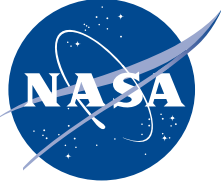


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Design Guide for Bearings Used in Cryogenic Turbopumps and Test Rigs

*H.G. Gibson
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December 2019

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Space Administration

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Disclaimer

This guide is prepared and presented as an aid to the designer or engineer working with cryogenic bearings. This guide has been written based on the experiences of engineers at NASA Marshall Space Flight Center and other referenced materials. Efforts have been made by peer reviews to ensure accuracy and eliminate errors. This is not a definitive design guide on cryogenic bearings. Persons selecting bearings are encouraged to consult the many textbooks, vendor catalogs, and other resources available. The use of companies, products, or persons' names does not imply the recommendation of that service or product over any other.

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LIST OF ACRONYMS AND DESIGNATORS

ABEC	Annular Bearing Engineering Committee
ADORE	Advanced Dynamics of Rolling Element Bearings, a bearing software analysis program developed by Dr. P. Gupta, Inc.
AFBMA	Anti-Friction Bearing Manufacturers Association
AISI	American Iron and Steel Institute
ASTM	American Society of Testing and Materials
CVD	chemical vapor deposition
DN	the product of bearing bore in millimeters times the rpm of the inner race
ESD	electrostatic discharge
FEP	fluoroethylene-perfluorpropylene
Fi	inner race designator
Fo	outer race designator
GOX	gaseous oxygen
ID	inside diameter
LH ₂	liquid hydrogen
LN ₂	liquid nitrogen
LOX	liquid oxygen
MoS ₂	molydisulfide
MSFC	Marshall Space Flight Center

LIST OF ACRONYMS AND DESIGNATORS (Continued)

OD	outside diameter
PTFE	polytetrafluoroethylene
RBEC	Roller Bearing Engineering Committee
STLE	Society of Testing and Lubrication Engineers

TECHNICAL PUBLICATION

DESIGN GUIDE FOR BEARINGS USED IN CRYOGENIC TURBOPUMPS AND TEST RIGS

1. INTRODUCTION

Cryogenic bearings are a unique and specialized area of the overall group of bearings that are used every day in industrial and aerospace applications. Cryogenic bearings operate in a unique environment that is not experienced by most bearing applications. The high speeds of turbomachinery, flow of cryogenic coolants, use of nonstandard materials, and lack of lubrication place unique demands on cryogenic bearings that must be met for the safety and success of the mission.

To meet the goals of safety and success, requirements are put on the designer, manufacturer, and user that are not normally applied to off-the-shelf bearings. The designer has to have knowledge of the operating conditions, rotational speeds, loads, stresses, installation methods, inspection criteria, dimensional requirements, and design and analytical tools. The manufacturer needs to be aware of the materials used for cryogenic bearings, special heat treatments required, cleanliness of the processes, and inspection techniques to ensure a good product. The user needs to be aware of the safe handling practices to eliminate corrosion and debris, correct installation and removal procedures, pre- and post-test inspections, and the documentation that follow the bearings.

This guide is based on the experiences of engineers at NASA Marshall Space Flight Center (MSFC) that have been involved in bearing research and testing along with specific bearing references that have been written. It is not meant to be a bearing design textbook for cryogenic bearing applications. These are available from many authors. Its purpose is to help the designer, manufacturer, or user in the application of cryogenic bearings to better understand the requirements placed on these bearings.

2. BACKGROUND

In the late 1970s, the Space Shuttle's turbopump bearings were not meeting the design requirements that were expected. A bearing tester was designed and built at MSFC that would use the flight bearings with cryogenic coolants and duplicating operating conditions as closely as possible. The tester was called the Bearing and Seal Material Tester. Testing began in the early 1980s, and continued with this rig until 1993. Some 25 separate builds of this rig were tested and run in either liquid nitrogen (LN₂) or liquid oxygen (LOX). Advanced bearing materials were run, new lubricating schemes were investigated, and internal geometries were changed. In 1995, a new test rig was designed that was capable of testing ball, roller, and hydrostatic bearings in either LOX or liquid hydrogen (LH₂). This tester was named the LH₂ Test Rig. To date, this rig has been assembled and run five times with combinations of ball and roller bearings with a hydrostatic bearing in one position. LH₂ has been the coolant throughout this test program.

Papers have been prepared and presented at conferences on the bearing work done at MSFC over the years. It is from this work and selected references that this design guide is prepared.

3. MATERIALS USED FOR BEARINGS

3.1 Rings

Materials selected for bearing inner and outer rings must be hardenable, machinable, wear resistant, and have the required fatigue strength and life to ensure successful operation of the rotating components. Cryogenic operation adds the requirements of resistance to stress corrosion cracking, dimensional stability at cryogenic temperatures, adhesion resistance, the ability to operate with a minimum of lubrication, and material compatibility with the coolant. The material normally selected for cryogenic use has been the American Iron and Steel Institute (AISI) 440C corrosion-resistant stainless steel in forged bar form. The process used to make 440C is vacuum induction melt with a vacuum arc remelt to ensure cleanliness of the steel. This material has shown that it has all the properties to operate successfully in high-speed turbopump environments in testing and turbopump applications. AISI specifications are available for 440C stainless steel and can be referenced as needed.

Another material that has been used successfully in bearings is Cronidur 30®. This is a corrosion-resistant steel developed by FAG Aerospace. It has lower carbon content than 440C and nitrogen has been added to the steel to improve its properties. It is reported to have better stress corrosion cracking properties and corrosion resistance than 440C. A greater fatigue life with higher Hertzian contact pressures is reported to be an additional benefit. Bearings from this material have been run successfully in LH₂ in the LH₂ Test Rig at MSFC and in turbopumps. This material is relatively new in the bearing field and proven results make it a candidate in the cryogenic bearing materials area.

Three other materials that can be considered for use in cryogenic bearings are BG42® (Timken Latrobe Steel), CRB-7®, and MRC®2001 (SKF Industries) powder steel. These materials have reported favorable characteristics, but have not been fully tested at MSFC. These materials are not as widely used as 440C and may be hard to find. More testing is needed before they can be recommended. For noncryogenic bearing applications, bearing materials normally used such as M-50 and 52100 can be used if they meet the operational requirements. An exception to this is hydrogen peroxide coolants, which is seeing some use as an oxidizer. Material compatibility with hydrogen peroxide is a big concern and must be verified by testing. The engineer needs to consult hydrogen peroxide guidelines and material compatibilities to become knowledgeable with necessary requirements.

The designer or engineer needs to know the end use of the bearings to decide the necessary documentation, testing, and inspection requirements for the processes. Bearing materials for use in manned flight applications must be highly documented and traced. Bearings for testing or nonflight applications may not need a detailed paper trail. For flight use, a chemical analysis along with a hardenability test, macrostructure exam, grain size microscopic exam, and an inclusion rating may be required on the raw stock. In addition, nondestructive tests for cracks and internal defects may be required. After the raw material is formed, it is assigned a batch or lot number and this number stays

with the material through final manufacture. In nonflight use, a certificate of conformity on the material may be substituted, nondestructive inspection results provided, and dimensional checks recorded. Nondestructive evaluation inspections of the finished parts should be at least a $\times 10$ visual inspection and a fluorescent penetrant exam. Eddy current and X-ray inspections can also be required depending on the application.

3.2 Balls

Balls are normally made of the same material as the rings. Some bearing companies buy balls from vendors that specialize in making very dimensionally accurate and precise balls. Balls of 440C steel are common and available in many sizes. Balls must meet certain Anti-Friction Bearing Manufacturers Association (AFBMA) and Annular Bearing Engineering Committee (ABEC) requirements for size and tolerances. The balls used in aerospace applications need to be grade 10 or better. It is important to match balls in a bearing by diameter. Balls of differing diameters will have unmatched rotational speeds and will cause detrimental loads on the cage. AFBMA and ABEC organizations have standards for matching balls in a batch or container as well as standards for individual balls. For example, the standard for grade 10 is an out-of-roundness limit of zero to 10 μin per ball and a ball-to-ball variation limit of up to 20 μin .

During the heat treatment of balls, a witness sample of the same material is put in the furnace and checked for hardness after the balls are removed. This provides a flat surface for the indenter to penetrate. Balls can have a one to two point higher hardness than rings due to the smaller mass of the ball. Inspection requirements on the balls should follow the AFBMA/ABEC guidelines.

