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FINAL REPORT

RESEARCH AND DEVELOPMENT OF MATERIALS FOR USE AS LUBRICANTS IN A LIQUID HYDROGEN ENVIRONMENT

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Propulsion and Vehicle Engineering Division
Engineering Materials Branch
George C. Marshall Space Flight Center
Huntsville, Alabama

Prepared by: W. H. Rempe
W. H. Rempe
Senior Experimental Engineer

W. E. Young
W. E. Young
Program Manager

Approved by: Carl R. Comolli
C. R. Comolli
Project Engineer

Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION
FLORIDA RESEARCH AND DEVELOPMENT CENTER

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FOREWORD

This report presents a summary of the work conducted under the Phase II portion of Contract NAS8-11537, sponsored by the George C. Marshall Space Flight Center of the National Aeronautics and Space Administration, Huntsville, Alabama. The work was administered under the technical direction of the Propulsion and Vehicle Engineering Laboratory, Engineering Material Division, with Mr. K. E. Demorest acting as Contracting Officer's Representative, and covers the period from July 1964 to November 1965. A summary of the work conducted under the Phase I portion of the contract, covering the period from August 1963 to July 1964, was presented in PWA FR-986, "Research and Development of Materials for use as Lubricants in a Liquid Hydrogen Environment," dated 18 June 1964.

ABSTRACT

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This report presents the results of work performed in the second phase of Contract NAS8-11537, "Research and Development of Materials for Use as Lubricants in a Liquid Hydrogen Environment." The objective of the program was to develop materials for use as lubricants in rolling-element bearings operating in a liquid hydrogen environment at DN levels up to 4.0×10^6 mm-rpm. In the first phase, the results of which were reported in Reference 1, 10 candidate lubricant materials were selected and evaluated in a ball-plate test apparatus. In the second phase, 10 additional materials were evaluated and from the 20 candidates, the four most promising were selected for testing in actual 80-mm bearings at speeds up to 50,000 rpm.

Author

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SECTION I
INTRODUCTION

The use of liquid hydrogen in advanced rocket engines and the inherent problems associated with the lubrication of high speed turbopump bearings operating in liquid hydrogen led to the initiation of Contract NAS8-11537, "Research and Development of Materials for Use as Lubricants in a Liquid Hydrogen Environment." Liquid hydrogen, with its extremely low temperature (-423°F) and high specific heat, makes an excellent bearing coolant. However, hydrogen has a low viscosity and is an active reducing agent that tends to remove protective oxide coatings, which makes it a poor lubricant. This makes it necessary to provide other means of lubrication for bearings operating in liquid hydrogen. The most common approach is to incorporate a solid lubricant as an integral part of the bearing retainer; the lubricant material is then transferred by the balls to the races. This method has proved highly successful in many applications such as the RL10 rocket engine in an advanced turbopump. In the latter the bearings have been operated in hydrogen for prolonged periods at DN levels above 2×10^6 mm-rpm. However, to extend bearing DN limits and/or increase life, new improved lubricant materials must be developed. For this purpose work on the first phase of the contract was initiated in June 1963.

The approach used in this program has been to evaluate selected candidate lubricant materials in a controlled simulated bearing environment, and then conduct full-scale bearing tests using the four most promising candidate materials as determined by the screening tests. The material evaluation was performed in a ball-plate test apparatus that simulates operating conditions in a rolling contact bearing. The ball-plate tester is a unique device that eliminates the cost of special test bearings and promotes a more accurate evaluation of a lubricant by eliminating the confounding effects of bearing structural considerations.

In the Phase I portion of the contract 10 candidate materials were selected and evaluated. Two of these materials, a bronze-filled Teflon (Salox-M) and a silicon-filled Teflon (Rulon A) showed excellent lubricating characteristics and wear resistance. In the second phase, 10 additional candidates were selected and evaluated in an attempt to find lubricants having a higher resistance to nuclear radiation than the Teflon-based materials. Three of these 10 candidates, a bronze-filled polyimide, a silver-polyimide-tungsten diselenide (Ag-Polyimide - WSe_2) composite and silver molydisulfide (Ag-MoS_2) showed promise.

Four of these five materials, Salox-M, bronze-filled polyimide, Ag-Polyimide - WSe_2, and Ag-MoS_2 were selected and tested in 80-mm bearings at speeds up to 50,000 rpm (4×10^6 DN). Three of the four materials operated successfully at 50,000 rpm for short periods. The bearing retainers that incorporated polyimide-bronze lubricant failed before reaching 50,000 rpm.

SECTION II
TEST APPARATUS

Comparitive evaluation of many bearing lubricants by testing actual bearings would be costly and time consuming. There are many interactions associated with an operating bearing that can affect the life of the bearing, but which have little or nothing to do with the lubrication process. A typical example is the structural limitations of a rotating retainer. It is advantageous, therefore, to develop a method of material screening that will provide a fundamental evaluation of a lubricant without the cost of parts procurement and testing of special bearings. However, this does not eliminate bearing testing, because lubricants developed and evaluated in simulating devices must ultimately be tested in actual ball bearings. The test apparatus used to select the most effective lubricants should eliminate as many variables as possible and still provide a realistic evaluation. Such a test apparatus can be considered as a fundamental evaluation tool making possible a test technique similar to tensile testing of materials, where material properties are evaluated prior to fabrication into an operational part.

A unique test apparatus of this nature has been used to evaluate lubricating oils for jet engines in an environment that simulates operating conditions including various levels of Hertzian stress (contact stress), slip in the ball-to-race contact zone, and ball rotational velocity in a retainer pocket. This ball-plate test apparatus, which was modified for cryogenic use, is shown in figure II-1. It consists of two counter-rotating test plates separated by three balls that are positioned 120 degrees apart by a retainer plate. On either side of the center housing, the shafts that drive the test plates are supported by oil-lubricated rolling element bearings. Axial loading of the three balls is applied by placing dead weights on the end of the lever arm, which is attached to the left shaft support assembly. This assembly is free to slide axially, thereby transmitting thrust directly to the rotating plate. An expanded view of the test plates, balls and retainers is also shown in figure II-1. The shaft bearings are lubricated with heated oil and the bearing compartments are separated from the test compartment by a twin face-seal system. Helium is injected between the seals to prevent oil-hydrogen mixing and possible freezing problems. The lubricant containing the ball pockets is made in the form of inserts, which are installed in the retainer plate.

The principle of using counter-rotating plates was adopted to limit rotation of the retainer to low speeds, thereby eliminating any structural problems associated with high stresses due to rotation. This arrangement also permits relatively low speed drives to be used to obtain tests at high equivalent DN values.

Failures are indicated by axial vibration measured by an accelerometer, which is located on the loading lever. This technique has proved highly successful, and small changes in the surface of a ball or race are easily detected. The millivolt output of the accelerometer is fed into an automatic abort system that shuts off the drive power when the output reaches a predetermined value.

The Herzian stress level can be adjusted by changing the included angle of the groove and/or the applied axial loads. For a constant applied dead weight on the lever arm, the Hertzian stress to which the ball and grooved plate are subjected is a function of the ball diameter and the groove angle. Ball spin will occur about an axis drawn from the grooved plate contact points through the ball center, and is controlled by varying the plate groove angle. In simulating these parameters, this test apparatus was designed consistent with a theory of failure that relates the principal causes of surface fatigue to (1) the level of Hertzian stress in the contact zone, and (2) slip in the contact zone.

Excluding the possibility of a retainer failure, a ball bearing will fail when a spot on the surface of one of its elements (ball, inner race, or outer race) ruptures from fatigue due to a vibratory stress associated with ball passing or rolling frequency. (The steady or constant stresses on these elements are insignificant.) The number of cycles that a material can withstand at various vibratory stress levels is usually depicted by an S-N curve (vibration stress level - life in cycles) similar to the solid line shown in figure II-2. If the material is subjected to a vibratory stress level, σ_1, then it will last 10^7 stress cycles and more. This value of 10^7 cycles is referred to as the runout life, meaning that if a stressed material lasts 10^7 cycles, it will not fail, no matter how many additional cycles are imposed. This is certainly not rigidly true, but is sufficiently accurate for this discussion. If the material is stressed to σ_2, the material will fail at 10^n cycles where $n < 7$. This curve is generally used in showing the fatigue strength of beams and other structural members when subjected to vibratory bending (tensile, and compressive) loads. However, the same effect is true with respect to a surface under contact compression; the only difference is that the surface does not experience complete stress reversals. The cycle goes from zero stress to maximum compressive stress and back to zero stress.

The dotted line on figure II-2 shows the effect of surface notching on fatigue life. The notching effect in ball bearings can result from surface fretting where spinning or slipping occurs in the contact zones. Under such conditions of contact zone slip, the material surface is damaged and the fatigue life of the material is reduced for any given stress level. Thus, as surface slip occurs and surface damage accumulates, the S-N curve shifts downward with increasing number of cycles. After the surface damage has evolved, the life would be 10^m at a stress level of σ_1 instead of the runout life that would be available with no slip (no surface damage). Figure II-3 shows a curve based on experimental data for the effect of life of an oil-lubricated ball subjected to a constant Hertzian stress and various ratios of spin velocity to roll velocity. For a cryogenic bearing application, the curve would move to the left because of the higher coefficient of friction and correspondingly greater surface damage at any given level of slip. This curve confirms the important effect of contact zone slip on life and in general, substantiates the conclusion that the prime factors governing the life of a bearing are the slip in the contact zone and the Hertzian stress.

Thus, for a true evaluation of a lubricant, the Hertzian stress in the contact zone, spin/roll velocity ratios, and ball rotational velocity must be defined and simulated in the test apparatus.

Using a mathematical model, these parameters were determined for bearings compatible with the requirements of advanced liquid rocket engines. The mathematical model predicts the bearing internal dynamics for any combination of speed and thrust load.

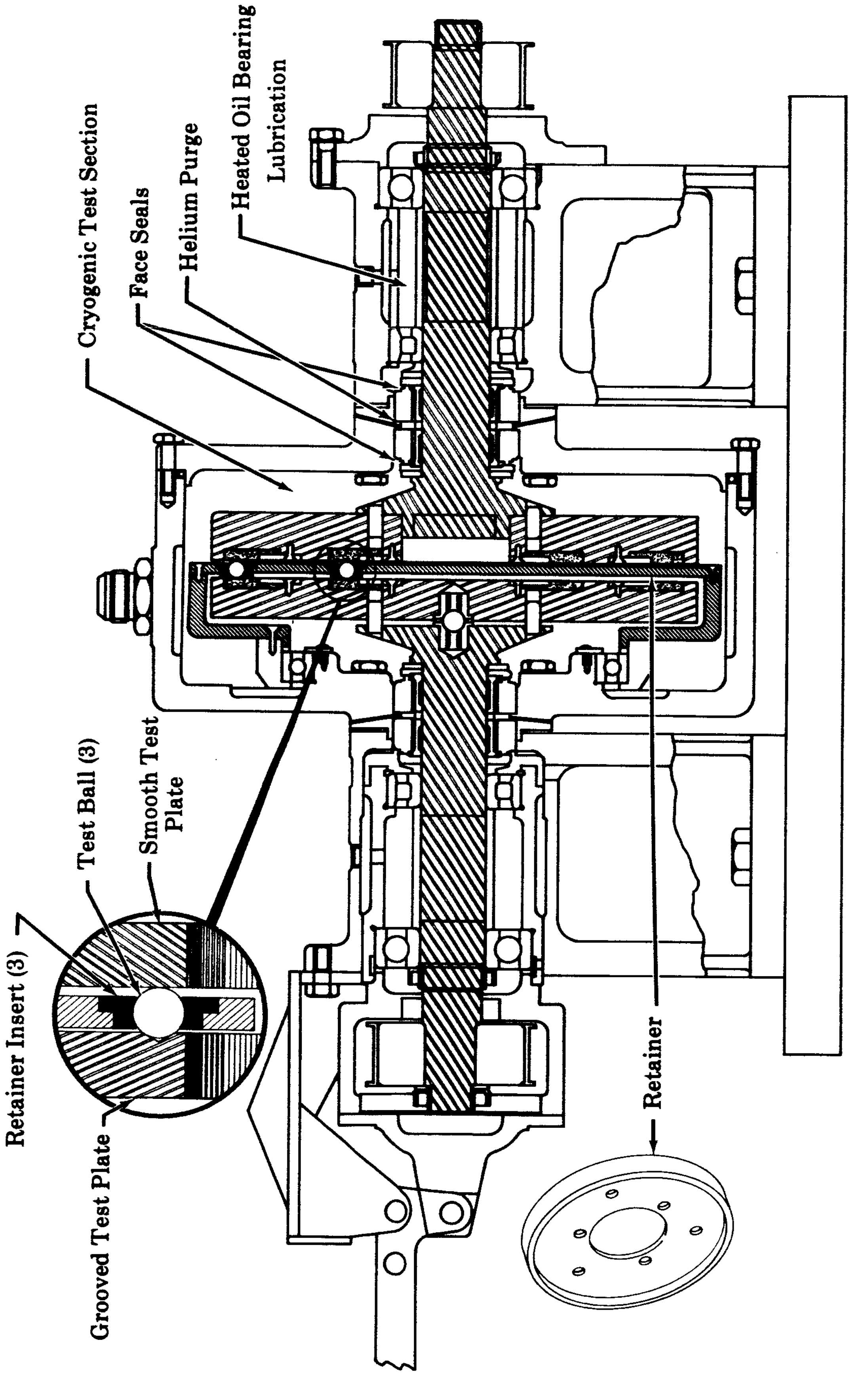

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Figure II-1. Ball-Plate Test Apparatus (Modified for Cryogenic Testing)

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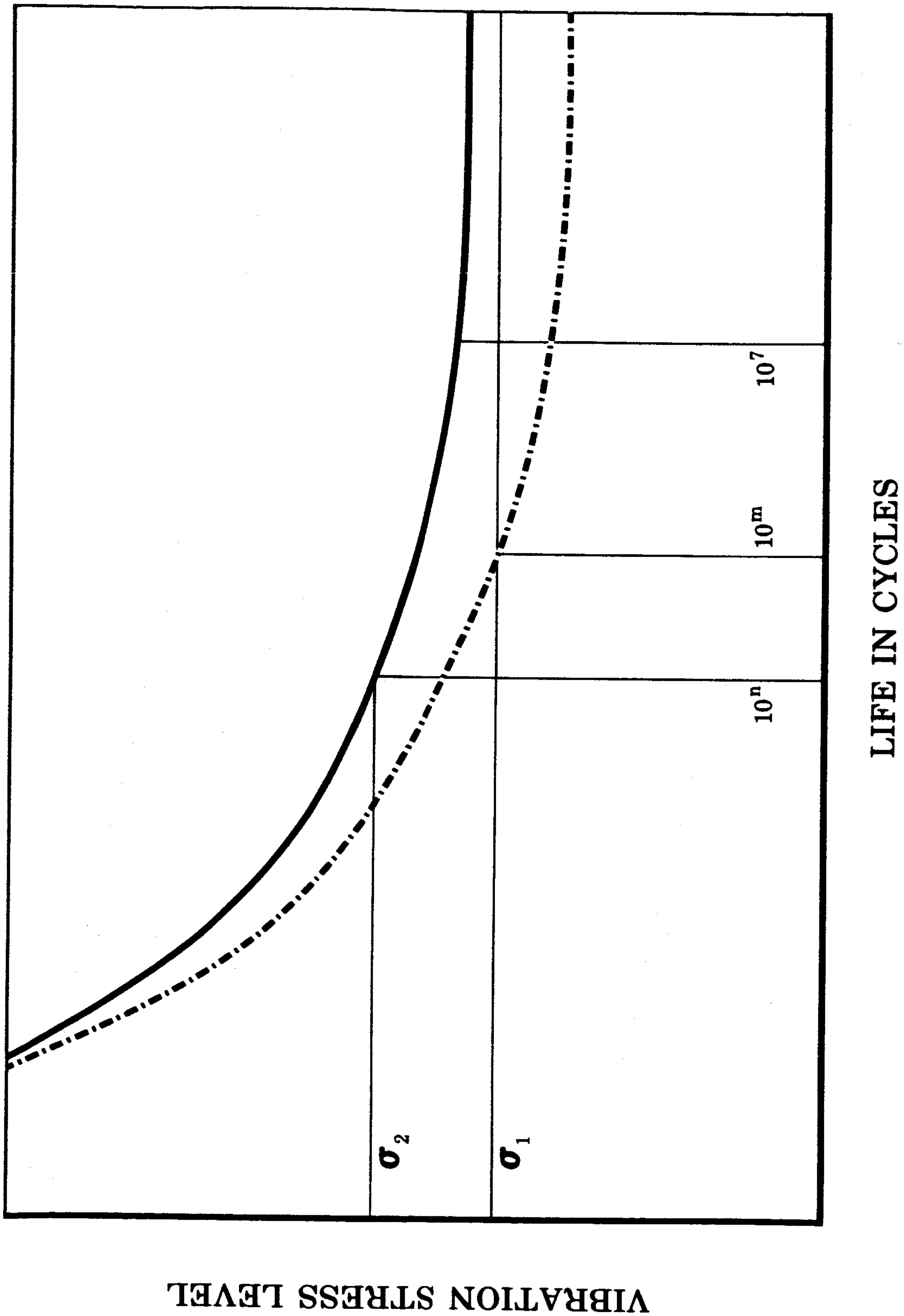

