HIGH TEMPERATURE BEARING AND DRY - LUBRICATION CONCEPTS PHASE I

FINAL REPORT

SEPTEMBER, 1982

August O. Weilbach

Sponsored by:

Defense Advanced Research Projects Agency (DoD)
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The Phase I program accomplished the study and functional evaluation of solid lubricant formulations in conjunction with the use of a specially designed and built test-bed bearing. The bearing was fabricated from hot pressed Silicon Nitride (HPSN).

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Special attention was given to problems associated with the design, material selection and lubrication of the ball-retainer. Much emphasis was also given to finding better methods of re-supplying lubricants to assure better bearing performance and to substantially extend the useful life of any type of solid lubricated bearing.

Investigations were made into ways of finding better methods of bonding the solid lubricants to the ceramic substrate materials without the loss of low friction qualities. Adhesion testing was accomplished by coating coupons of candidate materials. Rub-shoe tests were conducted using the same family of substrate materials; also some commercial and all the proprietary solid lubricant formulations were submitted to a series of wear tests.

The bonding of all useable formulations are expected to be enhanced by a metallization process of the substrate surface. The coating of substrate surfaces with metal thin films proved to be more difficult than anticipated and further work in this area is definitely needed.

The study presents several recommendations for work extending into Phase II. Appendix "A" presents a background history of past efforts to produce ceramic bearings and the use of solid lubricants.

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PREFACE

This report presents the results of a study and experimental work performed as a team effort by Mindrum Precision Products, Rancho Cucamonga, California and Helvart Associates, Fullerton, California. The work was performed for the Defense Advanced Research Products Agency (DARPA), Arlington, VA 22209 under contract N 00014-82-C-0248. Administration of the contract was performed by the Office of Naval Research, Department of the Navy, Arlington, VA 22217.

This publication is the final technical report on the program issued under the title "High Temperature Bearing and Dry Lubrication Concepts." The work was conducted during a period extending from March 1, 1982 to September 24, 1982.

The author wishes to acknowledge the contributions of W. T. Butek, who supervised the project, and of S. M. Grasso who designed, prepared the test stand, and conducted the experiments, also of W. A. James who worked on the preparation of the report. All three men are employees of Mindrum Precision Products. L. Sibley of Tribology Consultants, Inc. guided the manufacturing of the test bed ball bearing through a demanding schedule.

1.0 Introduction

1.1 Program Objectives

This study deals with all aspects of materials, design, manufacturing and testing for ceramic ball bearings. The emphasis is to establish criteria on the performance behavior and limitations of several proprietary high temperature lubricant formulations. A limited amount of performance testing was conducted through the use of a specially designed and manufactured all ceramic test bed ball bearing. The study is structured to fill voids in past similar efforts and contains several suggestions for improvement of ball retainer designs and methods for relubrication. This study endeavors to form the basis for further and deeper investigative work and to arrive at definite solutions under a Phase II follow up program.

The present study generally does not present discussion of advanced theories, which have been covered in many excellent books, reports, and studies. (See list of references in Section 9.)* Instead, it addresses itself to the practical aspects of solid lubricant and high temperature bearing technology.

1.2 Program Approach

The study emphasis is on high temperature lubricants. With this in mind, a review of existing technology was necessary to primarily find out if our plans and efforts projected duplication of current work, or if our approach consisted of new and promising avenues.

Differences and modifications as compared to the proposal are covered in the following section under 1.3 Proposal Conformity.

During the writing of the proposal and all during the study under this contract, it was very evident that some practical work and testing had to be done in order to confirm empirical results. This prompted the decision to obtain a specially designed ball bearing. It was, as the study shows, a good decision, despite the delivery problems for the bearing. To test the bearing lubricant combination, some test gear was needed. As described in 4.2.1 a simple and inexpensive set-up served the purpose well.

* Numbers in parenthesis refer to references at the end of the report.

1.3 Proposal Conformity

In order to facilitate comparison of the work as defined in the proposal with the actual work done under the Phase I contract, the main deviations, changes, and additions are outlined in the following:

- Lubricants listed as A, B, C, D, E, & G have been replaced with the following code numbers:

  A = 2-14 Basic Formulation,

  B = 2-14-S High grade formulation,

  C = 2-15-SI Subject to further study and analysis. Unsuccessful performance during Phase I,

  D = 3-14 Basic lubrication system

  E = 3-14-S Multi-layer system to 480 degrees C

  G = 3-15 High grade multi-layer system 815 degrees C

  4-1920 Full range multi-layer system Sub-zero to 538 degrees C. Tests and analysis are continuing.

- Tests at Cryogenic temperatures have been eliminated due to the availability of several suitable commercial products covering the cryogenic to 250 degrees C range.
- Kayon 2000 (Kayon is a trademark of Kennametal, Inc.) (Sialon) ceramic was not obtainable for the Phase I study.
- Modified Tilt-test superseded by bearing test.
- Test bed Bearing has been manufactured totally of Silicon Nitride and not Silicon Nitride races and Zirconia (Zirconium Oxide) balls as proposed.
- High density vs. porous ceramic materials study is deleted.
- Due to early negative results with Boron Fiber composites for use as ball retainers, no further investigation with this material has been conducted.
- Due to delivery problems and fixed price limitations of the contract only one ceramic test bearing was procured.
- The substitution of one ball bearing with a sleeve bearing of different ceramic material combinations were rescheduled because of unforeseen delays in funding for the Phase I study.

- Test equipment recommendations for Phase II were extended into a search of existing suitable and available equipment (see section 4.2.3).

2.0 Ceramic Bearing Considerations

2.1 Bearing Element Interrelationship

During the time of generating a proposal for the Phase I study and even more so during the study itself, it became obvious that solid lubrication has to be studied within the context of all bearing elements contained within a bearing assembly. Whether it be a change in race, ball, or ball cage material, the effect on the total may differ from one to the other. Changes in temperature for instance may not affect the balls or the races, but may well determine life and performance of the ball cage. Behavior of the lubricants is greatly dependent on load, speed, and temperature.

As discussed in Section 4.1 a combination of different materials may give a better performance than using the same ceramic for the races and balls. The above short references are indications that, like all precision mechanical systems, which a ball bearing really represents, all of the bearing elements, specifically the lubrication system, have to be studied, manufactured, and tested in concert with each other. This must be done under the purpose of this study, with due consideration to the high temperature requirement.

In order to cope with the many variables a baseline of some sort had to be established from which to work. This, in turn, led to the decision to design and manufacture a test bed bearing (Figure 1) that would permit easy interchange of parts and lubricants. This aspect of the study is described in detail under Section 4.2.2.

2.2 Ceramic Bearing Materials

Early efforts to produce ceramic ball bearings had to rely mainly on one material, aluminum oxide. The quality of other emerging ceramics such as silicon nitride, silicon carbide, zirconia and others was, at that time, insufficient to have them considered for the demanding requirements encountered in high temperature ball bearing applications. Today the quality and physical properties of candidate materials has improved to

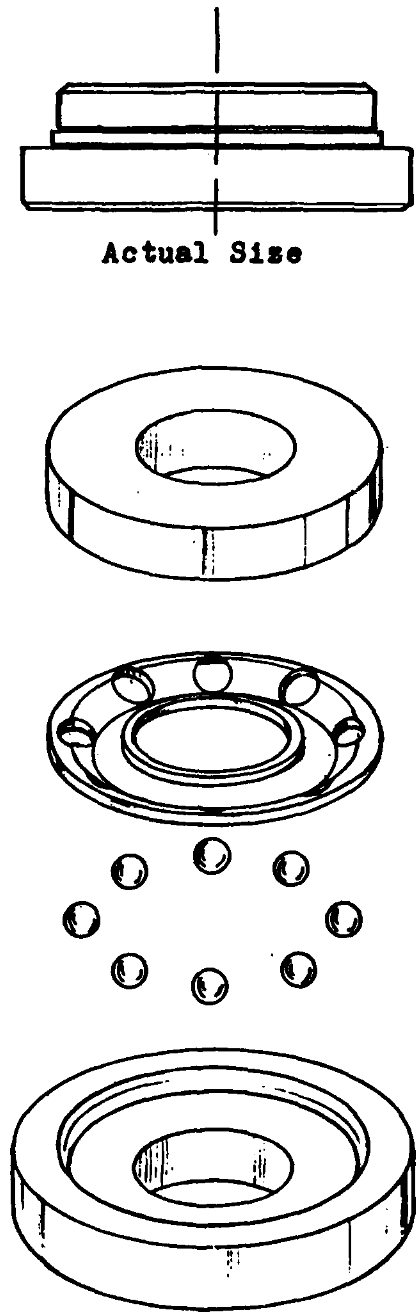


Figure 1. Test-Bed Bearing

a degree where the main attention has shifted from the bearing material to other concerns and problems such as ceramic bearing design and manufacturing methods. Still, the material problem is of paramount importance since some of the very desirable properties and characteristics inherent in ball bearing steels are missing or only present in insufficient amounts. Table No. 1 will highlight these differences and point out the relative values.

It is, for example, unlikely that the toughness of vacuum melted steel such as M-50 or 440 C can be obtained in ceramic within the near future. Also, the cost of the ceramic material will, for the time being, stay relatively higher compared to the price of even the costly steel superalloys. Nevertheless, the advantages of the ceramic materials are such as to fill urgent needs in advanced military and commercial equipment. Its resistance to high temperature and specifically its inertness to corrosive chemicals is unsurpassed. Its superior hardness in comparison to most metals makes it an excellent candidate for bearing application. Which one of the ceramics are the most suitable will be discussed in the following sections.

The mainstay material for most recent application has been hot pressed Silicon Nitride NC 132, produced by the Norton Company. The availability and consistent high quality of the product should hold it as the predominant choice for the near future.

2.3 Ceramic Ball Bearing Races

One of the advantages of ceramic material for ball bearing elements is the stability of the material. Even if metal bearings would accept the high temperatures it still would be nearly impossible to maintain the built-in precision of the individual parts. Hardness of ball bearing rated ceramics will stay substantially above that of steel (62 vs. 80 Rockwell C). The elastic limit or toughness of ceramic is below that of steel as represented in the different values representing the material's ductility. Some of these values are presented in Table 1. Another disadvantage is the relative difference in the thermal coefficient of expansion. It is desirable, and often necessary, to match expansion with the bearing supporting structure. Other considerations such as differences in thermal conductivity come into play. In balance for usage in temperatures above 350 degrees C, the ceramic ball bearing has to be considered.

Surface finishes obtainable on HPSN are comparable to finishes achieved on various steels. Nevertheless, to arrive at an equivalent finish level requires a much slower and

PROPERTY CHART			CERAMICS			METALS		
			Alumina	Silicon Nitride	Zirconia	M1	440C	17-4 PH
Mechanical	Density	gm/cm^3	3.3	2.8	5.6	7.5	7.7	7.8
	Hardness	HRA	87	94	80	56	58	60
	Compressive Strength	Kg/cm^2	28'000		20'000		4200	
	Young's Modulus	$10^6 kg/cm^2$	3.5	2.6	1.9	1.1	1.2	0.9
	Poisson's Ratio		.25	.25	.22			
	Tensile Strength	kg/cm^2	800	1000	1200	1800 (600)	5200 (2200)	4000 (1600)
	Fluxural Strength	kg/cm^2	3000	9500	10'000			
Thermal	Thermal Expansivity		7.7	3.3	11	6.5	5.6	6.0
	Thermal Conductivity	$\frac{cal.cm}{cm^2/sec/^{o}C}$	.06	.02	.004	.08	.12	.10
	Maximum Useful Temperature	^{o}C	1600	875	1500	450	500	500
	Thermal Shock Resistance		good	good	good	fair	fair	fair

Table 1. Properties of selected Ceramics and Metals

costlier process, and, depending on the type of solid lubricant applied, can actually be counter productive. In general terms, the surface quality for ceramic ball races can be degraded by 2-4 micro-inches, i.e., a 2 micro-inch finish can be 4-6. Too fast or overpolishing can result in the creation of surface stresses which are very detrimental to bearing life and performance. These stresses are known to induce spalling, cracking, and promote crack propagation. Overpolishing can also result in curvature deformation which in turn will create higher Hertzian contact stress. Such deformation has the tendency to induce excessive micro-slippage between balls and races.

2.4 Ceramic Bearing / Balls

In its selection and characteristics, the material for the balls is identical to the points discussed in the previous section. Material quality is even more important and equally so is uniform Size, Sphericity, and Surface quality (also referred to as the three S's). In using the ball retainer as the lubricant reservoir the balls act as the transfer mechanism to move solid lubricant from the retainer to the balls and subsequently to the inner and outer bearing races.

2.5 Ceramic Bearing Ball Retainers

2.5.1 Differences and special requirements

Within ball bearing element technology the ball retainer or ball cage has always presented special problems especially as related to high speed bearings. You may call these problems a "drag" on the performance of the bearings. Improper design, installation or material selection for the bearing has often been the cause of many catastrophic bearing failures.

With the advent of ceramic bearing and solid lubricant technologies the retainer has been given another purpose, namely to act as a lubricant reservoir. In that capacity, and lacking the separating layer and damping qualities of wet lubrication, the retainer is subject to considerable wear within the ball-pocket and on the interface areas with the lands of the inner and/or outer ball bearing races. The wearing of these contact areas generates substantial temperature rises. In addition, the sacrificial loss of material from the retainer presents a serious limiting factor to bearing life. Without damping as provided by wet lubricants, it is not uncommon for the retainer to go into high frequency oscillations.

The main purpose of selecting ceramic material for the bearings is to make them useful for high temperature applications and/or to withstand corrosive environments. These requirements also apply to the retainer. The following points establish the basic guidelines for the ball retainer improvements:

a) Select Materials (composites) with temperature resistance to 850 deg. C,

b) Select Materials with a very low coefficient of friction,

c) Select Materials with high tensile strength to sustain the substantial centrifugal forces at high RPM,

d) Search for Methods of reducing wear of the ball cavity by resupplying lubricant between the ball and the retainer cavity.

All the above points are much in the mind of anyone working on high temperature ball bearing technology and a number of ingenious methods have produced tangible progress. A review of literature and reports indicate the further need for research and improvements in several related areas. The following section represent and discuss some additional design ideas.

2.5.2 Ball pocket configuration

Whereas the motion of the ball against the race is and must be a rolling motion, the contact area between ball and ball retainer is purely frictional. Inertia, wind drag, and contact at the race land will result in substantially larger contact loads against the driven side of the retainer ball pocket than with that of lubricated bearings. This frictional contact is useful for rubbing off minute amounts of lubricant from the ball cage. Since the composite material retainer is normally manufactured as a ring with the required ball pockets drilled radially toward the center of the ball, contact within the cylindrical hole is very small and, depending on the accuracy of the hole, location may not be at the center of each hole wall. (See Figure 2.) To size and to equalize the contact-area location all high speed and high load bearings should be put through a run in sequence to conform the ball pocket with the matching ball diameter. After that, the required run-in time will be reduced and so will the wear within the ball pockets of the ball retainer. In all likelihood the larger ball contact will provide some damping and, in the process, reduce high vibration initiation of the

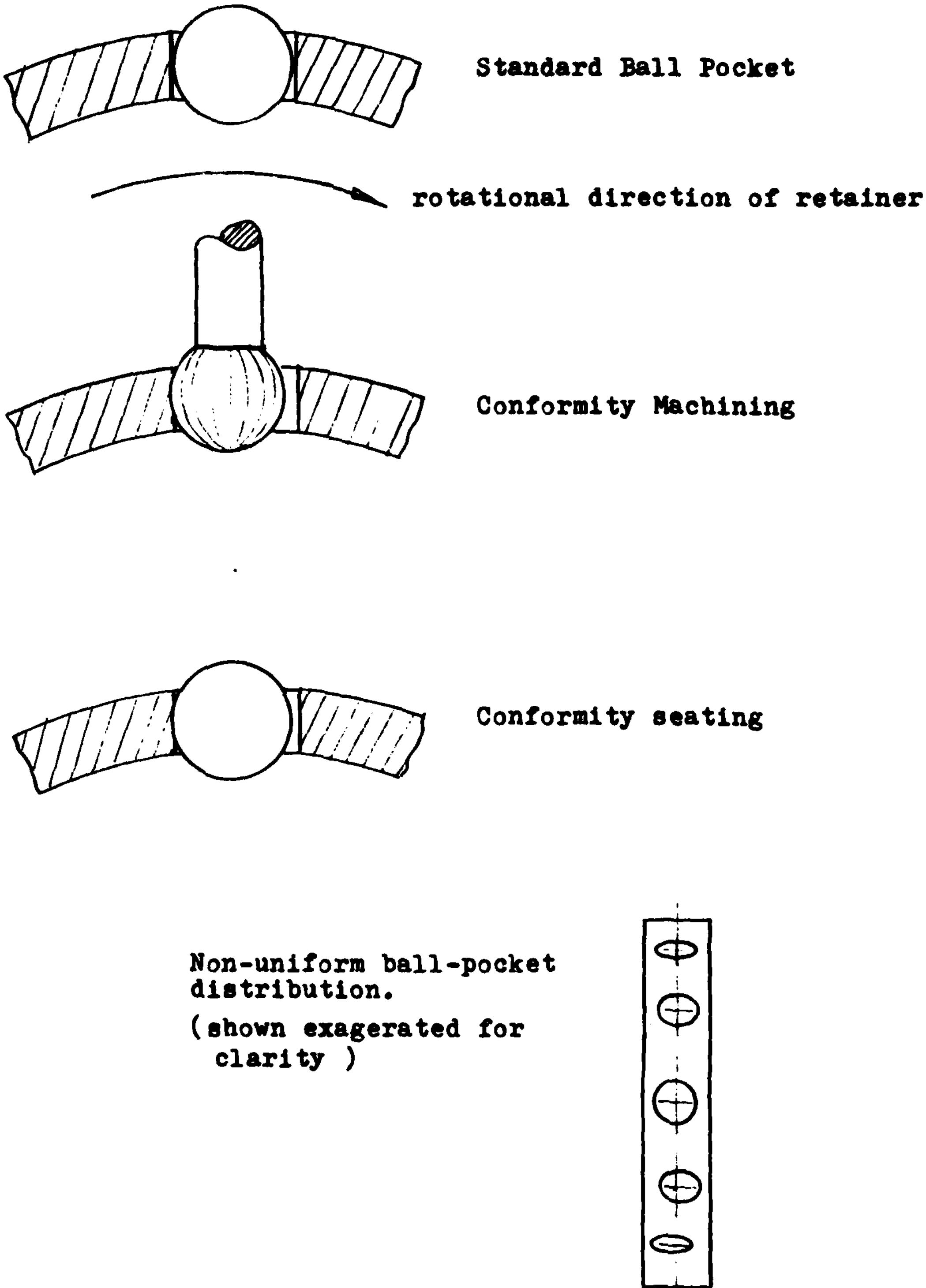


Figure 2. Ball Retainer/ Ball Pocket Conformity

ball cage. Such vibration, as is well known, adds additional stresses to the ball retainer that will greatly reduce its useful life. In regard to the solid lubricant formulations presented in this study, it should be pointed out that both the basic formulation 2-14 and the 2-14-S are suitable to be used as additional coatings for the ball retainer or as suitable lubricants to be used with inject relubrication. See discussion in sections 3.6.1 and 3.6.2.

2.5.3 Bearing retainer / Land contact area

Proper design of the ball retainer includes the necessary provision to run the retainer contact at the inner or outer race land (R39). For low and medium speed bearings and, depending on the retainer type, either the inner or outer race land is suitable to maintain guidance and contact. In fact, in order to reduce additional drag and friction in low and medium speed bearings, the retainer may only need ball guidance.

On high speed bearings where centrifugal forces will possibly result in dimensional changes of the retainer diameter, the choice is: (1) to compensate for the expansion before making frictional contact with the land of the outer race, or (2) to let the ring I.D. "float" on the O.D. land of the inner race. The choice is dependent on whether the inner or the outer race is the driven element. This in turn determines the relative amount of linear displacement of rotation and subsequently affects the choice of using the land of the inner or outer race. The frictional resistance between the ball retainer and the land will result in an increase of ball contact friction within the ball cavity. It is, therefore, of great importance to minimize the frictional contact load between the ball cage and the race land. Lubricant replenishment as discussed in section 3.6.2 and the promotion of an air cushion layer between the race land / retainer are important methods to be pursued further.

Two more points to be made in regard to the land contact areas: first, the ring surface of the outer or inner race providing the land has to be machined with a finish that has a pattern concentric and with the ring surface perpendicular to the axis of rotation; second, both the land and the ball retainer contact area need to be primed with lubricant and run-in before putting the bearing-assembly into final usage.

It is necessary for the ball retainer to ride the retainer periphery against the land of the inner or outer race. (See Figures 3 and 4.) Despite no-load conditions at the interface and the low friction characteristics of the retainer material the inevitable contact load and possible high