3.2.1 Silicon Nitride Balls

Silicon nitride is a powdered ceramic material that can be processed into rolling element form, either balls or cylindrical rollers. Silicon nitride was available in the 1970s but did not see use as a rolling element material due to inadequate performance. Advanced technology and manufacturing techniques has brought silicon nitride back as a consideration for bearing rolling elements. Silicon nitride balls are now becoming an off-the-shelf option and are being called 'hybrid bearings,' meaning they have steel rings and silicon nitride balls. Silicon nitride rolling elements have mechanical properties that are superior to steel. They have more wear resistance, lighter weight, give less thermal dimensional changes, and are more heat resistant. These properties make them an excellent choice where lubrication is marginal and rotational speeds are high. The American Society of Testing and Materials (ASTM) specification F 2094-01 covers the quality requirements, physical properties, mechanical properties, inspection requirements, and testing of silicon nitride used in bearing balls.

When silicon nitride balls are installed into a bearing, the ball-to-ring contact stresses increase due to the higher modulus of elasticity of the silicon nitride. Bearing analysis is necessary to investigate Hertzian stresses and the effect on fatigue life. A change in the curvature of the races to a more conforming radius can reduce the stresses but causes an increase in the heat generated by rolling friction. Because silicon nitride has better thermal properties than steel, this option is normally used.

MSFC has tested hybrid bearings with silicon nitride balls in two of their test rigs. Silicon nitride balls were installed in turbopump bearings and evaluated in LN₂, LOX, and LH₂. An in-house turbopump program named the Simplex Turbopump also used hybrid bearings. In all cases, an anomaly has been observed on the balls that has been called 'river marking.' The bearings were run at 30,000 rpm with approximately 400,000 psi contact stress. After an extended period of run time, indications were observed with a borescope on the surface of the balls. These indications did not have the common characteristics of cracks, i.e., narrow width with greater depth, but were shallow with width greater than depth. These marks increased in length as run time was accumulated, with smaller 'tributaries' branching off, running on the surface of the balls, hence the name river marks. No detailed engineering explanation has been given. It appears it could be a thermal and contact stress effect. More time was accumulated on these balls and balls with river marks pretest have been installed into the test rig and evaluated. This was done to investigate and prove the reliability of the silicon nitride material with river marks. No failures have been seen in any of the testing to date. No other literature on hybrid bearings has referenced these marks on silicon nitride balls. It appears that this is unique to the cryogenic environment that the balls operate in.

These surface indications have also been theorized to be from the Lichtenberg effect seen in insulating materials from high electrical discharges on the surface. Modern Lichtenberg figures have been created within solid insulating materials such as acrylic and glass. These figures and shapes are similar to those seen on the tested silicon nitride rolling elements (fig. 1).

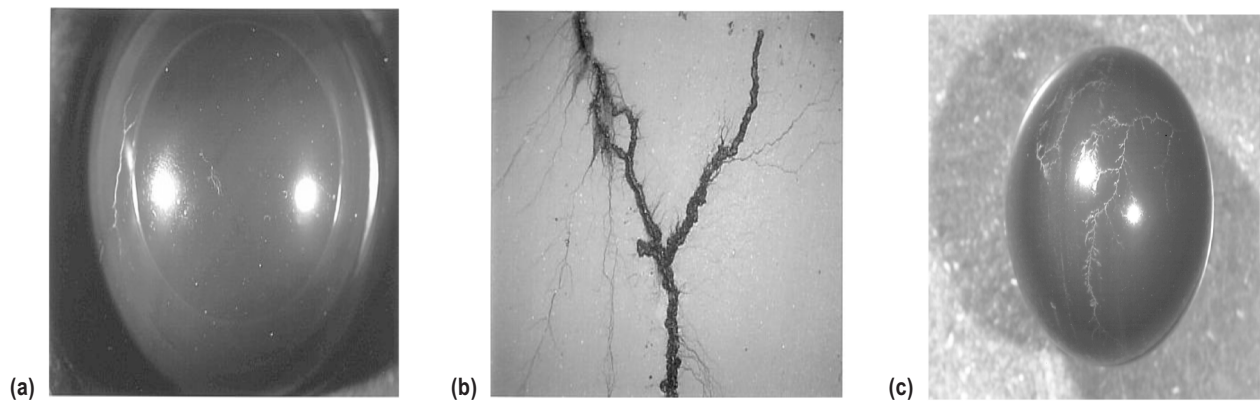


Figure 1. River marks: (a) on left side of ball, (b) closeup, and (c) severely marked silicon nitride ball.

Another concern of using silicon nitride balls in test rigs or turbopumps is electrostatic discharge (ESD). Silicon nitride is not electrically conductive. A rotor can become isolated from ground and an electric charge can build up on the rotor. This charge will jump across small gaps to another surface in test rigs or turbopumps creating small pits. ESD pitting has been seen in test rigs and turbopumps running hybrid bearings. Two theories of the cause of the buildup of static charges are: (1) the friction of the nonconductive balls rubbing a nonmetallic cage and (2) the flow of coolant in

the pump causes the static charge on the rotor. Designs for grounding the rotor to the housing have to be considered either through face seals, brushes, or some other means of contact. In LN_2 , ESD may not be a problem except for pitting the hardware, but in LOX, LH_2 , or other rocket propellants, this may be a major concern.

Even with the above concerns of silicon nitride balls, testing and turbopump use has shown the material to be superior to 440C balls. Silicon nitride rollers have also been tested and run in turbopumps with similar results. Hybrid bearings are now being offered as a standard bearing from many major bearing manufacturers.

3.3 Cage Materials

Bearings used in cryogenic applications are not lubricated by oil or grease. The cryogenic fluid is directed through the bearings to remove the heat that is generated. A transfer film from the cage to the ball sometimes provides lubrication that the bearing uses. Low friction coatings can be applied to the rings, balls, and cage; however, this coating is only good for startup of the bearing.

The standard cage material used for cryogenic service has been a fiberglass cloth coated with a thin layer of virgin, unplasticized, polytetrafluoroethylene (PTFE) polymer. This cloth is wound around a mandrel to give it an outside diameter (OD) and inside diameter (ID). These cylinders are then inspected for cleanliness, presence of voids or gas bubbles, tightness of the layers, and peel strength. Armalon® (American Durafilm) is one such product. The cloth has a Military Spec of MIL-Y-1140.

After the cloth cylinders are fabricated, they must be machined into cages. The ball pockets, IDs, ODs, and widths are machined to size. This process is done with no coolant to avoid contaminating the material. Machine feeds and speeds must be developed to cut the material and not damage the integrity of the cage.

A disadvantage of the Armalon cage material is the ends of the glass fibers rubbing on the balls in the pockets generating heat and wear. One solution to this has been the application of a sprayed-on or sputtered-on dry film lubricant in the cage pockets and the steel balls. The entire surface of the cage can also be coated to reduce any rubbing friction between the rings and cage. This coating will reduce friction during startup of rotation, but wears away quickly.

Another low-friction coating, tetra fluorethylene-perfluorpropylene (FEP), was tested at MSFC to evaluate its ability to lower the friction between the cage and rolling elements. This material was found to be unacceptable in cryogenic applications. The coating bubbled up from the surface of the cage during exposure to LN_2 in the test rig (fig. 2).

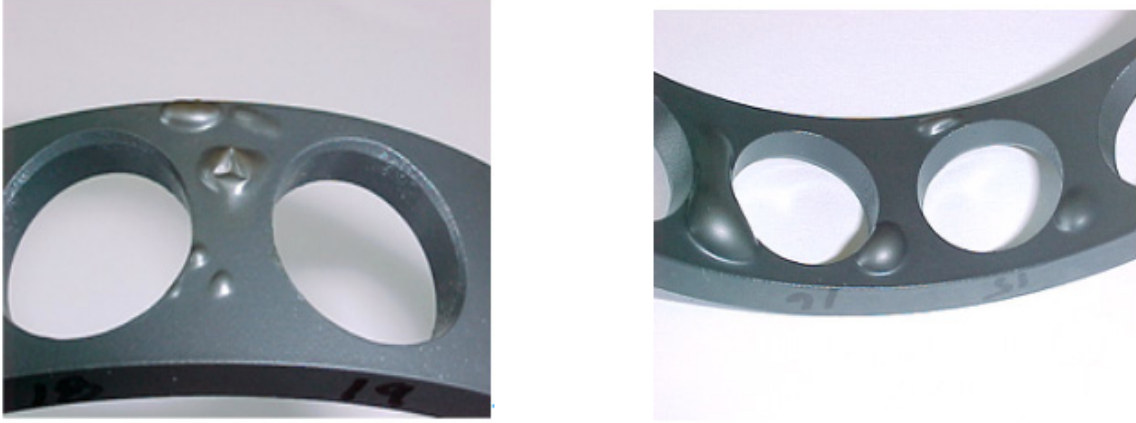


Figure 2. Bubbling of FEP coating on bearing cage after exposure to LN_2 .