Figure II-2. Typical Stress-Cycle Curves

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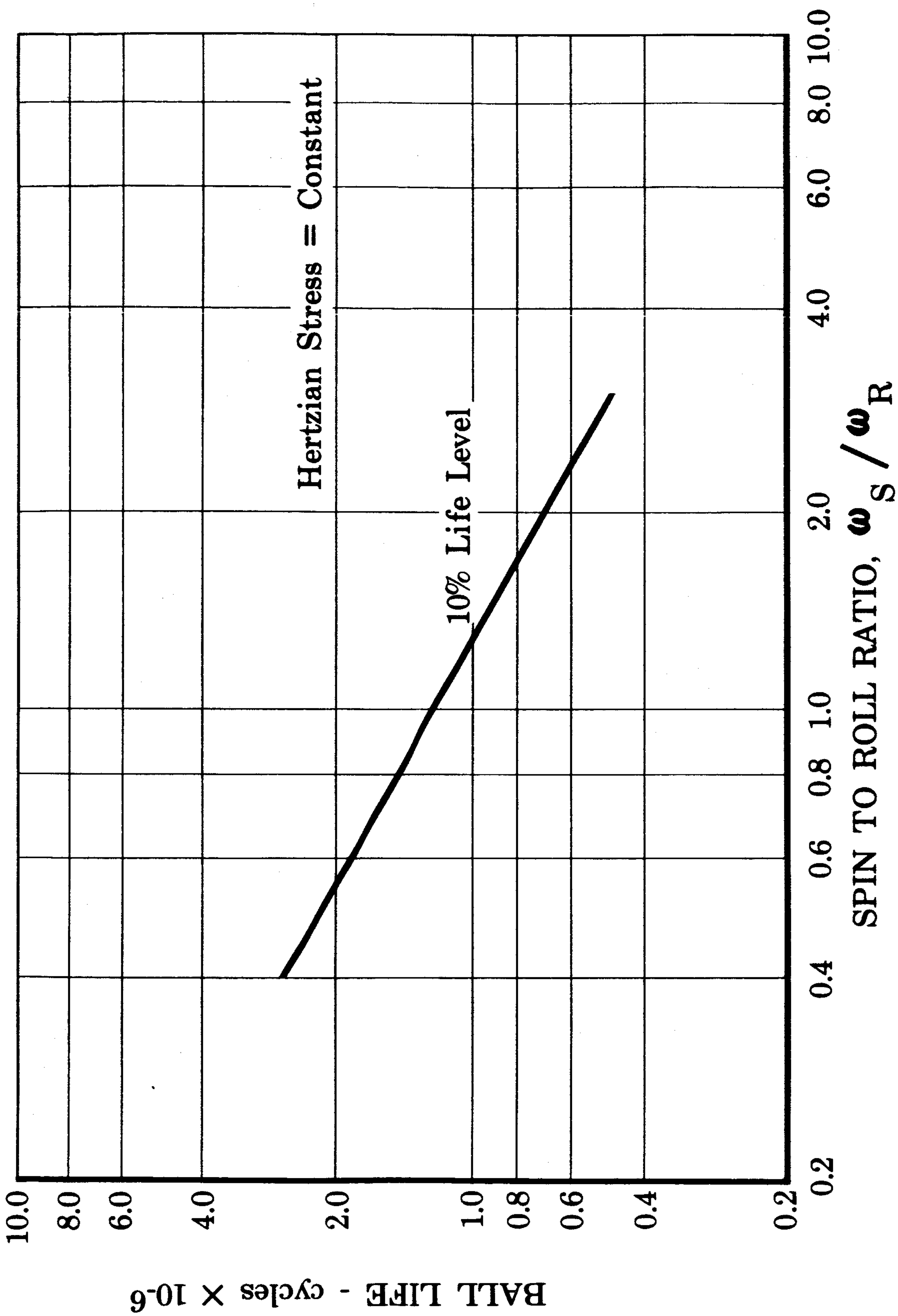

Figure II-3. Relationship of Ball Life to Spin/Roll Ratio

A study was conducted in Phase I to establish the required plate speed, ball diameter, included angle of the grooved ball track, and the radius of the grooved ball track required to simulate actual bearing conditions. In this program an inner race Hertzian stress of 250,000 psi was chosen since this level is considered near the maximum permissible value for reliable bearing design. A summary of the test apparatus parameters is presented in the following table.

Ball-Plate Test Apparatus Parameters

Equivalent DN value, mm-rpm	4,000,000
Plate speed, rpm	10,000
Ball velocity, rpm	221,000
Groove radius, in.	4.0
Ball diameter, in.	0.375
Plate groove angle, degrees	147
Ball spin velocity/ball roll velocity	0.295
Thrust per ball, lb	70

SECTION III
SELECTION OF CANDIDATE LUBRICANT MATERIALS

A. GENERAL

In the first phase of the contract a literature survey was conducted in which more than 50 technical articles were studied to determine the most likely candidate lubricant materials and/or lubricating systems. The survey covered various methods of providing lubrication such as powdered injection and flame-plated surfaces, as well as solid lubricant composites. The additional system complexity associated with powdered injection overshadowed possible advantages of this method, and the survey indicated that the metalized or flame-sprayed wear-resistant raceways by themselves would not be adequate for the required speeds and loads. As a result, no candidate representative of these lubrication methods were selected.

Table III-1 presents a tabulation of the lubricating materials that were selected for evaluation in the ball-plate tester. The first material, Rulon A (silicon-filled Teflon), was selected as the performance "standard" against which all the other materials were compared. The "standard" and the first ten candidate lubricant materials listed in table III-1 were tested in Phase I. A detailed discussion of the Phase I material selection and results of the evaluation tests is presented in Ref. 1. Candidate lubricants number 11 through 20 were tested in Phase II of the program and are discussed in Section IV of this report.

The testing in the Phase I portion of Contract NAS8-11537 produced several materials that showed promise as lubricants for use in a liquid hydrogen environment, the most outstanding of which was Salox M. This material, which is a bronze-filled Teflon, demonstrated a high degree of lubricity and wear resistance, but due to the presence of Teflon it has a relatively low resistance to radiation damage (10^5 ergs/gram). Recognizing that some future applications are expected to require satisfactory bearing operation in a liquid hydrogen environment at much higher radiation levels, the Phase II portion was directed toward development of lubricants with higher radiation resistance. Several of the materials selected in Phase II were refinements of composites that showed promise in the Phase I or early Phase II testing. The candidate lubricant materials selected for evaluation in Phase II can be categorized as follows:

1. Filled organic polymers
2. Dry lubricant coating
3. Metal composites.

These materials or material combinations possess varying degrees of resistance to nuclear radiation from 7×10^{11} ergs/gram (C) for the polymers to greater than 10^{17} ergs/gram(C) for the metal composites. Ergs per gram (carbon) refers to the energy absorbed by the sample. The above threshold dosage value is for static irradiation in air.

B. PHASE II CANDIDATE MATERIAL SELECTION

The following paragraphs discuss the selection of Phase II candidate lubricants, starting with No. 11.

1. Candidate No. 11 (Polyimide - Molydisulfide-Copper)

A MoS_2 - filled polyimide was tested during Phase I and demonstrated good lubricating properties, but rather poor wear resistance. It appeared from the results of these tests that the wear was due to the low thermal conductivity of the material, which resulted in a high surface temperature in the ball pocket and flowing of the material under load. The good lubricating qualities of this material made it worthy of consideration for further testing in Phase II. To improve the thermal conductivity and resistance to wear, copper was added to give a composition of 70% polyimide-10% MoS_2-20% copper. This material was selected as the first candidate for Phase II testing.

2. Candidate No. 12 (Nickel - Molydisulfide)

The next candidate selected was a metal composite of nickel and MoS_2. A similar composite (Ag - MoS_2) was tested in Phase I and displayed good lubricating characteristics, but poor wear resistance. A reduction in MoS_2 content was considered as a means of improving wear resistance, but data from wear rate tests conducted under a separate program revealed that a reduction in MoS_2 to as low as 5% had no significant effect on wear rate. Therefore, the silver was replaced with nickel as the matrix material in an attempt to improve wear resistance.

Table III-1. Summary of Candidate Lubricant Materials (Balls and Races are AISI 440C)

Candidate Number	Lubricant Material
A	Silicon-filled Teflon (40-60 by weight), (Rulon A)
1	Bronze-Pb-PTFE on steel strip
2	Bronze-filled Teflon (40-60 by weight), (Salox M)
3	MoS_2-filled Teflon (15-85 by volume)
4	Ag-MoS_2 Matrix (80-20 by volume)
5	Al-MoS_2 Matrix (80-20 by volume)
6	Ag-CaF_2 Matrix (80-20 by volume)
7	Boron Nitride
8	Silicon-filled Teflon insert (Rulon A) with MLF-5 coatings on races
9	MoS_2-filled Polyimide (15-85 by weight)
10	Silicone-filled Teflon insert (Rulon A) with Fluorocarbon telomer coating on races
11	Cu-MoS_2-filled Polyimide (20-10-70 by volume)
12	Ni-MoS_2 matrix (80-20 by volume)
13	Bronze retainer with MLF-5 coating on races
14	Bronze-filled Polyimide (20-80 by volume)
15	Bronze-filled Polyimide (30-70 by volume)
16	Unfilled Polyimide
17	Bronze-filled Polyimide (40-60 by volume)
18	MoS_2 impregnated Ag (sintered fibers)
19	MoS_2 impregnated Bronze (sintered fibers)
20	Ag-WSe_2-filled Polyimide (75-5-20 by volume)

3. Candidate No. 13 (Bronze - Molydisulfide - Gold - Graphite)

A dry film lubricant (MLF-5) was selected as the 13th candidate. MLF-5 is a composite of MoS_2 - gold - graphite and was bonded to the surface of bronze ball retainer inserts and to the plates with a sodium silicate binder. No information could be found on the application of coatings such as MLF-5 to high-speed bearings, but experience with this lubricant in high-load low-speed applications warranted a material evaluation test in this program.

4. Candidates No. 14 and 15 (Polyimide-Bronze)

The polimide-molydisulfide copper composite (candidate No. 11) was tested early in Phase II and, like the polyimide-graphite composite tested in Phase I, displayed good lubricating qualities but very high wear rate and indications of low strength. The good lubricating qualities and relatively high resistance to radiation damage (10^9 - 10^{11} ergs/gram) of the polyimide warranted an extra effort to develop a lubricant based on this material. This property, coupled with the outstanding results with the Teflon-bronze combination (Salox M) tested in Phase I, led to the selection of a bronze-filled polyimide as the next candidate. The optimum ratio of bronze to polyimide was unknown. However, the same ratio of bronze (20% by volume) as in the Salox M was selected as the 14th candidate and 30% bronze by volume was selected as the 15th candidate. Testing of both the 20% and 30% combinations established trends as to the ratio that will provide optimum performance.

5. Candidate No. 16 (Unfilled Polyimide)

Recent tests on bearings using unfilled polyimide as the lubricant and conducted in a vacuum environment at temperatures up to 700°F indicate that the polyimide can operate satisfactorily at high temperature. The results of these tests seemed to contradict the theory that the low thermal conductivity of the polyimide would result in high surface temperatures in the ball contact area. However, the addition of metal fillers increases the weight of the lubricant and also reduces the strength of the polyimide. Therefore, increases the stress in the lubricant retainer. For these reasons, the unfilled polimide was selected as a candidate material.

6. Candidate No. 17 (Polyimide-Bronze, 60-40%)

Tests conducted on the unfilled polyimide and the polyimide-bronze composites (numbers 14 and 15) of 80-20 and 70-30 ratios by volume showed that the wear resistance increased with an increase in the percentage of bronze. Therefore, a composite with 60% and 40% bronze was selected as the 17th candidate. Tests on composites numbers 14 and 15 also revealed that as the content of bronze filler increased, the strength of the polyimide-bronze composite decreased. Based on this, a further increase in bronze content was not considered.

7. Candidate No. 18 (Silver Feltmetal Impregnated with MoS_2)

The maximum resistance to nuclear radiation in a lubricant material will require the use of purely metallic or ceramic composites in which lubrication is supplied by such materials as MoS_2, the diselenides, or ditellurides impregnated in a base or matrix material.

The silver-MoS_2 combination tested in Phase I has been the most outstanding metal composite tested to date. This candidate has demonstrated excellent lubricating characteristics, but rather poor wear resistance. Various modifications to the composite and the way it is fabricated have been considered to improve wear resistance. One of the most promising modifications is to replace the current sintered method of fabrication with an impregnated wire-fabric method. This technique, which involves the weaving of very fine wire into a clothlike structure that can be impregnated with the desired lubricant, should provide the strength of the base metal that is often reduced in a sintered composite. Therefore, the 18th candidate was a silver wire-fabric impregnated with MoS_2. A wire-fabric density of 50% was selected for this test to provide a relatively high volume of lubricant at the ball contact surface.

8. Candidate No. 19 (Bronze Feltmetal Impregnated with MoS_2)

Selection of silver feltmetal impregnated with MoS_2 as the 18th candidate was the result of a search for a material with high resistance to radiation damage and with higher wear resistance than the sintered Ag-MoS_2 composite tested in Phase I. Friction and wear tests on the silver feltmetal composite and on bronze feltmetal impregnated with

MoS_2 showed the bronze composite to be at least as good as the silver. The bronze-MoS_2 composite was an attractive candidate because of the inherent self-lubricating qualities of bronze and the transfer lubricating ability of the MoS_2.

9. Candidate No. 20 (Silver-Polyimide-Tungsten diselenide)

This candidate (silver-polyimide-WSe_2, 75-20-5% by volume) was selected because of its high tensile strength (approximately 10,000 psi) and good resistance to radiation damage. Tests conducted by the manufacturer have shown no significant change in hardness, friction, or wear after 75 days exposure to approximately 10^{13} fast neutrons/cm^2 and 10^8 R/hour gamma radiation. Friction and wear tests indicate that this composite possesses good lubricating qualities and high wear resistance. The high tensile strength is an important factor in the selection of a retainer material for a high speed cage design.

Other metal composites, as well as several ceramics, were considered for candidate lubricant materials. Two metal composites, 90% MoS_2-8% Fe-2% Pt and 80% MoS_2-16% Fe-4% Pd, were evaluated as possible candidates. These two composites were found to be the best in friction and wear characteristics of over 200 combinations tested by Campbell and Van Wyk (Ref. 2) at ambient and high temperatures. They also operated satisfactorily in subsequent bearing tests. A sample was made of the MoS_2-Fe-Pt composite for investigation. The sintered composite was found to be extremely brittle even at room temperature, which makes it unlikely that this composite could operate satisfactorily at cryogenic temperatures. It is possible that improvement in the sintering process may provide the required ductility. However, this would require extensive development work and was considered to be outside the scope of this contract.

C. REFERENCES

1. PWA FR-986 "Research and Development of Materials for Use as Lubricants in a Liquid Hydrogen Environment," dated 18 June 1964.

2. Campbell, M. E., and J. W. Van Wyk, "Development and Evaluation of Lubricant Composite Materials," Journal of American Society of Lubrication Engineers, December 1964, pp 463-469.