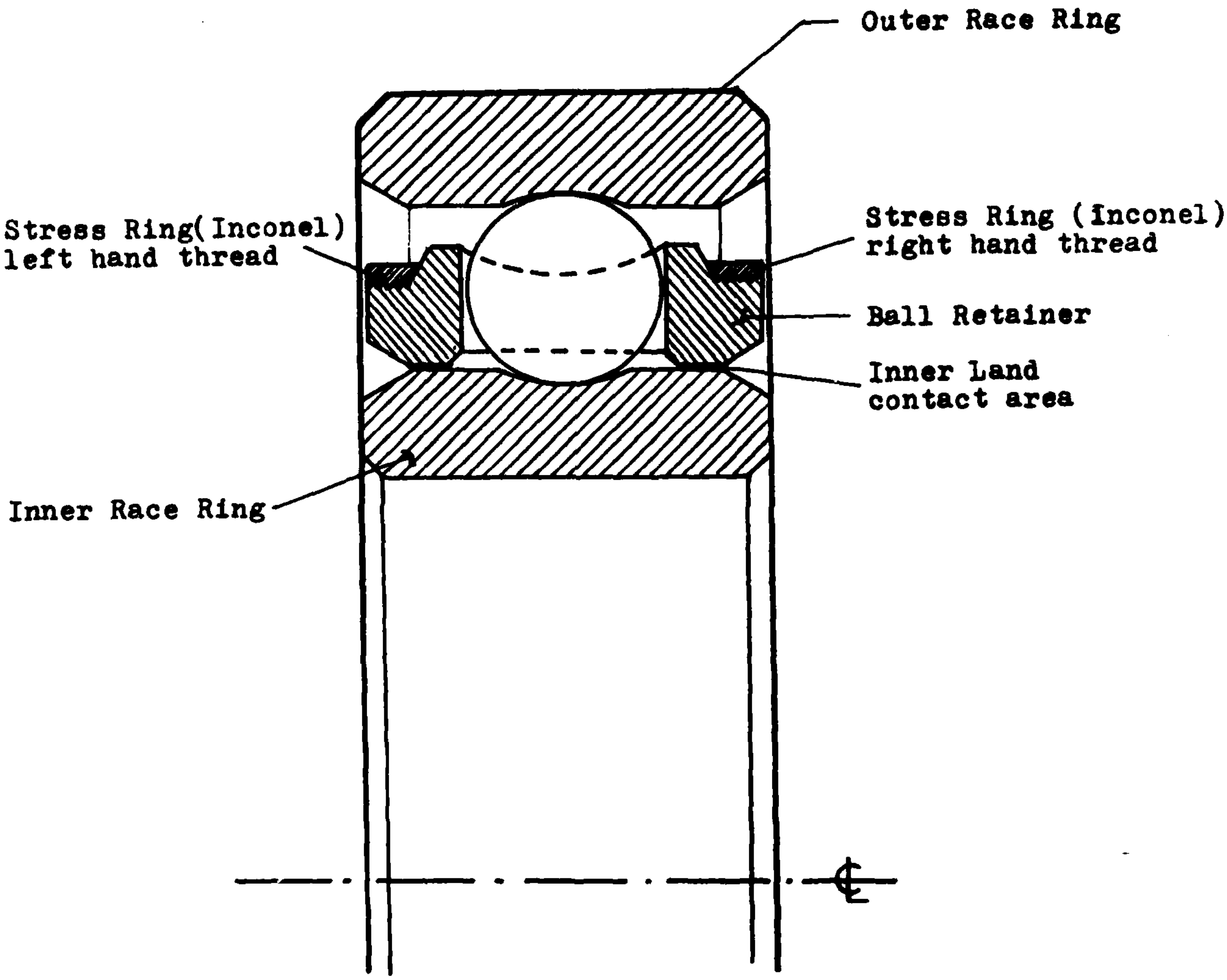


Figure 3. Ball Retainer / Concept A

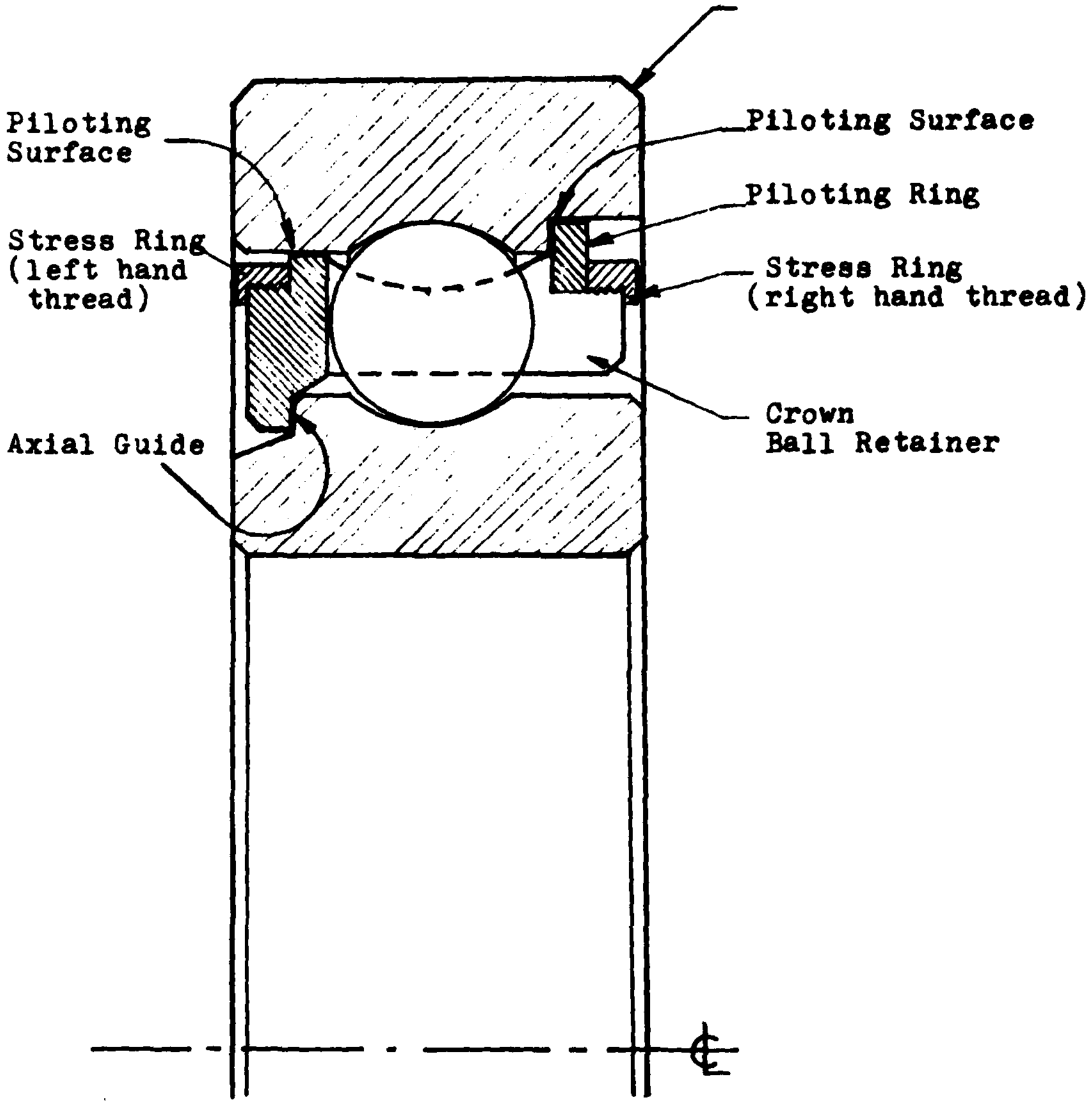


Figure 4. Ball Retainer / Concept B

frequency oscillations will build up temperatures in excess of ambient conditions and possibly exceed the considerable thermal build-up which occurs at the ball and race interface.

These conditions aggravate the temperature demands on the retainer material and will severely limit the temperature and speed capability of the bearing assembly.

Solid lubricants, as rubbed off from the retainer and as transferred to the race tracts, will thus have very limited life unless some provision for relubrication or special cooling would be provided. Another insufficiently explored possibility exists in the previously mentioned approach to provide an air or gas cushion between the land and the ball retainer contact area. If properly done, <u>cooling</u>, <u>relubrication</u>, and provision for an <u>air film</u> can be combined into one method. (See Figure 5.)

By slotting the land contact area of the ball-cage in a specific way to make it work as a turbine, the dry lubricant aerosol mix will be pulled in and through the bearing. Turbulence within the bearing will deposit some of the lubricant particles and at the same time pull out old debris. The gas will help the cooling and simultaneously provide a cushion at the critical contact area. The cushion will equally reduce the friction of the ball at the ball pocket contact area.

2.6 Ceramic Sleeve Bearings

Sliding bearings, of which sleeve bearings are one variety, can be classified as to their method of lubrication. Lubrication is a prime factor affecting the load carrying capacity. Compressed air is used as a cushion for high precision machinery. Hydrodynamic lubrication with greases and oils of a wide range of viscosity is used ranging from toys to very heavy machinery and in high precision equipment. Under inadequate air flow or break-down of an oil film, or where a bearing is loaded under stationary condition, starvation of the supporting films will result in solid contact of shaft and bushing. The starvation, in conjunction with fast equipment start-up, especially under load, can easily result in short or long term catastrophic bearing failure.

This study addresses itself primarily to high temperature bearing and solid lubricant problems, and this does not consider wet lubrication above the 300 degrees C level. By using a film of solid lubricant the chances of start up seizure are essentially eliminated. The sleeve bearing may be running on an air or oil film or, for a ceramic high temperature bearing, relying entirely on the solid lubrication. Obviously,

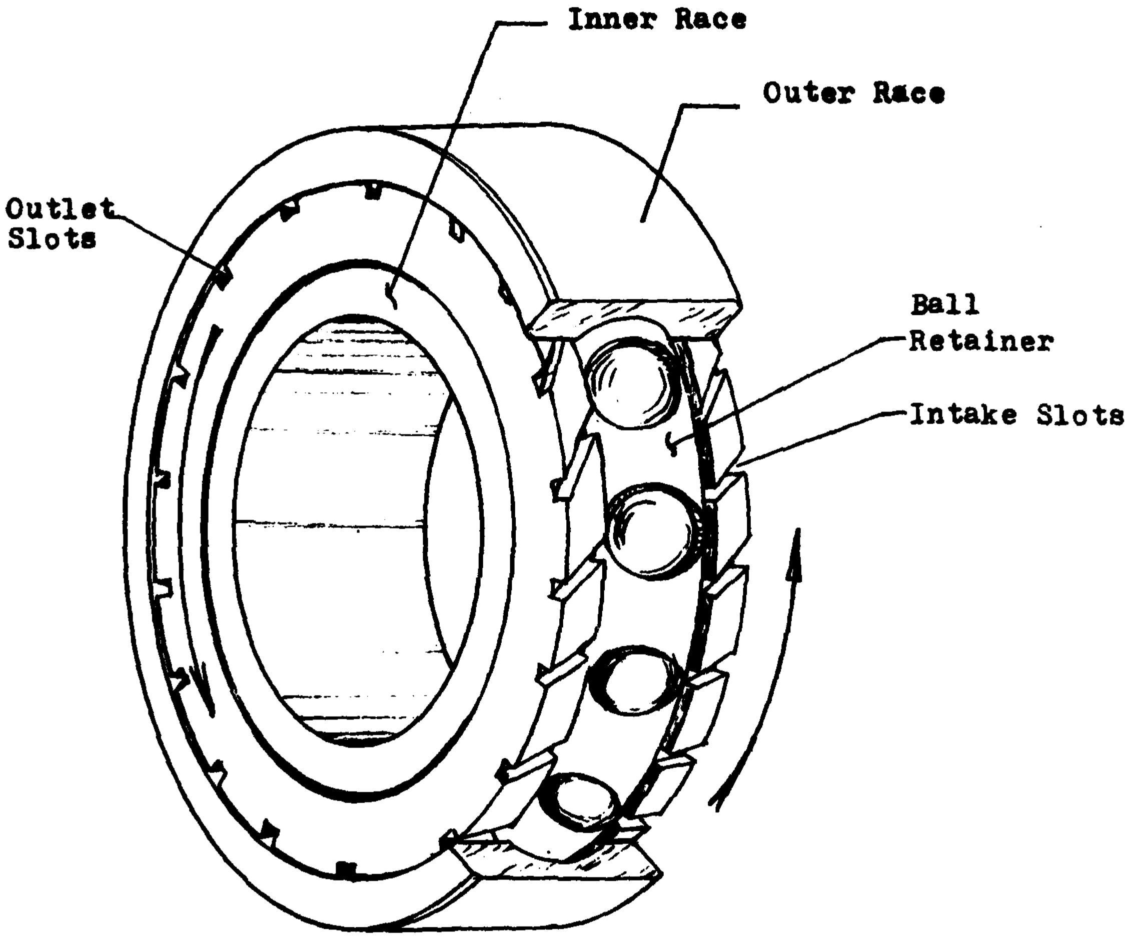


Figure 5. Ball Retainer / Air Bearing Concept

such bearings are well below the cost of ball or roller bearings and often could do the job of the more elegant ball bearing varieties. Still, their use, as attractive as it may be, needs careful analysis as to their advantages, and disadvantages.

Tests performed during the preparation of the proposal and during the early stages of this study were done by coating the shaft of a one-half inch diameter and two inch long air bearing. It resulted in a dramatic increase in free spinning time, starting at approximately 1000 rpm, from an average of 45 seconds to 150 seconds. The tests were performed with the parts made from aluminum oxide and manufactured with a contact surface finish ranging from 8 to 18 microns. Other valuable observations yielded results in respect to contamination tolerance. Introduction of a small amount of sub-micron size particle contaminant between the shaft and the sleeve bushing resulted in stickiness whereas the unlubricated assembly froze to where it was difficult to extract the rod. Radial clearance between shaft and bushing was .005 mm; the lubricant build-up amounted to approximately .002 mm.

From these experiments and tests it can be concluded that a dry-lubricant system (multi-layer) could closely approach the life of an oil impregnated type wet-lubricated bearing. Such a system can be more readily relubricated during periodic servicing or by feeding formulation through lubricant access holes and grooves. Simple tests will prove the validity of these methods.

3.0 Ceramic Ball Bearing Lubrication

3.1 Solid Lubrication Considerations

Much of the effort to provide a better and longer lasting wear surface between balls and races has centered around the deposition of super hard and tough thin films onto the races and the ball surface. This study presents alternatives that are suitable for both all-metal, hybrid or all-ceramic bearings. Since, especially with ceramic as a substrate, a bonding film to retain the solid lubricant material is necessary, one has to assure that the bonding layer itself has some lubricating or at least anti-galling properties. Most noble metals display suitable qualities and by being malleable can act as a retainer for the lubricating particles. Noble metal alloys can be formulated to match the thermal coefficient of the substrate, an important consideration for high temperature application. Several suitable dry-lubricants chelate readily to metal, but insufficiently to ceramic.

Therefore, special surface preparation is necessary. Since it is difficult to produce targets with a combination of precious metals and suitable dry-lubricants, and equally difficult to sputter or ion-plate from it, deposition has to be made sequentially. This technique though, provides an opportunity to achieve multi-layer coatings as explained in section 3.4.3. These coatings will accept lubricant transfer from a supply within the ball retainer, but are equally, if not more, suitable for inject lubricant replenishment.

3.2 Lubrication - Background

Solid lubricants have been available and have been used for many years. Molybdenum disulfide covers over 30% of the needs for the heavy industry and for usage at temperatures reaching from sub-zero to approximately 343 degrees C. It is a relatively plentiful and inexpensive lubricant with very good performance characteristics. It is supplied as a plain powder, mixed with aerosols, oils and greases and, in combination with suitable binders, has good adhesion to most surfaces. Other solid lubricants such as Tungsten Disulfide, in one instance marketed and applied under the name of Dicronite, has found good acceptance in commercial and aerospace applications. The latest edition of the "Encyclopedia of Chemical Technology" (R 37) by Kirk Othmer (page 507) lists a number of other commonly known and frequently used solid lubricants. Several of these have been somewhat modified and reformulated and sold under various trade names. Their usefulness at high temperatures is within a range from 150-450 degrees C in air and up to above 700 degrees C in vacuum or Nitrogen, but some of them have a relatively low load capacity within their temperature range. Requirements for solid lubricants considered under this study go beyond these parameters and are basically as follows:

1. High load capacity
2. Useful to temperatures of 815 degrees C (1500 F)
3. To be resistant to corrosive media and particle contamination nation
4. Humidity resistant
5. Replenishable
6. High lubricity (low friction)
7. Resistance to decomposition under long time storage

3.3 Materials

3.3.1 Listing of commercial solid lubricants

Dozens of commercial solid lubricant brands are available off the shelf. Most of them contain one or more of the materials listed in Table 2. Several exceed the properties as shown and most of them have a useful upper temperature range of between 250-400 C. They differ greatly in wear-ability, bonding quality, and in the coefficient of friction. Some others, such as a variety of Boron Nitride and Talcums, are missing from this list. Boron Nitride, for instance, is typically used as an additive in brake shoes to reduce burn out and "squeeking". Several varieties of the Talcum family and some rare earth compounds are also suitable for dry-lubricant applications. Most of the above, except the PTFE and FEP have been scrutinized for suitability in high temperature applications. With several materials, reprocessing and modifying of the compound resulted in an enhancement of properties such as adhesion and high temperature resistance. Better bonding normally is achieved with a secondary additive to the formulations. The amount and type of binder has a significant effect of lubricity.

3.3.2 Proprietary solid lubricant listing

The proprietary systems and formulations listed and tested under this study are only part of an effort to expand selection and performance characteristics of a series of solid lubricants with high-temperature resistance being only one of a number of important requirements..

As mentioned in section 1.3 the designation of the formulations and systems has changed from an alphabetical sequence into alphanumeric codes. The letters usually refer to undercoating and mixing elements.

Material	Acceptable usage temperature, °C: Min, In air	Min, In N_2 or vacuum	Max, In air	Max, In N_2 or vacuum	Av friction coefficient, f: In air	In N_2 or vacuum	Remarks
molybdenum disulfide, MoS_2	−240	−240	370	820	0.10–0.25	0.05–0.10	low f, carries high load, good overall lubricant, can promote metal corrosion
polytetrafluoroethylene (PTFE)	−70	−70	290	290	0.02–0.15	0.02–0.15	lowest f of solid lubricants, load capacity moderate and decreases at elevated temp
fluoroethylene–propylene copolymer (FEP)	−70	−70	200	200	0.02–0.15	0.02	low f, lower load capacity than PTFE
graphite	−240		540	unstable in vacuum	0.10–0.30	0.02–0.45	low f and high load capacity in air, high f and wear in vacuum, conducts electricity
niobium diselenide, $NbSe_2$			370	1320	0.12–0.40	0.07	low f, high load capacity, conducts electricity (in air or vacuum)
tungsten disulfide, WS_2	−240	−240	430	820	0.10–0.20		f not as low as MoS_2, temp capability in air a little higher
tungsten diselenide, WSe_2			370	1320			same as for WS_2
lead sulfide, PbS			480		0.10–0.30		very high load capacity, used primarily as additive with other solid lubricants
lead oxide, PbO			650		0.10–0.30		same as for PbS
calcium fluoride–barium fluoride eutectic, CaF_2–BaF_2	430	430	820	820	0.10–0.25 above 540°C; 0.25–0.40 below 540°C	same as in air	can be used at higher temp than other solid lubricants, high f below 540°C
antimony trioxide, Sb_2O_3							high load capacity, used as corrosion inhibitor in MoS_2 lubricants

reference (37)

Commercial solid lubricant materials

Table 2.

3.3.3 Solid lubricant testing

In general, testing of solid-lubricants is similar to testing of wet lubricating systems and includes the following methods:

1. Four ball test

2. Rub shoe test

3. Inclined plane test

4. Ball point linear friction test, and

5. Sleeve ball and roller bearing tests of various design and configuration.

From the above testing methods, the rub shoe test (Figure 6) provided a convenient system to simulate conditions as encountered between the ball retainer and the ring land it is in contact with. Speeds and loads can easily be changed and the parts can be used repeatedly. The rub shoe test is also very convenient for testing dry-lubricants to be used in conjunction with sleeve bearing applications. In this case, conditions are very realistic since loads per area unit and linear speed can be simulated to actual values.

Resistance to particulate contamination and chemicals is important. With the advent of ceramic bearings the lubricants are, and will be, subjected to considerably higher temperatures. Resistance of the ceramics to most chemical solutions will favorably affect the usefulness of solid lubricants with similar characteristics. Though most solid lubricants will work as well with metal, adhesion (bonding) to a ceramic substrate will present different problems than the absence of a generous supply of lubricant normally provided with a wet system.

Elliptical deformation at the ball and race contact area, graphically shown in Figure 7 and 8, also does affect lubricant testing and performance. This deformation is dependent on several factors and differs substantially between metallic and ceramic bearing material. Assuming the graphics represent an all steel bearing, a ceramic bearing will have a smaller contact area and, for the same load, will have a steeper load distribution. This is due to a higher Young's modulus, and other significant differences in physical properties of the ceramic material; generally resulting in an increase of Hertzian contact pressure. Partial compensation for such changes can be achieved by changes in the conformity ratio. The prudent approach is to increase the size of the bearing and/or the number of balls. All along one has to keep

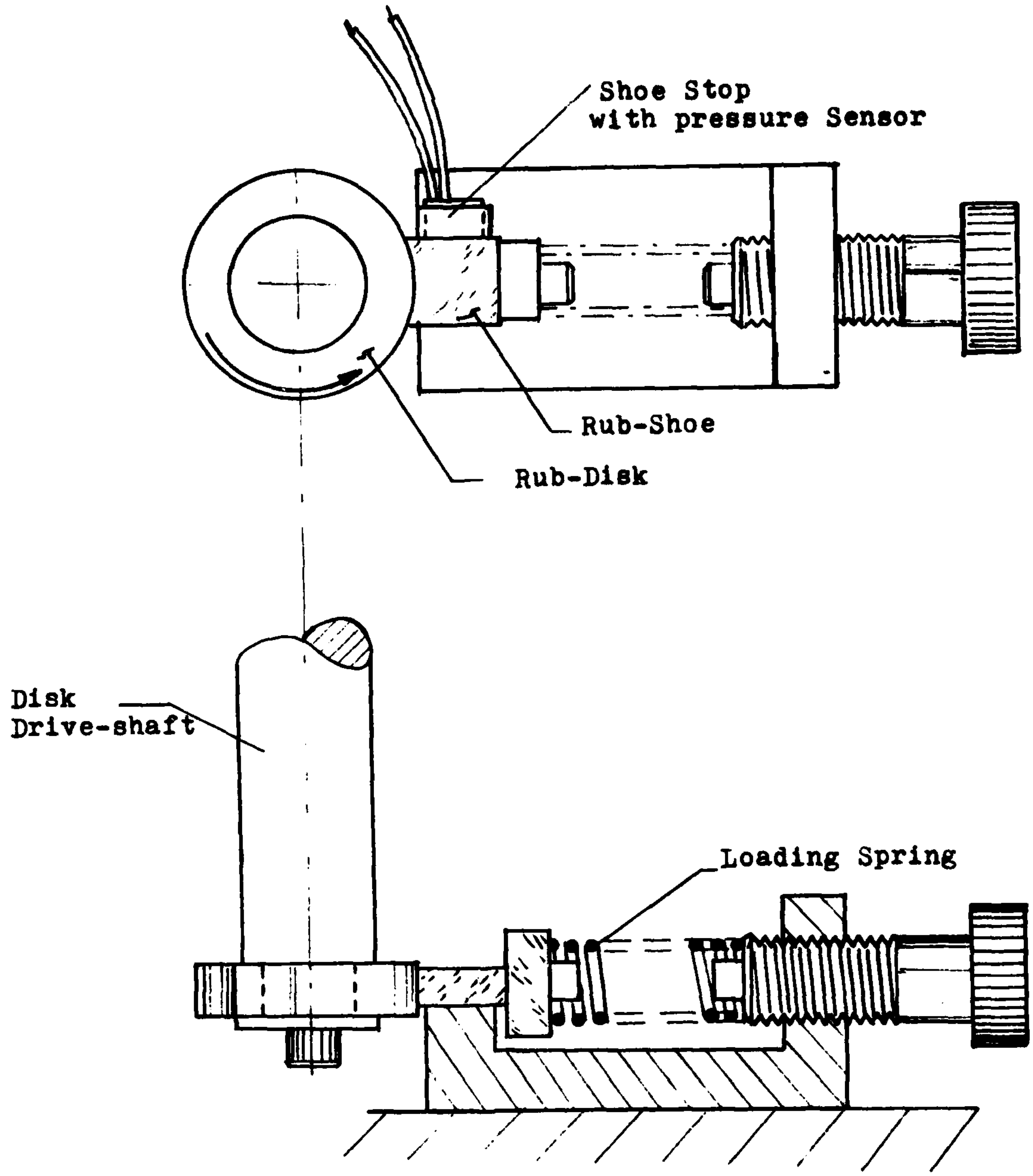
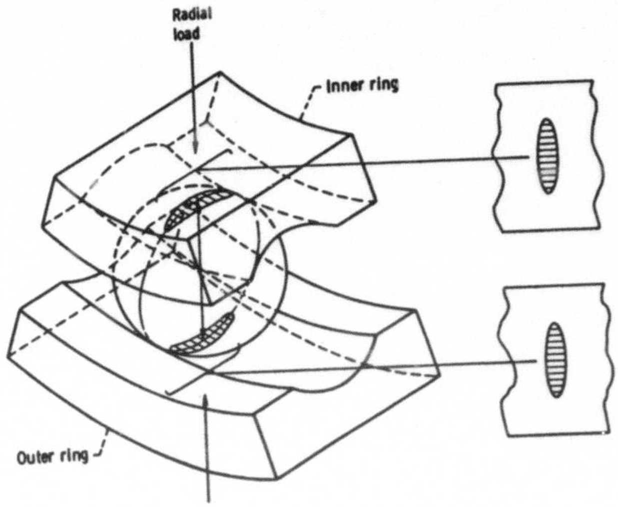


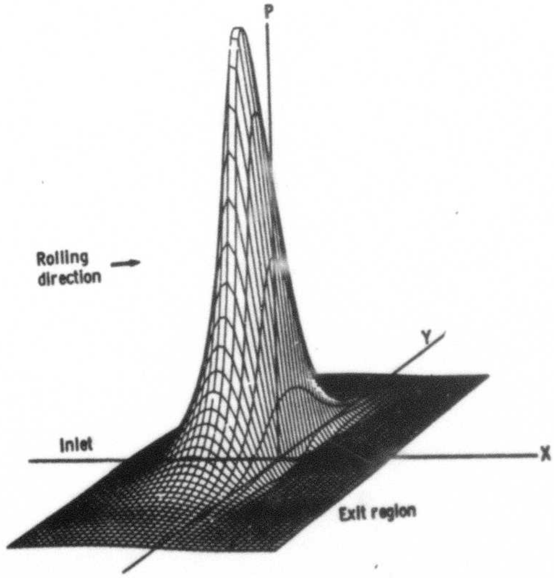
Figure 6. Rub-Shoe Test Set-Up

Figure 7.



Elliptical-Contact Deformation

Figure 8.



Three dimensional representation of pressure distribution as viewed from exit region.

in mind the substantial difference in the characteristics of solid lubrication and the hydrodynamics of wet lubrication. (R34)

Combinations of conditions such as high speed, high vibration, axial and radial load, high temperature and extended life is difficult, if not impossible, to simulate. Thus,in the final analysis,the proof of performance has to be provided with the actual hardware, proper test equipment, and close simulation of realistic working conditions.

3.4 Solid Lubricant Films and Coatings

3.4.1 Thin Films

Thin films (R38) in ball-bearing technology are typically applied to reduce friction, enhance hardness and wearability of ball and race surfaces and, in some cases, improve bonding with lubricants as transferred from the lubricating retainer. Within the applied techniques of this study the primary purpose of the thin film is to act as the primary bond to the solid lubricant formulation. Precious metals and precious metal alloys seem to be most suitable but have to be properly selected for specific substrate materials and for the type of lubricant to be applied. Concepts for three typical kinds of coating are shown in Figure 9.