Another material that has been tested and used successfully is PTFE filled with bronze powder. Salox® (Allegheny Plastics) is the product of this design that was tested. The bronze leaves a layer of soft metal on the balls and in the wear track as it transfers to the bearing. There are no fiberglass fibers in this material. This material is also available with molydisulfide (MoS_2), but bearing tests at MSFC showed no improvement with the MoS_2 additive. This bronze/PTFE material has to be supported in a metal housing or with the Armalon material. The bronze/PTFE material is not strong enough to be made into a bearing cage alone. Inserts of this material held in metallic bronze cages and inserts in Armalon supporting cages have been tested at MSFC. The retaining of the inserts in the cages needs to be designed carefully. PTFE materials will cold flow to relieve any installation stress. Ball pocket clearances can be reduced and the inserts can loosen in their cages from cold flow. Dimensional checks of the ball pockets should be done after the inserts have set in the cage for some time.

Carbon fiber cages were also evaluated. The strength of this material was not adequate and the manufacturing was not acceptable. The cages buckled with low force and the carbon layers did not stay adhered to one another. These cages were never installed into bearings and tested at MSFC.

If other materials similar to Armalon or Salox are considered, specifications exist for testing and verification of the candidate material. Federal spec FED-STD-406, "Plastics, Methods of Testing," ASTM Designation D790-92, "Standard Test Methods for Flexural Properties of Unreinforced and Reinforced Plastics," and ASTM Test Method D638, "Test Methods for Tensile Properties of Plastics" can be used to determine if the material is a candidate for bearing cages. Material compatibility with the coolant (i.e., LOX) must be verified also using test methods for the contact media used.

3.4 Liquid/Gaseous Oxygen Impact Testing

For use in LOX or gaseous oxygen (GOX), all bearing materials need to be LOX or GOX impact tested in that environment, including the materials for the races and balls, cage materials, and any lubricants used. NASA Standard 6001, test 13, provides a detailed description of what is required.

A synopsis of the method is given: vendors must provide 20 samples of the material approximately the size of a quarter from one batch of the material. The samples are placed in a small cup of LOX or GOX and a weight is dropped from a standard height on each sample. Twenty tests are run. If there is a reaction, 40 more samples are required from the same batch. Zero reactions in 20 or one reaction in 60 will pass. It is important to clean every sample for LOX service before testing. Contaminants can cause failures not related to the material. All materials for testing have to come from the batch or lot of material that will be used for bearing manufacture. This places a requirement on the vendor to make samples and hold the material while testing is completed.

3.5 Wear-Reducing Coatings

Coatings are used for lubrication, wear resistance, and corrosion protection. Coatings can be soft or hard and applied several different ways. Soft coatings flow into the surface imperfections and reduce the local stress levels. Soft lubricant coatings of MoS₂ reduce sliding and rolling friction, and hard coatings such as thin, dense chrome provide wear resistance and corrosion protection.

Soft coatings such as gold and copper will flow into the surface roughness asperities and lower the Hertzian contact stresses. Soft coatings have lower shear strength and will deform instead of the parent metal during rolling stress. MSFC has evaluated coatings using rolling contact fatigue test specimens. Some improvement in wear life has been determined. Work has been done by others with lead films and results of less noise and torque requirements over that of gold and silver have been reported.

Lubricating coatings are applied by spraying, dipping, or sputtering onto the active surfaces of the bearing. MoS₂ is usually the coating used for sputtering on the cage and balls. Hard coatings such as thin, dense chrome or titanium carbide can increase wear life and increase corrosion resistance. Some bearing companies are offering coatings of titanium carbide on the rolling elements as an option on their bearings.

The application processes for applying hard coatings varies. Ion implantation is one technique, but the minute level of surface penetration and cost prohibits this method. Ion plating is a technique that has given good results with titanium carbide on balls. Ion plating has two methods—chemical vapor deposition and physical vapor deposition. Both of these processes produce high temperatures and a reheat treatment is necessary. Hard coatings are brittle and need to be very thin to avoid flaking off. Thicknesses of 40 μm have shown good adhesion on test samples.

The use of coatings should be evaluated carefully. It is preferred to design the bearing with properties and materials so that coatings are not necessary. There are disadvantages that need to be considered before coatings are used. The quality of the application process varies from vendor to vendor and even with the same vendor on subsequent work. As coatings wear, debris is generated that may be degrading not only to the bearings but to the entire system. Paperwork, inspections, handling, and transportation issues may be time consuming. The additional processing may drive the cost to an unacceptable level and delivery time can become a factor. Experience needs to be gained and documented on the effectiveness of coatings before they are recommended.

4. BEARING DESIGN

Bearings are a major component in high-speed turbopumps and test rigs; their design cannot be taken lightly. The success or failure of the system can depend on the operation of the bearings. Bearings must provide axial and radial load support, give rotor-dynamic stability, prevent rubbing of shaft components, have sufficient fatigue life, allow transient movement, and align the rotating components, all without the benefit of lubrication. Many books and computer modeling programs have been written on bearing design. In many cases, the computer programs have been verified and improved by bearing testing. Because bearings can limit the speed of turbopumps, new designs and better materials are always being investigated. Bearing problems can better be prevented than solved. The designer needs to be involved with the upfront design and planning of the turbopump or test rig to better understand the requirements needed for successful design and operation of the bearings. Major bearing components are shown in figure 3.

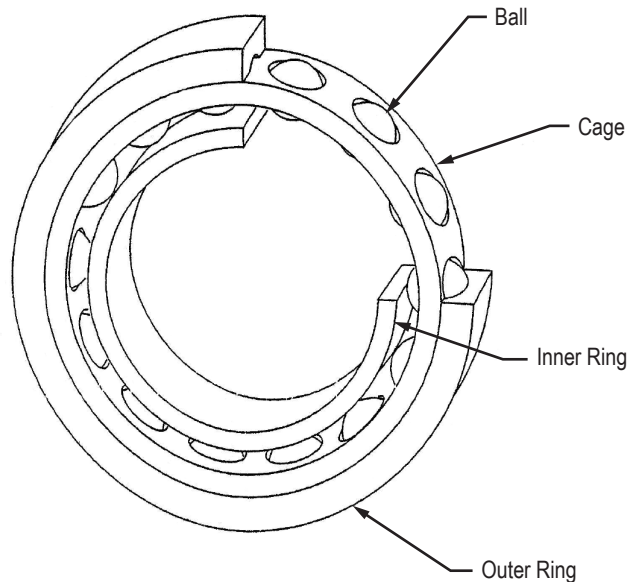


Figure 3. Bearing component hardware with commonly used names.

4.1 Rolling Element Bearings

The most basic rolling element design calculation is the DN value. This is the diameter of the bearing in millimeters times the rotational speed in revolutions per minute. This value only gets an engineer or designer into a type of bearing based on DN limits. DN values recommended for rolling element bearings are in the 1 to 2 million range. Above this value, centrifugal effects begin to be a limiting factor.

The direction of loading is also a factor in selecting the type of bearing. Conrad bearings can take axial loading in both directions and radial loads but are not widely used in high-speed applications. Angular contact bearings are the recommended bearing type for high-speed axial and radial loading applications. These bearings are separable which allows substitution of the ball cages for cryogenic use. This feature also allows inspection of the surfaces both pre- and post-operation.

Angular contact bearings (fig. 4) can support axial loads in only one direction, so they are used in pairs with the contact angles opposed to each other. The axial load needs to be low enough so that the balls do not make contact with the upper shoulder edge. The upper edge is normally relieved to prevent high stresses if this occurs. Preload springs are used to take up diametral clearance, set the contact angle, and prevent skidding of the balls. The springs also allow movement of the shaft without unloading or overloading the bearings. Split inner ring bearings allow thrust loading in two directions. A preload spring is also used with these bearings. If axial thrust loading is determined to be over the capacity of the bearings, a balance cavity or piston can be designed into the turbopump rotor to balance the forces on the bearing.



Figure 4. Angular contact bearing with silicon nitride rollers.

The major elements in rolling element bearings are inner and outer races, internal clearances, curvatures, surface finishes, stiffness, coolant flowpath, cage design, and rolling elements. Each one is addressed below.

