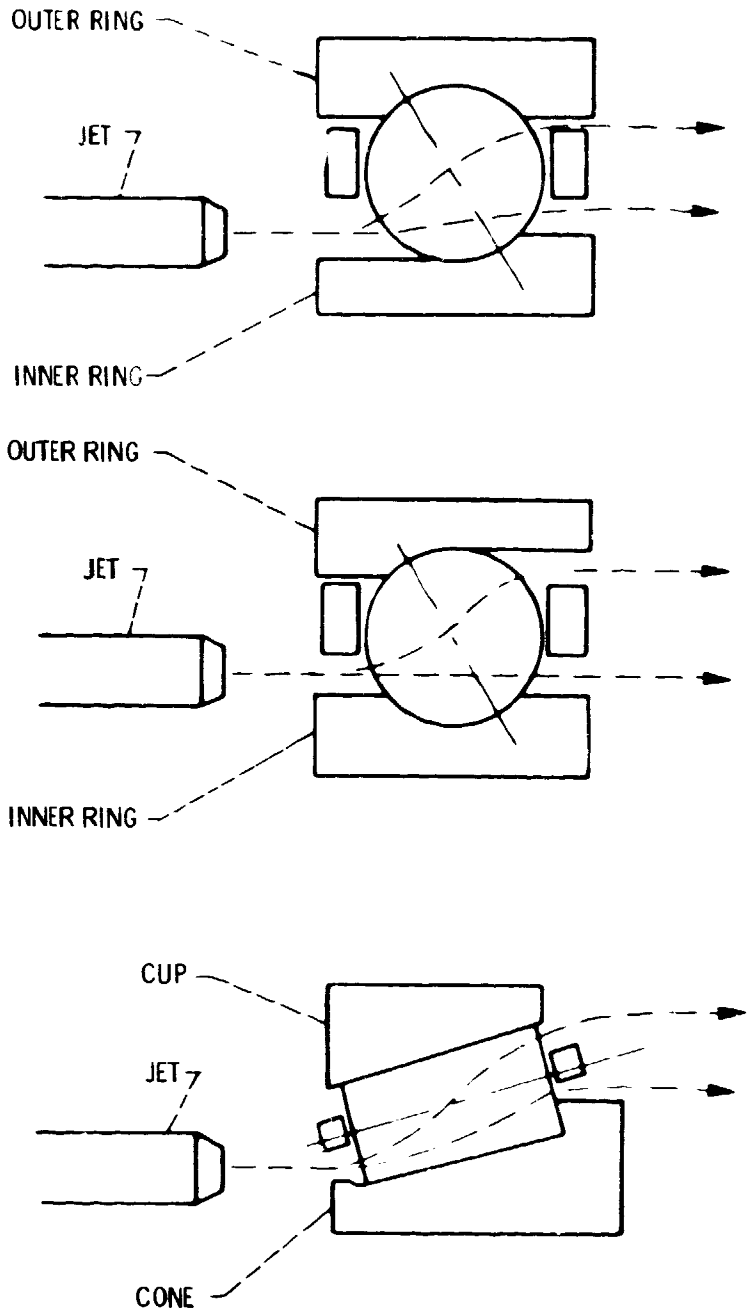
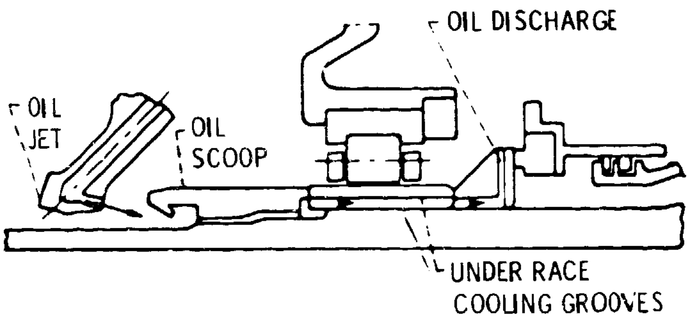
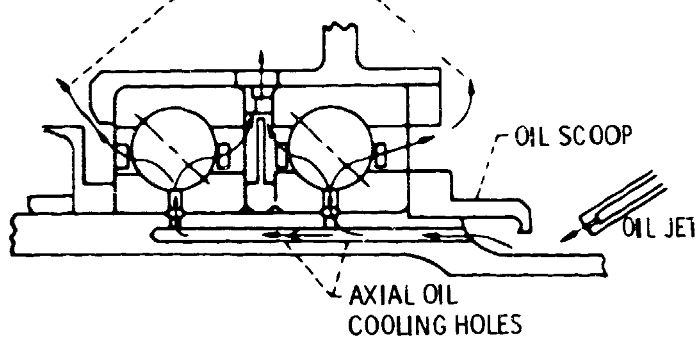


Figure 9. - Placement of jets for ball bearings with relieved rings and tapered-roller bearings.