SECTION IV
EVALUATION OF SELECTED MATERIAL CANDIDATES

A. GENERAL

The performance evaluation of materials is more meaningful if compared to a basic requirement or standard. Part of the work completed under Phase I of this contract was directed toward establishing a standard by testing materials that had demonstrated a degree of success in existing cryogenic bearing applications. The material combination that was selected from these tests was Rulon A inserts with AISI 440C balls and races. From these tests of Rulon A, a minimum level of lubrication and wear performance was established. For consideration in subsequent bearing tests, a candidate material should meet the following requirements:

1. Complete a minimum of 10 hours running at an equivalent DN value of 4×10^6 mm-rpm with no ball or plate damage
2. Retainer insert wear not to exceed 50%

$$\text{insert wear} = \frac{\text{wear diameter - insert ID}}{\text{insert OD - insert ID}} \times 100 .$$

Figure IV-1 is a closeup view of the ball-plate test apparatus used in these tests with the test section open. The three test balls can be seen in their retainer inserts. Six of the Phase I and Phase II candidates completed 10 hours at equivalent DN values of 4×10^6 mm-rpm with no damage to the balls and plates. Inspection of the balls and plates showed them to be in excellent condition. The wear tracks were highly polished and showed no evidence of fatigue. On the other hand, the retainer insert wear varied from low to very high with only four candidates meeting the 50% wear requirement previously mentioned. Wear in excess of 50% was considered unacceptable for an actual ball bearing retainer. The tests of all 20 candidate materials are summarized in table IV-1.

The Salox M displayed the lowest wear of any of the materials tested. The lubricating quality of the Teflon combined with the high thermal conductivity of the bronze provided this composite with a high resistance to wear as well as good transfer lubricating characteristics. The test results of all Phase I candidates are discussed in Ref. 1.

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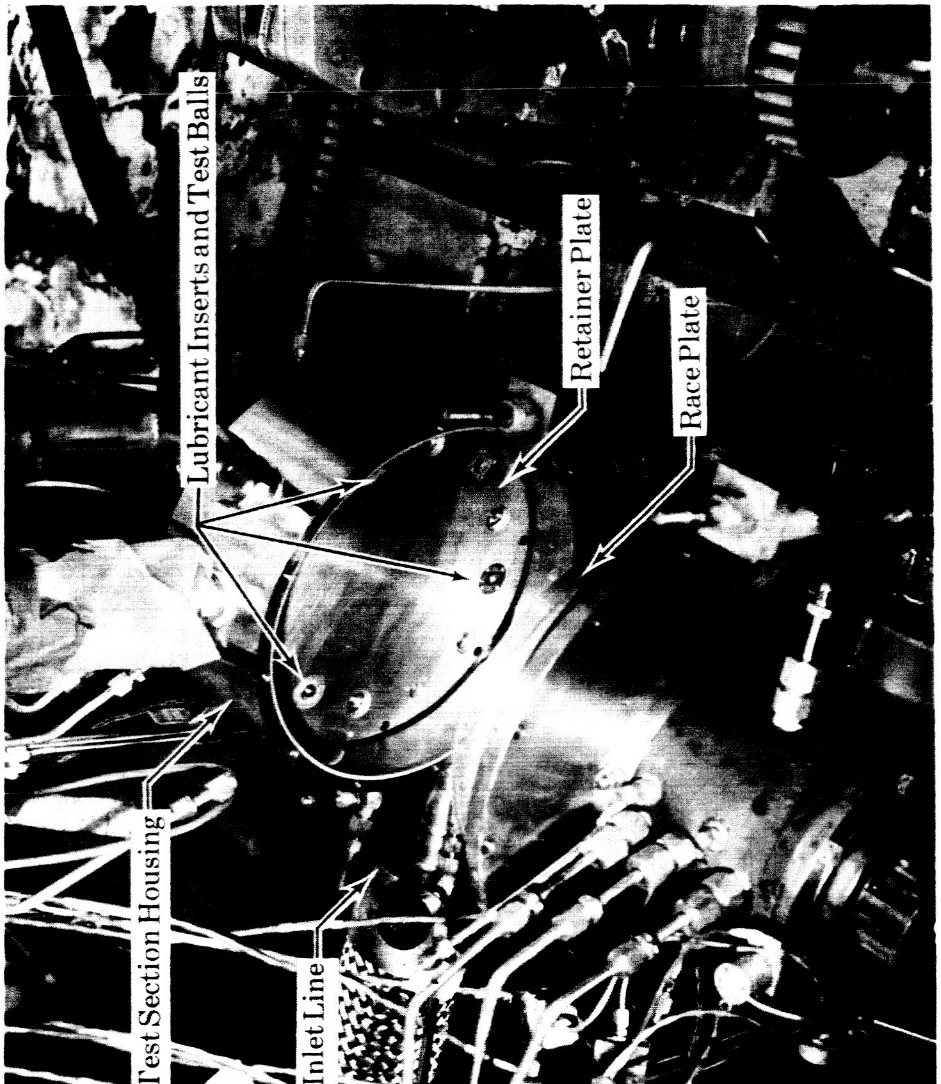


Figure IV-1. Closeup of Ball-Plate Test Apparatus

Table IV-1. Summary of Candidate Lubricant Materials

Candidate Number	Lubricant Material (Balls and Races are AISI 440C Throughout)	Limiting Test Condition DN Value x 10^{-6}	Hours to Failure (10 hr max)	Insert Wear
*A	Silicon-filled Teflon (40-60 by weight) (Rulon A)	4	10	Medium
1	Bronze-Pb-PTFE on steel strip	4	4.6	High
2	Bronze-filled Teflon (40-60 by weight) (Salox M)	4	10+	Low
3	MoS_2 filled Teflon (15-85 by volume)	2	5.5	Very high
4	Ag-MoS_2 matrix (80-20 by volume)	4	10	High
5	Al-MoS_2 matrix (80-20 by volume)	2	6.6	High
6	Ag-CaF_2 matrix (80-20 by volume)	2	3.5	Very high
7	Boron Nitride	2	5.5	Very high
8	Silicon-filled Teflon insert (Rulon A) with MLF-5 coatings on races	4	10+	Low
9	MoS_2-filled Polyimide (15-85 by weight)	4	4.7	High
10	Silicon-filled Teflon insert (Rulon A) with fluorocarbon telomer coating on races	4	4.5	High (coating built up)
11	Cu-MoS_2-filled Polyimide (20-10-70 by volume)	4	0.3	Very high
12	Ni-MoS_2 matrix (80-20 by volume)	4	3.0	Very high
13	Bronze retainer with MLF-5 coating on races	4	0.3	Very high (coating wore off)
14	Bronze-filled Polyimide (20-80 by volume)	4	9.0	High

*Reference material for comparison

Table IV-1. Summary of Candidate Lubricant Materials (Continued)

Candidate Number	Lubricant Material (Balls and Races are AISI 440C Throughout	Limiting Test Condition DN Value x 10^{-6}	Hours to Failure (10 hr max)	Insert Wear
15	Bronze-filled Polyimide (30-70 by volume)	4	10+	Medium
16	Polyimide (unfilled)	4	7.0	Very high
17	Bronze-filled Polyimide (40-60 by volume)	4	10+	Low
18	MoS_2 impregnated Ag (sintered fibers)	4	9.6	High
19	MoS_2 impregnated Bronze (sintered fibers)	4	0.2	Very high
20	Ag-WSe_2-filled Polyimide (75-5-20 by volume)	4	10+	Low

B. PHASE II CANDIDATE TEST RESULTS

Test results of phase II candidates are discussed in the following paragraphs.

1. Candidates No. 11 and 12 (Polyimide-MoS_2-Cu, and Ni-MoS_2)

The first test on each of these materials resulted in complete destruction of the material samples, making analysis of the type of failure virtually impossible. For the second test, the vibration abort limit was set at a level low enough to stop the test before the inserts were destroyed. With the lower abort limit, the polyimide-MoS_2-Cu material ran for approximately 1 hour and the Ni-MoS_2 composite ran for 12 minutes, compared to 3 hours and 17 minutes, respectively, for the first tests. In both cases, the material samples showed excessive wear and were broken out of the retainer plates. The balls and plates were badly damaged in both tests. The increase in wear rate of the polyimide-MoS_2-Cu over the polyimide - MoS_2 material tested in Phase I may be due to a failure to obtain a satisfactory bond between the constituents during fabrication. This is evidenced by the tendency of the material to flake or chip during testing.

The $Ni\text{-}MoS_2$ composite displayed much lower wear resistance than the $Ag\text{-}MoS_2$ combination of Phase I. Nickel is known to possess better-than-average wear resistance in high temperature applications where the environment maintains a continuous oxide film on the surface. The reducing action of the hydrogen in these tests tends to remove the protective oxide film, and the MoS_2 apparently cannot provide adequate lubrication. In addition, the nickel-MoS_2 was extremely brittle, leading to fracture-type failure.

2. Candidate No. 13 (Bronze-MLF-5)

For this test the plates and a set of bronze inserts were coated with MLF-5 using sodium silicate as the binder. The test was terminated after 5 minutes when vibration of the test apparatus exceeded the preset abort limit. Inspection of the parts revealed that the MLF-5 coating had worn off the inserts and plates in the ball contact areas, and the inserts were slightly worn. The poor performance experienced with this candidate material resulted in the decision that dry film lubricants applied in this manner should not be considered for further testing.

3. Candidates No. 14 and 15 (Polyimide-Bronze, 80-20 and 70-30% by Volume)

The first test of both candidates was terminated prematurely by a test apparatus drive belt failure; the 80-20% after 9 hours and the 70-30% after 2 hours. Failure of one of the two drive belts causes the insert carrier to accelerate. The centrifugal force on the balls caused by this rotation is resisted only by the lubricant insert, which wears away rapidly under the combination of ball velocity and load. In both instances, the test rig was shut down immediately and, even though the balls and plates were damaged, the wear on the inserts from normal operation prior to the failure could be determined. The 80-20% composite inserts showed relatively high wear and signs of excessive temperature at the ball-to-lubricant contact surface. The 70-30% composite inserts showed negligible wear in the normal ball contact areas. The second test of the 80-20% composite ended after 72 minutes when the test apparatus vibration exceeded the preset abort limit. The abort was caused by a sudden increase in vibration, which occasionally occurs when a small partical of the lubricant material breaks out and remains in the ball path. The test was not continued because, although the plates, balls, and inserts were in good condition, the wear rate on the inserts was excessive as in the previous test. The second test on the 70-30% composite completed 10 hours. The insert wear was moderate (figure IV-3) showing a marked improvement over the 80-20% composite.

4. Candidate No. 16 (Unfilled Polyimide)

This candidate was run for slightly over 7 hours before the vibration abort limit was exceeded. The inserts (figure IV-2) in this case were worn through and the balls and plates were discolored. There was no evidence of damage to either the balls or plates. The appearance of the inserts and balls indicated excessive temperature at the contact surface. This is essentially what had been expected and supports the decision to use a filler such as bronze to improve thermal conductivity.

5. Candidate No. 17 (Polyimide-Bronze, 60-40%)

This bronze-polyimide with 40% by volume of bronze successfully completed 10 hours at the equivalent DN of 4×10^6. The balls and plates were in excellent condition and the retainer insert wear was very low, second only to the Salox-M tested in Phase I. Figure IV-3 shows typical insert wear on this candidate.

6. Candidate No. 18 (Ag-MoS_2, Impregnated Feltmetal)

The first test on the MoS_2-impregnated silver feltmetal completed 9 hours and 20 minutes before being terminated by the vibration abort system. The inserts showed heavy wear, with the balls and plates in good condition except for a coat of silver. The second test on this candidate lasted only 1 hour and 30 minutes; again the insert wear was heavy and the balls and plates were well lubricated. Results of these tests were similar to those of tests conducted in Phase I on a sintered Ag-MoS_2 composite in that the candidate displayed excellent lubricating properties, but a comparatively low resistance to wear.

7. Candidate No. 19 (Bronze-MoS_2, Impregnated Feltmetal)

The two tests of the MoS_2 impregnated bronze feltmetal each lasted less than 20 minutes, at which time the vibration abort limit was exceeded. The lubricant inserts were badly worn and fractured in both tests, and the balls and plates were damaged. Low strength of the extremely fine wire used to make the feltmetal was apparently one of the major factors in the early failure; the material showed indications of "flowing" under the ball loading.