- Inner race—The preferred diameter of inner race ID should be the smallest diameter that will meet the shaft load, stiffness, strength, and bearing capacity requirements. This diameter is used in the

DN calculations and should be kept as small as possible. The bore interference fit should not be so tight as to create high hoop stresses on the ring when installed on the shaft. No splines, keyways, or retaining notches should be put on the shaft under the bore. The sharp edges of the inner ring bore are chamfered to help assembly onto a shaft. To ensure flatness of the inner ring against the shaft shoulder, the radius of the shoulder should be smaller than the radius of the ring bore. The bearing shoulder height should be sized so that the contact ellipse is contained in the raceway under all loading conditions. To prevent contact of the ball with the sharp upper shoulder under excessive loading, the upper edge is relieved.

- Outer race—Misalignment between the outer and inner race can cause ball wobble in the contact zone and high cage loads due to ball speed variations. Misalignment can come from manufacturing inaccuracies, mounting errors, support housing elasticity, and moments due to loads. Since the outer race is normally installed in the housing, the following guidelines have been set up to reduce misalignment:
 - Mounting surface dimensions should be consistent with the precision of the bearing used, AFBMA/ABEC 5, RBEC 5, ABEC 7, etc.
 - A taper tolerance of 50 $\mu\text{in/in}$ for interference fits in housing bores.
 - Surface finishes of 16 $\mu\text{in/in}$ AA in housing bores.
 - Fits should be analyzed at operating temperature to give retention but not to fracture any component.

In a positive, nonrotating ring application, anti-rotation tabs can be designed into the outer race to provide positive, nonrotation of the outer ring. This prevents fretting of the outer ring in the housing.

- Internal clearance—Internal clearance is a function of the outer and inner ring curvatures and ball diameter. Internal clearance sets the contact angle and the offset distance between the inner and outer race faces, sometimes called axial deflection or stick out of the bearing at assembly. The unfitted room-temperature clearance value has to be calculated with operating conditions in mind. All of these elements affect the internal clearance: interference fit of the inner and outer ring; centrifugal growth due to speed, thermal expansion, or contraction; and thickness of any coatings used. This clearance should never be negative for any case. Negative clearances will cause excessive heat generation and wear, resulting in a seized bearing.

Cylindrical roller bearings can have a slightly negative to zero clearance. This is to ensure that the rollers do not skew or skid under high-speed, lightly-loaded conditions. Coolant must be supplied to prevent excessive heat generation between the rollers and races.

- Cross race curvature—This is defined as the ratio of the race curvature radius to the ball diameter, labeled F_o for outer and F_i for inner races. This ratio can be different for the two races. Because of the race curvatures of an angular contact bearing, the balls experience differential speed, which

results in microscopic skidding and sliding. If the curvature is too conforming to the ball diameter, excess heat generation and skidding can develop; however, this will give the minimum stress and highest capacity. If the curvature is too open, high contact stresses will develop, lowering the fatigue life. In some extremely high-speed bearings, F_o is made less than F_i to reduce heat generation at the inner race. For inner race control (ball rolls at inner race), F_i is made less than F_o . This gives lower contact stresses at the inner race, increasing the fatigue life.

Computer analysis should be done using a program that can output heat generation in the contact zones. A test matrix can be set up comparing race curvatures to heat generation to fatigue life. The designer can select the best combination to give the desired results.

- Surface finish—Surface finishes for satisfactory performance can be met with today's machine shop grinding techniques. A fine surface finish promotes smooth bearing operation, increases fatigue life, and aids in the formation of transfer films. Circumferential surface finishes are in the range of 6 to 8 AA. Other noncontact surfaces do not need this fine of a finish. Lesser values of 16 to 64 AA can be used on noncontact surfaces. The finish in the axial and circumferential directions can be controlled by good grinding techniques, including feed rates, wheel selection, and coolant flow rates. This will also help to eliminate grinding burns that can reduce fatigue life and cause early failures.
- Radial stiffness—Turbopump shafts are supported at some point along their length by bearings. Rolling element bearings have a major influence on rotordynamic response of the rotor. Bearing stiffness depends on bearing size, contact angle, speed, mounting arrangement, and applied load. Radial stiffness influences the vibrations and critical speeds of the pump or test rig. Values for stiffness are 0.2 to 1.5 million for ball bearings and up to 2 million for roller bearings.

No simple formulas or life factors can determine the rotordynamic behavior of a system; every system is unique. Detailed computer models are used to calculate bearing stiffness values. Models have been updated to analyze a single bearing or a complete shaft system. This analysis is done to ensure that the pump will operate at all rpm and load values. For good rotordynamic design, bearings should be located as close as possible to the large rotating masses. Overhangs of turbines and impellers should be kept as short as possible. Accelerometers should be placed as close as possible to the bearing locations for data collection and analysis. Dynamic balancing of the shaft and rotor assembly helps to stabilize the system and eliminate unwanted vibrations. Radial proximity probes should be considered in the instrumentation package for rotordynamic analysis. Proximity sensors need to be calibrated at the operational temperature to give accurate data information.

If bearing stiffness is found to be inadequate, several options can be considered. The size of the bearing can be increased. The axial preload can be increased within the bearing's capacity. Contact angles can be decreased. Roller bearings might take the place of a ball bearing or be added. In the case of one bearing, two could be used or dual row rolling elements considered.

- Coolant flowpath—No oil or grease lubrication is allowed in bearings used in cryogenic turbopumps unless the bearings are in a separate, closed system. In today's turbopumps, the coolant is usually the propellant. Kerosene-based propellants such as rocket propellant or jet propellant provide good

lubricating properties. Low viscosity fluids such as LOX and LH₂ do not. These coolants do not lubricate, but are used more to remove the heat that is generated in the bearing.

The heat generated in a high-speed bearing comes from basically three sources: (1) From the rolling and spinning of the rolling elements on the rings in the contact zone, (2) ball-to-cage rubbing friction, and (3) churning of the coolant in the bearing. Most of the heat comes from the first two and can be addressed in the design phase. Computer analysis is done to select the best design with tradeoffs in heat generation versus contact stress levels. Cage materials should be selected that give the lowest frictional drag on balls and rings. The design of the cage should also allow the lowest restriction to the flow of coolant through the bearing.

Another concern with high heat generation is the auto-ignition temperature of the steel used in the bearing. This temperature must not be reached in LOX or LH₂ bearings. Values of auto-ignition temperatures are sometimes available in steel design handbooks.

Propellants are introduced in one or two ways, flooded lubrication or jet lubrication. Flooded lubrication is normally preferred. In a turbopump, some of the propellant is bled away from a high-pressure source, usually the impeller, and passed through the bearing. This fluid is recycled back into the flow stream at a low pressure point to reduce propellant loss. Angular contact bearings have a low shoulder that allows good flow of the propellant into the contact zone. The selection of an outer riding land separator helps to maximize the area of cooling. Positive pressure is used to provide propellant flow through the bearings. The pressure of the coolant has to be above the critical pressure at the predicted or measured temperature. If not, a vapor bubble can form in the contact zone, reducing heat transfer from the metal to the propellant. Contamination of the coolant has to be kept to a minimum. Debris in the bearings will decrease the life of the bearing and cause raceway damage. Passageways should be as large as possible so as not to block the flow of coolant. In most turbopump designs, no filters are used for the bearing coolant. Cleaning and handling processes for the pumps, test rigs, and bearings must control the level of contaminants. Water must be removed and kept out of cryogenic bearings. A dry purge gas, usually nitrogen (helium for hydrogen systems), is used during chilldowns, postoperations, and dormant times to keep moisture off of the bearing surfaces.

Jet lubrication is not used as often as flooded lubrication. With jet lubrication, the designer has to align the jet so the fluid enters the contact zone. The fluid must be free of contaminants so as not to plug up the orifices. Some jet designs use a hollow shaft and force the coolant through a ring mounted on the shaft. An advantage of jet cooling is that the flow can be directed through bearings in a parallel flowpath, reducing the heat input from one bearing to another as in a series flowpath.

Bearing heat generation can be monitored by installing thermocouples against the outer race of a bearing (nonrotating outer race). The end of the thermocouple needs to be in contact with the bearing outer race during operation. Bearings move axially during transients and can tilt and move during loading. Redundant measurements 90 degrees apart can also be used to help obtain the operating temperature. The clearance around the outer race can allow coolant to enter the space and the thermocouple will measure the coolant temperature instead of the bearing temperature. Data collected can be examined to determine if the thermocouple is measuring the bearing outer

race temperature or the coolant temperature. Another useful measurement is the temperature of the fluid entering the bearing and the exit temperature of the fluid. The heat input from the bearing can be calculated from the mass flow and the temperature rise.