3.4.2 Thick film

In this report thick film coating is defined as a film with a thickness of .0005 - .001 mm. The film can be a layer of pure material or a mix or formulation of different materials. Adhesion may be by a suitable blended binder or through a thin layer acting as a binder. In the latter case bonding is usually achieved by a baking process. The original thickness of the lubricating film can be substantial, .00050-.001 mm. The suitable thinner layer is obtained by burnishing methods. A micro-photograph of such a film is shown and explained in Figure 10. This particular sample, bonded and baked to a sputtered undercoat of silver, has survived temperatures of 870 degrees C and has been tested with an equivalent load of 25kg/mm sq.

A transferred distribution of the 2-14-S formula is shown on the surface of a steel bearing ball after approximately 5 hours of operation, this with a load of 15 kg per ball. (See Figure 11.) Performance optimization of the thick film is dependent on the following:

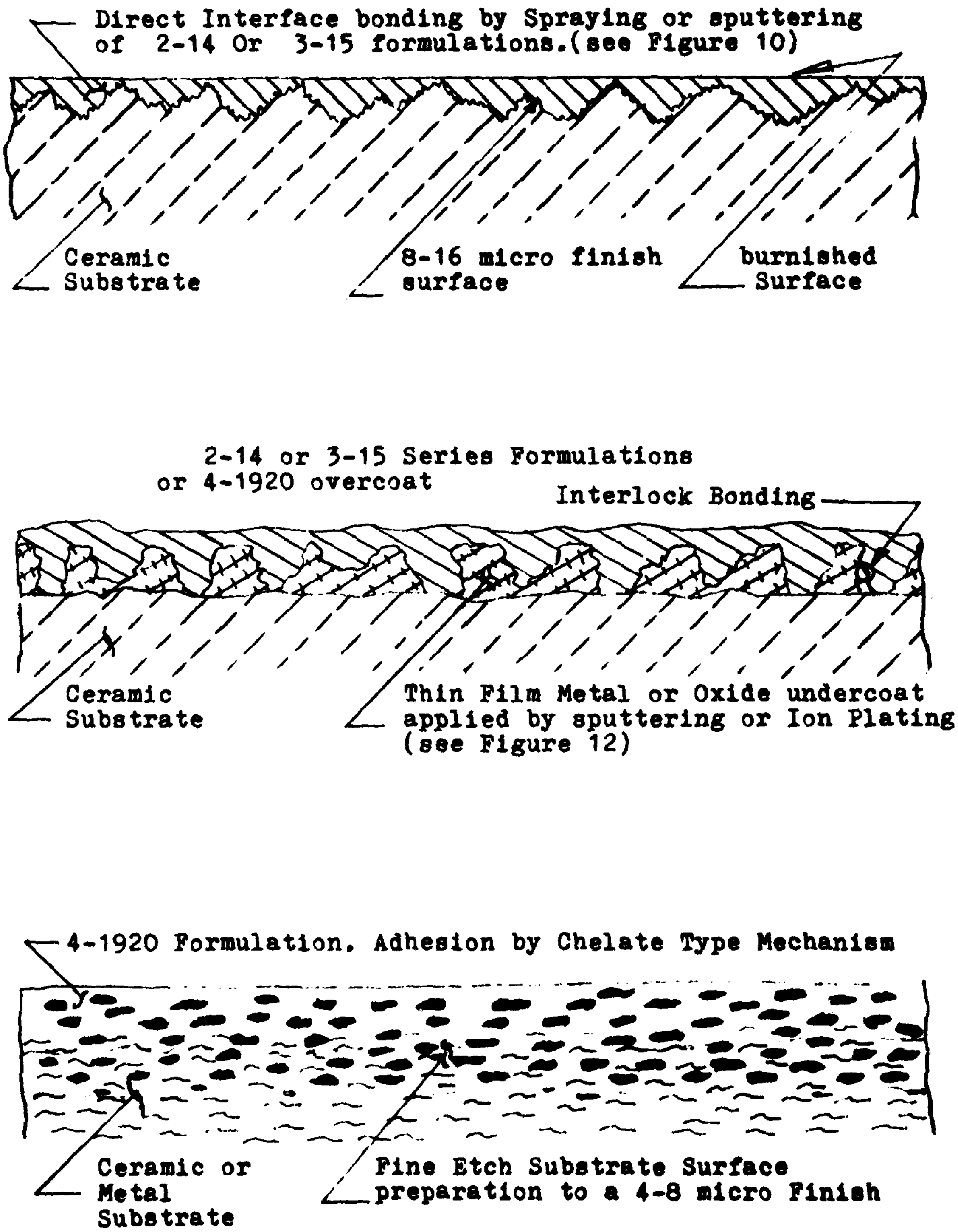
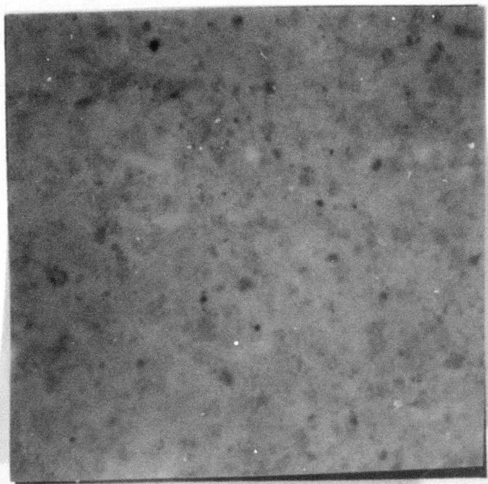


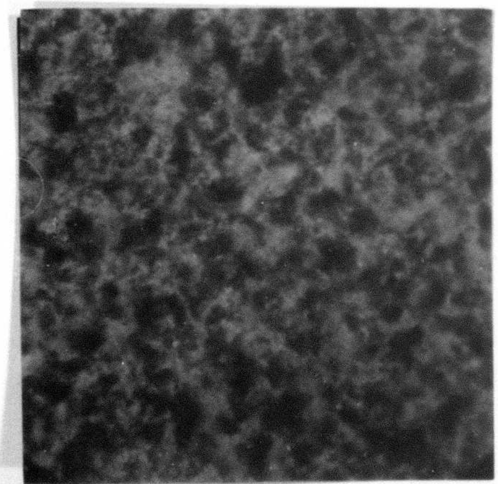
Figure 9. Solid Lubricant Systems / Cross Section Diagrams

Figure 10. 2-14 Formulation Coat on Alumina Al_2O_3
(16 micro finish surface)



100X

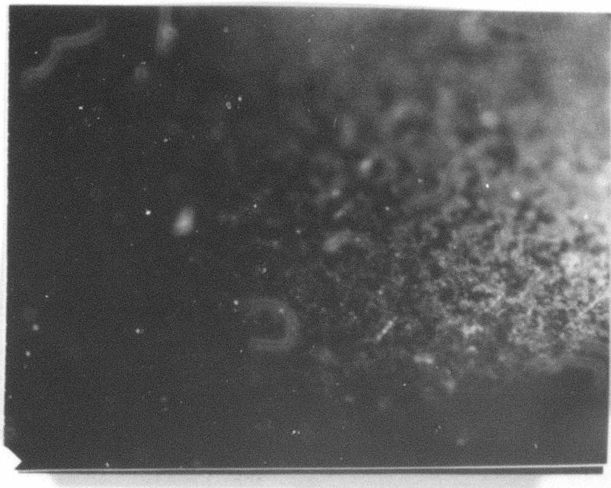
Uncoated Substrate showing Vitreous Binder Distribution



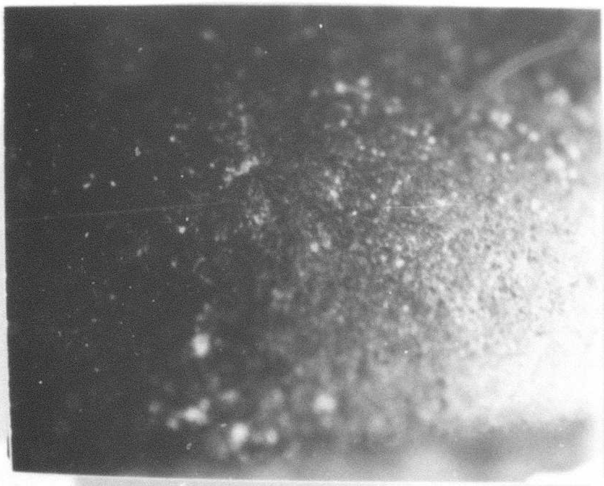
100X

Dark Field Illumitation

Substrate coated with 2-14 Formulation



Metal Ball Surface
with transfered
2-14 Solid Lubricant



Metal Ball Surface
with transfered
2-14 Solid Lubricant
including Phenolic-
Retainer Particles.

Figure 11. Ball Surface Photomicrograph

1. Substrate surface finish 4-8 micron

2. Substrate preparation by acid etching

3. Hardness and toughness of the binder coating

4. Formulation selection

5. Formulation thickness.

The thick film is an ideal lubrication method for sleeve bearings but also works well for ball bearings running at moderate speeds and loads and up to temperatures of 815 degrees C. Re-lubrication under running conditions is readily achieved.

3.4.3 Multi-film coating

A theoretically promising and partially tested method to build up more resilient layers with different characteristics, is by the application of multiple sandwiched layers of dry lubricants. (See Figure 12.) The major problem to accomplish the build-up is to obtain adequate bonding between the layers and to deposit them in the best performing sequence. Since we are applying the multiple coatings on the races only, and since eventual lubricant transfer from the races to the balls will occur, and is essential for proper bearing performance, one would like to see the coating of the ball by reverse lubricant transfer from the races. Lubricant supplied from the ball retainer should also be a contributor. It is desirable to eliminate the necessity for coating the balls. There are indications that priming a film transfer from the ball-cage is quite effective and can be achieved by a run-in operation.

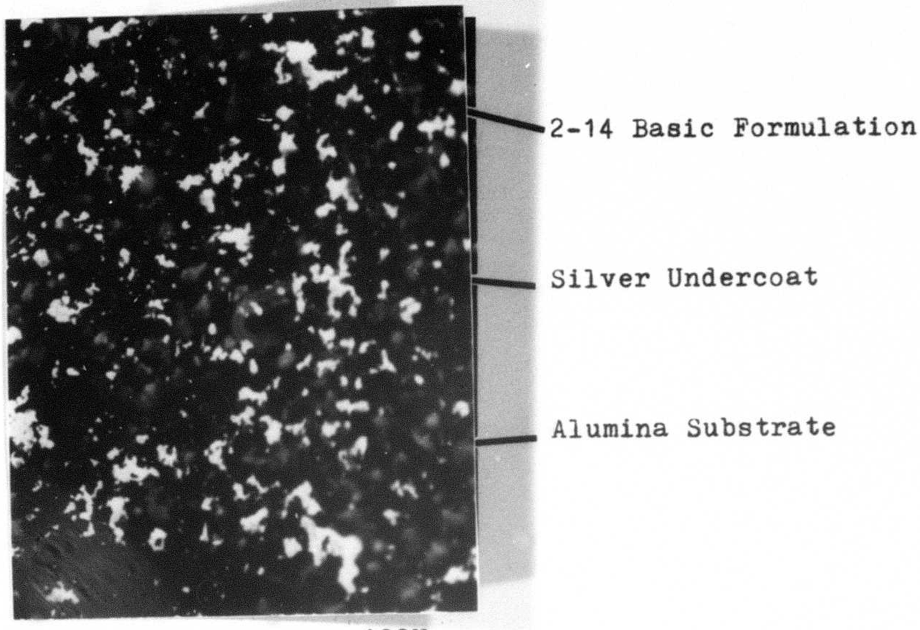
The multi-layer concept will reduce dependency on the ball retainer as a lubricant reservoir, and for life-limited applications, will simplify retainer design and fabrication.

3.5 Lubricant Bonding

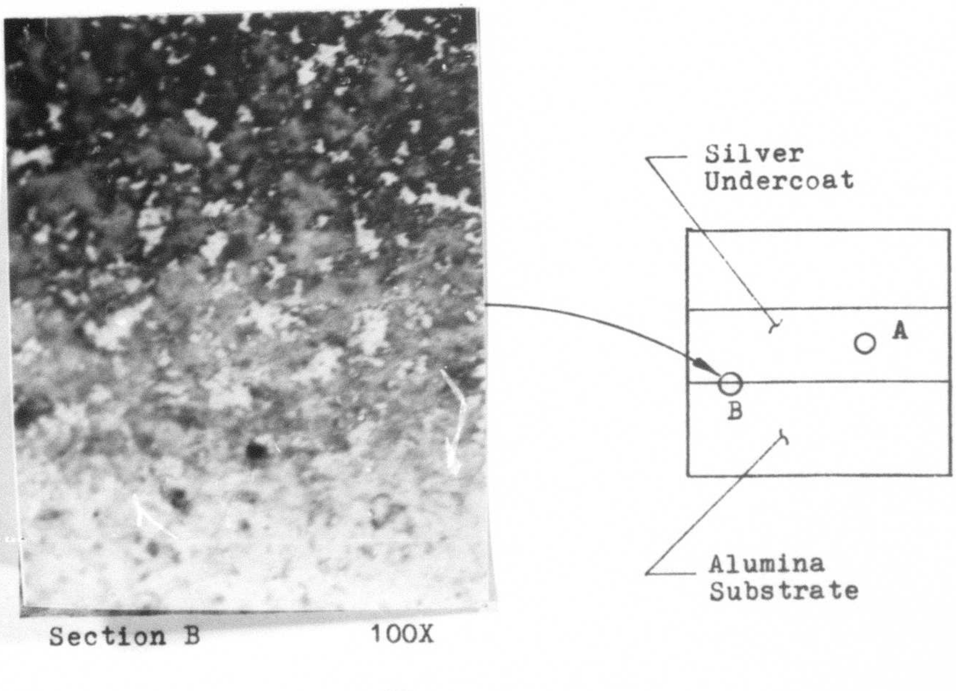
3.5.1 Interbond

Most commercially available lubricants are available in aerosol containers. This includes molybdenum disulfide, various grades of PTFE and boron nitride sprays. The sprays are formulated to provide interbonding for the lubricant and at the same time provide adhesion to the coated surface.

Figure 12. Formulation 2-14 on Silver Undercoat deposited on Alumina Al_2O_3 Substrate



Section A 100X



Section B 100X

Binders commonly used for such applications are aluminum phosphate, magnesium silicate (Talcum) and coloidal alumina. Others are proprietary formulations containing reactive ingredients to strengthen the bond on specific substrate materials such as metals or plastics. Metals that readily form surface oxides under normal ambient conditions generally provide a better substrate bond for the lubricant.

3.5.2 Bonding especially as related to ceramic substrate materials

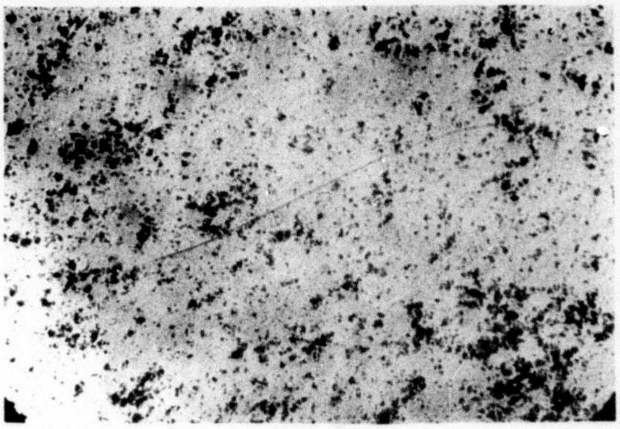
Most ceramic materials suitable for bearing applications are difficult to bond to. Special surface preparation (fine etching) and the deposit of intermediate films are necessary and is a technology in itself. Special attention needs to be given to the quality of the substrate ceramic material. Inclusions and/or pores result in crater formation after deposition of the metal undercoat. (See Figure 13.) In contrast to microcircuitry applications, films for bearing usage have to withstand continuous high temperatures, high concentrated loads, and severe wear and abrasion. In addition, bonding layers also must have lubricating properties and "fuse" to the solid lubricant layer. Experience has shown that Silver is a very suitable metal for this application, though more research is needed to improve the interbond between Silver and Silicon Nitride. Silver alloys and other noble metals may help that problem. Ion-implanted metals into the immediate substrate sub-surface should be seriously considered under the Phase II effort.

It seems to be a logical step to intermix some thin film material with the lubricant formulation. Tests and experiences have shown this to be very helpful. The addition of thin film material to the relubrication formulations will also benefit from this approach.

Tests and experiments have shown that when exposed to temperatures in the 980 degrees C (1800 F) range some solid lubricant or lubricant additives such as Boron Nitride react with metals such as Nickel or Platinum. Lead Oxide behaves in a similar manner and we feel the range of possibilities has not been fully researched. This interaction and reaction can well be used as a binder technique.

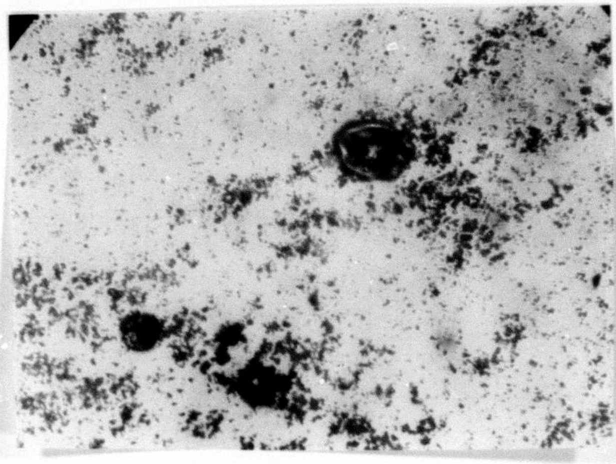
Binding and adhesion can also be obtained by ion plating or sputtering methods. In the latter case the target composition becomes of prime importance. Discussions with sputtering target experts indicate that unless produced and tested, target behavior during the sputtering process is difficult to predict. (R36)

Figure 13. Ion Silver Plate on Silicon Nitride



A

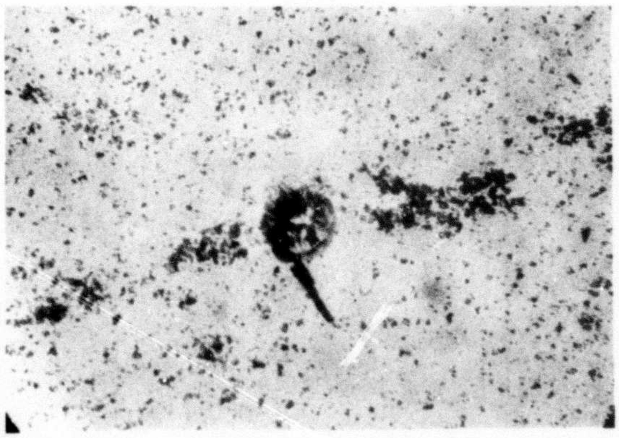
Silver on
on Norton NC 132



B

Craters due to
Substrate Porosity

Silver on
HPSN- Toshiba



C

Silver Coat defect
due to material
Inclusions

Silver on
HPSN - Source unknown

Some materials including Tungsten disulfide and metal free phthalocynamine (see Appendix C, p. 3) bond themselves to metal by a chelate or similar interaction (R5, 6, 10). Tungsten disulfide (under the trade name of Dicronite) deposition is presently used as a solid lubricant by several ball bearing companies and finds ready application to lubricate gears, pistons, valves, screws, bolts, etc. Its present drawback is its temperature limitation of 468 degrees C. Reprocessing of Tungsten disulfide by the addition of 2-14 formulation gives an encouraging indication of no loss of adhesion (eutectic processes) (Figure 14) and will boost its temperature limit by a considerable amount, possibly exceeding the 750 degree C limit. One of the significant advantages of Tungsten disulfide and related coatings is its self-controlling thickness which amounts to an average of .005 mm. The other advantage will be its usefulness for injection re-lubrication purposes since the particle size is in the submicron range. The discovery of these and other interesting characteristics of these formulations came late into Phase I study. We recommend further thorough investigation during the Phase II program.

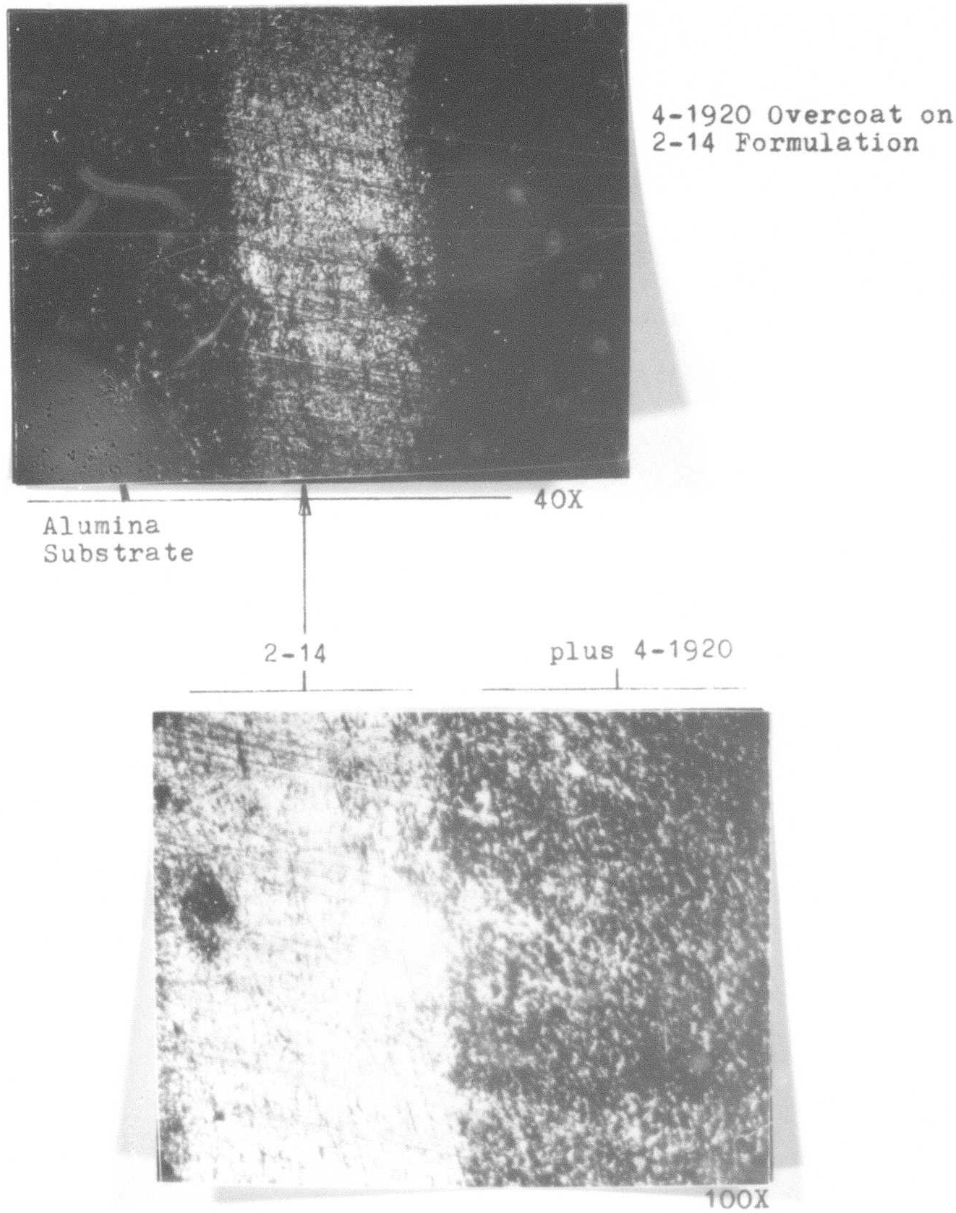
Out of several materials investigated for bonding properties, some of them were indicated to be "neutral" as far as lubricating qualities are concerned. Neutral, in this sense, is having no particularly low friction quality but on the other hand, exhibiting no galling under load. Several oxides including lead oxide, and indium oxide,fall into this category. The absence of a truly lamellar structure is probably the cause of it, but in mixing it with solid-lubricant materials at a 1:10 ratio the performance is remarkably good.

3.6 Solid Lubricant Replenishment Methods

3.6.1 The ball retainer as a solid lubricant reservoir

Within this report much reference is being made to the need and ability to replenish lubricants. Re-lubrication is needed to prolong bearing life and to increase bearing efficiency by reducing temperature build-up which is generated by ball micro-slippage and due to friction at the retainer and race-land interface. Presently the ball retainer is the main source of additional lubricant. Other methods of re-lubrication are needed and some concepts are described in the following:

Figure 14. Formulation 2-14 on Alumina(Al_2O_3)Substrate covered with Formulation 4-1920 Overcoat



3.6.2 Lubricant injection methods

By using a ball retainer concept as shown in Figures 3 and 4 and as described in section 2.5.3, a method to inject a gas and finely dispersed solid lubricant mixture onto and into the ball-bearing components becomes a practical and feasible concept. (Figure 15) The optimum injection angle needs to be determined for each individual application. What is important, however, is that the injection velocity be roughly equal to the linear rotational speed of the bearing.