(a) Cylindrical roller bearing.



(b) Ball thrust bearing.

Figure 10. - Under-race lubrication system for main shaft bearings on turbofan engine.

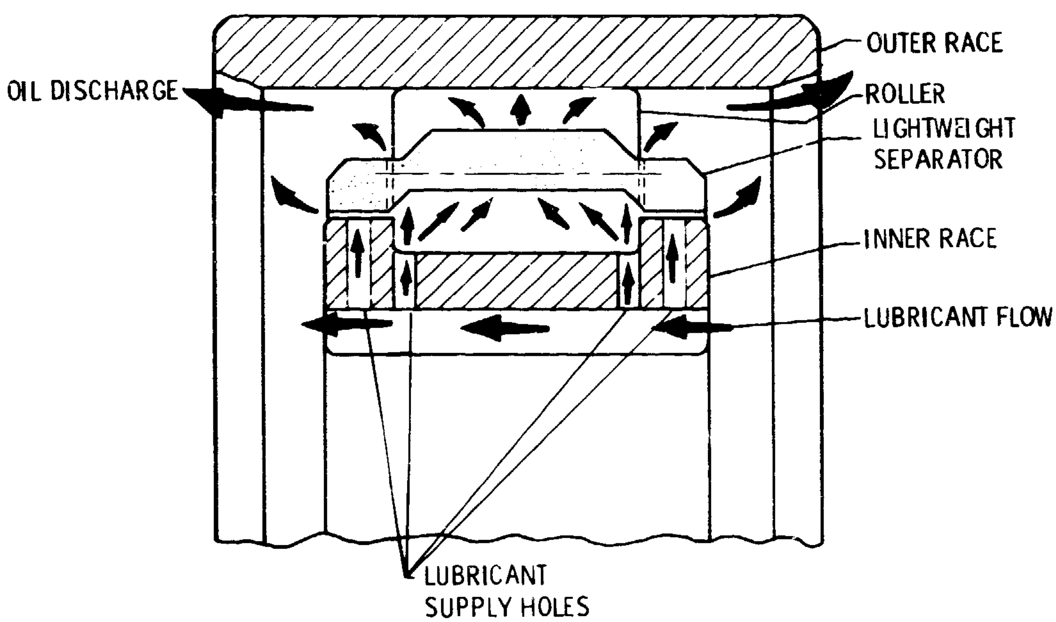
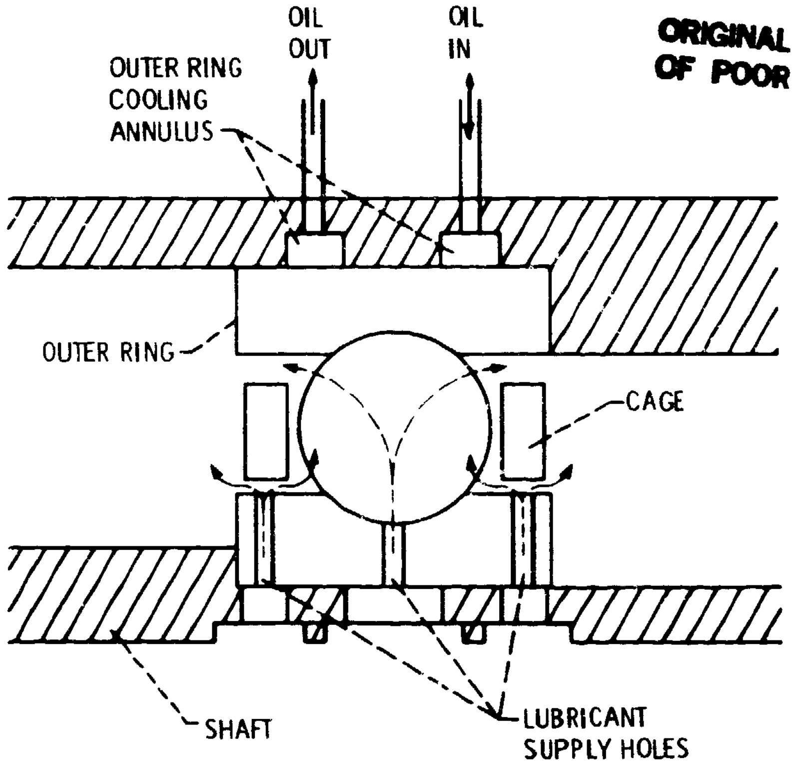
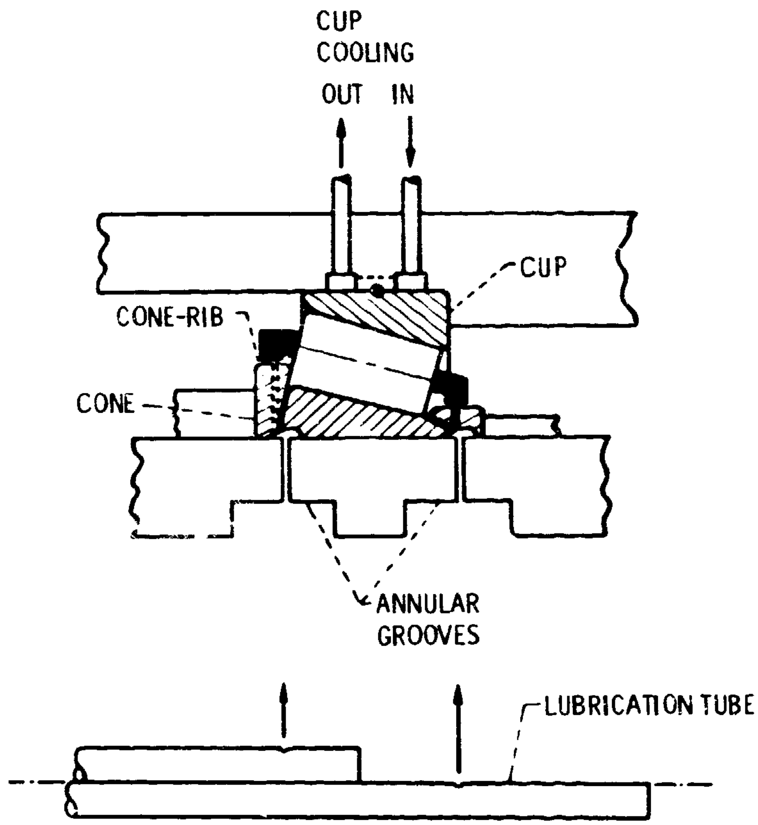