- Cage design—The cage or separator has to withstand ball loads, centrifugal hoop stresses, provide lubrication to the bearing, keep the rolling elements separated, all with having a minimum cross section to allow coolant flow through the bearing and minimum weight.

Ball to cage loading can be high enough to lift the cage upwards on the trailing side of the pocket and push the cage down on the leading edge of the pocket. This creates flexural loads in the cage. To prevent this, the cage to ball contact should be at the pitch diameter of the bearing. Ball to pocket clearances also have to allow advancing and retarding of the ball due to misalignment and loading. Guidelines for ball to pocket clearances are 0.025 to 0.035 in/in of ball diameter. In some cases, such as misalignment tolerance, the ball pocket can be elongated in the circumferential direction to allow more ball movement in the pockets.

Web thickness is selected to support the ball loads and resist wear. Web thickness is the circumferential distance between balls. Web thickness should be a minimum of 0.015 in. The web dimension along with cage circumference determines the number of balls that will fit into the bearing. The ball number and size must support the static and dynamic loading experienced by the bearing.

Cages need to have a thin radial cross section to let the coolant flow through with minimal restriction. The cross section also has to be strong enough to resist any cage loading from misalignment. A value that has been published for minimum radial cross section nonmetallic cages is 0.090 in. In order to get the coolant into the inner race contact zone, most cages are outer race riding. Some designs have used inner race riding cages with slots or outer race riding with cutouts on the outer race for cooling. The effect of shrinkage must be considered for all designs. The outer race or inner race can cause increased drag on the cage if the dimensions are not correct at operational temperatures. It is desirable to have very slight rubbing on the cage from the bearing ring for cage stability. Values of thermal contractions for some cage materials are shown in table 1.

Table 1. Thermal contraction values for PTFE bearing cage material.

Contraction of PTFE materials from 70 °F. to -423 °F	
100% PTFE	0.0215 in/in cage diameter
15% glass fiber, 85% PTFE	0.0084 in/in cage diameter
25% glass fiber, 75% PTFE	0.0090 in/in cage diameter
65% bronze, 35% PTFE	0.0140 in/in cage diameter

A one-piece cage is usually preferred. If a metallic shroud is used, considerations for the rubbing on the bearing ring must be analyzed. A method to hold the nonmetallic part in place has to be designed along with cutouts for the ball pockets to keep the balls from rubbing the metal. Rivets

must be secured so as not to loosen. The materials used have to be analyzed so as not to loosen or create forces on the parts due to thermal expansion or contraction. Salox inserts in both bronze and Armalon cages gave good results at MSFC in bearing testing. The bronze cage design had outer race riding pads to reduce the friction between the outer race and cage. Cage width should not be so wide as to rub any item close to the bearing. Cages are normally the width of the bearing race.

- Rolling element design—Rolling elements are sized to be large enough to support the loads the bearing will experience, but not so big as to decrease capacity due to centrifugal effects. The size must also allow minimum cage radial cross-sectional dimensions and give sufficient flow area for coolants. The number of elements should be maximized, but not so many that the cage web thickness is reduced below 0.150 in. Rolling elements are either made at facilities that specialize in balls and rollers, or the bearing company may have its own source.
- Balls—It is very important that ball sizes be controlled in a bearing. Normally, when a bearing is made, the rolling elements stay with that bearing and are not interchanged with other bearings. Ball diameter variation can create differential speeds in a ball train, causing acceleration and deceleration of the balls. High cage pocket impacts can result in wear or failure of the cage. Ball speed variation is always present in a highly radial-loaded angular contact bearing, so the ball diameters have to be closely matched.

The AFBMA has set standards for ball sizes and grouping of balls in lots or batches. Some example specifications are given in table 2.

Table 2. AFBMA standards for bearing ball sizing.

AFBMA Grade	Maximum Out-of-Roundness per Ball (μin)	Maximum Size Variation in a Batch (μin)
5	5	10
10	10	20
25	25	50

For cryogenic high-speed bearings, a minimum ball grade of 10 is used. The AFBMA specification also covers surface finish on the balls. Balls that are not smooth will result in vibration, noise, and a shorter life of the bearing. A finish of 1 μin AA is used for grade 10 balls and 4 μin AA for rollers. Cylindrical rollers are covered under the AFBMA specification also and similar requirements are imposed for size.

- Rollers—Rollers have additional requirements that balls do not. Roller misalignment (skewing) with the races can cause high edge stresses. Roller skewing with the cage also causes high friction and stress. To reduce the stress, rollers are typically crowned on the ends and the edges are rounded. The transition line between the flat section and the crown is smoothed and blended. Published values for the corner radii are 0.020 to 0.030 in.

Rollers to guiding race end clearances are in the range of 0.0002 to 0.0020 in with the race guiding edges controlled to 0.0002 in parallelism. Roller length within a bearing is controlled to 0.0002 in to ensure uniformity. The stackup forces on the guiding race and cryogenic thermal contractions must be analyzed to ensure that the race will not pinch the rollers during operation. Roller bearings normally have a slight negative internal clearance at operation temperatures. This is to help guide the rollers in the race. Analysis has to be done to ensure that the interference will not lock the rollers against the race at chilled temperatures. A roller length-to-diameter ratio of 1 is suggested with an upper limit of 1:3 used for a maximum.

Roller bearings (fig. 5) do not react to axial loads very well. The sides of the rollers will rub the races and shorten the life of the bearing. Published values for guidance race shoulder height are 14% to 20% of roller diameter. In operation, some axial loading may be seen in the bearing. In the design phase, a ball bearing should be placed in close proximity to the roller bearing to react any axial loads present.



Figure 5. Roller bearing with silicon nitride rollers.

4.2 Hydrostatic Bearings

MSFC has not tested hydrostatic bearings but is aware of their relevance for future vehicle engine turbopumps. Turbopumps for future space applications will be required to meet higher life goals, have higher payload capacities, run at higher speeds, and be lighter in weight than the currently used turbopump designs. The rolling element bearing DN values have been a limiting factor. To overcome this, new turbopumps are being designed and tested with hydrostatic bearings. Hydrostatic bearings eliminate balls and rollers, thereby increasing speeds and reducing maintenance inspections

and bearing replacements. Rotor damping, high stiffness, and high load capacity can be obtained in hydrostatic bearings. Disadvantages of hydrostatic bearings are the need for a high-pressure supply, avoiding metal-to-metal contact at startup and shutdown, loss of pump efficiency due to bleed-off, and lack of a database on the operation of hydrostatic bearings in low viscosity, cryogenic fluids.

The fluid supply pressures needed to support the rotor come from a high-pressure source within the turbopump. A pressure tap is placed in the pump from a high-pressure source, usually an impeller or volute cavity. The rotor assembly will tend to move around until high enough fluid pressure is provided to center the rotor. Displacement probes and pressure measurements are beneficial in analyzing the behavior of the bearing and rotor during startup, steady state, and shutdown.

To overcome the rubbing issue, hybrid systems are being designed using a ball bearing to support the shaft system during the startup and shutdown transients. The ball bearing is taken out of the system after the hydrostatic bearing pressures build up to support the rotor. Realizing that all sliding and rubbing cannot be eliminated, rub-resistant materials are tested and selected for the bearing and journal. Wear-resistant coatings may also be applied to the potential contacting surfaces. The harder surface is normally the rotating shaft with a wear-resistant coating applied either to the shaft or on a sleeve fitted over the shaft in the journal region.

Hydrostatic bearings for cryogenic use require large supply pressures for operation. Clearances are small to reduce leakage. Pocket recess depths can influence rotordynamics. The relationships of pressure, flow, load, and geometries are complex and detailed computer model analyses are required for a complete design. A search for a database for hydrostatic bearings using cryogenic fluids gives limited information. Many papers have been written on hydrostatic bearing designs and computer codes have been built and provided analytical results. Texas A&M has done work with hydrostatic bearings in water and published their results. There has been some work by Boeing/Rocketdyne and the Air Force Research Laboratory with hydrostatic bearings. The need now is to run instrumented test rigs and pumps and validate the computer models for accuracy and update as test data are obtained. Table 3 lists general starting points for designing a cryogenic hydrostatic bearing. Figure 6 shows typical nomenclature for a hydrostatic bearing design.

Table 3. Design points for hydrostatic bearings.