Since exit turbulence at the injection nozzle-tip will, in all probability, be substantial and unpredictable it is very helpful to put a rotational "twist" to the ejected air and lubricant mixture. Work done some years ago, to control and stabilize the jet, resulted in a performance as demonstrated in Figure 16. The picture is a high speed photograph in Schlieren projection and shown in roughly actual size. Without the twist flow, heavy dispersion would normally occur a short distance from the nozzle tip. The techniques to achieve the twist are well known. One must, however, be aware that conditions are heavily dependent on pressure and on the type of carrier media plus particle size and mass of the injected material. It can be assumed that for proper performance two of three nozzles around the bearing periphery will be adequate. Since the system is designed for high temperature it will be worthwhile to consider making the injector tip of ceramic material.

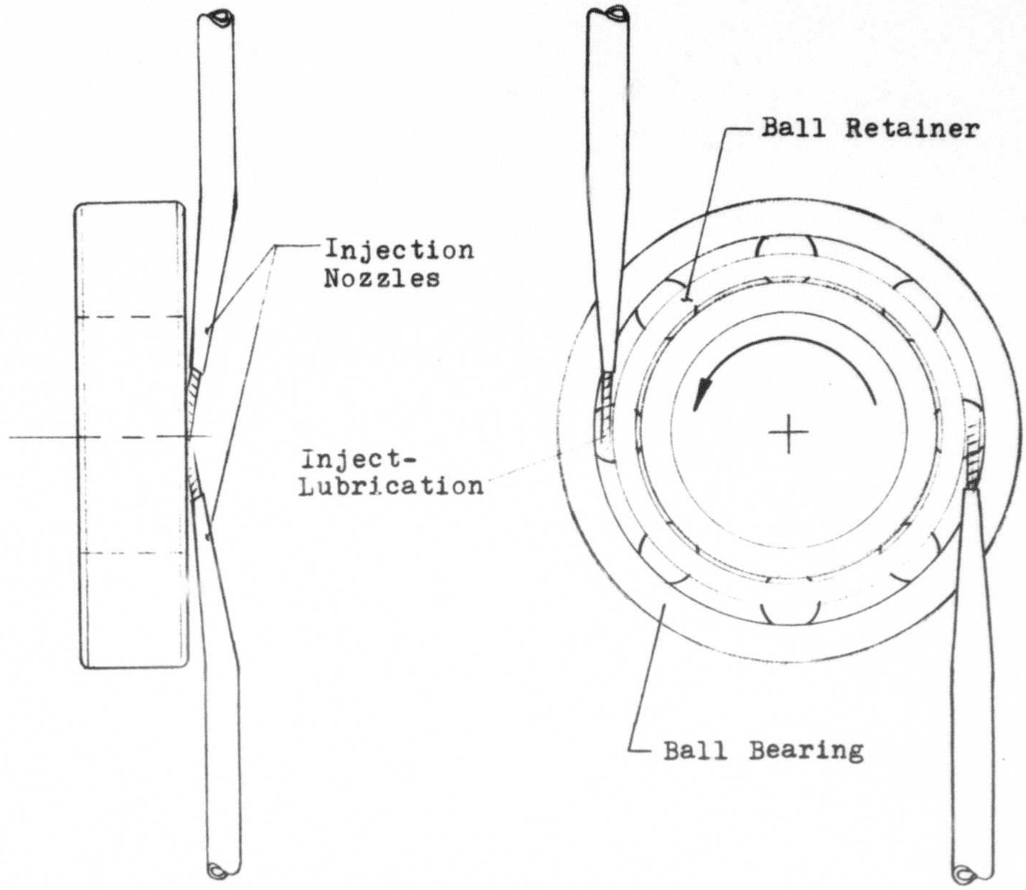


Figure 15. Inject Lubrication Method

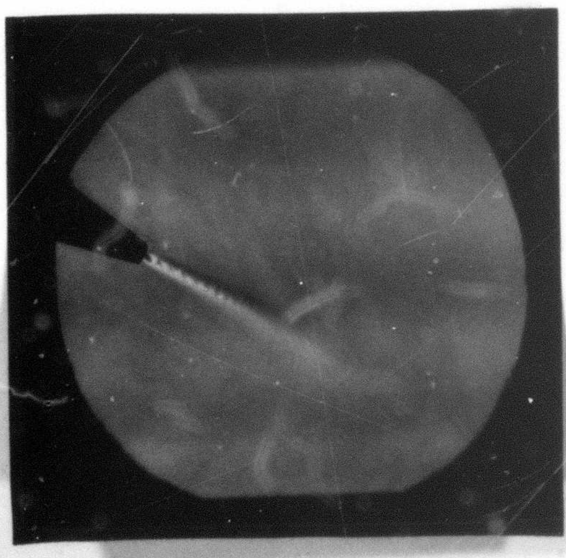
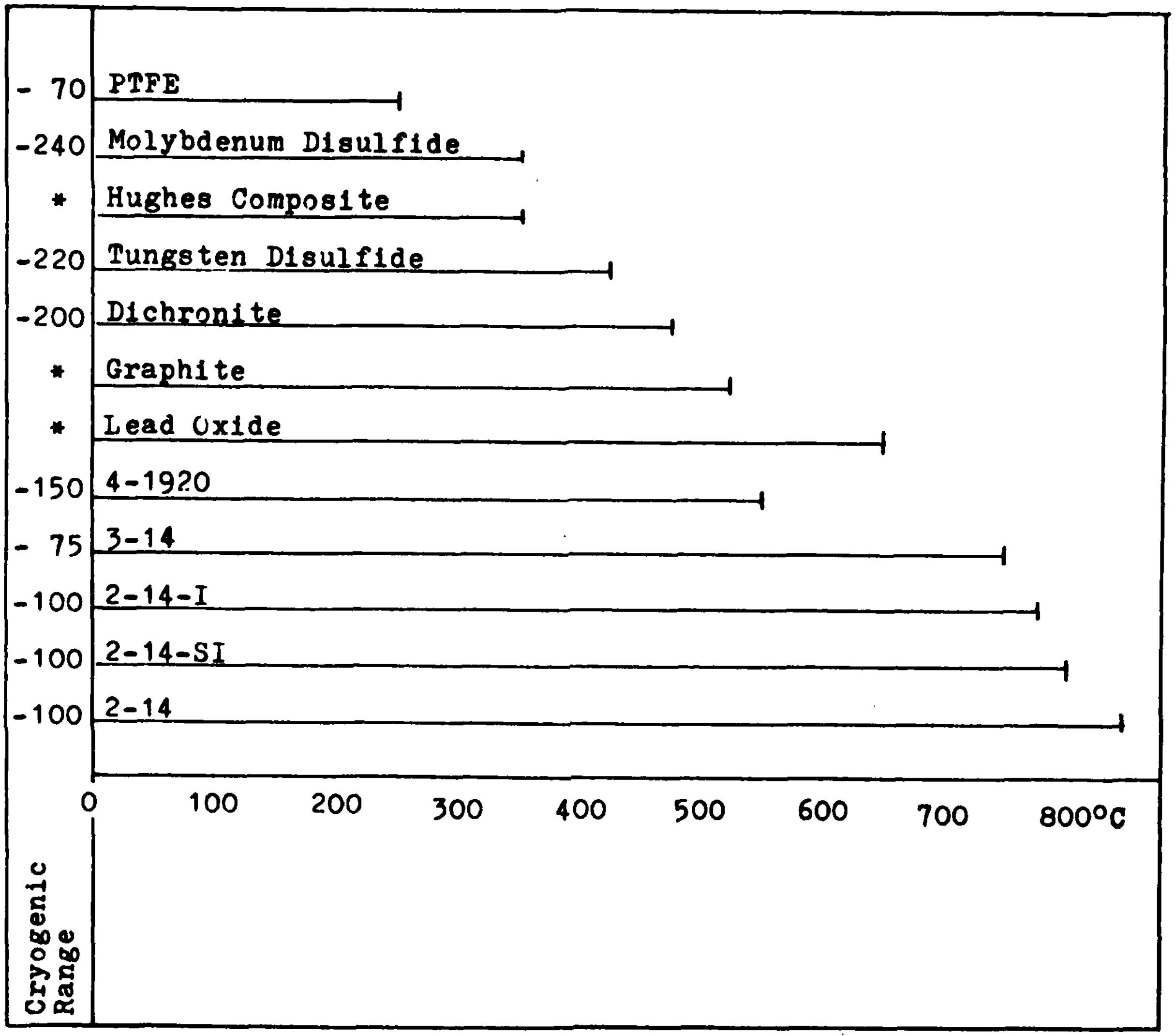


Figure 16.
Injection Twist shown
in Schlieren Projection

3.7 Temperature Range of Commercial and Proprietary Solid Lubricant Formulations

Table 3.



Notes: 1. Dicronite is a modified Tungsten Disulfide
2. Series 2,3 and 4 do not contain any PTFE or Molybdenum Disulfide
3. PTFE is listed for reference only

3.8 Solid Lubricant Resistance to contaminants

3.8.1 Particle contaminants

It is a well known fact that wet lubricated ball bearings show a considerable resistance and tolerance to contamination of all sorts. Many cars manage to make a final destination by the "throwing" of some penetrating oil and lubricating oil into a frozen fan bearing or idler pulley, etc., or a 10-speed bike has been fixed by spraying the gears, chains and linkages with a "Molly" aerosol.

Both of the above fixes indicate a difference in the lubrication approach and a difference in lubricant behavior. In the first case not only dust but also the temperature of the car's engine had contributed to the bearing failure. In all probability the bearing is heavily damaged but the introduction of an oil film will lower the friction and subsequently the temperature for a limited period of time. In the second case, all original lubricant was well mixed with the road dust and after a rain or a few cleanings of the bike most of it had pretty much disappeared. A dry Molly spray, especially in corroded areas, will adhere rather well and after a run-in and a re-spray, operational life and performance of the re-lubricated parts is rather remarkable. New contaminants, such as grime and dust have less tendency to adhere.

Tests with used ball bearings have provided interesting information and results. After cleaning all bearing elements, races, balls, and retainer were "dusted" with various types of dry lubricants. In one case, a ball bearing with rather severe corrosion damage, and pretty much frozen by corrosion debris, was washed out with a corrosion inhibiting solution, rinsed in alcohol, and dried. Subsequently it was dusted with formula 2-14 solid lubricant while slowly rotating. After a short run-in period the bearing was spun at 3000 rpm under an axial load of approximately 50 kg. for a duration of 20 minutes. Inspection under low magnification evidenced two major facts:

1) rough corroded areas on balls and races were pretty much filled with lubricant material and the lubricant was burnished down to the basic contour of the surfaces.

2) some excess lubricant that had worked out of the ball retainer showed traces of residual rust that had been previously embedded in the retainer.

After repeated high load runs the bearing kept on performing well.

This example does not mean a final solution to the solid-lubricant performance problem. It does, however, establish a trend in the right direction and would indicate that preventive service (re-lubrication) not only could prolong bearing life by a substantial amount, but also gives an indication that, with the proper dry-lubricant, a certain tolerance to contamination can be obtained. In order to substantiate this claim a brand new bearing was lubricated by the same method using formulation 2-14-S. This formulation shows good adhesion to substrate metal and ceramic materials. It has been demonstrated that by a run-in sequence a protective and lubricating coat can be formed on the ball and race surfaces.

Subsequently, a small amount approximately .5 gm. of granite flour was added to the formula for the re-lubrication sequence. The bearing was then run under previous conditions. Much of the contamination problem originates from fine dust during extended storage and also from field service. Although total protection from particle contamination is difficult to achieve, any amount of resistance of the lubricant to such contaminations will be beneficial.

3.8.2 Moisture and Chemicals

The moisture (water) absorption of the dry-lubricant base material before it is processed and treated into the 2-14 formulation is in excess of 2% weight gain within 170 hours and at 25 degrees C, and at 80 - 100% relative humidity and is only slightly lower in the finished formulations. Bake-out at 175 degrees will reduce it to about .4%. The addition of binders and various other additives have, so far, not changed this undesirable characteristic. The real effect on the performance needs further study. Further work with the formulations will eventually reduce water absorption to a negligible level.

The basic formulation of 2-14 has a very low oxidation rate in air with a weight loss of less than .005 mg/cm2/hr up to a temperature of 700 degrees C. Above this temperature the rate will increase and reach about .01 mg/cm2/hr at a temperature of 1000 degrees C. These rates are dependent on the percentage of additives in the mix and can be further decreased by the use of small quantities of precious metal alloys and base metals.

A similar effect is achieved in regard to resistance to chemicals such as hydraulic fluids, solvents, alcohols and low concentration acids. Normally the baking-out of such chemicals does not seem to affect performance of the lubricants, though it is best to test for it. Some additives affect the lubricating qualities to some degree. In such a case a multi-layer system (section 3.4.3) ought to be considered. Besides adding to the thickness, a deteriorated layer will, after burn-off, have one or more back-up layers that will last until re-lubrication or supply from the retainer will take over.

4.0 Tests and Test Methods

4.1 Material Test Criteria

Properties and characteristics, as described in manufacturer's sales literature, determine the first line selection for a high temperature ball bearing material.

The ideal material will have:

High mechanical strength at elevated temperatures,

Excellent resistance to thermal shock,

Good thermal conductivity and low thermal expansion, and

High corrosion resistance.

Some of the high grade ceramic materials exhibit some or/all of these properties.

The following steps consist of the verification of published data by testing for the specific requirements associated with high speed and high temperature bearing technology. Information as to which is the best method to fine grind and polish such materials is essentially unobtainable. So is information on how to metalize such surfaces. Little is known about the wear and frictional performance of material combinations such as zirconia against silicon nitride. The list for a need of answers is a long one.

As explained in section 2.5.1, the need for different and superior ball retainer materials is urgent; also methods for better adhesion of solid lubricant formulations. The general requirements for properties usually follow the ideal material properties shown above.

It is obvious that test methods such as the rub-shoe test described in Section 3.3.3 are of great value to predetermine the characteristics of bearing and retainer materials, and to a degree, their inter-reaction with solid lubricants. Within this study, we found the rub-shoe test very helpful to test adhesion of lubricant undercoats and overcoats; also friction-testing commercially available solid lubricants. Since the rub-shoe presents a proportionally smaller contact area equal to: Rub-disc diameter divided by length of shoe curve, it is desirable to have the shoe as the test specimen.

4.2 Bearing Test Equipment

4.2.1 Test-Rig / Phase I Study

Testing was accomplished with a simple self-aligning set-up shown in Figure 17. The lower spherical end of the drive shaft rotates the upper race by frictional contact within a matching cavity machined into the adaptor plate which is seated on the upper race ring. The bearing itself is held within a shallow recess of the bearing support plate. The main support plate made of insulating material will also accommodate the heated chamber during elevated temperature testing of the bearings.

Thermocouples within the chamber allow for measurement and control of oven temperatures. A separate sensor which is in contact with the underside of the lower race permits monitoring of the lower race temperature in close proximity to the ball track. The above assembly is installed on the table of a Rockwell model 15 drill press. Four speed ranges are provided by a stepped diameter pulley drive.

4.2.2 Test-Bed Ball Bearing

In order to subject the various solid lubrication formulation to realistic loads and environmental conditions the Mindrum/Helvart team decided during the proposal effort to obtain a fully ceramic ball bearing. Both cost and delivery quotations went beyond the original estimates. After award of the Phase I study program it was decided the bearing was essential to our effort. (See Figures 1 and 18) The manufacturing of the ceramic balls and races was subcontracted through the services of Tribology Consultants, Inc., Paoli, PA. The special interchangeable ball retainers were produced at the Mindrum Precision Products facilities. Inclusive to the subcontract is a report by Lew Sibley presenting a review of past and present ceramic bearing and solid lubricant technology. (See Appendix C.) The report also includes a further description of material characteristics of the test-bed bearing. A critical review of the report in the light of our own findings is presented in section 4.4.3.

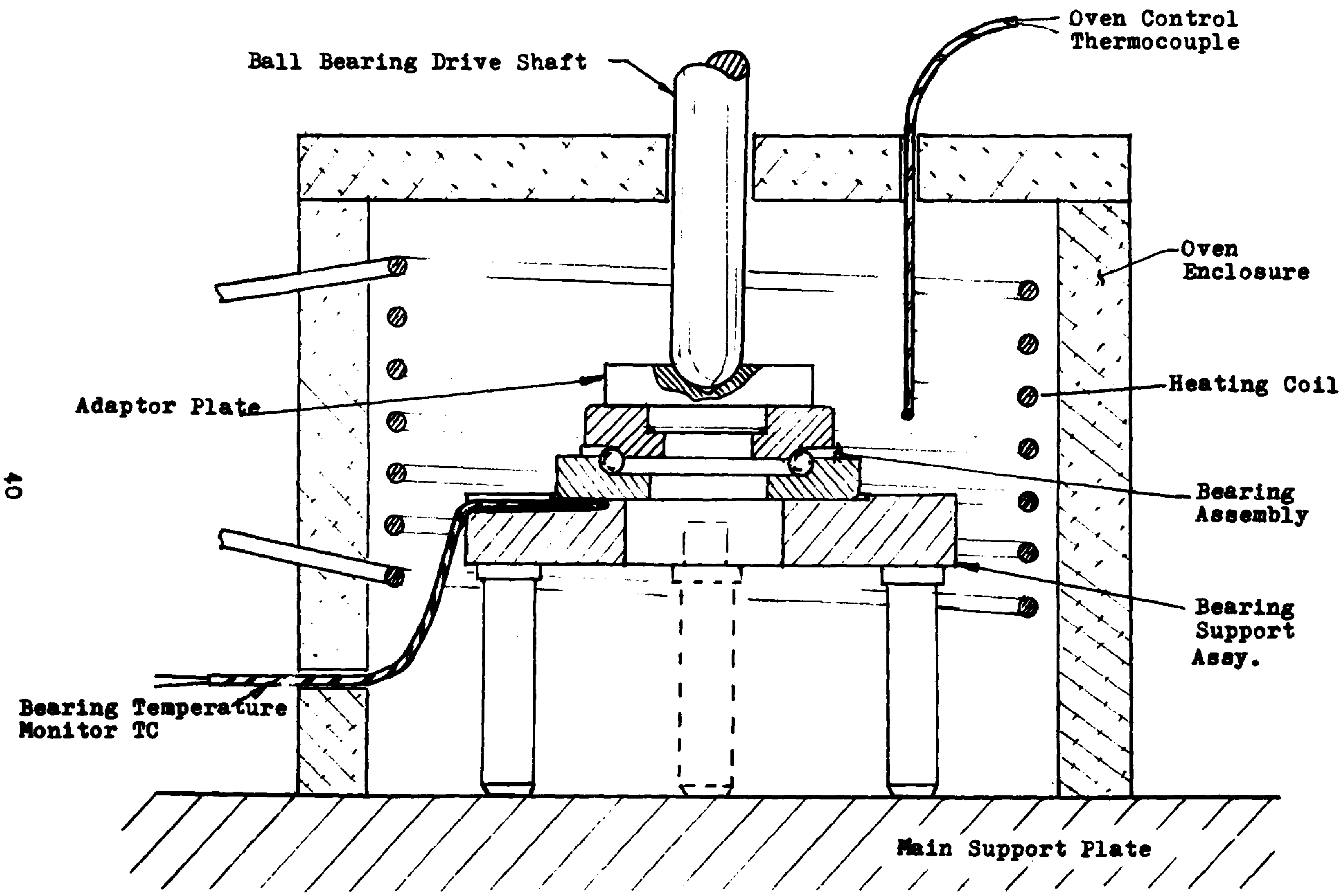


Figure 17. Test-Rig / Phase I Testing

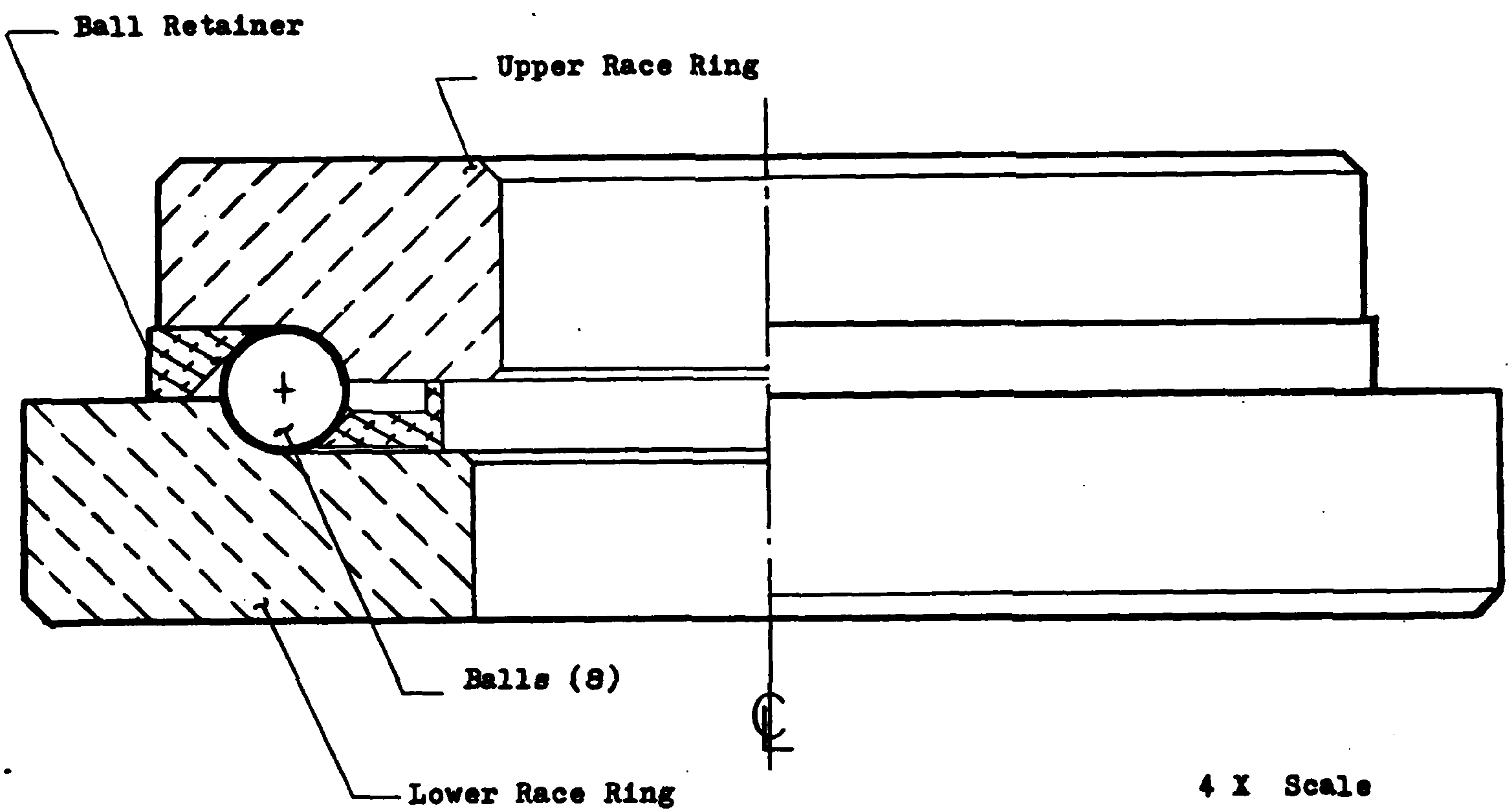


Figure 18. Test-Bed Bearing Assembly

Despite its high quality the bearing is not meant to prove out a new design but, rather, to be a test vehicle to test performance and behavior of the solid lubricants. All parts within the bearing are readily replaceable, however, in the case of the balls it is advisable to replace all eight of the set. The tests were performed in a custom built but basically simple set up as described in section 4.2.1. Its only drawback is the lack of high rotational speed capability which in turn precludes full testing of the ball cage quality. Other high speed ball cage concepts are described in paragraph 2.5.3 of this report. Due to funding and time limitations, the manufacturing and testing of these and other concepts will have to be executed during the Phase II program.

The test-bed bearing is basically designed to accept thrust loads which best match the characteristics of the test equipment. The races were designed and manufactured with an actual contact angle of approximately 38 degrees and will accept substantial loads. Since the tests are performed inside a heated chamber, distortions and misalignments of the insulating drive shaft and the set up itself will exert a certain amount of radial loading too.