FE 48042

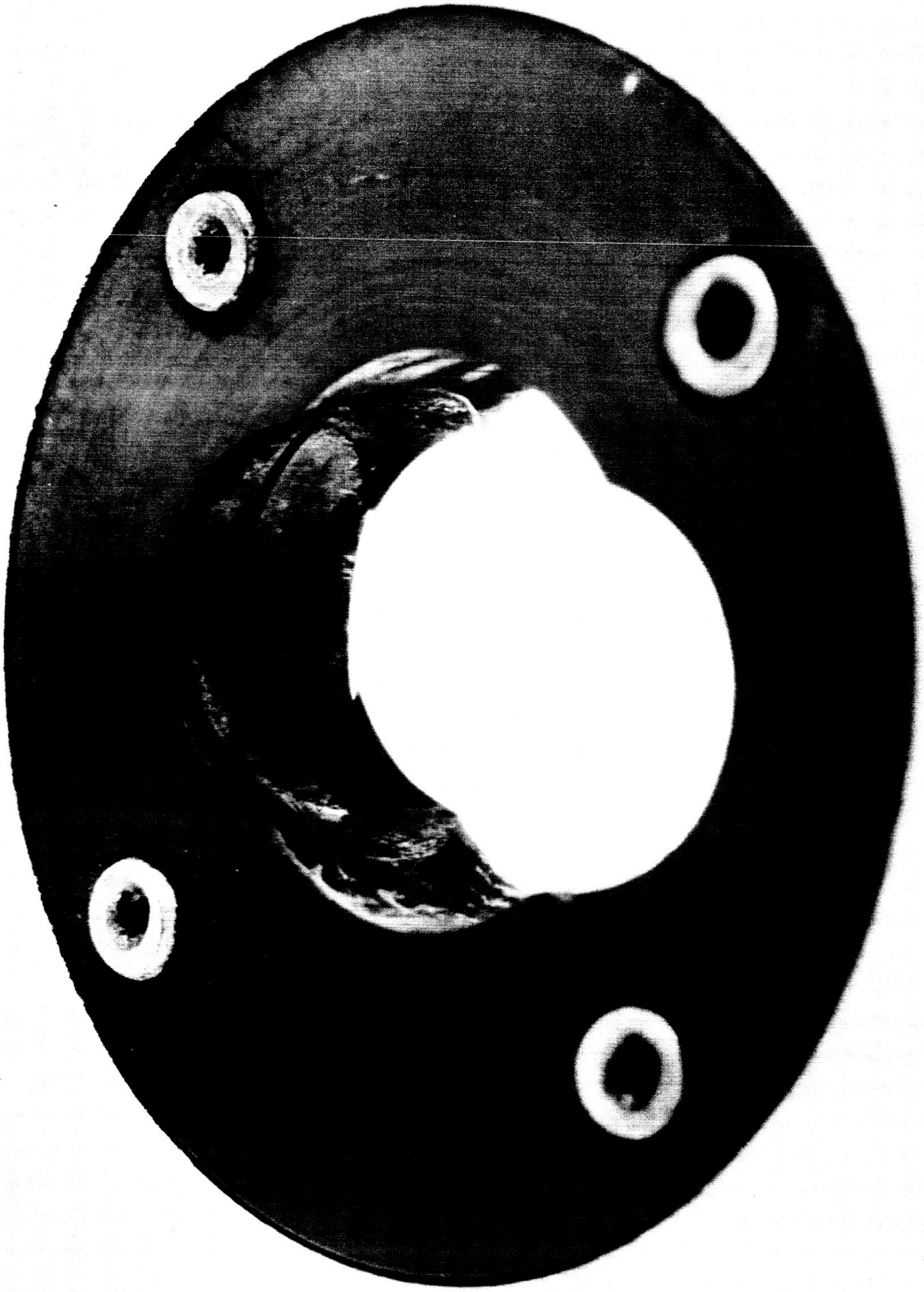


Figure IV-2. Unfilled Polyimide (SP-1) Insert after 7 hours at 4×10^6 DN

FE 47914

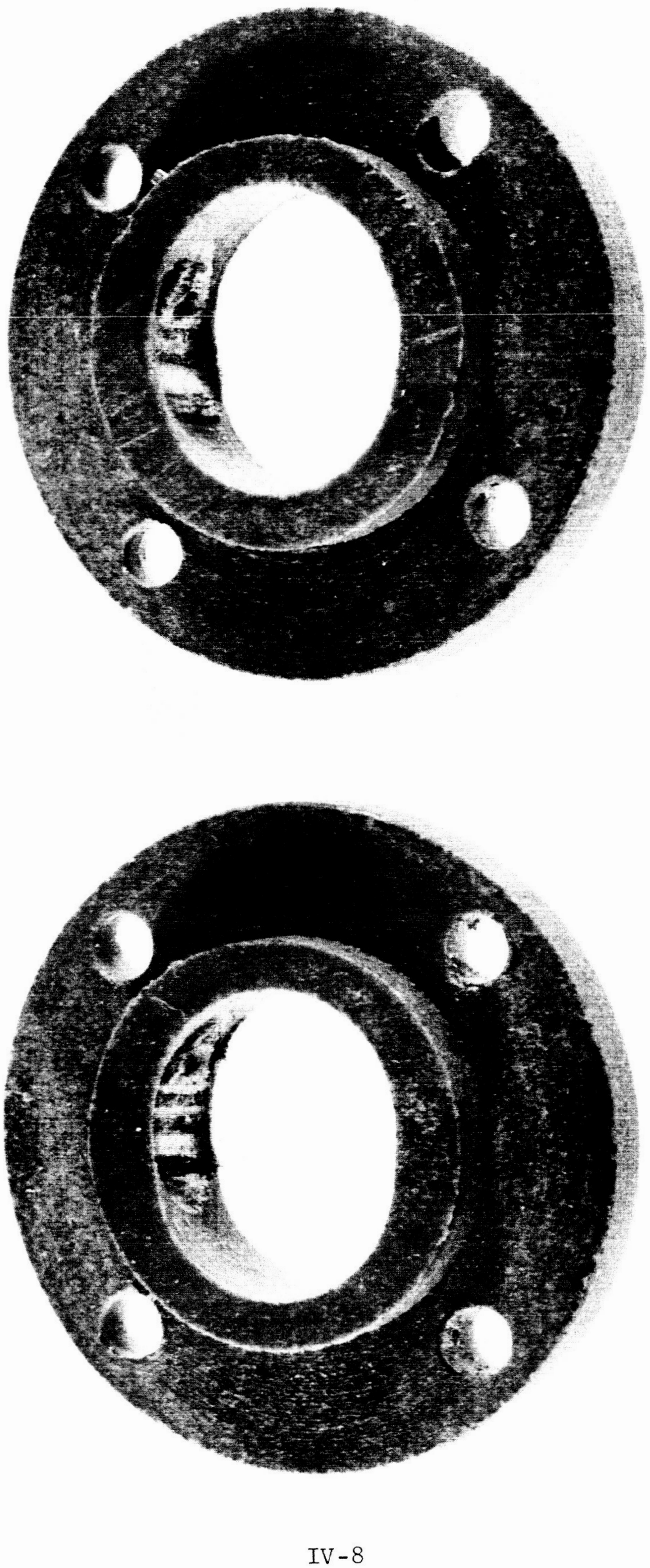


Figure IV-3. Polyimide-Bronze (70-30%) Inserts after 10 hours at 4 x 10^6 DN

8. Candidate No.20 (Ag-Polyimide WSe_2)

The first test on the Ag-polyimide-WSe_2 successfully completed 10 hours, while the second test was terminated after 9 hours 18 minutes by a vibration abort. As in the case of the polyimide-bronze (80-20) test, the abort was caused by a sudden increase in vibration. The lubricant inserts from both tests (shown in figure IV-4) showed only moderate wear, slightly heavier than that of the polyimide-bronze (60-40). The balls and plates were well coated with silver and in excellent condition.

9. Related Lubricant Tests

A series of tests of 55-mm bearings with Salox M cages in liquid hydrogen were conducted under an independent research program. In this series of tests, the 55-mm bearings were run at speeds up to 40,000 rpm (2.2×10^6 DN). In the first tests of the series the Salox cages showed unexpectedly high wear after a very brief operation at 40,000 rpm, and the Salox material in each case showed slight indication of overheating. On subsequent tests the coolant flow rate was increased and the bearings operated successfully at 40,000 rpm for 10 hours. This resulted in a decision to repeat the baseline tests on Salox M in the ball-plate tester.

In the Phase II portion of this contract several steps were taken to reduce hydrogen consumption during the ball-plate testing. One of the changes was the relocation of a Rosemount temperature probe used to measure the rig discharge temperatures. This probe is monitored during each test, and the liquid hydrogen flow rate is set to maintain liquid flow out of the rig. The relocation of the probe resulted in a significant (approximately 15%) reduction in LH_2 flow rate. The results of the 55-mm bearing test raised a question as to whether or not the LH_2 flow rate to the ball-plate tester had been reduced enough to affect the results of the tests. Therefore, as part of the same independent research program, a set of Salox M inserts was procured and installed in the ball-plate test apparatus and a 10-hour test was conducted at the equivalent of 4×10^6 DN. The Salox M inserts showed a marked increase in wear over the Phase I results and were considerably worse than the polyimide-bronze tests in Phase II. There was slight discoloration of the balls and races, whereas in Phase I the balls had an almost new appearance after the test.

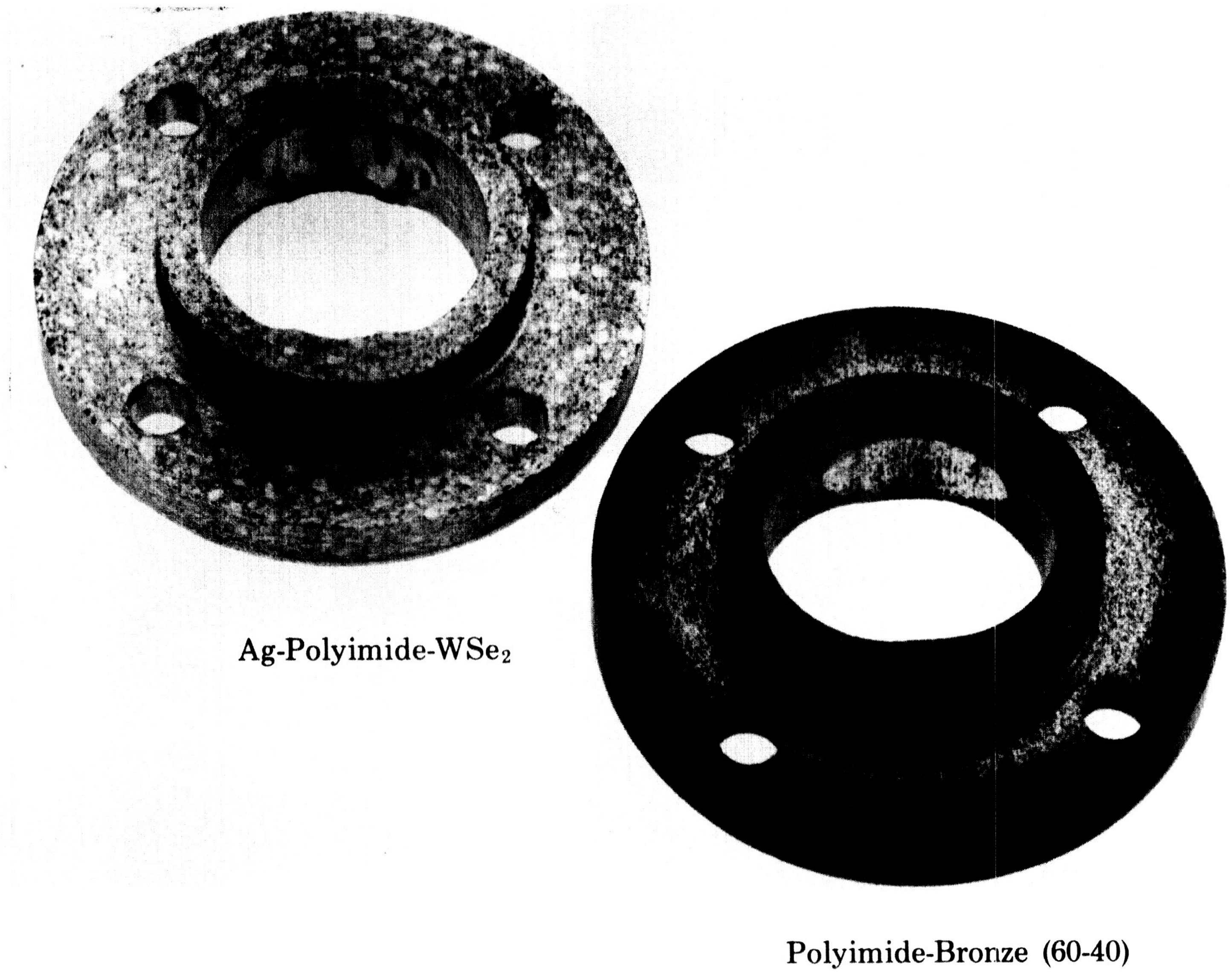


Figure IV-4. Retainer Inserts after 10 hours in Ball Plate Tester at Equivalent DN of 4×10^6 mm-rpm

FD 14329

To confirm the results of this test and to ensure that the increased wear was a function of the liquid hydrogen flow and not an undetectable change in the rig or its operation, a second test was conducted on the Salox M with the higher flow rates used in Phase I In this test the insert wear was reduced by approximately 50% over the first test, but was still slightly greater than in Phase I. Figures IV-5 through IV-7 show the Salox M inserts from the test in Phase I and the two tests in Phase II, respectively. The reason for the increase in wear between Phase I and this last test has not as yet been explained, but the material still exhibited excellent wear characteristics and it was decided to use it as a baseline for the evaluation of candidates in Phase II.