Load capacity	$0.25 \times (P_{\text{supply}} - P_{\text{sump}}) \times \text{length of bearing} \times \text{diameter of bearing}$
Pressure ratio	$(P_{\text{recess}} - P_{\text{sump}}) / (P_{\text{supply}} - P_{\text{sump}}) = 0.5$
Outer land to bearing length, a/L ratio	$a/L = 0.25$
Length-to-diameter ratio	$0.25 < L/D < 1$
Radial clearance	0.001 in/in of shaft radius
Number of pockets	6 to 8
Depth of recess depth	3 to 10 times clearance
Max eccentricity ratio	0.5

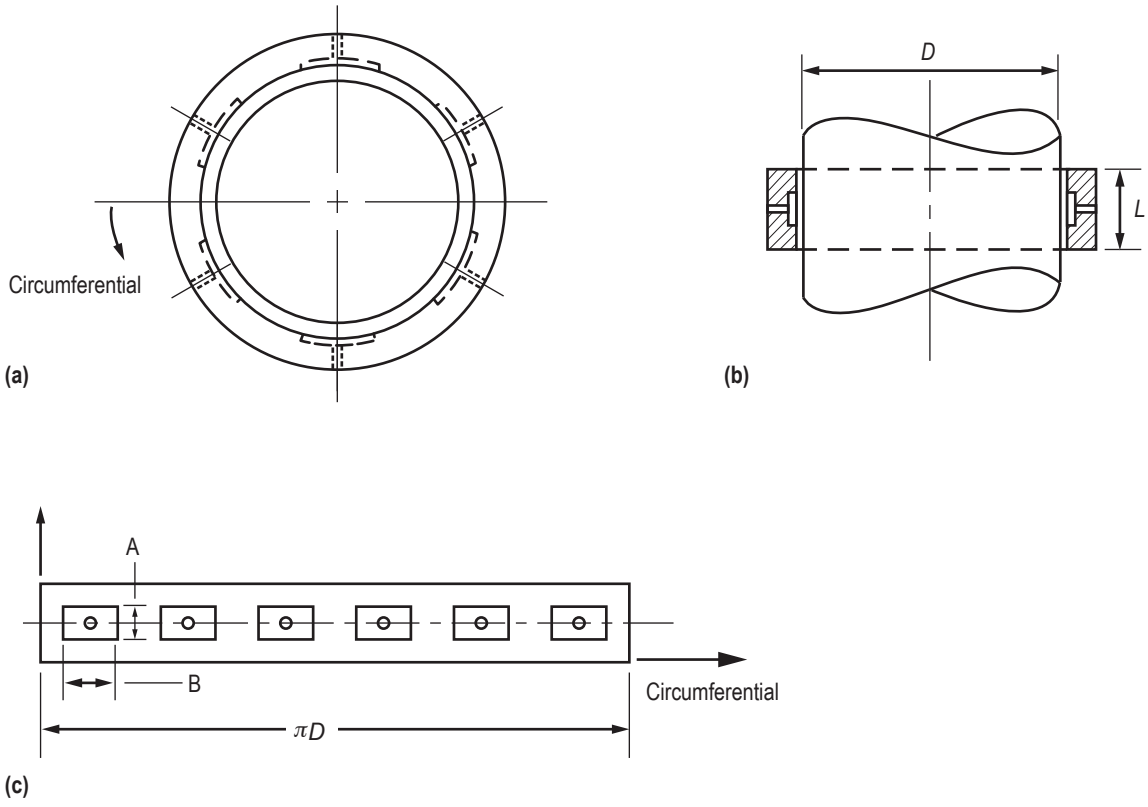


Figure 6. Hydrostatic bearing nomenclature: (a) Aerial view, (b) cross section, and (c) unwrapped view.

5. DESIGN AND ANALYTICAL TOOLS

The design and analytical tools available to the designer or engineer range from the simple diameter times rpm (DN value) to detailed computer simulation programs such as Advanced Dynamics of Rolling Element Bearings (ADORE). All the large bearing manufacturing companies have an engineering section in their product catalog or a separate guide that can be obtained by contacting them. This information gives product descriptions including materials, dimensional sizes, load capacities, cage materials, applications, mounting, and other bearing information. These guides are good for learning bearing terminology and getting ‘in the ball park’ on bearing size and load ratings.

Static and dynamic load capacities based on bearing rolling element size, number of elements, and contact angle can be calculated from textbook formulas. Fatigue life calculations can give a B-10 life rating of a bearing. This calculation is based on statistics that 90% of the bearings will meet the fatigue life requirements. High-speed turbopump bearings seldom reach this life. These bearings fail due to wear more than fatigue. Textbook calculations assume good lubrication, which is not the case in cryogenic turbopumps. Failure of bearings is also accelerated by debris either generated internally or not filtered out of the system.

A more detailed design and analysis can be done by referring to the library of books that are available on bearing design. Tedric Harris’s *Rolling Bearing Analysis* is updated regularly to contain the latest information on new materials and analysis. Another book for bearing design is *STLE Life Factors for Rolling Bearings* by Erwin Zaretsky. This book addresses new approaches for predicting bearing life for new and refurbished bearings. *Ball and Roller Bearings, Theory, Design, and Application* by Eschmann has bearing theory and specific applications for bearings.

For high-speed cryogenic bearing applications, more analysis is needed than the textbook formulas can provide. Computer modeling programs for bearing systems have been developed over the years and refined by testing. These models take the bearing’s dimensions and operating conditions and provide a large amount of output including fatigue life, spin/roll ratios, stresses, heat generation, cage loads, and other useful information. These computer programs allow fast parametric studies of interacting elements and allow the designer to pick the best combination of geometry for the application. They can also modify the loading to the bearings and see heat generation and life changes. Some of these models can be built to include the entire rotating assembly. A large number of hydrostatic codes are available to the designer also. Some of the programs allow a combination of rolling element and hydrostatic bearings on the same shaft to be analyzed.

Following is a listing of some of the computer analysis codes:

- Rolling element bearings:
 - A.B. Jones Systems Program, A.B. Jones Inc.

- SHABERTH, developed by SKF for the Navy.
 - SINDA, added thermal interaction to SHABERTH.
 - BAT (Bearing Analysis Tool), developed by NeXolve (formally SRS Technologies), advanced graphical interface user tool for bearing analysis.
 - ADORE, PKG Inc., dynamic solution to bearing/cage interactions.
 - COBRA, J.V. Poplawski and Associates.
- Hydrostatic, hydrodynamic, and journal bearings:
 - Hydrothrust, Hydroseal T, Hydroflex, Texas A&M, Luis San Andres.
 - Rotordynamics Seal Research, Loomis Calif., numerous codes and tools.

Many of the computer programs require very detailed input data including environments, loading, and dimensions of the bearings. If a bearing is being procured from a vendor, all the dimensions may not be available. Standard bearing catalogs will have OD, bore, width, load capacities, and materials, but not necessarily ball diameters, race curvatures, contact angles, internal clearances, cage dimensions, and other needed information. Some vendors are willing to supply this information if they know how the information is being used. The engineering section should be contacted for this information.

Output from computer analysis codes will be comprehensive. Some codes have the option of simple outputs and others will give solutions with thermal analyses. Experience with the codes will be beneficial. For parametric analysis or relative comparisons, the output data can be plotted using a spreadsheet program or some other graphical tool.

6. FINDING A VENDOR

Bearings used in cryogenic coolant turbopumps or tests rigs are not normally off-the-shelf items. The loads and speeds of the bearings require contact angles, race curvatures, and diametrical clearances that are not standard. The size of the bearing bore, OD, or width may not be a standard size. The environment that they operate in eliminates the commonly used bearing steels of M50 and 52100. Lubrication issues may require that the bearing be separable for replacing the cage. No currently known bearing vendor has LOX cleaning and testing facilities. Bearings used in cryogenic applications require a subzero quench as part of the heat treatment. A heat treat facility may have to be found separate from the manufacturer. The quantity of bearings will be small if a test program is underway. The requirements for quality control and inspections may be to a degree that the vendor does not normally have. Packaging requirements for cleanliness may not be what the vendor is normally accustomed to doing.

All of the above requirements will make it difficult to find a vendor for cryogenic bearings. Some of the larger bearing companies have smaller shops for nonproduction bearings. The willingness of the supplier may be determined by the economy and what interest they have in supplying the bearings and sharing the data. The buyer needs to be clear to the vendor what the end use will be. In some purchases, more than one supplier may be required; e.g., one for the bearing races and balls, one for heat treating, one for the cage materials, and another to machine the cage. The buyer may have to coordinate some or all of the individual purchases and transportation. Delivery times ranging from 6 months to a year can be expected.