The bearing (similar to a regular thrust bearing) is easily disassembled for the purpose of checking retainer lubricant and ball and bearing race conditions. "Flow", adhesion or other condition of the lubricant can readily be observed and photographed under a microscope. We think this to be an all important feature. Results of these observations are presented in Appendix A.

All tests performed at this time are done at very moderate speeds not exceeding 2000 rpm. It must be noted that the safe load-carrying capacity of ball bearings is reduced as speed increases and it suggests that this behavior is probably related to the known effect of very rapid reversals of stress. Hertzian contact pressure as applied by the thrust loads have been computed not to exceed 280 KSI.

The original intent to manufacture the balls and races from dissimilar ceramic materials to better frictional properties (see Section 4.1) had to be abandoned because of cost and material procurement problems. The test bed bearing, however, consists of different grades and manufacture of Silicon Nitride. The balls are Norton NC 132. The outer ring is a standard grade Toshiba HPSN and the inner race is manufactured from a high grade Toshiba HPSN. The observed differences of this arrangement are described in Appendix A.

4.2.3 Test equipment for Phase II

For reasons of expediency, cost, and availability, and in general to keep within achievable goals, tests for the Phase I study were performed by relatively simple methods. the most complex equipment is the test-stand described in section 4.2.1. No doubt, interim or Phase II work will require more advanced and sophisticated test gear. As evidenced from government and commercial industry reports, numerous equipment has been designed and manufactured to test metal and ceramic bearings. Under the assumption that we will proceed with Phase II, we found it worthwhile to make a survey for suitable and available equipment within a convenient distance from our location. The use of such facilities would be preferable to our designing and constructing a new test-rig or equipment.

Based on this survey we found suitable facilities and capabilities at SOLAR TURBINES INC., in San Diego, California. Their expertise in turbine technology makes the company an attractive choice. Bearing investigations have been conducted under both government and company sponsored projects. A number of current computer programs are specifically dedicated to bearing technology. Equally valuable are their capabilitiy, facilities, and equipment to perform material evaluations to obtain surface reaction characterization. A closer description of SOLAR is presented in Appendix B.

4.3 Assembled Test-Bed Bearing Tests

4.3.1 Bearing Lubricated / Metal & Hybrid Bearings

The decision to manufacture and to use a ceramic ball bearing as a test-bed for solid lubricant formulations did not eliminate the necessity to evaluate the test equipment. In order to not unduly risk the valuable ceramic bearing it was thought best to use metal bearings of comparable size for the set-up and run-in period. This exercise proved to be a very worthwhile addition to the testing program, for it provided data about solid lubricant formulation behavior in relation to plain untreated metal substrates. It also gave us an indication of the beneficial effects of re-lubrication which, at this point, was performed with simple methods with both wet and dry mixes. The approach and results are listed in the test log as part of Appendix A. The test bearings were a set of SKF 7206/CP4 DGA angular contact units with a phenolic ball retainer. The open area between the ring and the retainer provided a viewing port to observe lubricant wear and distribution and made it easy to "dust" all components as needed. The monitoring of the outer race temperature under varying speeds and loads provided a delta value-base that permitted a degree of compensation for changes in ambient and oven temperature. These set-up tests also gave emphasis to the need for a run-in period of a dry lubricated bearing. It also was made evident that the phenolic retainer made a very good substrate for the lubricant formulations. The phenolic material, unfortunately, has serious temperature limitations below our goals and this, in turn, limited the test runs. Formulations, at the beginning, were applied onto the balls and races without a binder. The coatings did an excellent job. From the data it is obvious that the lubricating qualities of the formulations even without the benefit of a binder undercoat were substantial.

4.3.2 Lubricant replenishing in metal test bed bearing

Early tests were conducted under the assumption that an impregnated phenolic retainer within a metal bearing would hold a fair amount of lubricant reserve and this provided some significant information as to the efficiency and the need for lubricant replenishment. To establish a baseline, one bearing was run completely unlubricated. It resulted in immediate unacceptable noise and drag and, if continued, would have quickly destroyed the bearing. After cleaning out retainer material that efficiently bonded itself to balls (See Figure 10) and the races the bearing was "primed" with formulation 2-14 and tested under ambient temperature and with axial loads of up to 25 kg.

4.3.3 Ceramic TEST BED Bearing

The work with the ceramic test bed bearing is detailed throughout this report. Some special observations in relation to gained experience through these tests need to be made. The metal bearings, SKF (7206/CP4 DGA), being open and unsealed, were easy to re-lubricate, but also were easily contaminated by particles from test equipment and the general environment. In contrast, the test bed bearing, by virtue of the extending retainer between the lands of the upper and lower races gave excellent protection from such contamination. It also required disassembly for re-lubrication, which proved to be somewhat time consuming. Since the retainer is supported on the I.D. lands, it is possible to open the gap at the O.D. periphery sufficiently to permit injection to the ball pockets. This approach will be tested in further work.

Low and medium temperature testing was performed by using a retainer fabricated from SP-1 Vespel (Vespel is a trademark for polyimide produced by the DuPont Company). Vespel has a safe upper temperature limit of 250 degrees C, has excellent wearability and high strength, and, in contrast to

FEP composites, will not creep under load. For temperatures above the medium level, two grades of Boron Nitride have been tested successfully. Grades A and 26 made by the Carborundum Company, will readily accept several of the dry lubricant formulations. The materials themselves exhibit good low friction qualities. Yield strength of these boron nitride materials is relatively low and the reinforcing of the ball retainer periphery as shown in Figures 3 and 4 will be necessary for high speed applications.

Wear marks within the pockets, indicated that ball contact within the ball pockets was not uniform. The problems resulting from such uneven contact can be rather serious. As observed and written up in several reports, such conditions will initiate retainer vibration and, by our experience, will also affect proper tracking of the balls within the race, resulting in bearing noise, excessive retainer wear, and uneven loading of the retainer periphery against the bearing lands. Also it affects proper lubricant distribution by not allowing the roll-in of lubricant under the high load ball contact areas and will initiate an excessive lubricant build-up in low or no contact locations.

An important discovery was in the observation that lubricant (2-14) adhesion differed between the three grades of hot pressed silicon nitride. As pointed out by the Tribology Consultants, Inc. report (See Appendix C) the balls of the test bed bearing are made of Norton NC 132 material whereas the upper and lower race are made of a Toshiba commercial HPSN and an experimental grade respectively.

It was observed that the Norton material accepted metallic additives (Silver) more readily, and the experimental grade showed a grayish appearance on the ball contact race area. The Polishing quality of all components is nearly equal. The rings show evidence of forced polishing, which, in all probability, resulted in some surface stress. This condition was partially remedied by annealing both rings at 1,000 deg. C for one hour and by overnight cooling down in the annealing oven.

The constant "graying" of one of the balls created the suspicion of a size difference. What was first thought to be abrasion due to oversize was eventially determined to be a heavier lubricant build-up due to the undersize condition of that particular ball. The difference was in the order of .001 mm. This occurence indicates the need for extremely even ball size and sphericity.

The starting torque value of the test bed bearing was checked each time the bearing was removed for re-lubrication or observation and checking of the solid lubricant. It was a consistent occurrence that after a one to two hour test the lubricant at the ball contact areas showed interference fringes and the balls themselves had a blue cast. Removal of the dry lubricant formulas by ultrasonic cleaning in alcohol was unsuccessful. The extremely tough, thin film was finally removed by washing in a 30% nitric acid solution. This washing procedure also removed any residual Silver. Repeated cleanings always resulted in appearance of the original high polish on the balls and race-tracts.

We anticipate that eventually higher loads and speeds will affect this condition. It is important to find a way to obtain a very thin, but fully metallic undercoat with excellent adhesion. This will result in better binding and distribution of transferred or supplied lubricant.

4.4 Auxiliary Test Methods / Starting Torque Test

A neglected, but simple, test to check bearing performance, and at the same time, lubricant performance under ambient condition, is to measure starting torque by methods as shown schematically in Figure . The method is applicable to both radical contact and thrust bearings, but it is necessary to apply a fixed preload to obtain firm ball contact. The test is especially significant to measure the effect of run-in performance and condition. The measurement of starting torque of the bearing in an unlubricated condition is important. In method A it is advisable to move the rotating race to a different angular starting position, usually about 90 degrees apart and take the average reading. With method B the torque watch adapter can be designed such as to form the preload weight including the weight of a torque watch gauge. A Waters Manufacturing Company gauge with a range of .35 - 45. grm.-cm will be quite suitable for 1-1/2 to 2-inch diameter bearings. Other gauges with different ranges and digital read out provisions are commercially available.

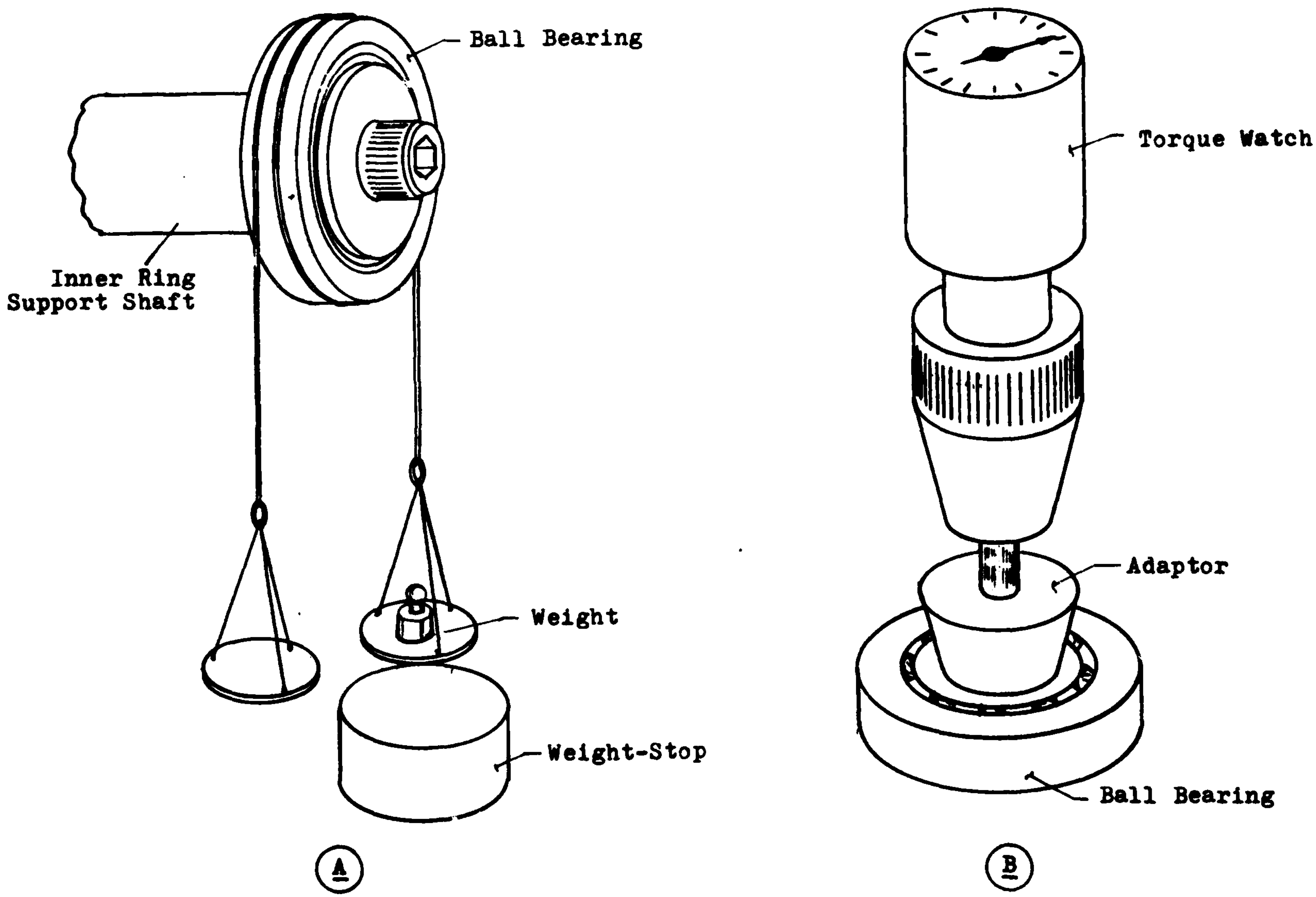
4.5 Bearing Test Results

4.5.1 Test Data

The test-data as accumulated and shown in Appendix A of this report was obtained on the equipment described in section 4.2.1. Test conditions as recorded for each test run are as follows:

- Drive rod speed in revolutions per minute (rpm)
- Linear speed, Meters per second

Figure 19. Starting-Torque Measurement Methods A & B



- Hertzian Stress at the ball and race contact area
- Lubricant System or formulation
- Lubricant replenishment method, if any
- Comments. Will list special events and observations.

4.5.2 Test Set-Up / Conclusions and Recommendations

As repeatedly mentioned within this report, it is difficult to come up with a clear-cut separation of what is being tested when running an assembled and lubricated bearing though its paces. In the final analysis, the test equipment itself may have to be scrutinized. It is not the purpose nor does it fall under the charter of this contract to review and critique the numerous existing test facilities. We would like to recommend such a review, for little is known of the specific differences and advantages of each test set-up and facility. Most of the domestic ball bearing manufacturers have developed refined techniques to check, analyze and record bearing performance. In high speed and high temperature tests two reasons are presently the main cause for not obtaining better and more reliable results of bearing behavior and performance:

a) Equipment induced vibration transmitted into the test article.

b) Lack of reliable temperature information at the ball and race interface.

In both cases the test parameters may be exceeding what is shown on the monitoring instrumentation and thus, in case of failure, the actual conditions may disagree with recorded data by a substantial margin.

Returning to the condition of our ball bearings after testing, we are assuming that the parts and materials used, including the lubricants, are sufficiently good performers. Upon a possible failure, however, it is very difficult to firmly establish which part and what particular condition was the first and primary contribution to the failure. One exception to that will be a ball-cage failure. Its destruction or excessive wear are conclusive evidence of its deficiencies.

4.5.3 Review of the Tribology Consultants' Report

This report, presenting a good background and history of ceramic bearing technology, also gives urgency for the need for further advances in ceramic bearing design and materials. Several companies are in the process of finding ways to lower the cost of materials such as HPSN and at the same time improve its properties. The application of relatively accurate preform manufacture will eventually result in considerable time savings in the production of the high precision balls and rings for the races. It is doubtful that prices will, in the near future, be competitive with steel bearings. However, the ceramic bearing is not meant as a replacement for the steel bearings. The idea is to replace metal where thermal conditions will exclude the use of metal bearings and wet lubricant systems. The report also does not reach beyond the present ball-retainer technology in that retainers are limited by the usage of composites at the 345 degrees C level. The air bearing concept as shown in Figure 9 of Appendix C is an attractive one. Its design though needs to be incorporated into a non-metallic retainer and needs a method to hold the two halves together. Above all, bearing and cage life need to be extended by methods of relubrication as described in section 3.6.2. The solid lubricants themselves have to be able to sustain temperatures of 815 degrees C and this, unfortunately, leaves out good performers such as molybdenum disulfied, and PFTE, etc. Our experience also has shown that vitreous inclusion, as desirable as they are for bonding, have a detrimental effect to low friction qualities and will bleed out at the higher temperatures encountered at the ball and race interface contact areas.

Regarding the temperature levels mentioned for several solid lubricant formulations, such as Tungsten diselenide with gallium indium eutectic, one must realize that oxidation and other causes of lubricant breakdown do occur much sooner than the temperature environment would indicate. The cause is at the immediate interface of ball and race where temperatures affecting the lubricant coating and lubricant particles is substantially higher. It is not unreasonable to assume that a bearing functioning within a 600 degrees C environment, running under load at 60,000 rpm, will experience a Δ T of 200 degrees C. Methods to more precisely determine events at the ball and race contact area are still in the early stages of development, and even more so are systems that will permit continuous monitoring of bearing conditions. (R 35)

In regard to hybrid bearings, it must be realized that they, too, present only an intermediate step on the way to a true high temperature bearing. Another factor neglected in many reports is the requirement for resistance to moisture and chemicals. Even stainless steels have only limited resistance to either. A degree of moisture absorption must be expected with most composite materials. It is an area that has not previously been given sufficient attention, and is given further consideration in Section 3.8.2 of this study. The report indicates a 45 degree contact angle for the bearing whereas the real value is near 38 degrees. This change does not affect the level of contact stress. The report contains little reference to solid lubrications.

5.0 Report Summary

The studies and the tests performed under this program provide significant guidelines on how to proceed under a Phase II effort. The excellent results obtained by life testing of four proprietary solid lubricants should not hide the potential differences and possible problems that might occur at high rpm. Several of the lubricant systems and formulations have been subjected to static testing at the 800 deg. C level and this without significant loss of frictional quality. A 10 to 20% temperature rise is expected when running a solid lubricated bearing at high speeds and loads.

The ball retainer still needs further attention, especially in regard to strengthening available high temperature materials and in reducing frictional contact at the ring lands by using air-bearing concepts for the interface areas.

Much attention has been given to lubricant replenishment. The study and the tests give all indications that proper relubrication can solve several inherent problems of the present concepts.

Further work needs to be done to improve the deposition of metallic films to act as binders for the lubricant.

Lubricant formulations used in this study have not been optimized. Optimization will be a better approach than to increase formulation selection. The report suggests the improvement of some commercially available lubricants by methods learned under this study. This has to be done in cooperation with the manufacturers of the specific solid lubricants.

Tests to improve contaminant and moisture resistance have been only partially successful.

Resistance to chemicals need further investigation, although tests with alcohols and diluted acids were successful.

6.0 Technical Recommendations

6.1 Ball retainer design, material, and test

In reviewing problems with existing ceramic ball bearing technology, it is evident that the ball retainer presents a special challenge for innovative solutions. It is as much a material as a design problem and each of these has to be handled by its own merits. Since the retainer also plays an important role in providing back-up lubrication and, in the case of re-lubrication, is acting as a distribution equalizer, its role as a bearing element is very important. Section 2.5.1 and 2.5.3 refer to the special problems and Figures 3 and 4 describe suggestions for improvements. As a recommendation we promote the testing of:

A. Different high speed retainer designs by themselves without being part of the bearing assembly. Spinning a retainer at progressively higher rpm under various ambient conditions will permit observation of dimensional changes exerted by substantial hoop stresses.

B. As discussed in section 2.5.3, a portion of the retainer will ride against the land of the inner and/or outer race under purely frictional contact. This condition can be simulated by running the retainer inside a fixed ring. Injection of lubricant can be simulated, as can the method of using a small gap as an air bearing support. (See Figure 3 and 4.)

C. Either A or B test can be pushed to failure. The data from the fail or near fail points can be used to verify computed theoretical values.

The advantage of running such tests is in their simplicity and without the use of costly finished bearings. Predictability of life and performance of a complete bearing assembly is thus greatly enhanced.

The Mindrum/Helvart team has the capability and knowledge and would be glad to design and manufacture the necessary equipment to run tests as described above.

6.2 New Concepts for Ceramic Ball Bearing Design

Temperature limitations of customarily used ball retainer materials and the special characteristics of ceramic ball bearing materials limit the design flexibility incorporated into all metal bearings. There is, for instance, a need for deep groove bearings and the assembly of such bearings will require the use of crown type retainers. Concepts such as described in section 2.5 and shown in Figures 3 and 4, are not only useful for the usage of crown ball retainers but, at the same time, will provide the necessary strengthening of the retainer periphery by the addition of high strength high temperature metal rings. These rings will take the brunt of centrifugal loads exerted at high rotational speed. It is recommended such design avenues be further pursued. The search for lighter and stronger ball retainer materials must continue as a parallel effort.

Priority should be given to concepts that will allow the land contact to area to run as a air bearing. In order to relieve the ball retainer from its function as primary lubricant reservoir, re-lubrication techniques need to be studied and tested. At high speeds and temperatures re-lubrication will extend bearing life and at the same time enhance performance and reliability. By reducing the dependence on the retainer for permanent lubrication, more attention can be given to reinforcing of retainer materials, which in turn, will add to design flexibility.

6.3 Substrate Metallization

Thin film metallizing of ceramic bearing materials will need further development. Methods to activate the substrate surface in order to form an inseparable bond should be given priority. This process forms the basis to interface with the 2-14 and 2-14-S series formulation.

6.4 Ball and Sleeve Bearing / Mounting techniques.

The low coefficient of thermal expansion of most ceramic materials and especially of Silicone Nitride presents installation and functional problems that need to be resolved in the near future. Convoluted shaft diameters and convoluted seats to match the bearing O.D. are one solution. High or low expanding metal (invar) inserts or adaptor rings are other possibilities. In any case the problem is an urgent one and needs early attention.

6.5 Phase II and Interim Work

In order to proceed testing on a continuous basis, it will be necessary to prepare or analyze bearings while the other is on the test stand. It provides for a more efficient use of personnel and equipment. All selected bearing designs should be manufactured in quantities of three to assure availability of one back up unit in case of catastrophic failure. Three sets of balls of all selected sizes need to be on hand. This in turn suggests that all bearings be designed for one size of balls. Substantial savings will thus be achieved. The I.D. and O.D. of all bearing types should be identical for the same reason.

All bearing material needs to be analyzed for voids, inclusions and other defects. This is to be done on the raw stock and also after rough grinding of the rings.

APPENDIX A

SUMMARY

The following represents a summary of all significant test events and conclusions. The detailed test-log and other pertinent material are contained in Volume 2 of this report.

- Equipment check out and preliminary testing were performed with a SKF 7206C/R4 angular contact bearing. Observations from the procedures are discussed in section 4.3.1. Because of the limitations of the phenolic retainer material, test temperatures with this bearing were not allowed to exceed 160 deg. C. Work with this bearing provided the first chance to test out the beneficial effects of lubricant replenishment.

- Over-lubrication resulted in a temporary increase of bearing noise and contamination by retainer material (phenolic). This occured at the higher temperature levels resulted in a marked increase of starting torque (from .6 grm.-cm to 1.8 grm.-cm).

- Some problems with the equipment bearing drive shaft were observed and corrected.

- Installation of the ceramic test-bed bearing required a new drive shaft adaptor which engaged and rotated the upper race strictly by frictional contact of the rod tip within the spherical cavity of the adaptor. (See Figure 17.)

- The ceramic bearing had rotated over 1.25 million revolutions at termination of the first series of tests. This was done under various speeds, loads, and temperatures. Also it was lubricated with three different solid lubricant formulations.

A second series of tests (presently in progress and all log information extending beyond the writing of this report) will be presented in a separate volume as an addition to this appendix.

- Highlights of lubricant performance were:

  Lubricant doped with silver will exhibit higher load capacity and allow longer intervals between lubrication.

  Run-in periods of approximately thirty minutes, and under constantly increased loads (up to 25 kg.), result in very uniform distribution of the lubricant and in eventual formation of a thin and very tough film coat. This film, visible by the merits of interference color pattern, is difficult to remove by ordinary cleaning methods, such as ultrasonic agitation. We recommend the study of deposition of this film by sputtering or other suitable methods.

It is known from coupon tests that a thin film silver deposit would further enhance adhesion of such a film. Several attempts, though, to obtain a reliable silver coat upon a silicon nitride substrate have proven more difficult than anticipated.

- Distribution of the solid lubricant over the ball surfaces proved to be equally uniform and its adhesion adequate for present testing conditions.

- The film and particle distribution is visible under the microscope by the existence of very small specs of silver. We recommend, in further work, to include formulation improvement by particle size control and a more homogeneous intermixing of all particles. The smaller the particle size the better the intermixing. Experience has shown that mixing of a slurry and subsequent drying and then regrinding is a good way to obtain the needed results.

- Particle size will also beneficially affect re-lubrication processes. During the various tests, where periodic injections with extremely small amounts of lubricants are customary, the procedure will permit an almost indefinite running of the test specimen. After extensive cleaning of the bearing elements, the balls and the race tracts showed no sign of wear or degradation. If anything, they exhibit an improvement of the contact surface (two previously small defects were more visible).

- The ball pocket wear marks have now shown an appreciable increase in size since the original run-in amounting to about 25,000 revolutions, a proof of the effectiveness of the lubricants.

- Periodic starting torque tests resulted in a slow but steady torque decrease during the last test sequence. As measured with the torque watch (see section 4.4) a low reading of .45 grm.-cm was recorded.

APPENDIX B

SOLAR TURBINES INCORPORATED

APPENDIX B

BACKGROUND, EXPERIENCE, AND FACILITIES

Formation of Solar Turbines Incorporated

Solar Turbines Incorporated, a subsidiary of Caterpillar Tractor Company, was founded as a research, development, and production company to provide high-performance services and products. Since this founding over fifty years ago, Solar has maintained active research and development programs in support of the aircraft, aerospace, and turbomachinery fields. Solar's policies, procedures, and business practices are in alignment with the policies of Government procuring agencies and defense contractors associated with these fields.