Figure 11. - High-speed roller bearing with under-race lubrication and cooling.

ORIGINAL PAGE IS OF POOR QUALITY



(a) Angular-contact ball bearing.



(b) Computer optimized tapered-roller bearing.

Figure 12. - Outer-race cooling with under-race lubrication for high-speed bearings.

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM-87107		

4. Title and Subtitle	5. Report Date
Design and Lubrication of High-Speed Rolling-Element Bearings	
	6. Performing Organization Code 505-33-7C
7. Author(s) Erwin V. Zaretsky	8. Performing Organization Report No. E-2670
	10. Work Unit No.
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135	11. Contract or Grant No.
	13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546	14. Sponsoring Agency Code

15. Supplementary Notes

Prepared for the Original Equipment Manufacturing Design Conference, Philadelphia, Pennsylvania, September 9-11, 1985

16. Abstract

The speed capability of rolling-element bearings has increased from speeds of less than two million DN to speeds of three million DN. The life and reliability of these bearings have also increased where they are equal to, or greater than, those of bearings with limited speed capability. However, high-speed bearings are not "off-the-shelf" bearings which can be readily ordered from a manufacturer's catalog. Design parameters must be carefully chosen and optimized based upon sophisticated bearing computer programs. Material and lubricant selection must be integrated into the bearing design. Bearing thermal management must be implemented through proper lubrication and cooling. Parameters which can be used to design, specify, and lubricate high-speed bearings are presented and discussed.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Rolling-element bearings High-speed bearings Lubrication	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of pages	22. Price*
Unclassified	Unclassified		

*For sale by the National Technical Information Service, Springfield, Virginia 22161

END

DATE

FILMED

NOV 6 1985