Cleanliness of bearing components is vital to successful operation. The supplier needs to have procedures in place to ensure that the components are free of contaminants before being sent to the user. Bearings need to be inspected in a clean environment with safeguards to ensure no debris is allowed on components or in packaging. Bearings for cryogenic applications are made of corrosion-resistant steel and are sometimes packaged dry in inert gas-filled bags and heat sealed. If components are separated and wiped with a preservative oil, the oil must be removed before inspection and use. Oil should never be applied to composite cages as it will wick into the material and cannot be removed.

Once a bearing design is chosen, an engineering drawing is made with all the specifications listed including all material requirements, geometry and dimensions, quality inspections, LOX/GOX testing requirements, heat-treating requirements, labeling, and packaging instructions clearly defined, and any notes that will be beneficial to the manufacturer. To reduce cost and delivery time, the vendor may have quality checks, inspections, packaging methods, and material specifications that will be acceptable. The vendor may convert the supplied drawing to one specifically for their floor operations. The designer needs to review the vendor's drawing before approval to proceed. A good relationship with the vendor is a necessity to ensure that what is received is what was required.

7. INSTALLATION AND REMOVAL

Bearings with rotating inner rings are designed with an interference fit between the inner ring bore and shaft, which prevents fretting caused by rotation of the ring relative to the shaft. This interference must be tight enough to prevent fretting but not so tight as to cause failure of the inner ring by hoop stresses. These bearings are not normally pressed onto the shaft in order to prevent galling of the contacting surfaces. Bearings are heated in order to expand the inner ring bore and the shaft (or sleeve) is chilled in LN₂ to shrink the OD. The elevated temperature must not be so high as to temper the bearing steel or damage the cage material. Bearings are heated in a temperature-controlled oven or induction ring heater. Bearings should be handled carefully using noncontaminating gloves. Shafts or sleeves that are chilled in LN₂ must be handled with care coming out of the LN₂. Cryogenic gloves will be needed for handling. Ice will start to form on the parts very quickly once they are out of the nitrogen. Assembly of the bearing on the shaft or sleeve must also be done quickly. Ice can cause a lack of flatness of matching faces and create misalignment. A small press should be used to apply light pressure to the bearing until the temperatures normalize. It may be possible to eliminate the LN₂ chill by using a cold freezer instead. A dimensional thermal analysis can be done to calculate the temperature gradient needed for assembly. The part can be placed in a plastic bag to eliminate ice buildup on the part while in the freezer. This eliminates the hazards of using LN₂.

After assembly operations, the shaft assembly is placed in a temperature-controlled vacuum oven where moisture is removed by a bakeout operation. Minimum recommended time in the vacuum oven is 4 hr at 150 °F. Before the bearings are placed on the shaft or sleeve, a lubricant to reduce friction should be applied to at least one of the contacting surfaces. A burnishing of MoS₂ with a clean, lint-free cloth should be used as a minimum. The best method is to use a LOX-compatible, heat-cured dry film lubricant; however, using dry film lubricants requires several operations that can be time consuming. The area to be sprayed needs to be grit blasted for the lube to adhere. This requires that the unsprayed areas be masked off using a low residue tape. After grit blasting, the part must be cleaned before spraying. After spraying, the part must be placed in a circulating air oven and heat cured. The coated surfaces need to be burnished when the part is removed from the oven. A tape test is then required to check for quality of the application. The part is then recleaned and ready for use. The advantage of this method is that the lubricant will stay on the parts for several cycles before it needs reapplying. MSFC-RQMT-1282, Rev. A, covers the requirements for dry film lubricants.

Press assembling is the least preferred method for installing bearings on a shaft or sleeve. A very thin layer of grease can be used in applications requiring pressing. The grease must be compatible with the operating environment of the tester or turbopump. The Bray 600 series of greases with MoS₂ have been used successfully on test rigs and turbopumps. Care should be taken so as to not use an excessive amount of grease. All components should be cleaned of excess grease to prevent contamination in other areas of the pump or test rig. The grease should not be between faces of parts so as to cause stackup problems. Only the bearing component that is assembled with the interference fit should have force applied to prevent brinelling of the race. The face of the ram and part has to

be parallel so forces are applied evenly. If threads are on the shaft, they have to be protected from deformation from the force of the press.

In some turbopumps and test rigs, additional clamping of the race onto the shaft is done to ensure there is no rotation of the ring. Clamping nuts or tie bolts are used to keep the race from creeping during rotation. The force must be great enough to withstand vibrations, torques, and centrifugal forces that could loosen the clamping force. Retainers or locks are used to prevent loosening of the nuts. Snap rings and lock wires are not used.

The bearing orientation, assembly drawings, and assembly procedures need to be understood before assembly operations so any puller grooves, load angle orientations, and spacers are correct when assembled. Removal and reinstallation is time consuming and creates unnecessary wear on components.

Bearing removal is usually done with a hydraulic press. Bearings with a puller groove designed into a ring can be disassembled using a jaw-type puller or press (fig. 7). Tooling is designed to remove the bearings using a press that pulls or pushes on the tight component to prevent overloading the rolling elements and race. Angular contact bearings and roller bearings can come apart during disassembly. Any records of ball or roller locations can become unusable. In all work, the bearing components should be kept clean and protected from damage. Press forces can be high and parts can loosen suddenly and fall to the floor.

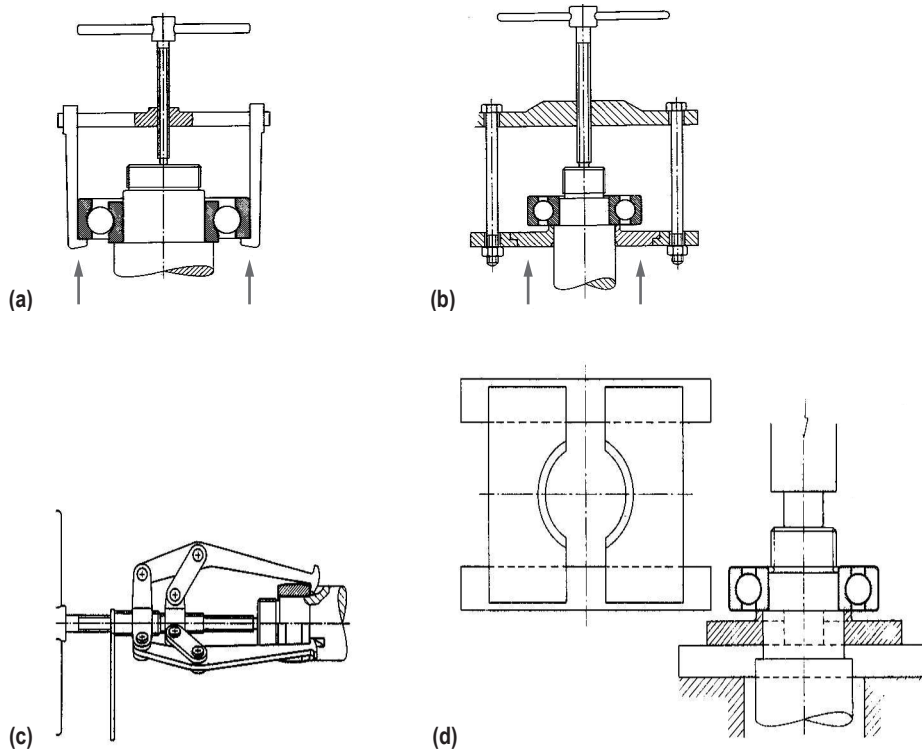


Figure 7. Methods to remove bearings from the shaft: (a) Incorrect way, (b) correct method, (c) puller with three jaws, and (d) hydraulic press method.

8. PRETEST, TESTING, AND POSTTEST INSPECTIONS

Bearings should be inspected and documented before use. The handling of these bearings should be by powder-free gloves. Oil from fingerprints can etch the surfaces of the bearing. The inspections should be done in a clean room environment or in an air-flowing clean bench. Visual inspections using a microscope with photographic capability is beneficial for before-and-after comparisons. Rolling element diameters can be measured in three planes by accurate metrology instruments. Rolling elements can be matched up to individual cage pockets and followed throughout testing. Individual bearing parts can be weighed to calculate an approximate volumetric wear rate. Race curvatures can be measured, surface finishes recorded, and roundness values taken by using electronic metrology equipment. The AFBMA has guidelines for measuring bearing components that can be followed.