Solar is the largest producer of small and medium industrial gas turbine engines. Its expertise in this field goes back over twenty-five years with the successful introduction of small APU turbine engines. More recent engines, which have captured a large portion of the domestic as well as foreign market, are the Saturn, Centaur, and Mars industrial turbine engines. Presently entering service are both Centaur and Mars combined-cycle units.

While advancing the turbine engine technology in-house, Solar has also performed research programs for defense contractors and the major governmental agencies. These programs typically include contracts to develop small high speed turbine engines, small steam generators, and bearing development.

High Speed Bearing Research and Development Activity

In 1971 the Research Department, Applied Sciences activity was broadened to include specifically identified bearings, seals, and lubrication research and development work. The primary objectives of this effort have included data acquisition and reduction, and development associated with the most efficacious application of hydrodynamic journal bearing, lubrication, and rolling contact bearing technology in Solar's high speed turbomachinery. Continuously from that time appropriate test rigs and supporting equipment have been designed, fabricated, or accumulated and successfully operated to provide a sophisticated, broad-base flexible research and development bearings, lubrication, and seals activity. The development aspect is strengthened through, whenever possible, the use of standard Solar turbomachinery components or adaptations thereof in test rigs. This approach has not only proven very cost effective but has often provided data on and insight into the operation of turbomachinery components not otherwise obtainable. During 1976, in the development test cell areas, a separate Bearing and Seals Research Laboratory was established commensurate with the increasing levels of activity in the fields of bearings, seals and lubrication.

Figure 1 shows a schematic of a bearing test rig based on the Solar Titan Engine, capable of steady-state operation at main rotor shaft speeds up to 75,000 rpm. The current configuration of the rotor with a new dummy turbine wheel which, with other changes, permits operation at up to 100,000 rpm. The radial faces of the wheel were manufactured to hydrodynamic thrust bearing standards to permit evaluation of such bearings.

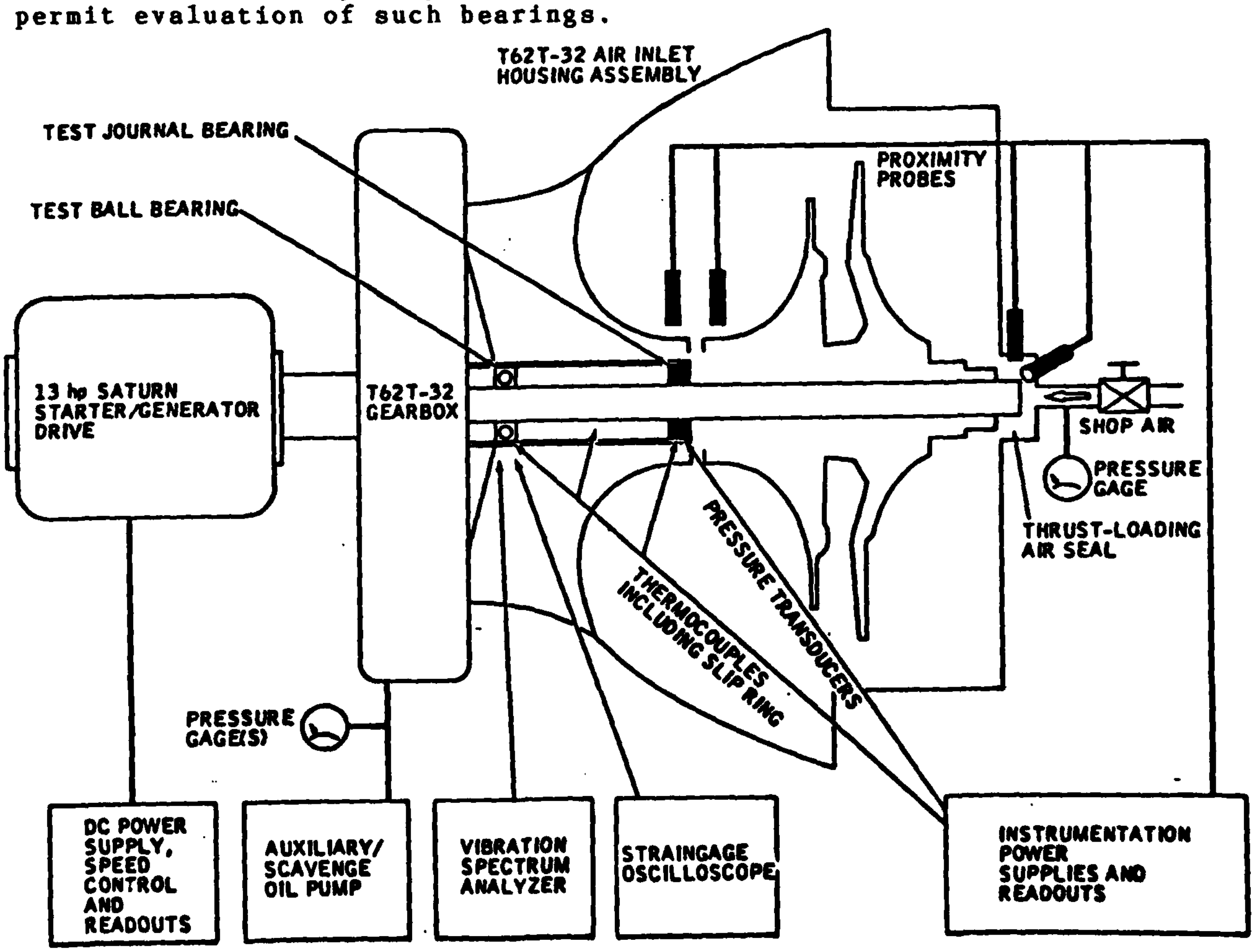


Figure 1. Titan Bearing Rig Schematic

Figure 2 shows a schematic of a bearing rig based on the Solar Gemini engine and capable of operation to at least 115,000 rpm. This is a specially-instrumented version of, but otherwise similar to, two other such rigs, all of which are employed in a U.S. Army MERADCOM program (contract DAAG53-76-C-0228). These rigs, driven by air turbines, are capable of untended running with automatic control and data acquisition.

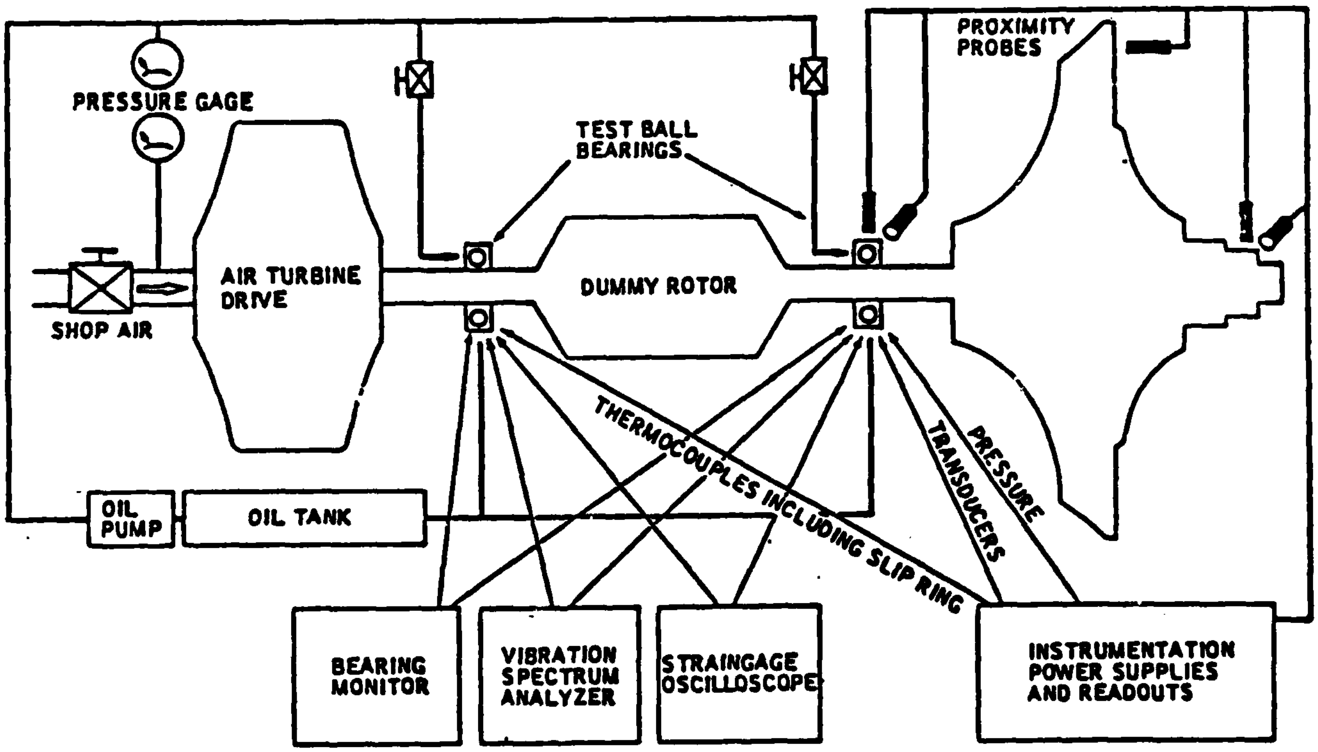


Figure 2". 10 kW Gemini Bearing Rig Schematic

The Analytical Department in Engineering has an active computing group which has developed an extensive library of Fortran IV computer programs in gas turbine technology covering areas of stress, vibration, combustion, heat transfer, aero-thermodynamics, bearings, and seals. The personnel of the computing group have unique capabilities for developing and maintaining sophisticated programs. These include a generalized data reduction and utility program, transient temperature analysis, performance synthesis for multi-stage radical and axial compressors, gas peak computer analysis, dynamic response of ducting systems, a radical compressor design program, finite element structural analysis of an integral rotor wheel, three dimensional stress analysis, and exhaust emission analysis. Several application oriented programs are in use. These include the "Continuous System Modeling Program" (CSMP), the "General Purpose System Simulator" (GPSS), the "Project Control System" (PCS), and the NGPA Phase Equilibrea Program.

Materials Evaluation

Solar has fully equipped mechanical testing, chemistry, physics and metallographic laboratories to perform evaluations on conventional engineering and state-of-the-art materials including virtually all alloy systems, ceramic materials, and plastics.

Of special importance to the proposed program are the metallographic facilities which include an AMR 1200 Scanning Electron Microscope (SEM) interfaced with a PGT 1000 Xcel Energy dispersive Xray Analyzer (EDX). This unit is particularly useful for characterizing surface reactions (wear, corrosion/erosion) and will provide important data on the effect of the various euvironments on the life of the selected seal materials.

Note: The above information is published with approval of Solar Turbines International, San Diego, CA 92138.

HIGH TEMPERATURE BEARING AND DRY-LUBRICANT APPLICATIONS

TRIBOLOGY CONSULTANTS INC.

Lewis B. Sibley

Introduction

Ball bearing shaft support systems are used extensively in a wide variety of turboequipment, often under minimal lubrication conditions without sustaining excessive bearing damage. Silicon nitride ball bearings recently have been shown to be especially tolerant of the high speed stresses and surface speeds in turbines which are sometimes abusive to standard steel bearings. In addition, silicon nitride bearing contact surfaces operating dry have a very low wear rate, approaching that of oil-lubricated steel. The churning of the lubricating oil in ball bearings operating at high turbine speeds results in bearing friction losses at times as much as an order of magnitude (ten times) larger than the inherently low rolling ball-race and self-lubricating cage friction in ball bearings operating without circulating oil. It is apparent, therefore, that oil-less silicon nitride ball bearings obviously are the ideal solution for advanced turbomachine shaft support.

Silicon nitride ball bearings are currently being developed for specific turbine equipment such as the 93,500 rpm MERADCOM 10KW turboalternator and the 60,000 rpm cruise missile gas-turbine engine mainshaft. It is appropriate that the results of these on-going programs be utilized.

Significant progress is now being made in developing high-volume low-cost methods of bearing quality silicon nitride preform manufacture, which most likely will reduce the cost of production silicon nitride ball bearings to at least as low as that of current steel bearings used in turbines. Advanced ceramic coating techniques are also under intensive development for use on steel bearing ring races to be run with silicon nitride balls.

Background

The low density of silicon nitride balls compared to steel (40%) results in lower internal bearing centrifugal loads and skidding in high-speed bearings, thus resulting in much less severe stressing of the lubricant and even lower bearing friction loss. In addition, silicon nitride has three times the hardness of hardened bearing steel (Rc 80), does not gall and its dry friction coefficient is much lower than that of steel, making it much less prone to surface distress under marginal lubrication conditions.

The primary purpose of any lubricant in a shaft-supporting bearing system is to prevent asperity contact between mating parts and thereby prevent excessive wear. In oil-lubricated bearings substantial friction loss occurs by shearing of the oil films and churning of the excess oil in the open spaces of the bearing. Even in ball bearings, as shown in Figure 1, excessive oil churning and viscous friction loss occurs in front of the ball-race and cage contacts, for which calculation methods have been developed (1)*. Experiments with high-speed thrust-loaded ball bearings have shown that almost a factor of ten reduction in bearing friction loss can be realized by reducing these churning and viscous losses with minimal oil-mist lubrication (2). Even greater friction savings should be possible using dry solid lubricants.

Successful dry film performance depends both on the tenacity with which the solid lubricant material adheres to the bearing material substrate and on the ability of the solid lubricant to develop a smooth low-friction low-wear surface at which relative motion occurs with the mating surface. The normal oxide film on polished bearing surfaces made of very hard materials can perform as a solid lubricant if the contact stress conditions and sliding speeds are not too severe. The generation of the smooth low-wear contact surfaces occurs during initial operation of the bearing. Life of the bearing is defined not in terms of fatigue cycles of the bearing material as in oil-lubricated rolling bearings, but in terms of the ability to form successful smooth low-wear contacting surfaces which last until the supply of the self-lubricating solid materials forming these surfaces is exhausted.

*Numbers in parentheses refer to list of references at the end of this report.

The ultimate ability of candidate solid lubricant and bearing materials to produce successful self-lubricated surfaces can only be determined by tests run under conditions simulating the intended bearing application service conditions as closely as possible. Some test acceleration can be obtained by increasing speeds and loads as long as the basic solid lubrication and wear mode does not change as a result of the accelerated test conditions, compared to service conditions. It is important, therefore, that any test program be planned to include careful examination, analysis and interpretation of the bearing wear surface condition after test in terms of the expected performance characteristics of the solid lubricant materials used.

The development of solid lubricants and dry wear resistant materials for high temperature bearing applications has been underway for many years (3-20). The most successful solid lubricant materials used in many commercial applications are molybdenum disulfide, graphite and Teflon. However, a wide variety of other layer-lattice structure or soft solids has long been identified as having good solid lubrication capabilities. One such material not yet commercially developed but shown in laboratory testing to possess outstanding lubrication properties and thermal stability to operating temperatures greater than 1000°F is the layer-lattice organic solid, metal-free phthalocyanine, which is believed to attach itself to metal surfaces by a chelate mechanism identical to that by which hemoglobin holds iron in human blood (5,6,10).

The bearing industry markets several types of dry bearing materials based on molybdenum disulfide, graphite, Teflon and other plastics used as solid lubricants. Woven glass fiber reinforced Teflon bearings are fabricated by bonding a stiff metal backing to a thin composite layer of the soft solid-lubricating Teflon reinforced with a hard glass fabric so that very thin films of the Teflon lubricate the glass fibers with a minimum of deflection, plastic flow and wear of the assembly. Similar types of Teflon-based bearings consist of a thin metal-backed layer of sintered tin bronze impregnated with a mixture of Teflon and molybdenum disulfide. Laboratory tests have demonstrated higher load capacity and longer service life for the fabric reinforced compared to the impregnated composites. Sliding friction coefficients for these materials varies from 0.03 to 0.1.

The maximum operating temperatures of these Teflon-based materials is 400 to 500°F (200-260°C). Impregnated graphite plain bearings having much higher temperature capability have been commercially developed (14) and used extensively in high-temperature seal applications (11), but the wear life of the best graphite bearings is an order of magnitude lower than that of the Teflon-based materials at lower temperatures. However, the graphites have excellent self-lubrication and transfer film characteristics that are required for successful high-temperature dry rolling bearing cages, and their well-engineered commercial availability under careful quality controlled manufacturing conditions makes them especially attractive for high-temperature prototype dry rolling bearing components.

Another promising solid lubricant bearing material for high temperatures is the extensive current development of graphite fiber reinforced polyimides (21,22). Preliminary data indicate a temperature capability of these materials to 600°F and a wear life of the same order of magnitude as the best glass fabric reinforced Teflons at lower temperatures. Alternative candidate solid lubricant rolling bearing cage materials developed for high temperatures include a molybdenum or tungsten disulfide compact in a columbium-molybdenum matrix known as Molalloy for operation up to 800°F (16), and tungsten diselenide with gallium-indium eutectic in an intermetallic complex for operation up to 1300°F (18). Also, a calcium and barium fluoride compact with silver and glass in a nickel-chromium matrix has been tested successfully in Stellite bearing cages at 1500°F (12,17). Wear life data for Molalloy indicate a service life comparable to the Teflon materials at lower temperatures and a friction coefficient ranging from 0.03 to 0.25. Wear rates of the other higher temperature materials apparently are considerably higher.

The fluoride compact material is also available commercially as a plasma-sprayed coating. Another such lubricant coating similar to Molalloy that has achieved an outstanding performance record in a wide variety of tests and applications is Vitrolube NPI1220, which is a low-melting-point glass bonded mixture of molybdenum sulfide, graphite and other soft constituents found to enhance the oxidation and dry lubricating properties of the film.

Other coatings with potentially superior dry wear characteristics are an especially tenacious electroplated and heat treated very thin chromium known as Armoloy or Nobilizing and a material known as Kaman DES which is a chromium oxide coating applied in such a way that an intermetallic bond is formed with the bearing surface. Armoloy and Noblizing have corrosion resistance exceeding that of conventional hard chromium plating, a surface hardness up to 70 Rc, reduced surface friction and wear, a maximum surface build-up of 0.00025 inch that can be controlled to less than 0.0001 inch, and adherence to the treated surface which exceeds that attainable with conventional chromium plating. Both processes are applied to finished components without affecting the heat treated structure of the substrate, or adversely affecting surface finish down to 4 microinches, rms, roughness. Both processes have been tested on heavily loaded bearing contact surfaces and found to exhibit behavior consistent with the normal response of conventional bearing steel surfaces to rolling contact stressing. Both coatings displayed the ability to deform around surface discontinuities without cracking, chipping or peeling at maximum Hertz contact pressures of at least 480 ksi.

In addition to Armoloy and Noblizing, other candidate hard coating processes have been developed recently with potentially improved wear resistance for dry operation with silicon nitride balls. One of these is Chemetal's CM500 coating by controlled nucleation thermochemical deposition (CNTD) of tungsten and tungsten carbide with a surface hardness of over 85 Rc. A sample of this coating on AISI 1018 steel examined recently shows excellent adherence and such a fine structure that the carbides cannot be resolved by electron microscopy at magnifications of at least 20,000X. Also, this coating is reported to have run successfully on poppet valve seats at Rocketdyne. The coating process requires heating for several hours at 600 to 700°F, which has no effect, of course, on the secondary hardening M-50 tool steel which is tempered at 1025°F and used extensively for turbine ball bearings.

Chemical vapor deposited (CVD) refractory coatings have been developed, also having potentially improved performance capabilities. Titanium carbide, with a very thin underlayer of chromium carbide, has been used successfully on 440C steel instrument bearings operating dry (23). The CVD processing temperature is about 1800°F (1000°C), which means that the coating process must be integrated into the heat treat cycle of the bearing steel. Hughes reports only a small yield of useful bearings of very small instrument size from this process. In addition, the successful instrument bearings have run under only very small loads, so this coating has not yet really been demonstrated under the rolling contact conditions needed for turbomachine bearing surfaces.

Other promising CVD coatings are titanium nitride and titanium diboride, which is used commercially very successfully on highly stressed tool surfaces. Titanium diboride coatings on steel should have been better adherence than titanium carbide, since it has better matched thermal expansivities with steel. Even with M-50 steel, of course, it will be necessary to integrate the coating process with the steel heat treatment, but since M-50 steel is air hardenable, this may not be too formidable.

In a lightly loaded rolling bearing, experience has shown that a very thin lubricant coating applied to rolling elements and races can be enough. However, as load increases and longer life is desired, lubricant must not only coat the bearing track surfaces but also must be available for replenishing purposes as lubricant is lost due to wear. The amount of lubricant (thickness) that can be "stored" on rolling element and race surfaces without interfering in the satisfactory operation of the bearing is very limited. Consequently, the technique which has been employed to provide sustained lubrication in a rolling bearing depends heavily on transfer of dry lubricant from the cage as a source of supply. This requires very careful compromise between the lubricant function and the structural requirements needed to serve as a ball separator. The proposed effort will be directed to the engineering application of existing candidate cage materials. New materials will be incorporated later when they become available.

Tool steel bearings were run at moderate speeds and loads successfully dry almost two decades ago at temperatures up to 1000°F using steel shrouded graphite cages (24). Impregnated graphite materials have excellent self-lubrication and transfer film characteristics that are required for successful high-temperature dry rolling bearing cages. Table 1 shows the results of these early tests in which over 100 hours bearing life was demonstrated dry at 750°F.

Under a recent NAVAIR subcontract to Norton (30), silicon nitride was tested in highly loaded (400 ksi) rolling-spinning contacts operating dry over the temperature range from ambient to 1000°F using a rolling four-ball tester. This testing has demonstrated low enough wear coefficients in some tests with graphite cages up to 600°F (wear coefficient of 10^{-6}) to predict satisfactory turbine engine dry running roller bearing wear life.

Dry bearing analysis work is also in progress as a part of the Hughes dry lubricant development program (23). Existing computer codes will be modified on the Hughes program to provide optimized characterization of the dynamic and thermal behavior of solid lubricated bearing systems. Parametric studies will be run to define solid lubricant property "windows" for optimum bearing behavior to guide the Hughes advanced cage material development for use in the main shaft locations of the F-107 cruise missile engine shown in Figure 2. Weight saved with silicon nitride ceramic bearings, by eliminating the bearing fluid lubricant storage and delivery system, results in substantial performance dividend and cost reduction. Concern over lubricant deterioration after long periods of missile storage is eliminated.

Silicon nitride has been applied to the angular-contact ball bearings in a typical high-speed turbine (25) shown in Figure 3. Since the present all-steel ball bearings in this turbine are limited in life and speed capabilities and require copious quantities of circulating oil lubrication to perform satisfactorily in service, silicon nitride material is considered to offer substantial advantages over steel for these bearings. Two bearings with silicon nitride balls completed 813 hours testing successfully at 93,500 rpm in this turbine, the last 23 hours of which one bearing operated inadvertently without a cage, allowing the balls to run together, without any significant damage to the bearing surfaces (26). Two more bearings with silicon nitride balls have recently completed over 1000 hours testing time in this turbine successfully and are currently continuing to run in this endurance test (27).

Test Bearing Ring and Ball Materials

Silicon nitride from a variety of sources has been tested for rolling bearing application (28, 29) and the most successful materials to date are those processed by hot pressing, especially Norton's Grade NC-132. Table 2 presents typical properties data on bearing quality hot pressed silicon nitride and inspection data on the Norton NC-132 material used for the balls in the test bearing. Similar data are also given for two grades of Toshiba hot pressed silicon nitride used for the inner and outer rings of the test bearing, respectively.

Fracture sections of the Toshiba and Norton materials are shown in Figures 4 and 5, respectively, indicating an acceptable grain structure (28), although some minor yttria segregation is noticeable in the standard Toshiba material, possibly affecting performance. Hardness indentations under light load (500 g) have been shown to produce cracking at the corners in the Norton material but not in the Toshiba samples, as shown in Figure 6, indicating higher fracture toughness for the Toshiba material (30).

Test Bearing Design and Manufacture

The design of the ceramic test ball thrust bearing was finalized according to the drawing shown in Figure 7. The design selected is a modified basic 7005-size angular contact ball-bearing internal geometry (20 mm bore x 55 mm OD x 17 mm width with 36 mm pitch diameter) which normally contains 13 balls of ¼in. diameter with a phenolic cage. Since the rings are ceramic, their sections have been increased by making the bore of 7004 size and the OD of 7006 size with an approximate 7009 size total width. The initial test bearing contains a set of eight 0.187 in. diameter balls.