FE 42225

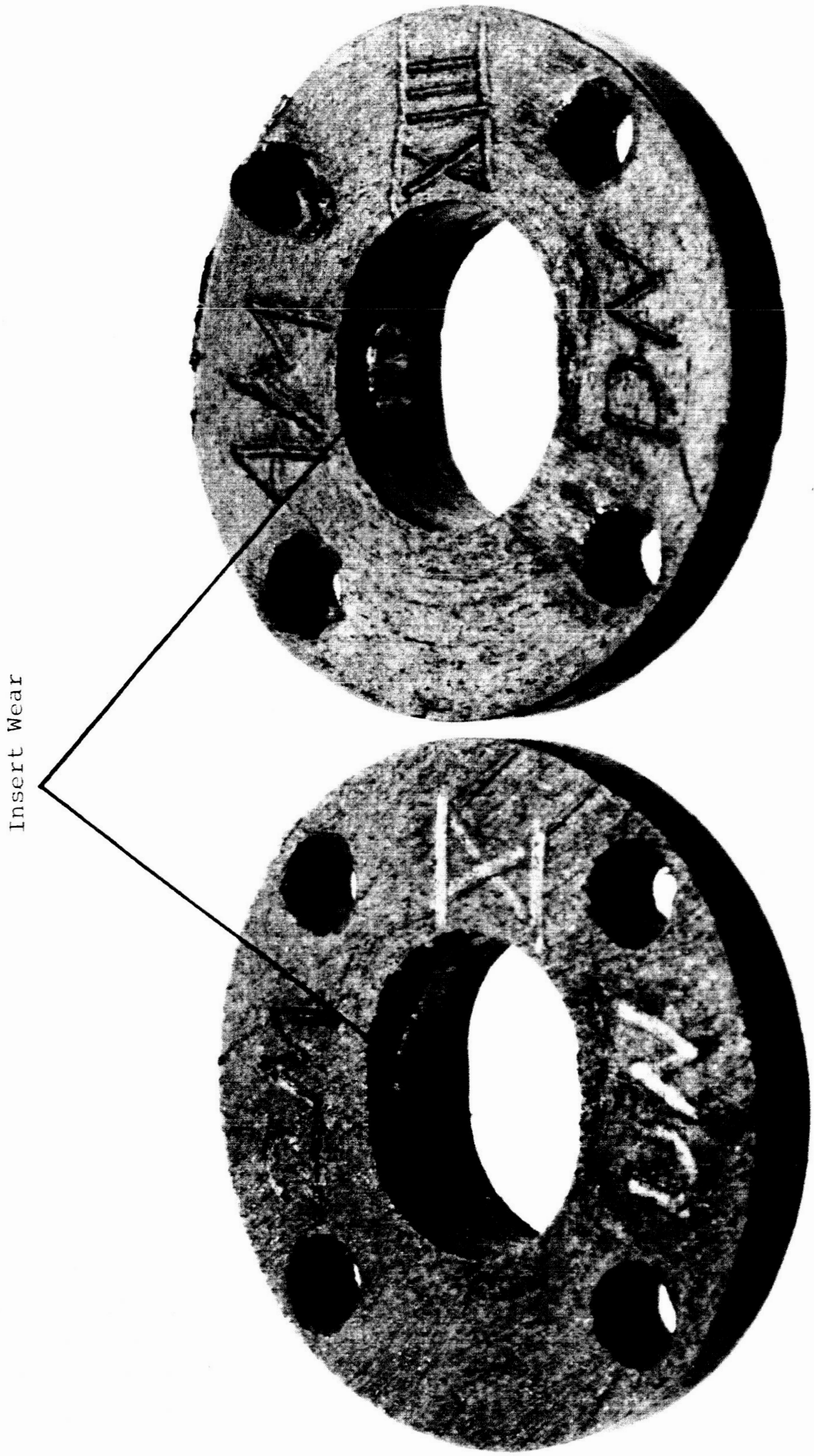


Figure IV-5. Salox M Inserts from Phase I Test

FE 48041

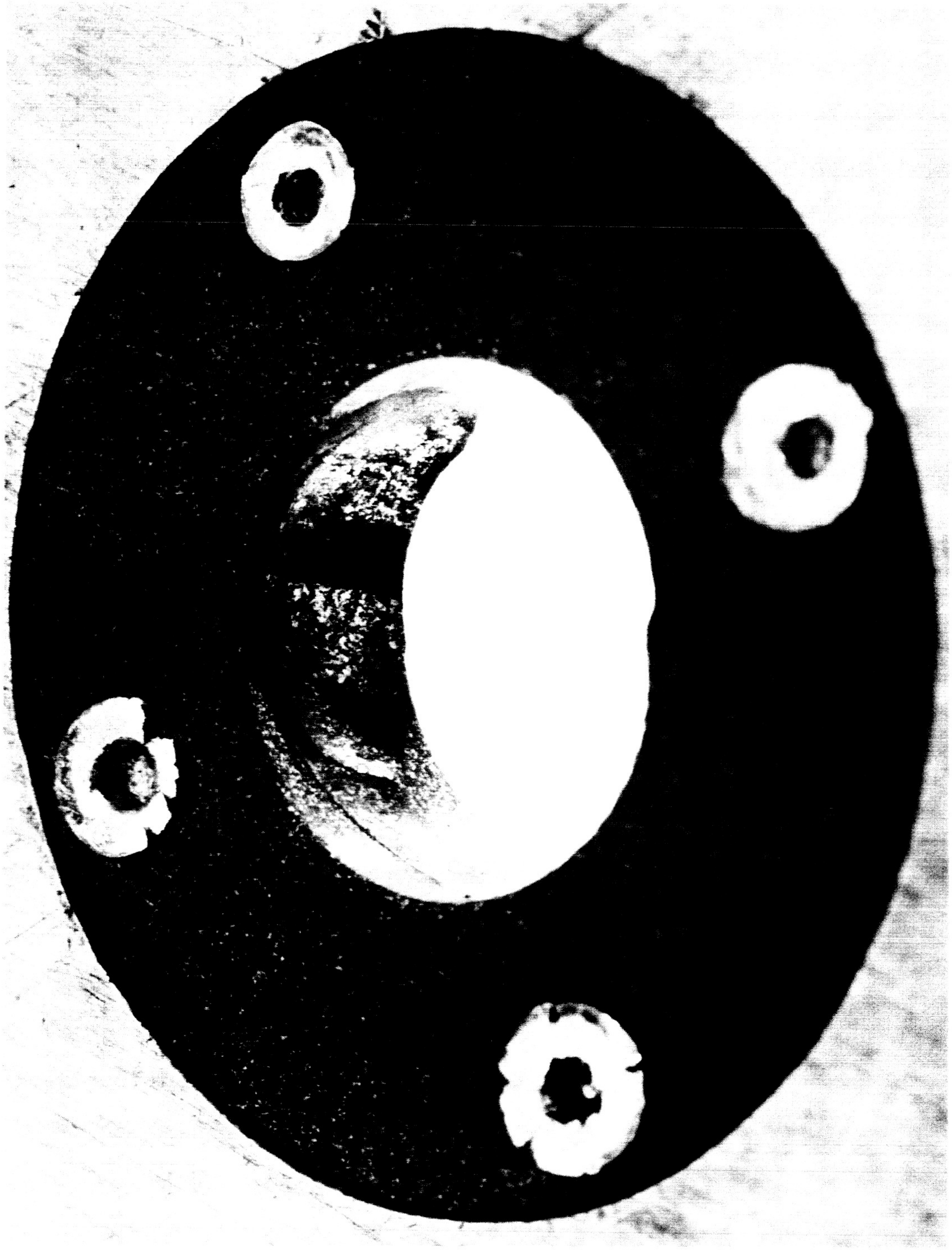


Figure IV-6. Salox M Insert from Phase II Test at Reduced Flow Rate

FD 8193A

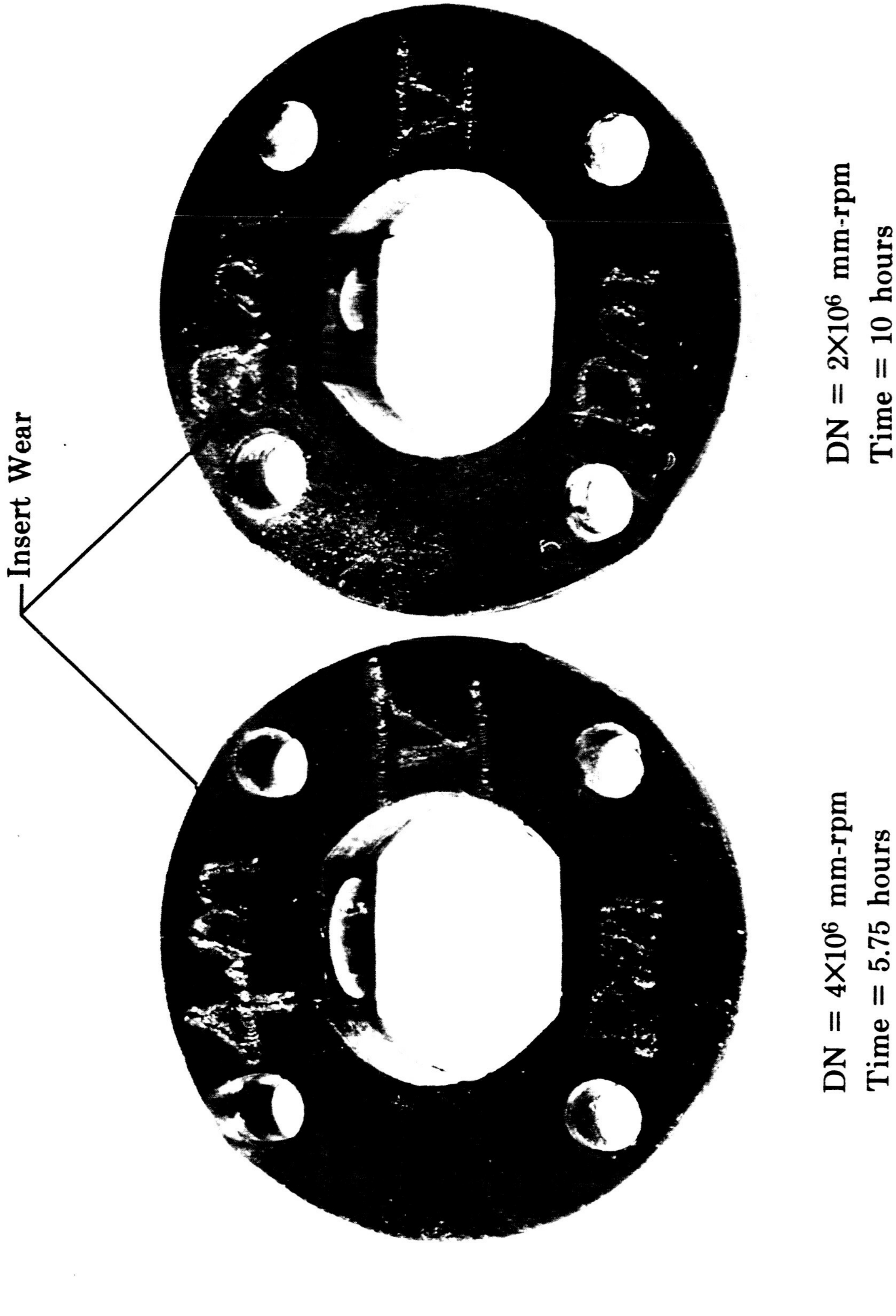


Figure IV-7. Typical Insert Wear of Standard Material (A)

SECTION V
SELECTION OF BEARING LUBRICANTS

The results of the lubricant material evaluation in this program showed that for rolling contact bearings operating in liquid hydrogen at DN values up to 4×10^6 mm-rpm under Hertz stress levels up to 250,000 psi and spin/roll ratios up to 0.30, the most effective lubricant material combinations were as follows in order of decreasing level of overall performance:

Rank	Retainer Material
1	Bronze-filled fluorocarbon composite (20-80 by volume) Salox M
2	Silicon-filled fluorocarbon composite (40-60 by weight) with MLF-5 coating on plates
3	Bronze-filled polyimide composite (40-60 by volume)
4	Ag-WSe_2 - filled polyimide composite (75-5-20 by volume)
5	Silicon-filled fluorocarbon composite (40-60 by weight)
6	Ag-MoS_2 matrix (80-20 by volume)

From this list four materials were selected for further evaluation in actual bearing tests. The lubricant materials ranked first, second, and fifth demonstrated outstanding lubrication and resistance to wear in Phase I tests. However, these composites are filled Teflons and cannot be considered satisfactory for applications where radiation environments exceed 2×10^6 ergs per gram (C). This radiation level is generally accepted as the critical dosage above which the organic compound, Teflon, loses its strength. These three filled-Teflon composites show promise of providing satisfactory cage materials for bearings where the radiation environment is low or moderate. One filled-Teflon composite, Salox M (Rank 1), was selected from these three for further evaluation.

In Phase II ball-plate testing, the bronze-filled polyimide composite (40-60 by volume) displayed adequate lubricating qualities as well as excellent wear resistance, second only to the bronze-filled Teflon. The use of polyimide in place of Teflon as the base material provides a substantially higher resistance to radiation damage (approximately 10^{11} ergs/gram (C) as compared to 10^6 ergs/gram (C) for Teflon). Screening tests on the three bronze-filled polyimide composites (80-20, 70-30, and 60-40) showed a marked increase in wear resistance and a decrease in tensile strength with an increase in bronze content. A further increase in the percentage of bronze was not attempted because (1) it would further reduce the tensile strength of the composite, and (2) the 60-40 composite displayed a very low wear rate. Therefore, the polyimide-bronze (60-40) composite was selected as the second candidate for bearing testing.

The wear rate of the fourth-ranked lubricant composite, silver-tungsten diselenide-polyimide (75-5-20 by volume) was relatively high but, its lubricating qualities were excellent. The high silver content is also expected to improve the resistance to radiation damage over other composites having high polyimide content. This material was the third selected for further evaluation.

The sixth-ranked lubricant material (silver-molydisulfide) is an all-metal composite that will provide the maximum resistance to radiation damage (up to 10^{17} n per sq cm). Both the sintered matrix composite tested in Phase I and the impregnated feltmetal tested in Phase II provided excellent lubrication for the 10-hour tests, but they displayed low wear resistance. The Ag-MoS_2 lubricant composite was selected for bearing tests to further evaluate a highly radiation resistant lubricant. The sintered Ag-MoS_2 material was selected over the impregnated feltmetal because it was less difficult to fabricate into a bearing cage.

In summary, the four materials selected from the candidates screened during Phase I and Phase II in the ball-plate test apparatus for testing in the 80-mm bearings are:

1. Bronze-filled PTFE (20-80 by volume) Salox-M
2. Bronze-filled polyimide (40-60 by volume)
3. Ag-Wse_2-polyimide (75-5-20 by volume)
4. Ag-MoS_2 matrix (80-20 by volume)

SECTION VI
BEARING AND TEST APPARATUS DESIGN

A. BEARING DESIGN

The four most promising lubricant materials from the ball-plate testing were incorporated in 80-mm bearings and tested in a liquid hydrogen environment for 10 hours or failure, whichever occurred first. The 10-hour limit was selected as a practical design life for most rocket engine applications and provided a firm basis for planning the test program. The design of these bearings consisted of two individual, but inter-related efforts: (1) ball-raceway geometry optimization, and (2) structural optimization of retainer configuration. Previous experience in the design of cryogenic bearings for moderately high DN levels in the RL10 and high pressure hydrogen turbopumps was applied to the 80-mm bearing design. However, new problems arise in the high speed operation of large diameter bearings, the most critical of which is inner race centrifugal growth. This growth can adversely affect the internal geometry of the bearing. Treating the inner race as a thin ring, the growth can be expressed as a constant x $(DN)^2$ x diameter, which is derived from the equation;

$$\Delta R = \frac{SR}{E} \tag{1}$$

where:

S = Rotational stress, psi

R = Radius, in.

E = Modulus of elasticity

S is calculated from the equation:

$$S = \frac{\gamma}{g} (R\omega)^2 \tag{2}$$

where:

γ = Density, lb/in^3

g = Gravitational constant

ω = Speed, rad/sec

Substituting in equation (1)

$$\Delta R = \frac{\gamma/g \ (R\omega)^2 \ R}{E} \tag{3}$$

or

$$\Delta R = \frac{\gamma/g \ (D\omega)^2 \ D}{8E} = K(DN)^2 D \tag{4}$$

where:

D = Diameter, in.

DN = Bore diameter in mm x speed in rpm

K = Constant (includes conversion factor for inches to mm and rad/sec to rpm)

From this it can be seen that the centrifugal growth problem becomes greater at the larger bore sizes for constant DN levels.

Figure VI-1 describes the static contact angle vs internal clearance and the inner race growth vs speed for an 80-mm bearing. The use of these contact angle curves enables an estimate to be made of the required static contact angle to provide a reasonable contact angle at high speed.

From figure VI-1, a static contact angle corrected for inner race growth can be determined. This represents the angle that results from reducing the internal clearance by the inner race growth at 50,000 rpm. From past experience it has been determined that the corrected static contact angle should be maintained as low as practical without drastically reducing internal clearance. From RL10 experience an angle of 20 degrees was assumed and the bearing geometry optimized through a series of calculations in which each factor is varied independently and the results compared to obtain a balance of Hertzian stress, spin/roll ratio, and heat generation for maximum bearing life. These calculations are made using IBM 7090 and 1620 computers. The ball diameter (0.5312 in.), number of balls (20) and ball pitch diameter, E (4.030 in.) were selected to provide ample space for an adequate cage structure.

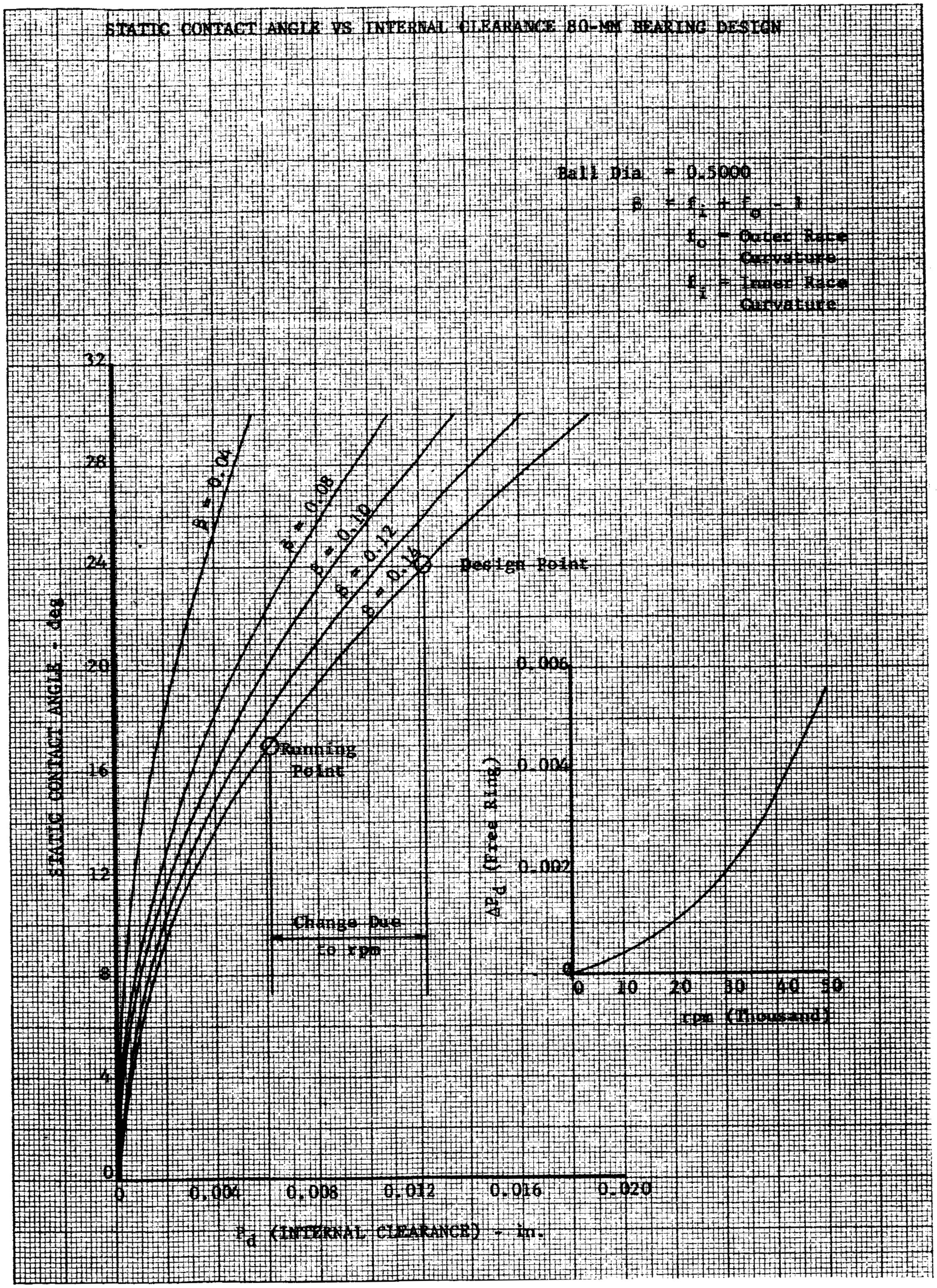


Figure VI-1. Static Contact Angle vs Internal Clearance 80-mm Bearing Design

DF 43846

Due to the fact that pure rolling occurs at the outer race, increases in outer race radius of curvature (f_o) beyond 52% have little or no effect on heat generation or spin/roll ratio. The only effect is on Hertz stress. As the curvature is increased, Hertz stress increases. Because the outer race curvature was maintained at the standard 52 percent.

Increasing the inner race radius of curvature reduced the difference between inner and outer race contact angle, spin power (heat generation), and spin/roll ratio. However, Hertz stress increases, which reduces bearing B-10 life. The design point was, therefore, selected to maintain inner race Hertz stress equal to or less than the outer race stress. A review of the optimization indicated the static contact angle could be reduced to decrease spin/roll ratio and heat generation without seriously affecting stress levels. The angle was then changed from 20 to 16.5 degrees, and suffucient data were processed to assure that the optimization results did not change significantly at the low contact angle. Inner race curvature was set at 62 percent. With the 16.5-degree contact angle. The static contact angle was determined to be 24 degrees from figure VI-1.

Figures VI-2 through VI-5 represent the characteristics of the selected design. Figure VI-2 shows the minimum preload for the geometry selected to be 65 lb/ball at 50,000 rpm, which results in a nearly equal inner and outer race hertz stress as shown in figure VI-3.