Test rigs or turbopumps should have an inspection port to view bearings between tests. An instrumentation port may be used in the absence of a dedicated port. A borescope with still picture or video capability can document the condition of the bearings for test-to-test comparisons. Views are normally limited to cage and ball surface inspections with parts of the race being visible. The contact surfaces or the races may not be visible in all inspections. It is beneficial to have thermocouples on the bearing outer race (if not rotating) and upstream and downstream of the coolant flowpath. Pressure measurements in this area can determine the quality of the coolant and whether or not the bearing is pumping the fluid. If silicon nitride rolling elements are used, special attention should be placed on the condition of the balls or rollers. The ‘river mark’ surface features that have been observed at MSFC are not known to be detrimental to performance at this time, but they will increase in length during testing. These indications should be documented if seen. If an angular contact bearing is wearing severely, the preload can be lost. If this happens, the rolling elements will begin to slide instead of roll. This will generate heat in the bearing. Steel balls will begin to look dull and start to show colors ranging from gold to black, depending on the temperature. Streaks will appear on the balls where they are skidding. An indication of good transfer film lubrication from the cage to the balls will be seen as a white color from PTFE cages or a bronze color in a Salox-type cage.

Posttest inspections follow the pretest inspection plan with documentation to report the final condition of the bearings. Clean handling practices should also be followed. Attention should be given to the wear bands or tracks on the inner and outer races. Bands should be concentric around the rings. Ball wear will cause the band to be wider as the ball changes contact angle. An eccentric band can indicate heavy radial loading on the bearing as the balls are forced down on one side and climb back up on the other. Eccentric bands can also indicate misalignment of angular contact bearings used in pairs. Deadband clearances allow the angular contact bearings to act as ‘book-ends’ that can lean together or in opposing directions. Spalling of the inner and outer races should be documented. A holding fixture can be made that will hold the bearing races under a microscope at an angle and rotate the races so the wear tracks can be seen clearly. Balls can be put in a white container (such as a coffee cup) so the light can illuminate the surface. A tent of white paper can be

constructed to help view ring and ball surfaces. The color or tint of the balls should be noted. Dull surfaces can indicate thermal effects, with gold to black colorations if the temperatures are getting high. Ball separators should be inspected for rubbing in the pockets and rubbing from the guiding race. Heavy rubbing on the fore or aft end of the pocket can indicate the ball pushing or retarding the cage. If coolant flows through the bearing, some side pocket rubbing will be visible. Debris damage will be seen as pitting on the race surfaces. The examiner should be sure the bearings are clean for inspections. Some handling debris or lint can stick to the bearing surfaces and appear as anomalies.

Ultrasonic cleaning of rolling elements should be done in separate containers. The vibrating motion of the elements against each other will create ‘worm trails’ on the surfaces of the balls or rollers if cleaned together in one container.

Roller bearings can have roller end wear from rubbing the guide flanges. The race with the guide flanges should be examined for fatigue spalling at the edges where the rollers roll. The flanges can also show signs of pitting from debris. Roller cage pockets should be examined for uneven wear that indicates roller skewing.

Hydrostatic bearings under operating pressure should have no contact. The wear in these bearings occurs during startup and shutdown. The contacting surfaces of the bearing and journal are examined for scratches, roughness, abrasion, and transfer of metal. The bearing orifices should be checked for debris to ensure that they are not plugged. In a highly instrumented test rig or turbopump, data can be plotted to verify the operation of the hydrostatic bearing. Rotor speed, hydrostatic bearing supply and sump pressure, and rotor radial proximity probe data can be plotted to investigate the operation of the bearing.

Hertzian contact failure atlases are available to help determine failure modes. These atlases have photos of wear and failures of bearings along with descriptions of the conditions seen on the bearings. *Failure Atlas for Hertzian Contact Machine Elements* by Tallian is one source of information. Some bearing design textbooks have a section on bearing failure modes along with pictures of failures. *Rolling Bearing Analysis*, 4th edition, by Ted Harris has a chapter on failure mechanisms and analysis. Major bearing companies have brochures showing bearing failures. Barden Bearing has a brochure entitled “Bearing Failure: Causes and Cures” and NTN Bearing Company publishes catalog No. 626, “Bearing Failures, How to Prevent Them.” Searching the Internet using ‘bearing failures’ or ‘Hertzian contact stress failures’ will bring up pages of Web sites with information. Technical libraries should have a selection of bearing design and failure guidebooks for the designer or user to consult.

Some examples from references are shown in figure 8.

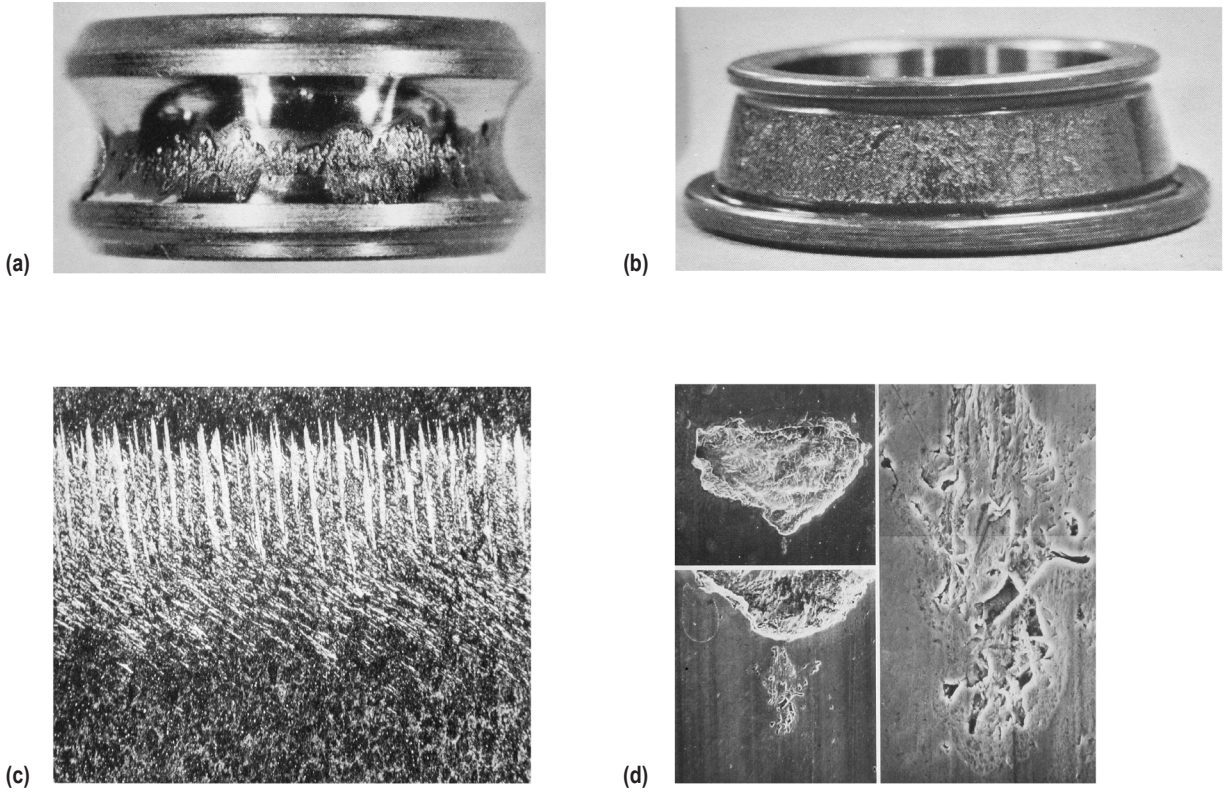


Figure 8. Examples of damage to rolling element bearings: (a) Fatigue spalling of inner race, (b) spalling of roller bearing inner ring, (c) white stripes seen on inner race—strips are normal to rolling direction, and (d) closeup of spalling on inner race (rolling direction is bottom to top).

9. SUMMARY

Cryogenic bearings are a unique subset in the overall field of bearings. They operate in an environment different from other bearings. It is the author's intent that this guide will help the designer or engineer understand the design issues, procurement, assembly and disassembly, operation, and inspections required for a successful program. As new bearing materials, analytical tools, and hydrostatic bearings are developed and proven, design guides based on practical experiences will be beneficial for the engineering community. Comments and additions to this guide are welcomed.

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14. ABSTRACT Cryogenic bearings are a specialized area in the overall group of bearings that are used in industrial and aerospace applications. Cryogenic bearings operate in an environment that is not experienced by most bearings. High speeds of turbomachinery, cryogenic coolants, nonstandard materials, and lack of lubrication place demands on these bearings that must be met for mission success. This guide is based on the experiences of engineers at NASA Marshall Space Flight Center that have been involved in bearing research and testing along with specific references that have been written. It is not meant to be a comprehensive design textbook for cryogenic bearing applications. Its purpose is to help the designer or user to better understand the requirements placed on bearings in these applications.					
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