A 45 degree contact angle design was selected to maximize the ability of this basic ball thrust bearing to carry combined radial and thrust loads. Ordinarily a maximum ball groove conformity of 54% (ratio of groove cross radius to ball diamter) is used for steel ball thrust bearings and a closer conformity is needed to obtain the sameball track width with ceramic materials having higher Young's modulus than steel. However, since the contact spin microslip, which is the main stress factor on solid lubricant films in angular-contact ball bearings, increases with closer conformities, a 54% conformity was selected also for the ceramic bearing to minimize this microslip yet still maintain maximum possible load capacity.

The AFBMA load capacity based on Lundberg-Palmgren theory, (31), suitably adjusted for the different modulus and Poisson's ratio of silicon nitride compared to steel, (28), is given on the drawing in Figure 7 together with the factors used to compute AFBMA rated fatigue life under combined load conditions (32). The test bearing parts were ultrasonically machined from hot pressed billets and finished by standard bearing manufacturing processes, except that special abrasives were used in some finishing operations.

For high elastic modulus ceramic materials, the ball loads and contact stresses in the test bearing can be computed manually to good accuracy assuming a constant contact angle under load. For example, under a pure thrust load of 267 N (60 lbs), the normal ball load is $267/(8 \sin 45^{\circ}) = 47.2$ N (10.6 lbs) for which the maximum Hertz contact stress is 3.1 GPa (448 ksi). If a radial load of 40 N (9 lbs) is superimposed on the 267 N thrust load, the maximum ball load is $267/(0.7693 \times 8 \sin 45^{\circ}) = 61.4$ N (13.8 lbs) for which the maximum Hertz stress is 3.7 GPa (534 ksi) and the minimum ball load is 0.589 times the maximum load or 36.1 N (8.1 lbs).

High-Speed Bearing Design

The most recent computer analysis codes for computing the performance of high-speed ball bearings are used to optimize the design of silicon nitride bearings. The reduced density and Poisson's ratio and increased Young's modulus of silicon nitride are used properly in the computer programs to calculate the reduction in centrifugal forces and redistribution of the loads between the balls in the bearing resulting from these changes in material properties. However, an adjustment to the computer calculated fatigue lives is required by the need to account for the change in contact stress and stress volume resulting from the change in elastic properties of the silicon nitride material from those of steel.

In order to assess the operating characteristics of the selected design silicon nitride element bearing in the 10 kw turbo-alternator shown in Figure 3, computer program SHABERTH for SHAft/BEaRing/THermal interactions has been run, as given in Table 3 from (25). The two bearings on the simulated 10 kw turbo-alternator shaft in this analysis are each given a negative mounted diametral clearance calculated to simulate the minimum preload applied by the Belleville washer. The dimensions of the standard design cages and the ring steel material with life factors expected on the basis of test results under well-lubricated thick EHD film conditions are entered as input in Table 3.

For the detailed lubrication analysis in SHABERTH the maximum specified surface roughness and typical surface asperity slope characteristics for the raceways and balls in this size bearing are provided as input data. Also given as input are the properties of the MIL-L-7808G lubricant. Estimated values of asperity friction coefficient, percent lubricant in the cavity for the balls to plow through, and film replenishment layers on the ball and raceway tracks between contacts are given as input. A reduction in the percent of lubricant in the cavity was used in Table 3 to simulate a minimal lubrication system. It was also assumed that the mean operating temperature of all components in the system is 93.3 degrees Centigrade, except the bulk oil temperature which is assumed to be 68.3°C. Also in Table 3 are given the properties of the bearing materials used in the analysis.

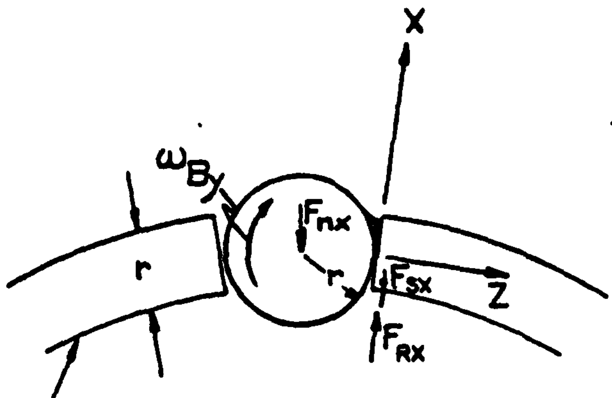
The dimensions of the 10 kw turbo-alternator shaft, with the location of the two bearings and the two radial loads corresponding to the alternator rotor weight and the compressor/turbine wheel weights, respectively, are given at the end of the input data in Table 3. Each bearing location is defined as the intersection of the plane of the inner ring groove curvature centers with the shaft center.

The bearing system output in Table 3 shows the computed deflections and reaction forces and moments at each bearing. The bearing lives in Table 3, of course, must be adjusted for the material property effects explained above, and thus result in recomputed lives of 46,576 hours for bearing No. 1 and 20,411 hours for bearing No. 2, which are about three times the SHABERTH predicted fatigue lives of all-steel bearings.

Thefrictional heat generation rate is computed by SHABERTH at all ball-race and cage contacts, as well as the rolling-element drag or plowing friction. The contact friction computations in SHABERTH include hydrodynamic losses in the oil films outside the elastic ball-race contact areas. Therefore, it seems likely that considerably smaller bearing frictional heat generation rates than those shown in Table 3 are possible and of course desirable from the standpoint of avoiding skid damage and thermal imbalance.

Table 3 also gives the results of a SHABERTH analysis of the ring mounting fits. The initial mean mounting fit pressure of the inner ring on the shaft computed by SHABERTH (3380 psi or 23.3 N/mm^2) is reduced under operating conditions by the combined effects of the centrifugal expansion of the inner ring and the ball loading to 1750 psi (12.1 N/mm^2). This results in a total mean reduction in the bearing radial clearance of 0.00017 inch (0.0042 mm), most of which is caused by the interference fit on the shaft. SHABERTH also has the capability of predicting the transient thermal behavior of high speed bearings as shown in Figure 8, in which the computed bearing clearance is plotted against time after an abrupt increase in speed of a solid lubricated hybrid silicon nitride ball bearing (33).

For high speed solid lubricated ball bearings it will be important to provide the means for guiding the solid lubricant cage to minimize friction at the guide contact surfaces so that the ball pocket contact loads will not be excessive and the heat generation in the bearing will be minimized. A suggested method for achieving these results is to use a self-acting air lubricated bearing design for the cage guide surfaces, as shown in the cage design drawing given in Figure 9.



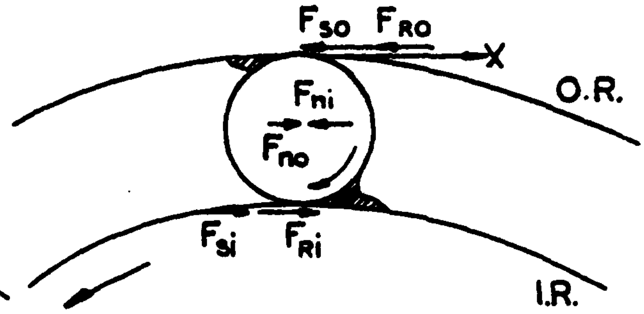
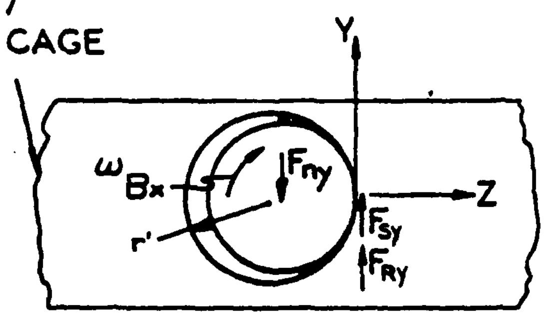
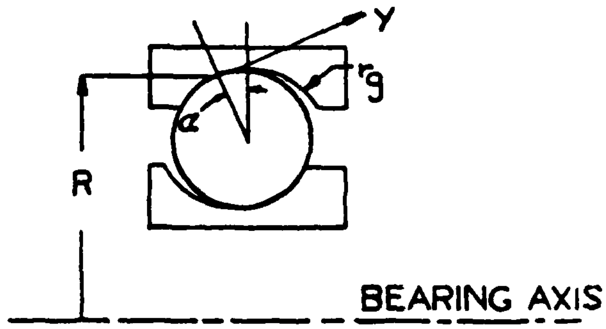


FIGURE A

FIGURE B



$R_X = r, \quad R_y = (1/r - 1/r')^{-1}$



$R_y = (1/r - 1/r_g)^{-1}$

$R_x = (1/r - \cos\alpha/R)^{-1}$

FORCES ON FLUID

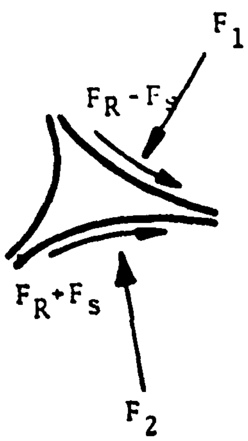


Figure 1. Sketches of Oil Viscous Forces at Ball-Race and Cage Contacts that Contribute Significantly to High-Speed Ball Bearing Friction Losses.

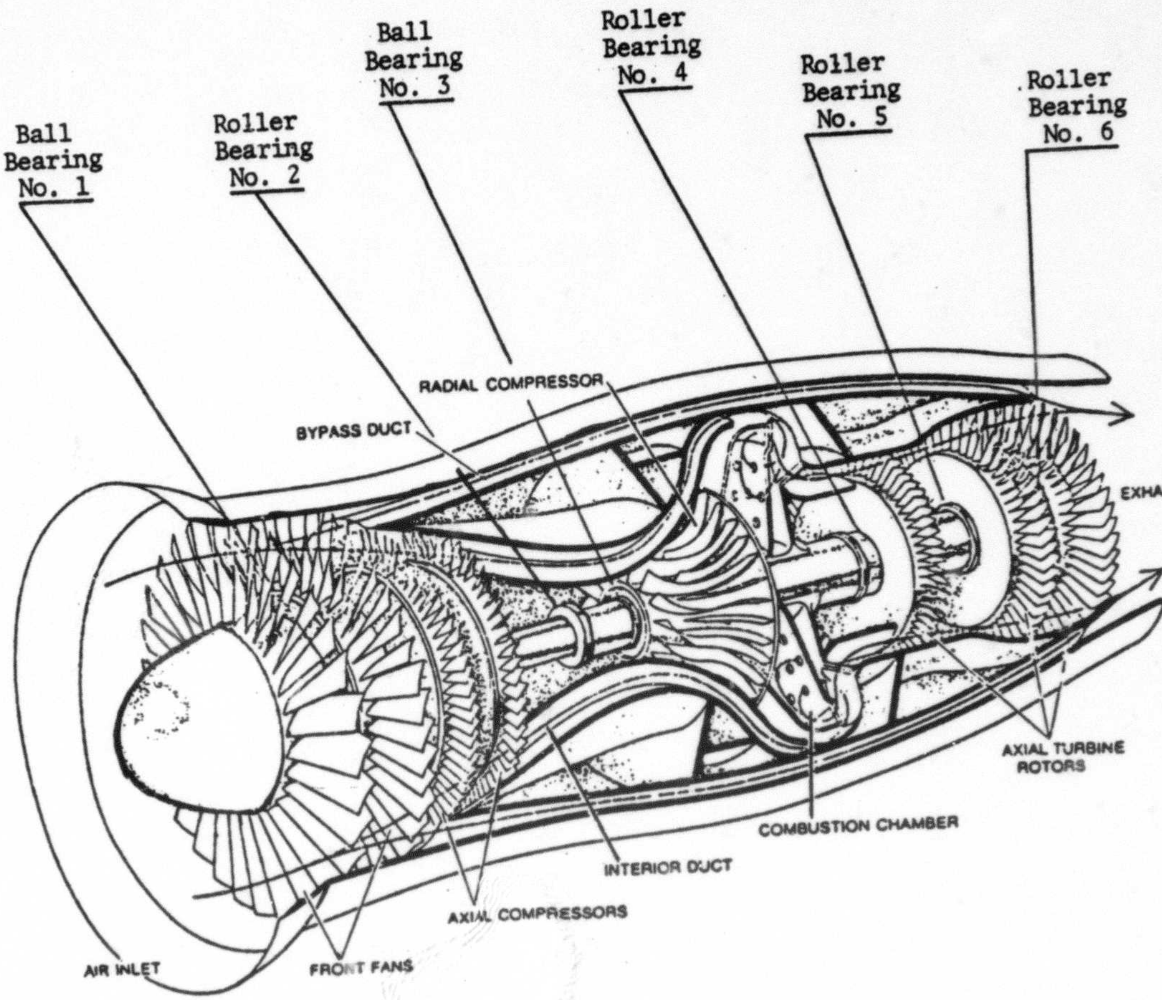


Figure 2. Cutaway Section of F107 Cruise Missile Engine Showing Mainshaft Bearing Locations

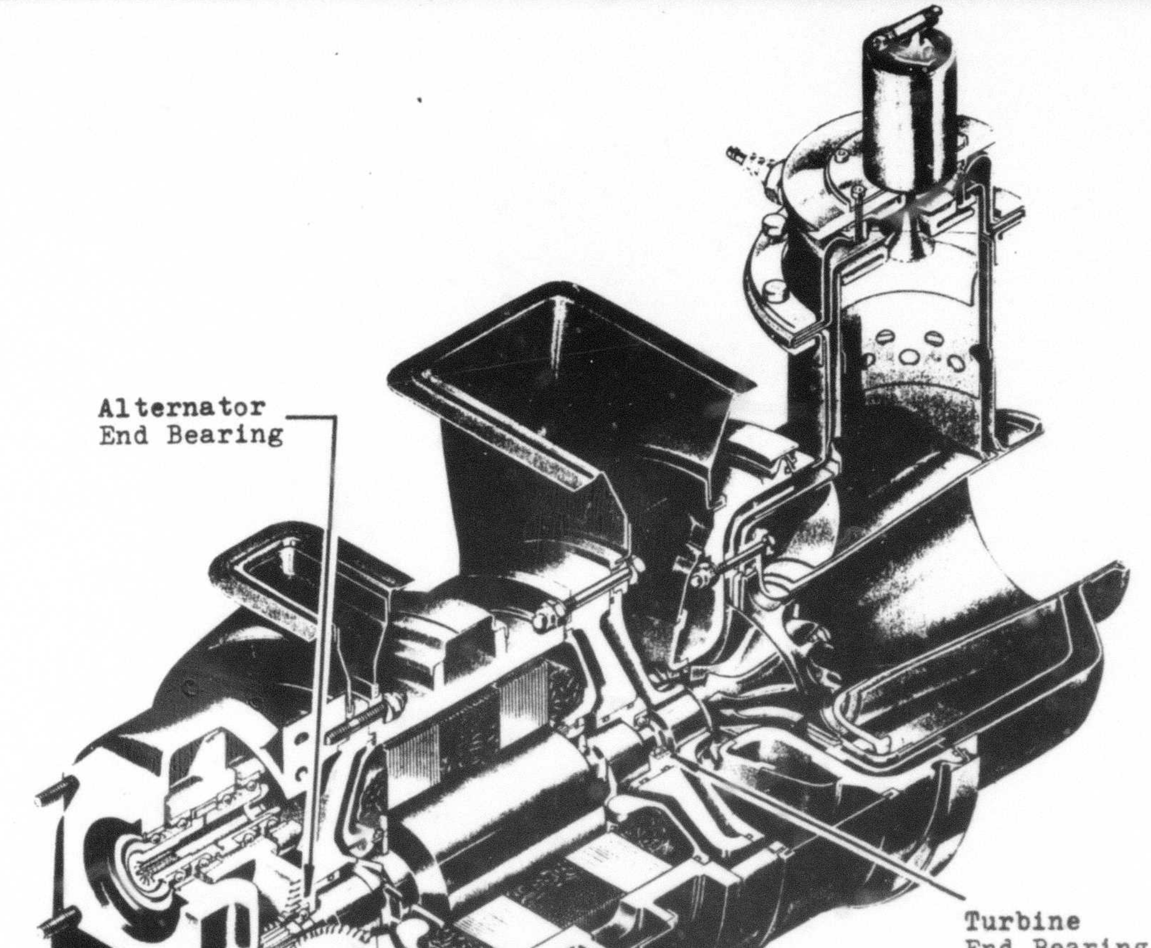
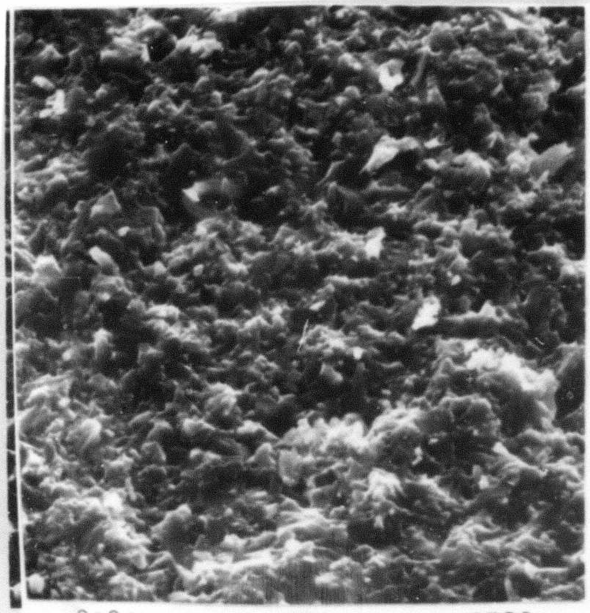


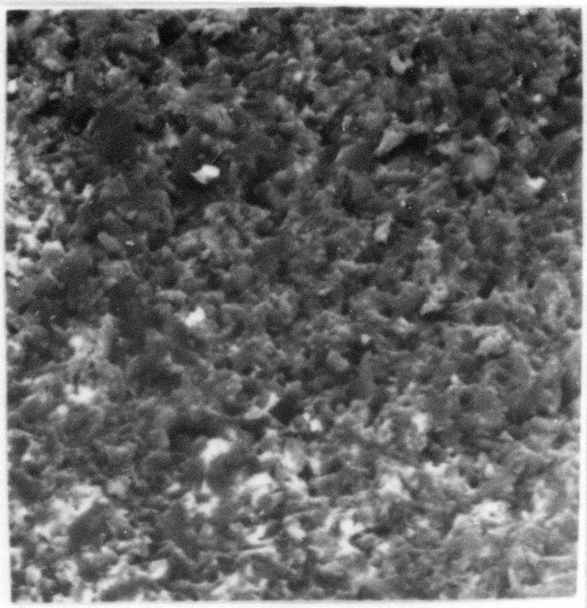
Figure 3,

Cutaway Section of
MERCADOM 10 KW ALTERNATOR
Showing Ball Bearing Location

23

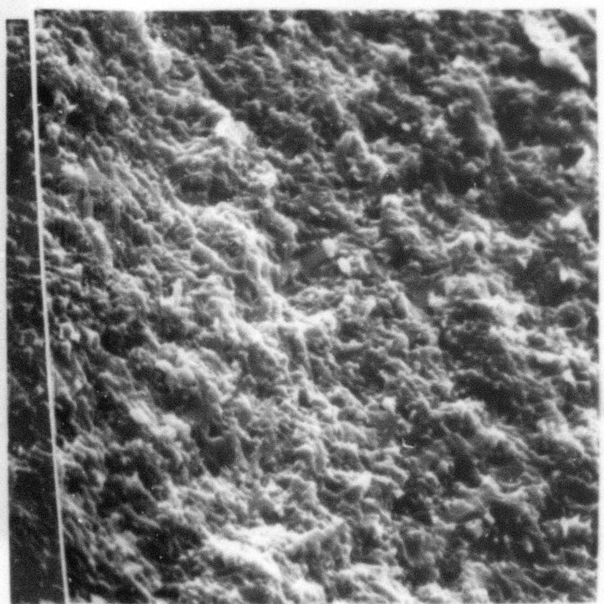


8289 SE 2500x



8290 BSE 2500x

Toshiba Standard Silicon Nitride

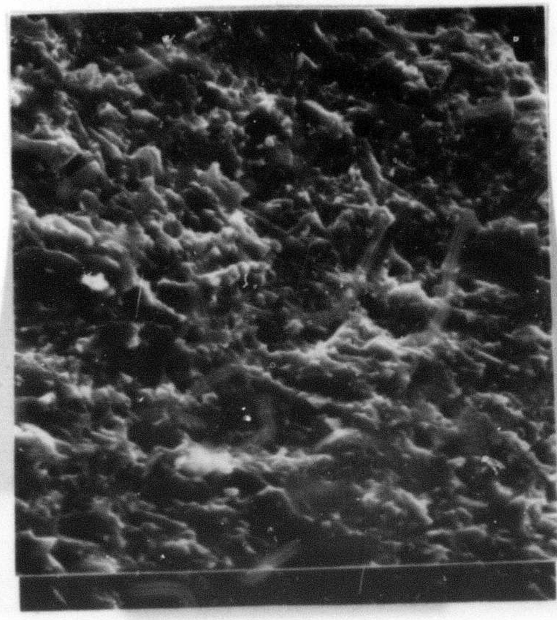


8287 SE 2500x

8288 BSE 2500x

Toshiba Experimantal Silicon Nitride

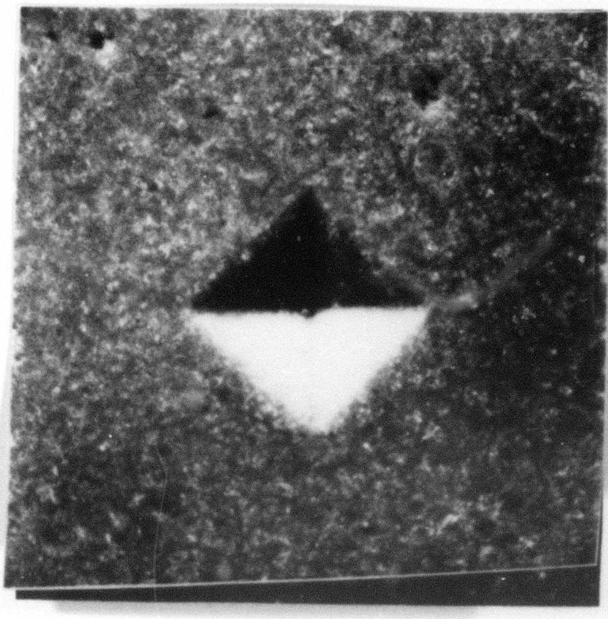
Figure 4. SEM Micrographs of Toshiba Fracture Surfaces
SE = Secondary Electron Image
BSE = Backscattered Electron Image



8291 SE 2500x

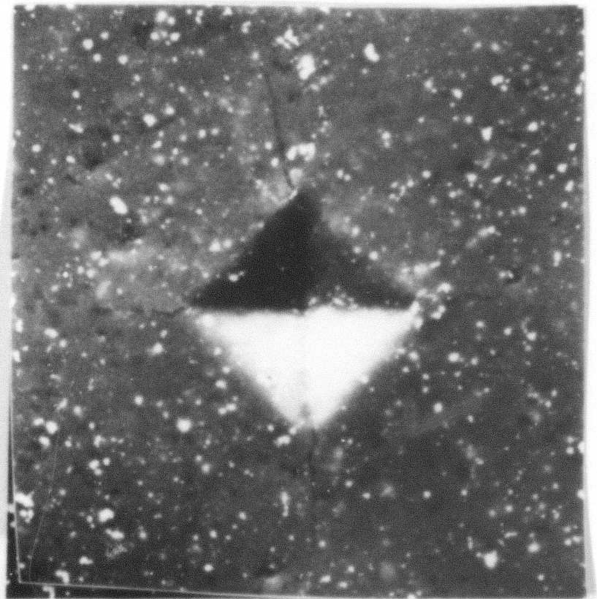
8292 BSE 2500x

Figure 5. SEM Micrographs of Norton NC132 Fracture Surfaces
SE = Secondary Electron Image
BSE = Backscattered Electron Image