Figure VI-4 shows that the spin/roll ratio is nearly constant above approximately 30,000 rpm. This can be explained by the fact that the difference in contact angle between inner and outer race is nearly constant above this speed, as shown in figure VI-5.

The bearing geometry was selected as follows:

Bore dia	- 80 mm, OD - 125 mm
Outer race curvature	- 52% of ball diameter
Inner race curvature	- 62% of ball diameter
Contact angle	- 24° (static)
Ball diameter	- 0.500 inch
No. of balls	- 19
Ball pitch diameter	- 4.030 in. (approx. 102 mm)
Series	- extra light

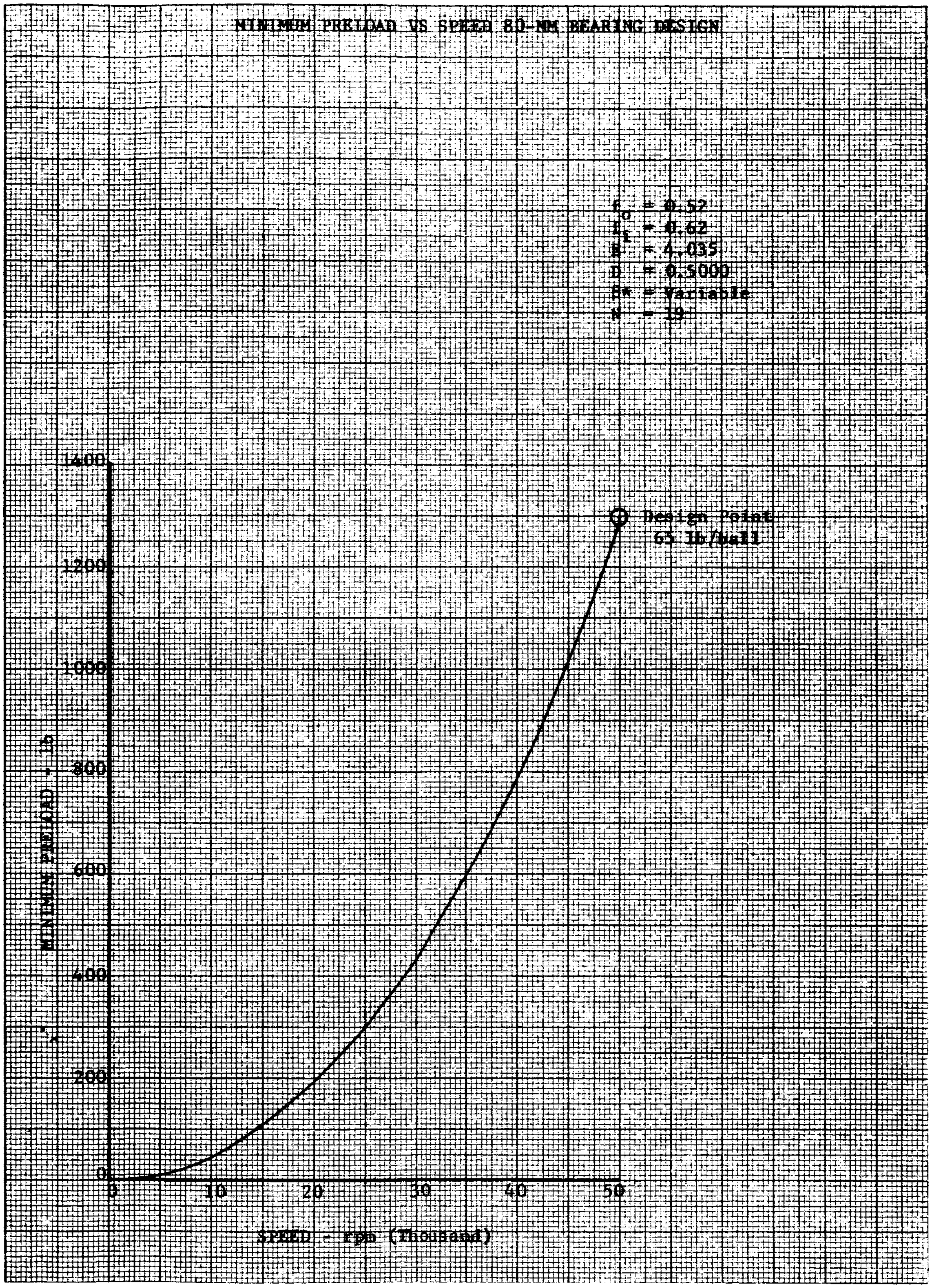


Figure VI-2. Minimum Preload vs Speed 80-mm Bearing Design

DF 43847

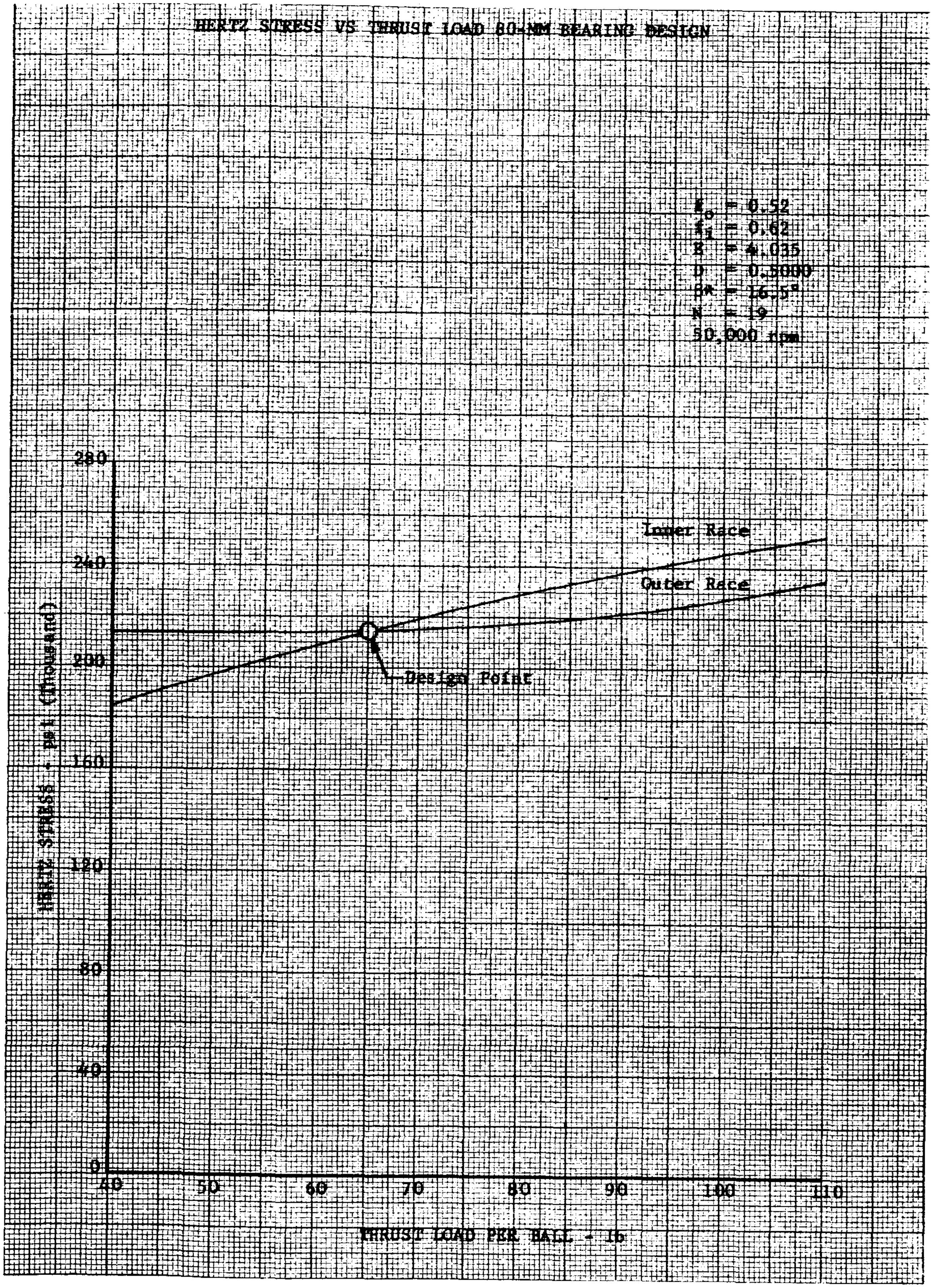


Figure VI-3. Hertz Stress vs Thrust Load 80-mm Bearing Design DF 43848

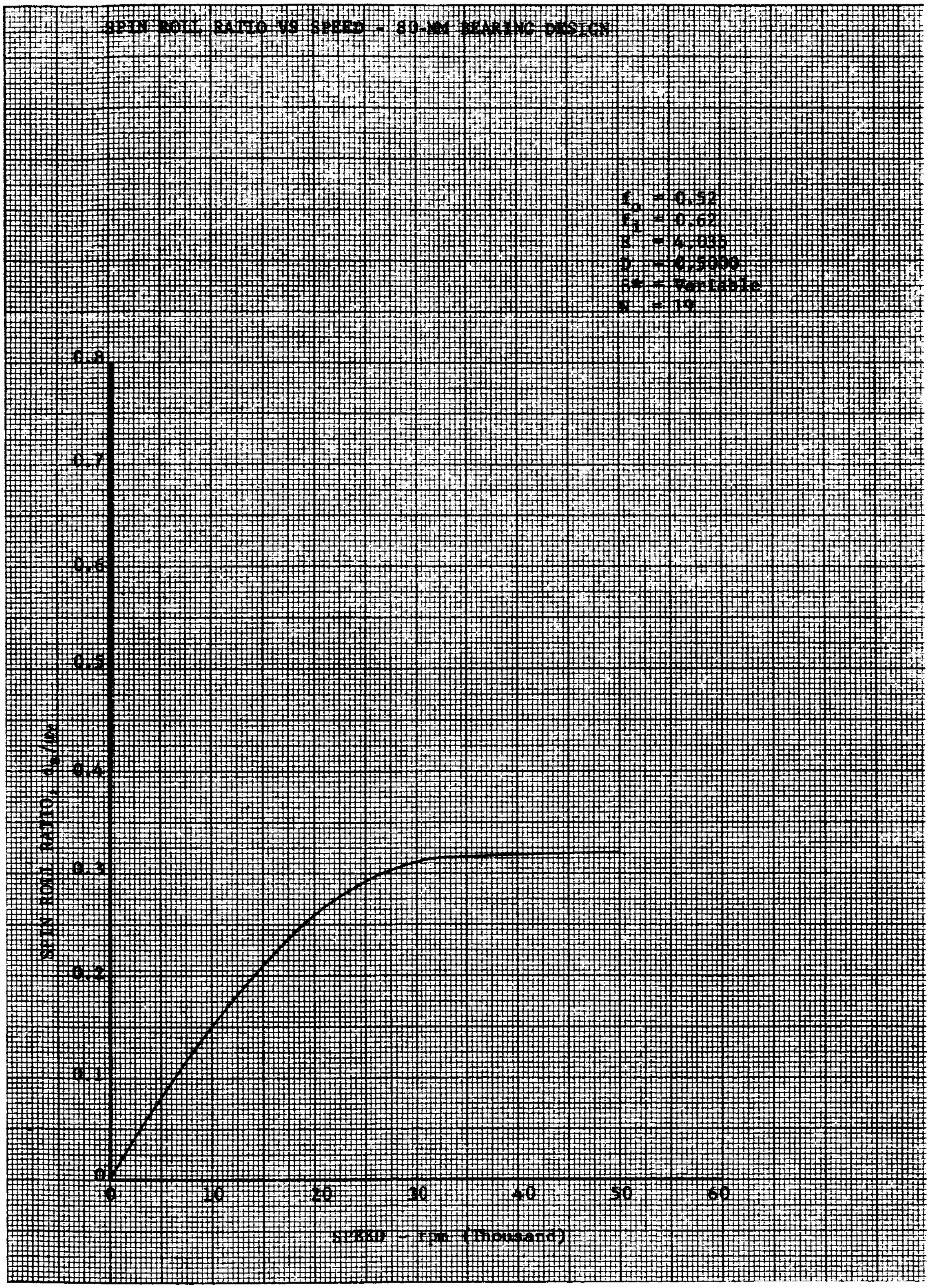


Figure VI-4. Spin Roll Ratio vs Speed
80-mm Bearing Design

DF 43849

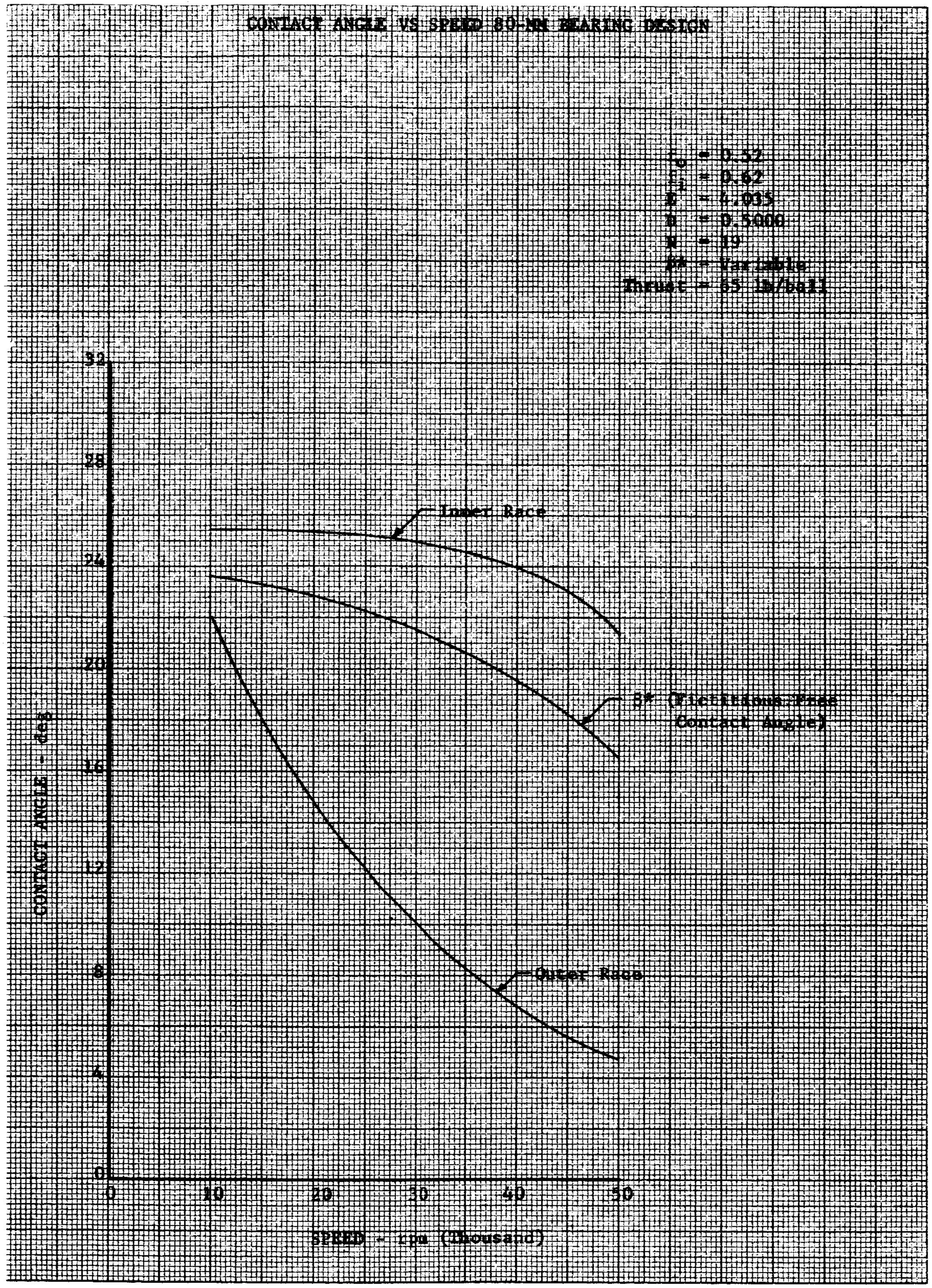


Figure VI-5. Contact Angle vs Speed
80-mm Bearing Design

DF 43850

With the bore diameter and speed fixed by the DN requirement, an extra light bearing series was selected to provide ample space for a sturdy cage design. The cage design, which is inner-land riding, incorporates a ring of lubricant material riveted into a one-piece high-strength steel U-shaped shell. Figure VI-6 shows a disassembled 80-mm bearing.