8315 2000x

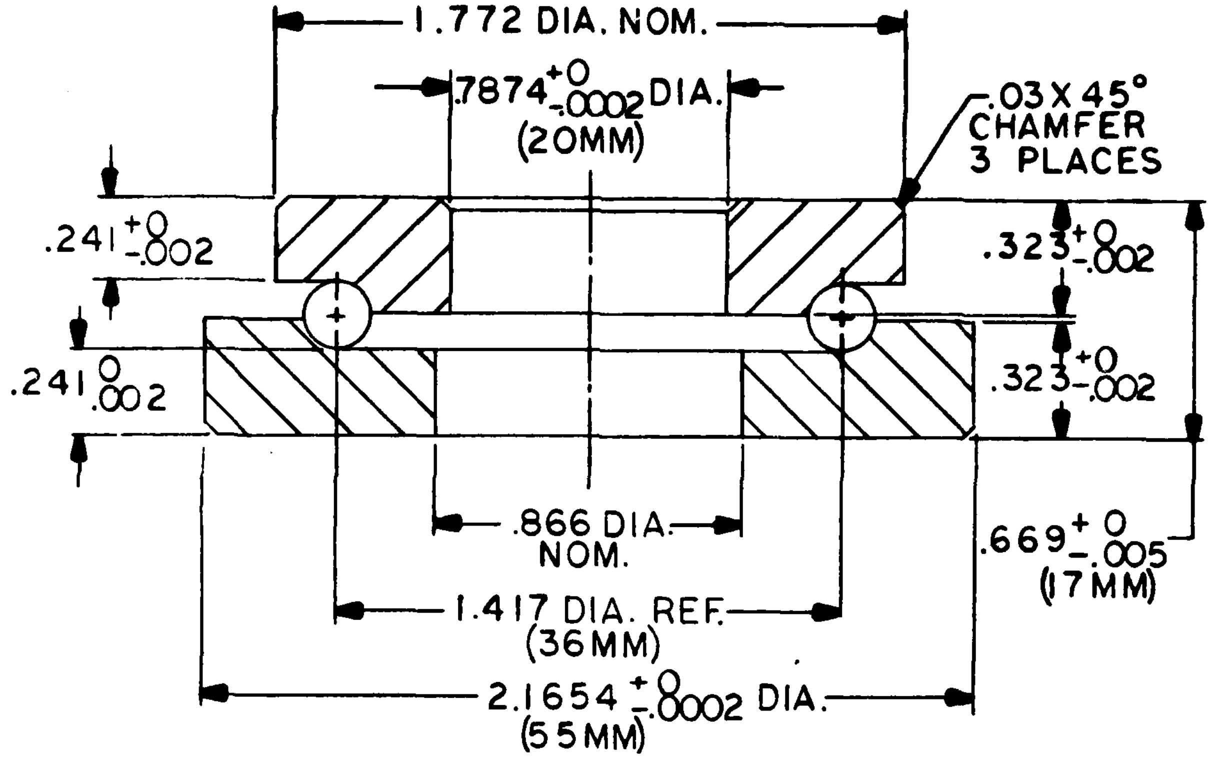
Toshiba Silicon Nitride



8311 2000x

Norton Silicon Nitride

Figure 6. SEM Micrographs of Toshiba and Norton Materials with Vickers Indentations Produced by a 500 Gram Load



BEARING DATA

BEARING NO.	SPEC 7005/AA
BEARING TOLERANCES	ABEC 5
NO. OF BALLS	8
BALL SIZE	.187 ±.000025
AFBMA CAPACITY	405 LBS. (1820N), e=1.25, x=.66, y=1
OUTER RING MATERIAL	TOSHIBA STANDARD HPSN
INNER RING MATERIAL	TOSHIBA EXPERIMENTAL HPSN
BALL MATERIAL	NORTON NC132 HPSN
CAGE	SUPPLIED BY HELVART ASSOC.
CONTACT ANGLE	45°
GROOVE CONFORMITY	54%

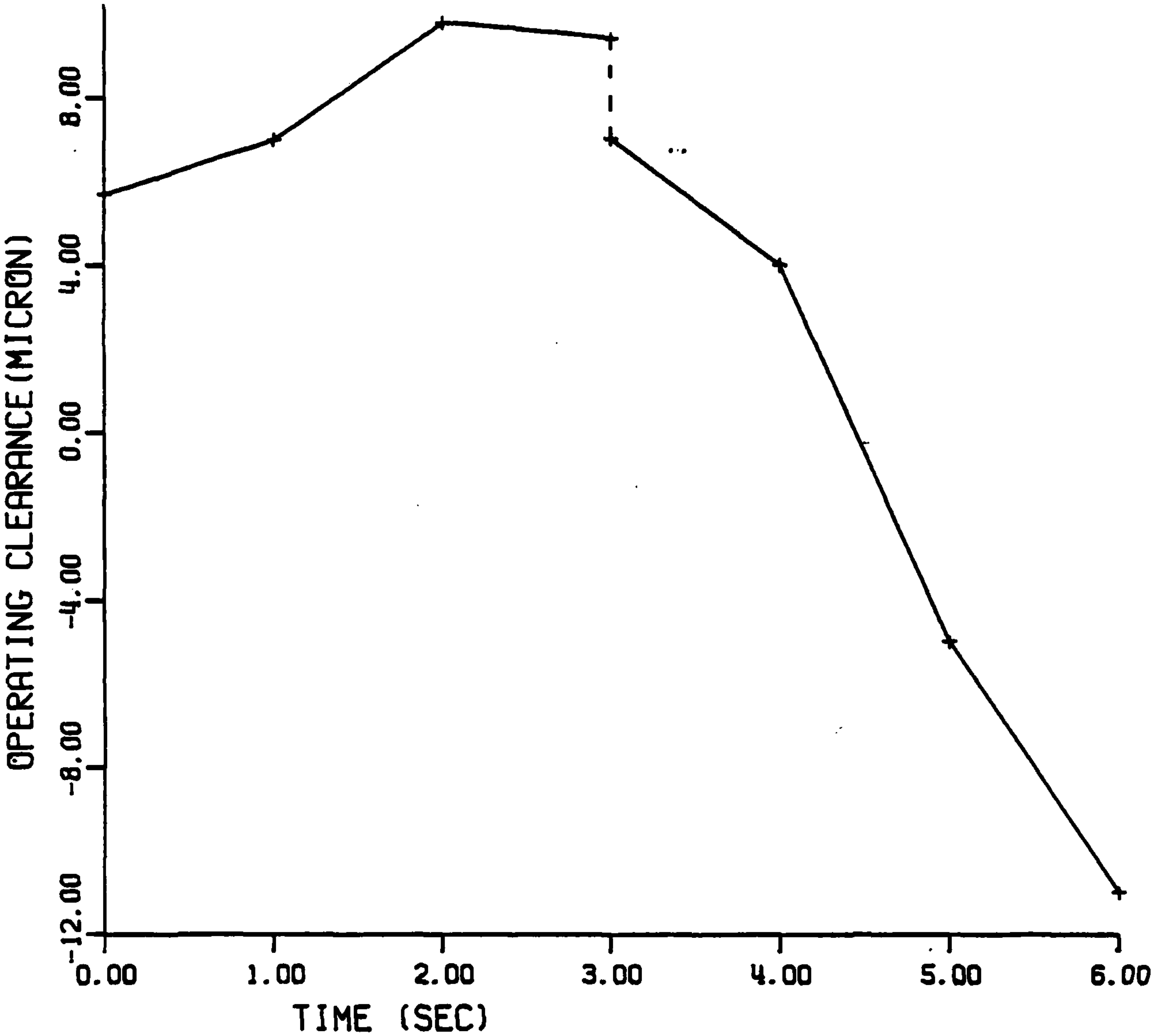


Figure 8. 7205 Hybrid Bearing Operating Clearance vs. Time (Data Calculated by SHABERTH Time Transient Thermo-analysis)

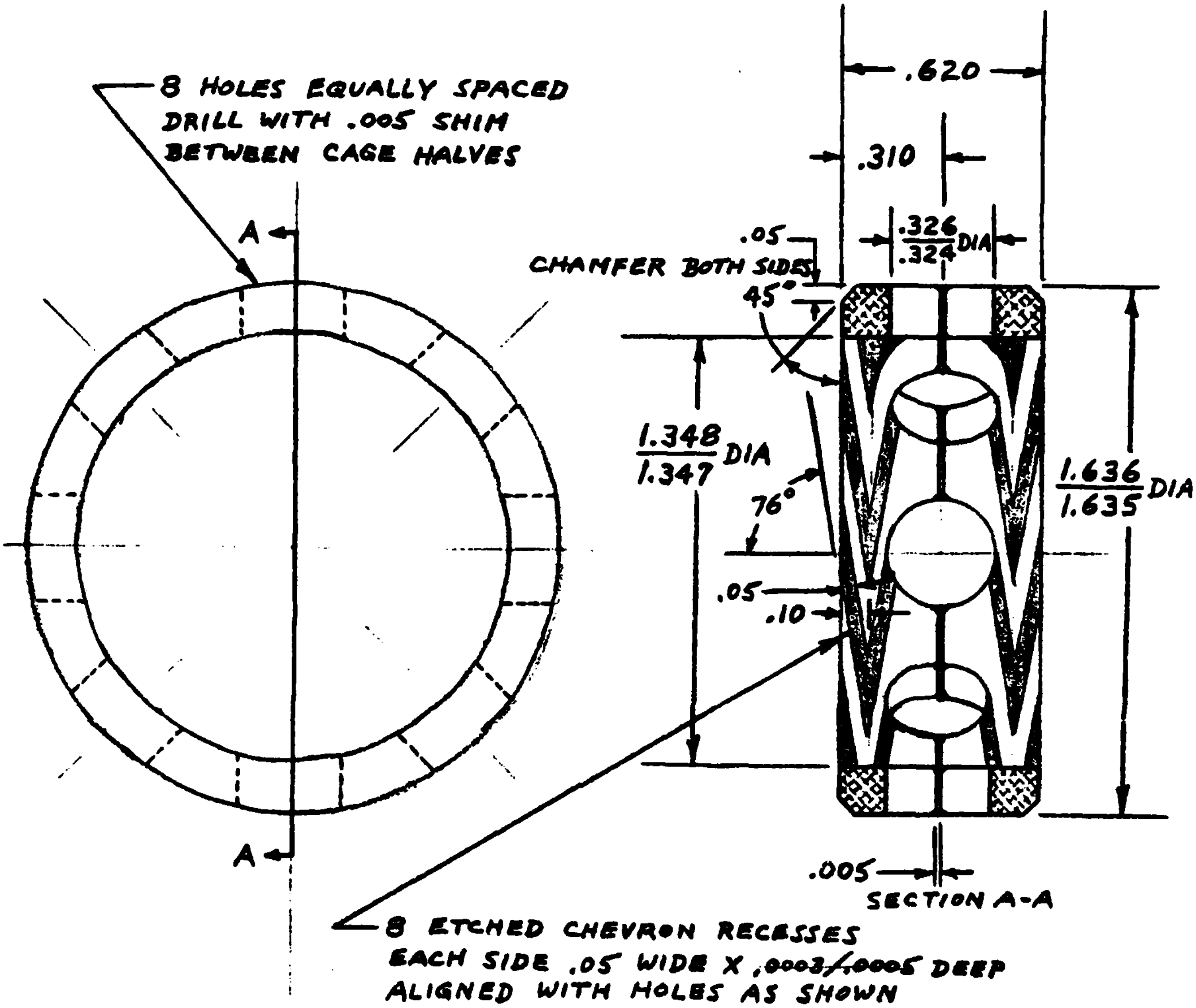


Figure 9. Solid Lubricated Air Bearing Guided Cage Design for Oil-Less Turbine Ball Bearings

TABLE 1

HIGH TEMPERATURE PERFORMANCE OF 25 MM. BORE M1 AND M50 STEEL BALL BEARINGS USING VARIOUS METHODS OF LUBRICATION OPERATED IN N_2 OR AIR ATMOSPHERE

Cage Type	Temp. °F	Thrust lbs. (Test Conditions)	Speed rpm	Atmosphere	Hrs. Run	Remarks
Brg. 452833 (Basic 6205) M50 steel coated with MoS_2 Dry Film Lub.						
Bronze filled	1000	50	2350	N_2	1.0	Brg. O.K.
with NAMC-AML-23A	"	75	5350	"	7.5	Cage failed, rings worn
Brg. 455760 (Basic 7205) M1 steel coated with MoS_2 Dry Film Lub.						
NAMC-AML-23A	1000	50	2350	N_2	1.0	Brg. O.K.
in M1 Steel shell	1000	175	5350	"	7.4	Brg. O.K. Lub. on cage broken off
Carbon-graphite	1000	50	2350	N_2	0.2	Cage failed
in steel shell	"	"	"	"	0.3	" "
	600	"	"	"	1.0	Brg. O.K.
	"	175	5350	"	5.2	Heavy ring & cage wear
	1000	150	"	"	11.9	Some wear, cage cracked (carbon graphite)
Brg. 455760 (Basic 7205) M1 Steel						
M1 Steel Brg. & Cage, coated	1000	50	2350	Air	1.0	Brg. O.K.
with Silver & Gold Oxides	"	175	5350	"	0.17	Very noisy, slight wear
M1 Steel, Belco	600	50	2350	Air	1.0	Brg. O.K.
5016 graphite grease	"	175	5350	"	7.1	Brg. seized Rings O.K. cage worn
Brg. 455760 (Basic 7205) M1 Steel						
Purebon P 03XHT	750	50	2350	N_2	100.	Brg. O.K.
Ring of 416	1000	"	5350	"	13.	Cage & stainless ring)
Stainless on O.D.	750	"	2350	"	100.	Brg. O.K. broken)

TABLE 2

Bearing Grade Hot Pressed Silicon Nitride
Material Properties and Inspection Data

Average Physical Properties for Bearing Calculations:

Young's modulus	310 GPa (45 x 10^6 psi)
Poisson's ratio	0.26
Density	3.2 g/cm^3
Thermal expansivity	2.9 x 10^{-6} per °C
Thermal conductivity	7.3 Cal/s·m°C (18 Btu/hr·ft°F)

Inspection Data on Test Materials:

	Norton NC-132	Toshiba Standard	Toshiba Experimental
Modulus of rupture, MPa (ksi)	866 (126)	785 (114)	951 (138)
Density, g/cm^3	3.25	3.25	3.21

Program SHABERT Analysis of Silicon Nitride Bearings of Mean Design in a10 KW Unit with Minimal Lubrication and 25 Kg (220N) Preload, including Mounting Fit calculations

INPUT DATA

*** S H A B E R T H / A B R ** TECHNOLOGY DIVISION S K F INDUSTRIES INC. ** S H A B E R T H / A B R ***

SOLAR GAS TURBINE TWO BEARINGS LEVEL 2 SOLUTION SIN BALLS

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	11	.000	20.244	-.001	21.590	93500.	0.
2	11	.000	20.244	-.001	-21.590	93500.	0.

Cage Data

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	OUTER RING LAND RIDING	.317500	.7600	23.1140	.102	.001000
2	OUTER RING LAND RIDING	.317500	.7630	23.1140	.102	.001009

Steel Data

BRG. NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50	15.000	M-50	15.000
2	M-50	15.000	M-50	15.000

Rolling Element Data

BEARING NUMBER (1) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
4.7625	.530	.517

Surface Data

BEARING NUMBER	CLA ROUGHNESS OUTER	INNER	ROLL. ELM.	RMS ASPERITY SLOPE OUTER	INNER	ROLL. ELM.
1	.13	.10	.03	1.500	1.500	1.500
2	.13	.10	.03	1.500	1.500	1.500

Lubricant Data

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	(98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152
2	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152

Lubrication and Friction Data

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	INNER	ASPERITY FRICTION COEFFICIENT
1	.50	.1000-03	.5000-04	.10
2	.50	.1000-03	.5000-04	.10

Table 3.

Fit Data and Material Properties

BEARING NUMBER	COLD FITS - (+= TIGHT) SHAFT	HOUSING	EFFECTIVE WIDTHS SHAFT	INNER RING	OUTER RING	HOUSING
1	.0050	-.0010	16.0000	8.0000	8.0000	8.0000
2	.0050	-.0010	16.0000	8.0000	8.0000	8.0000

BEARING NUMBER	SHAFT I.D.	BEARING BORE	EFFECTIVE DIAMETERS INNER RING AVE. O.D.	OUTER RING AVE. I.D.	BEARING O.D.	HOUSING O.D.
1	.000	12.000	17.000	24.000	28.000	44.500
2	.000	12.000	17.000	24.000	28.000	44.500

BEARING NUMBER (1)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	310264.0	204083.0	204083.0
POISSONS RATIO	.3000	.3000	.2600	.3000	.3000
WEIGHT DENSITY	7.806	7.806	3.190	7.806	7.806
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00000320	.00001224	.00001224

SIMPLE TEMPERATURES

BRG	SHAFT	I. RING	I. RACE	ROLL EL.	O. RACE	O. RING	HOUSING	BULK	FLANGE
1	93.50	93.50	93.50	93.50	93.50	93.50	93.50	68.50	.00
2	93.50	93.50	93.50	93.50	93.50	93.50	93.50	68.50	.00

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X-- Y.

21 GEOMETRIC SECTIONS 3 LOAD SECTION(S), 2 BEARINGS, MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	RIGHT	OUTER DIAM. LEFT	RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	RIGHT		BEARING SEAT POS.ERR	DEFL/FOR	ANG.ERR	DEFL/MOM
1	.0	.0	.0	.0	7.0					1				
2	34.0	.0	.0	7.0	12.0					2				
3	34.0	.0	.0	12.0	12.0					3	.000	0.00	.0000	0.00
4	43.0	.0	.0	12.0	15.0					4				
5	47.0	.0	.0	15.0	30.0					5				
6	64.0	.0	.0	30.0	43.0					6				
7	76.0	.0	.0	43.0	49.0					7				
8	89.0	.0	.0	49.0	53.0					8				
9	97.0	.0	.0	57.0	59.0					9				
10	104.0	.0	.0	64.0	64.0					10				
11	135.0	.0	.0	64.0	64.0	35.3				11				
12	164.0	.0	.0	64.0	64.0					12				
13	172.0	.0	.0	62.0	62.0					13				
14	173.0	.0	.0	62.0	57.0					14				
15	186.0	.0	.0	57.0	44.0					15				
16	195.0	.0	.0	44.0	42.0					16				
17	207.0	.0	.0	42.0	33.0					17				
18	213.0	.0	.0	33.0	30.0					18	.000	0.00	.0000	0.00
19	222.0	.0	.0	30.0	16.0					19				
20	[illegible]	.0	.0	15.0	12.0					20				
21	270.0	.0	.0	12.0	.0	10.0				21				

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD, PLANE X-- Z.

1[illegible] GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS, MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = -4.450+01

Table 3 (Continued)

Bearing System - Output Data (Metric Units)

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (N-MM)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	-1.033-03	2.731-04	4.157-10	8.906-13	5.259-06	222.	12.1	-.385	-1.57	-4[illegible].3
2	-1.008-03	5.101-04	3.872-10	1.076-13	1.162-06	-268.	32.5	-.308	1.52	1[illegible]5.

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	4.557+04	3.414+04	2.046+04	1.62	1.29	.566	.483	15.0	15.0
2	3.081+04	1.774+04	1.232+04	1.60	1.26	.562	.483	15.0	15.0

FRICTIONAL HEAT GENERATION RATE (WATTS) AND FRICTION TORQUE (N-MM)

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	23.6	0.000	34.2	0.000	14.3	5.07	24.2	102.	10.4
2	26.4	0.000	41.4	0.000	13.9	4.70	23.8	110.	11.2

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.137	.122	.762	.572	.840	.853	.125	6.763-02	.574	.564
2	.135	.120	.756	.571	.843	.852	.129	7.121-02	.624	.640

	FIT PRESSURES (N/MM2)				BEARING CLEARANCES (MM)			SPEED GIVING ZERO FIT PRESSURE
BRG.	SHAFT-COLD.	OPER.	HSG.-COLD.	OPER.	ORIGINAL	CHANGE	OPERATING	SHAFT-INNER-RING (RPM)
1	23.5	12.1	0.030	0.000	-1.395-03	-4.235-03	-5.735-03	1.319+05
2	23.5	12.2	0.000	0.000	-1.395-03	-4.202-03	-5.702-03	1.325+05

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

LOCATION	TEMPERATURES (DEGREES.C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1 - OUTLET	93.300	.8975	3.504	3.144	.1062-01
INLET	93.300	.8975	3.504	3.144	.1062-01
HULK	69.399	.9152	5.656	5.176	.1231-01

C A G E D A T A METRIC UNITS

	CAGE RAIL - RING LAND DATA				CAGE SPEED DATA					
BRG.	TORQUE (N-M)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	-6.03	24.2	0.000	0.000	4.054+03	3.871+04	4.016+03	3.835+04	.991	.418
2	-5.78	23.4	0.000	0.000	4.024+03	3.842+04	3.982+03	3.803+04	.990	.407

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-21427.135	1416.806	-172.637	21474.619	4006.365	176.22	-179.54
32.73	-21423.262	1432.184	-175.354	21471.797	4007.268	176.18	-179.53
65.45	-21419.161	1441.026	-180.446	21468.339	4008.006	176.15	-179.52
98.18	-21432.243	1452.137	-187.943	21482.203	4012.161	176.12	-179.50
130.91	-21454.927	1459.007	-194.989	21505.363	4017.937	176.11	-179.48
163.64	-21478.227	1459.970	-197.435	21528.715	4023.219	176.11	-179.47
196.36	-21495.906	1453.390	-199.852	21545.910	4026.527	176.13	-179.47
229.09	-21499.835	1441.060	-195.886	21548.965	4026.207	176.17	-179.48
261.82	-21492.215	1427.981	-190.078	21540.398	4022.998	176.20	-179.50
294.55	-21472.376	1416.753	-181.362	21519.428	4017.295	176.23	-179.52
327.27	-21450.193	1413.412	-175.393	21497.558	4011.413	176.25	-179.53

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CASE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.039	70.224	43.158	1486.170	1386.319	.0433	.0917	16.98	28.75
32.73	-.076	69.813	42.741	1483.258	1382.443	.0433	.0919	16.95	28.82
65.45	-.057	68.719	41.642	1475.444	1370.482	.0433	.0923	16.86	29.01
98.18	-.044	67.420	40.294	1466.087	1355.540	.0432	.0928	16.75	29.27
130.91	-.041	66.325	39.133	1458.140	1342.393	.0430	.0932	16.64	29.51
163.64	-.050	65.747	38.485	1453.864	1334.933	.0429	.0935	16.58	29.65
196.36	-.068	65.817	38.510	1454.378	1335.230	.0428	.0935	16.57	29.66
229.09	-.091	66.503	39.197	1459.412	1343.117	.0429	.0932	16.63	29.52
261.82	-.113	67.647	40.378	1467.736	1356.474	.0429	.0927	16.74	29.29
294.55	-.124	68.913	41.713	1476.834	1371.262	.0430	.0922	16.85	29.04
327.27	-.119	69.706	42.773	1483.492	1382.786	.0431	.0918	16.94	28.83

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-21134.041	-1863.345	186.959	21216.853	3964.349	-174.96	179.49
32.73	-21117.920	-1896.142	192.033	21203.744	3963.737	-174.87	179.48
65.45	-21123.307	-1928.282	205.508	21212.731	3964.318	-174.78	179.44
98.18	-21152.595	-1944.477	223.822	21243.331	3977.217	-174.74	179.39
130.91	-21194.556	-1953.874	241.202	21285.794	3987.681	-174.73	179.35
163.64	-21238.019	-1941.451	252.233	21328.098	3996.773	-174.78	179.32
196.36	-21258.715	-1914.285	252.471	21356.360	4031.468	-174.85	179.32
229.09	-21276.639	-1883.000	242.035	21361.171	4030.074	-174.94	179.35
261.82	-21256.543	-1851.673	225.740	21338.262	3992.102	-175.02	179.40
294.55	-21214.335	-1834.334	205.343	21294.531	3938.466	-175.06	179.45
327.27	-21169.656	-1833.195	191.439	21250.261	3970.399	-175.03	179.48

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CASE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.042	81.210	56.799	1572.617	1519.888	.0424	.0884	-18.32	-27.70
32.73	-.027	82.306	55.607	1565.000	1509.181	.0429	.0887	-18.26	-27.82
65.45	.002	79.028	52.579	1545.821	1481.273	.0428	.0895	-18.09	-28.16
98.18	.013	75.493	48.940	1522.418	1446.280	.0426	.0906	-17.86	-28.63
130.91	.018	72.549	45.909	1502.640	1415.786	.0424	.0915	-17.64	-29.07
163.64	-.030	71.052	44.260	1491.962	1393.627	.0422	.0921	-17.51	-29.34
196.36	-.041	71.144	44.291	1492.603	1378.446	.0421	.0921	-17.50	-29.35
229.09	-.069	72.827	45.986	1504.278	1416.575	.0421	.0915	-17.63	-29.09
261.82	-.103	75.759	49.024	1524.272	1447.101	.0422	.0906	-17.85	-28.65
294.55	-.115	79.240	52.634	1547.199	1481.792	.0425	.0895	-18.08	-28.18
327.27	-.097	82.119	55.635	1565.716	1503.427	.0427	.0887	-18.25	-27.83

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