B. BEARING TEST APPARATUS DESIGN

The bearing test apparatus, shown in figure VI-7, was designed to test 80-mm bearings completely submerged in liquid hydrogen at 50,000 rpm. The test apparatus is a modification of an existing P&WA rig design that was used successfully for several years in testing smaller cryogenic bearings. The test apparatus incorporates a small radial-inflow turbine drive using high pressure nitrogen or hydrogen gas as the working fluid. The bearings are loaded axially by a piston pressurized with gaseous hydrogen. Figure VI-8 shows a schematic of the test stand installation. Liquid hydrogen is supplied to the bearings through fittings at the center of the housing and discharged outboard of both bearings. Speed is controlled by a remotely operated valve in the turbine GN_2 or GH_2 supply line, and piston loading is controlled by maintaining the required differential between housing pressure and piston supply pressure. Speed is measured with a magnetic pickup and six pins located on the bearing shaft. Accelerometers are used to measure radial and axial vibration, and bearing outer race temperatures are recorded. A helium seal dam prevents mixing of liquid hydrogen and the turbine drive gas.

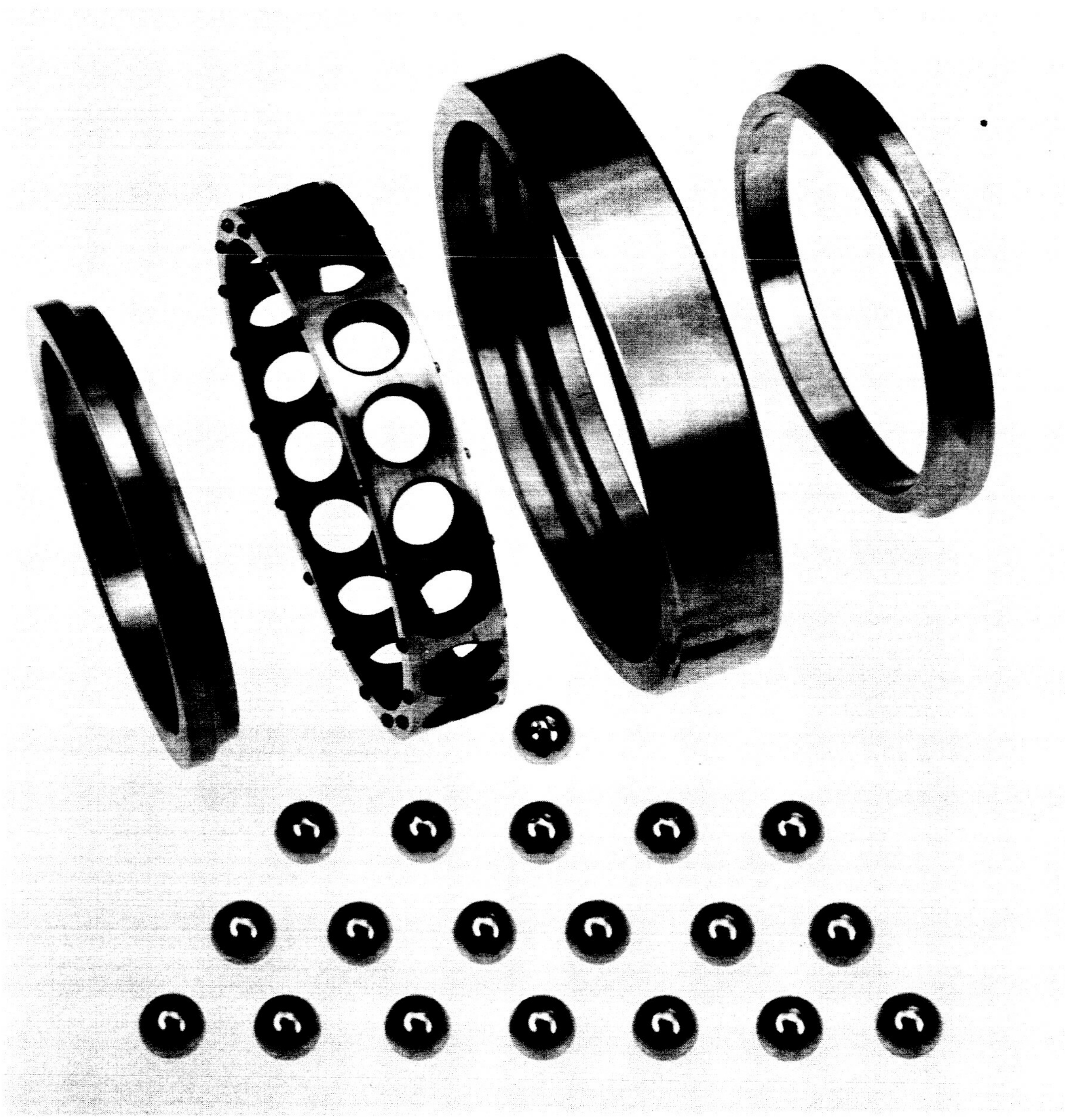


Figure VI-6. 80-mm Bearing with Salox M Cage FE 49721

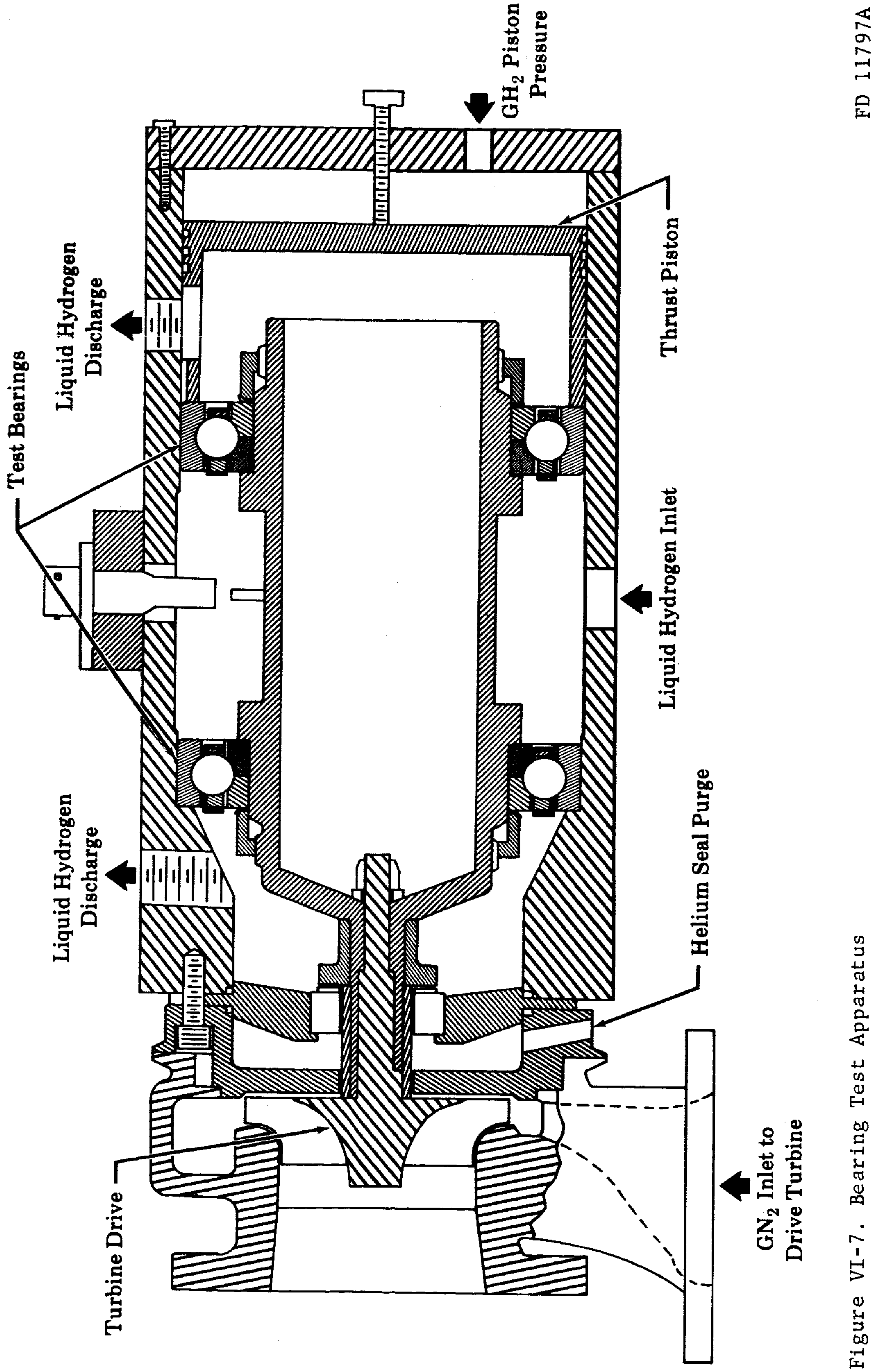


Figure VI-7. Bearing Test Apparatus

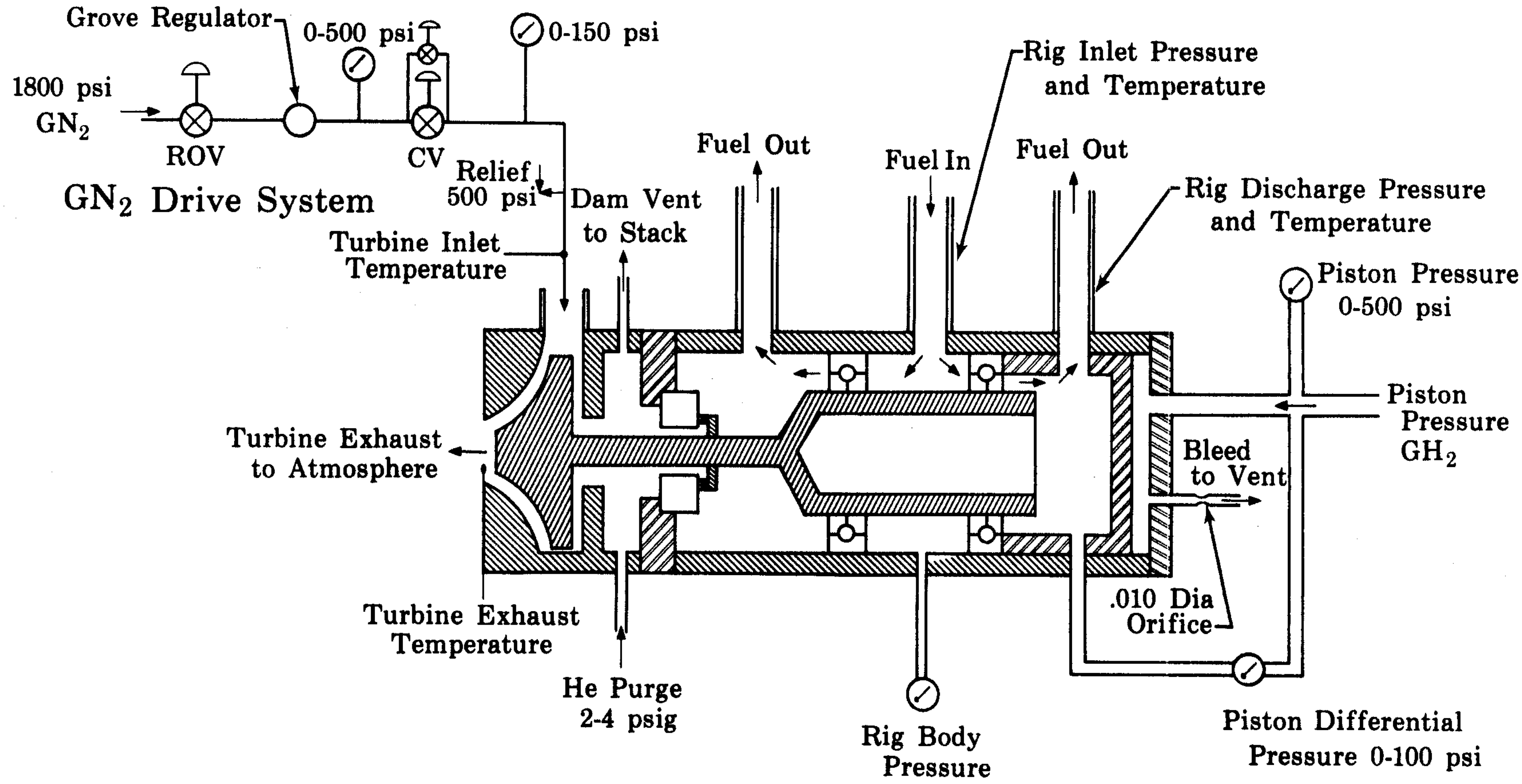


Figure VI-8. Test Stand Schematic for 80-mm Bearing

FD 11798

SECTION VII
BEARING TESTS

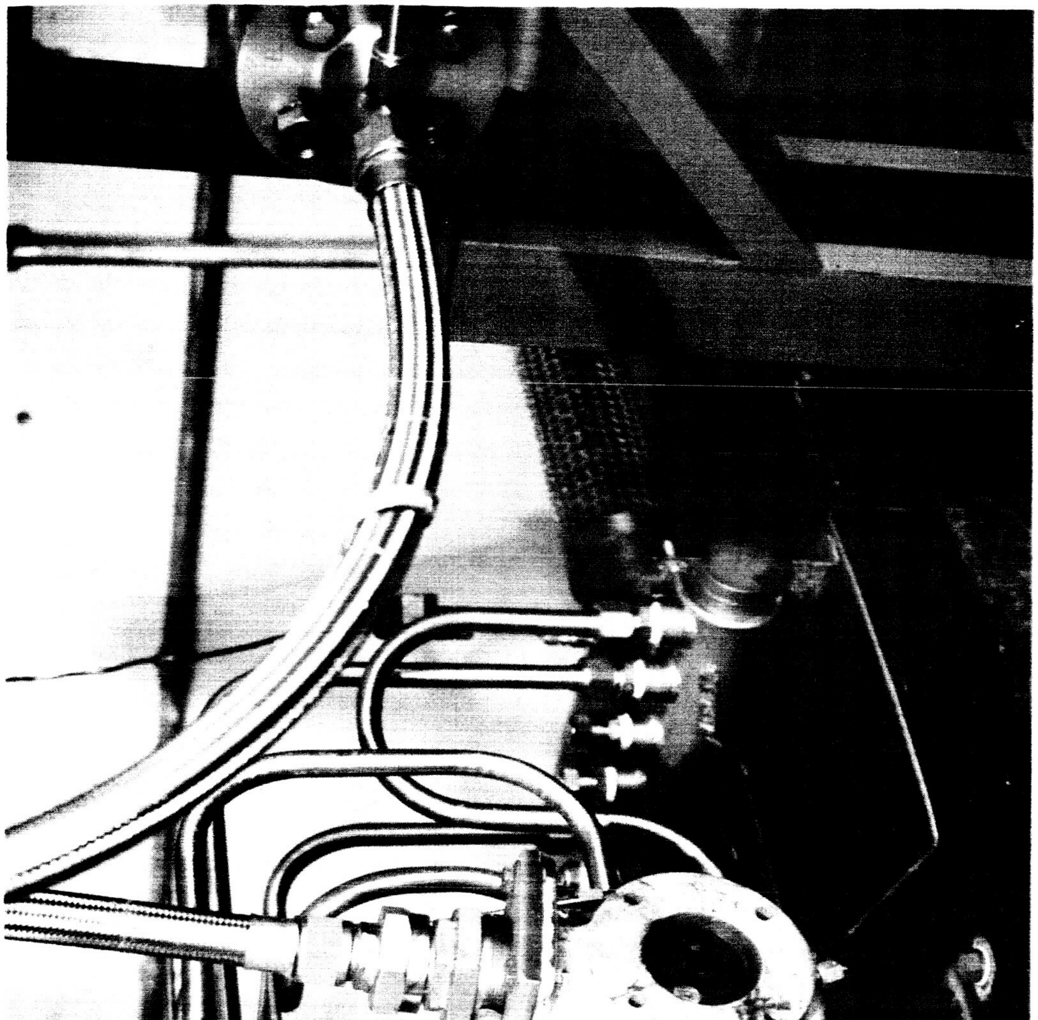
A. TEST RIG SET-UP

The bearing tests were conducted at FRDC's rocket component test area "B" on test stand B-13. The test rig (figures VII-1 and VII-2) was supported on a steel base plate with flexible mounts. Liquid hydrogen was supplied at 100 to 130 gpm to the test rig through vacuum-jacketed lines from a 7000-gallon roadable dewar. The LH_2, after flowing through the test bearings, was piped to vent stacks and burned. The gaseous nitrogen and hydrogen for the turbine drive were supplied from 1800 psia evaporators with the flowrate controlled through remotely operated valves. The bearing test rig was instrumented to indicate shaft speed, inlet and discharge pressures, LH_2 flowrate, temperatures, and vibration levels. Bearing failure was indicated by radial and axial vibration measured by three accelerometers mounted on the test rig, by bearing outer race temperatures, and by an increase in torque (as reflected by an increase in drive gas pressure required to maintain a constant speed).

B. BEARING TEST RESULTS

1. General

Three of the eleven sets of 80-mm bearings completed short duration running at 50,000 rpm (DN value of 4.0 million). The bearing cage materials used in these tests were Ag-polyimide-WSe_2, Ag-MoS_2, and Salox M, with the minimum wear rate being demonstrated by the latter material. A summary of the bearing tests outlining candidate materials, running time and post-test conditions is given by table VII-1. Because of the high wear shown on the initial tests of the silver-molydisulfide and the silver-polyimide - tungsten-diselenide retainer rings, the contractor and the NASA contracting officer mutually agreed to stop further testing on these materials and to conduct additional tests of bearings with Salox M and polyimide-bronze retaining rings. This decision was influenced by the high potential of Salox M and polyimide-bronze as indicated by the ball-plate tests, as well as by the fact that test rig problems early in the program may have affected the results of the Salox M tests. The bearing tests, according to the lubricant contained in the bearing retainers, are discussed in the following paragraphs.



FE 50292

Figure VII-1. Front View of 80-mm Bearing Test Rig Installed in Test Stand

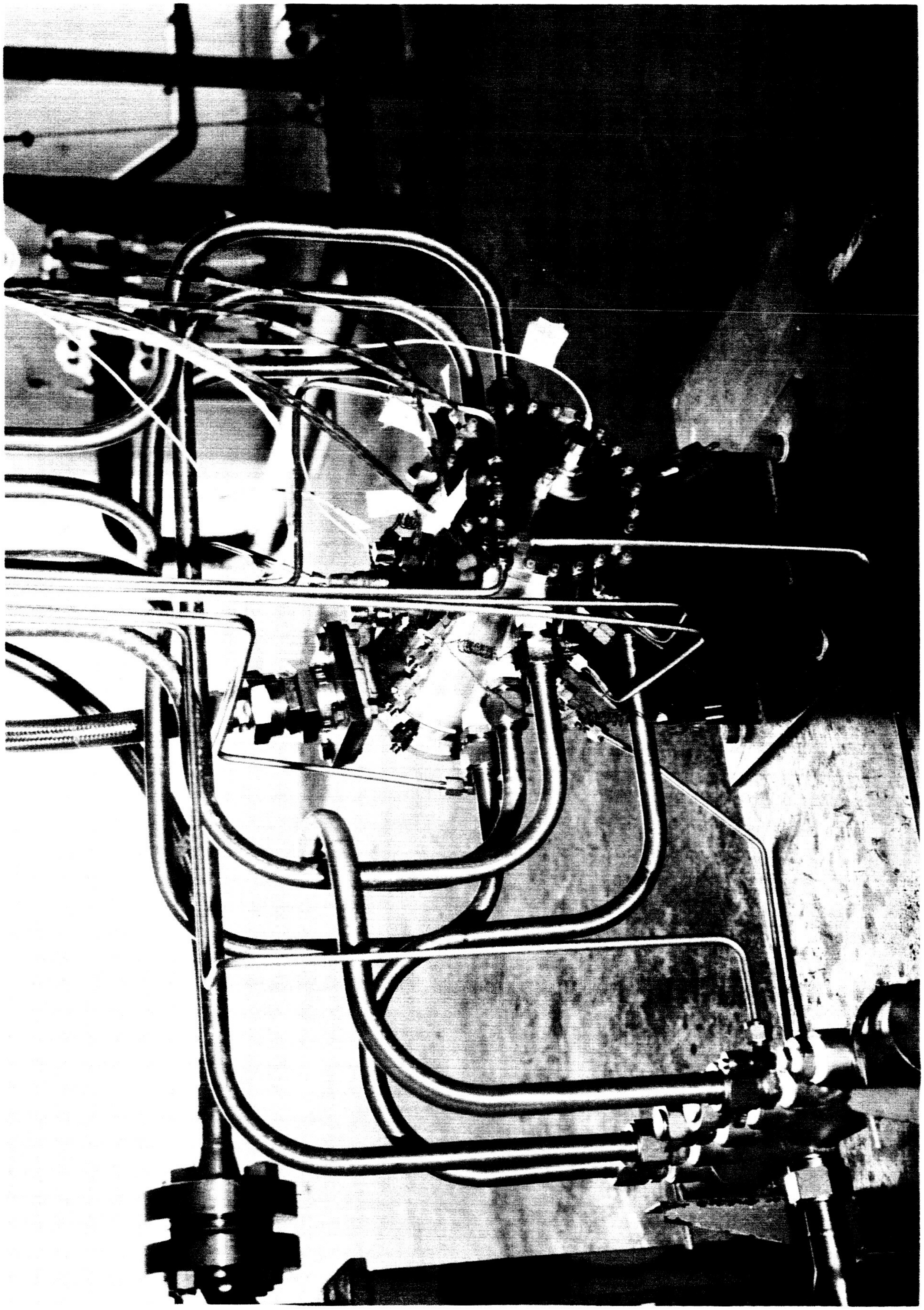


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Figure VII-2. Rear View of 80-mm Bearing Test Rig Installed in Test Stand

Table VII-1. Bearing Test Results, Contract NAS8-11537

Test No.	Bearing Lubricant	Duration	Results
1	Salox M (Bronze Filled Teflon)	25 min at 10-20,000 rpm 13 min at 30,000 5 min at 40,000	Bearing inner race retaining nut failed at 40,000 rpm. Bearings and cages damaged.
2	Salox M	13 min at 10-20,000 8 min at 30,000	Rear bearing failed at 30,000 rpm. Cages rubbed on outer race lands.
3	Salox M	33 min at 10-20,000	Reduced cage ID radial clearance and balanced cages. Front bearing failed. Indications of uneven LH_2 flow distribution.
4	Polymide-bronze (60-40% by vol)	27 min at 10-20,000	Added pressure balance holes to front of bearing shaft. Test terminated by test rig failure, (speed pickup tooth broke off). Bearings reused in test No. 11.
5	Polimide-bronze (60-40% by vol)	20 min at 10-20,000	Rear bearing failed. Races pitted, cages worn heavily.
6	Polimide-bronze (60-40% by vol)	83 min at 10-20,000 1 min at 40,000	Front bearing failed at 40,000 rpm. Cage wear heavy on ID and in ball pockets.
7	Ag-Polimide-WSe_2 (70-20-5% by vol)	2 min at 10-20,000 5 min at 35,000 3 min at 50,000	Bearings failed after 3 min at (4×10^6 DN). Thrust load 900 lb cages badly worn and expanded.
8	Ag-MoS_2 (80-20% by weight)	2 min at 10-20,000 4 min at 50,000 overspeed to 62,000	Thrust load varied widely due to chattering carbon face seal allowing overspeed to 62,000 rpm during initial acceleration. Bearings completed 4 min at 50,000 rpm (4×10^6 DN). Cage wear was moderate to heavy on ID and in ball pockets.
9	Salox M	2 min at 10-20,000 6 min at 40,000	Bearing outer race temperature increased to -200°F at 40,000 rpm. Cages rubbed on outer race. Ball pocket wear was heavy.
10	Salox M	1 min at 10-20,000 5 min at 40-45,000 3 min at 50,000	Completed 3 min at 50,000 rpm (4×10^6 DN)-front bearing damaged - rear bearing in good condition.
11	Polimide-bronze (60-40% by vol)	5 min at 5-10,000 1 min at 20-30,000	High bearing torque limited speed to 30,000 rpm. Cage ID rub on inner race. These bearingswere previously run in test No. 4.

2. Salox M (Bronze-Filled Teflon)

Five tests were conducted with bearings that used Salox M as the lubricant. The first test, of 45 minutes duration, was stopped when a bearing inner race retaining nut failed at 40,000 rpm. The failure was attributed to excessive stress concentration in the wrench slots. Both bearings were badly damaged and the Salox M material was split radially through the ball pockets into three approximately equal sections. The splitting was attributed to the thermal shrinkage of the Salox M between rivets combined with the centrifugal force due to rotation. To relieve the overstressing, the Salox rings in the remaining cages were intentionally split in the same pattern. In addition, tapered grooves were added on the ID of the Salox rings to increase cooling flow to the inner race and to provide a slight hydrodynamic lift to reduce the bearing pressure of the cage on the inner race. The locking nuts were redesigned for the second test, and the LH_2 inlet and discharge flow lines to the rig were enlarged to increase flow rate.

The second test failed after 8 minutes duration at 30,000 rpm and 1200-lb axial load on the bearings. The bearings showed indications of heavy cage rub on the outer race. The case-to-inner-race clearance was originally established to provide a 0.010-0.018-inch clearance at operating temperature, considering shrinkage of the Salox M cage. With the Salox M material split into three sections, the "hoop" in the cage was destroyed; therefore, the cage shrinkage was greatly reduced with a resultant increase in ID clearance at operating temperature. Slight wear of the cage ID then allowed the cage to rub on the OD. All succeeding Salox cages were modified to remedy this situtation by removing the Salox M material from the stainless steel shell and installing new blanks. These blanks were also split into three sections; however the ID was reduced to provide the required inner race clearance. The cages were balanced to within 0.001 oz-in. and the tapered grooves were incorporated on the ID, as in the second set of cages.

In the third test, the bearings were accelerated to 20,000 rpm and test conditions were allowed to stabilize. During the subsequent acceleration to full speed, vibration increased rapidly at approximately 30,000 rpm. The test rig was disassembled to inspect the bearings. The front bearing cage (turbine end) showed heavy ball pocket wear in four locations as well as indications of overheating. The rear bearing cage was in good condition with light wear in the ball pockets. The results indicate that the total LH_2 flow to the rig was not being equally split between the two bearings. Leakage of helium past the face seal on the turbine end could have restricted the flow out of that section of the rig and reduced flow through the front bearing. To balance the discharge pressure on both bearings, a set of holes was drilled in the web of the bearing shaft that tied the two compartments together.

In the fourth test, the bearings could not be accelerated beyond 40,000 rpm because the torque increased rapidly above 30,000 rpm. Inspection of the bearings revealed that the balls and races were heavily coated with bronze; the retainers showed high lubricant wear and cold flowing of the lubricant in the ball pockets. The turbine shaft had rubbed on the carbon seal housing. The rub marks covered approximately a 90-degree section on the shaft, indicating a whipping action or a rotating radial load. The turbine shaft was nickel-plated before the last Salox M test to ensure a tight fit at all conditions between the turbine shaft and the bearing shaft.

The fifth Salox M test was of 9 minutes duration, of which 3 minutes were at 50,000 rpm (4×10^6 DN). The test was terminated because of high vibration and an increase in the front bearing outer race temperature. The webs between ball pockets of the stainless steel retainers in both bearings had bowed slightly from the centrifugal loads. The balls and races from the front bearing were coated with bronze and the cage showed heavy wear. The rear bearing, except for the cage bowing was in very good condition. The Salox M bearings from the last test are shown in figure VII-3.

Figure VII-3. Salox M Bearings After 3 Minutes at 50,000 rpm FD 14327

3. Polyimide-Bronze (60-40)

Four tests were made with the polyimide-bronze lubricated bearings. These bearing cages were all individually balanced to within 0.001 oz-in. In the first two tests, the turbine was driven with GN_2 and the maximum shaft speed attained was 20,000 rpm. In the third and fourth tests, GH_2 was used to drive the turbine; a maximum speed of 40,000 rpm was achieved before the bearings failed. The bearing preload, which was 1200 lb in the first three tests, was reduced to 900 lb in the fourth test in an effort to achieve 50,000 rpm.

The first test was terminated after 27 minutes at speeds up to 20,000 rpm because of the loss of a speed pickup lug. The bearings were in good condition and were reinstalled, with the races reversed, for the fourth test. The high torque that prevented the bearings from reaching 50,000 rpm was apparently caused by the cage rubbing on the inner race. In each test, the cage ID wear was heavy; in the bearings that reached 40,000 rpm, the rubbing was severe enough to cause heat-cracking on the surface of the inner race. In each instance, the polyimide-bronze lubricated well but wore heavily in the ball pockets because the cage was driven against the balls by the contact with the inner race.

4. Silver-Molydisulfied

One set of bearings was tested with Ag-MoS_2 cages. During acceleration to design speed, the piston and rig pressures varied radically due to chattering of the carbon face seal. The pressure fluctuations allowed low differential pressure on the loading piston and reduced preload on the bearing. The resulting low bearing preload during acceleration allowed the shaft speed to reach 62,000 rpm. The duration of the overspeed condition was approximately 1 second. The speed was then adjusted to 50,000 rpm and the bearings ran for an additional 4 minutes at a bearing thrust load of 900 lb. High vibration indicated bearing failure. Examination of these bearings (figure VII-4) revealed a heavy coating of silver on the balls and races. Both retainers rubbed against the bearing outer race lands and showed high wear in the ball pockets. The retainers exhibited diametral growth due to high centrifugal forces. The front bearing and its retainer were badly damaged. The rear bearing was in unusually good condition, considering the high speed to which it was subjected. Although the rear bearing retainer showed OD rubbing had occurred and the ball pockets were heavily worn, neither the retainer nor bearing was seriously damaged.

5. Silver-Polyimide - Tungsten Diselenide

One test was made with bearings using the Ag-WSe_2-polyimide lubrication.

These bearings were tested for a total of 8 minutes duration, of which 3 minutes were at 50,000 rpm. High vibration indicated failure of the bearings. The balls and races of both the bearings were damaged. Wear in the ball pockets of both retainers was high and the OD of both retainers showed indications of rubbing on the lands of the bearing outer races (figure VII-5).

Front

Rear

Figure VII-4. Silver-Molydisulfide Bearings After 4 Minutes at 50,000 rpm. FD 14328

Rear Front

Figure VII-5. Silver-Polyimide-Tungsten Diselenide Bearings After 3 Minutes at 50,000 rpm FD 14330

SECTION VIII
CONCLUSIONS

1. Four materials have been developed that possess the potential of providing adequate lubrication for bearings operating in a liquid hydrogen environment at DN levels to 4×10^6 mm-rpm.

   These four materials are:

   a. Salox M (bronze-filled Teflon)
   b. Polimide-bronze (60-40% by volume)
   c. Silver-polyimide - tungsten diselenide (70-20-5% by volume)
   d. Silver-molydisulfide (80-20% by weight).

2. The four materials, in the order listed above, have an increasing resistance to damage by nuclear radiation from 2×10^6 ergs per gram (C) for the Teflon-based Salox M to 10^{15} ergs per gram (C) for the silver-molydisulfide composite. The life expectancies of bearings containing these lubricants, based on lubricant wear rate, vary inversely with the resistance to radiation damage.

3. The successful use of these materials in an actual bearing application will depend on the development of lubricant retainers capable of withstanding the rotating stresses associated with high DN operation.

4. Results of the 80-mm bearing tests were clouded by problems encountered in the test rig and the bearing cage designs. The time and scope of this contract did not permit a step-by-step development of a cage design for each lubricant that would have provided a true evaluation of its lubricating capability.

5. The excellent condition of the rear bearing after the last Salox M test, which operated at 4×10^6 DN for 3 minutes, is an indication of the potential of this lubricant composite to perform satisfactorily for longer periods.