AD NUMBER

ADB046587

NEW LIMITATION CHANGE

TO

Approved for public release, distribution unlimited

FROM

Distribution authorized to U.S. Gov't. agencies only; Test and Evaluation; OCT 1979. Other requests shall be referred to Air Force Materials Laboratory, Nonmetallic Materials Division, Fluids Lubricants and Elastomers Branch, ATTN: MBT, Wright-Patterson AFB, OH 45433.

AUTHORITY

AFWAL ltr dtd 3 Aug 1983

AD B046587

AFML-TR-79-4091, Part II

LEVEL (2)

SELF-LUBRICATING COMPOSITE BEARING MATERIALS — PART II: CORROSION PREVENTIVE SOLID LUBRICATION OF THE ASALM MISSILE HYDRAULIC ACTUATOR BEARINGS

HUGHES AIRCRAFT COMPANY
Culver City, California 90230

December 1979
TECHNICAL REPORT AFML-TR-79-4091, Part II
Final Report for Peiod 1 June 1978 to 15 May 1979

Distribution limited to the U.S. Government agencies only; test and evaluation, October 1979. Other requests for this document must be referred to the Air Force Materials Laboratory, Nonmetallic Materials Division, Fluids, Lubricants and Elastomers Branch, AFML/MBT, Wright-Patterson AFB, Ohio 45433.

AIR FORCE MATERIALS LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
AIR FORCE SYSTEMS COMMAND
WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

NOTICE

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data, is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission to manufacture, use, or sell any patented invention that may in any way be related thereto.

This technical report has been reviewed and is approved for publication.

B. D. McConnell

B. D. MCCONNELL
Project Engineer

R. J. Benzing

R. J. BENZING
Fluids, Lubricants and Elastomers Branch

FOR THE COMMANDER

F. D. Cherry

F. D. CHERRY, Actg Chief
Nonmetallic Materials Division

"If your address has changed, if you wish to be removed from our mailing list, or if the addressee is no longer employed by your organization please notify AFML/MBT*, W-PAFB, OH 45433 to help us maintain a current mailing list".*

Copies of this report should not be returned unless return is required by security considerations, contractual obligations, or notice on a specific document.

AIR FORCE/56780/24 April 1980 — 210

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER AFML-TR-79-4091, Part II	2. GOVT ACCESSION NO. AD-B046587L	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) SELF-LUBRICATING COMPOSITE BEARING MATERIALS PART II. CORROSION PREVENTIVE SOLID LUBRICATION OF THE ASALM MISSILE HYDRAULIC ACTUATOR BEARINGS.		5. TYPE OF REPORT & PERIOD COVERED Final Report 1 June 1978 - 15 May 1979
		6. PERFORMING ORG. REPORT NUMBER FR 79-76-1146
7. AUTHOR(s) M.N. Gardos J.J. Bohner		8. CONTRACT OR GRANT NUMBER(s) F33615-76-C-5082
9. PERFORMING ORGANIZATION NAME AND ADDRESS Hughes Aircraft Company Culver City, CA 90230		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Project No. 7343 Task No. 73430209
11. CONTROLLING OFFICE NAME AND ADDRESS Air Force Materials Laboratory, Fluids, Lubricants and Elastomers Branch (MBT) Wright-Patterson Air Force Base, Ohio 45433		12. REPORT DATE December 1979
		13. NUMBER OF PAGES 177
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report) Distribution limited to U.S. Government agencies only; test and evaluation, August 1979. Other requests for this document must be referred to the Air Force Materials Laboratory, Nonmetallic Materials Division, Fluids, Lubricants and Elastomers Branch, AFML/MBT, Wright-Patterson AFB, Ohio 45433

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Self-lubricating Composites
Wide Temperature Range
High Load
Graphite Fibers
Self-lubricating Compacts
Molybdenum Disulfide
Antimony Trioxide
Roller Bearings
Hydraulic Actuators
Ceramic Coatings
Titanium Carbide
DC Magnetron Sputtering
Friction
Wear
X-Ray Analysis
Corrosion Resistance

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

(Attached)

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

ABSTRACT

This report deals with the efforts expended in a 14-month program to develop a corrosion-resistant hard coating and a solid lubricant top coat. The objective of these coatings was to afford hydraulic control actuator roller bearings corrosion resistance for a 10-year storage life and to allow bearing operation for 1000 seconds with an applied load of 31,581 Newtons (7100 lbs) over a temperature range of -54°C to 316°C (-65°F to 600°F).

By the use of exposure to aggressive chemical media, candidate ceramic hardcoat materials were evaluated, leading to the use of nichrome and titanium carbide in a two-layer structure. Similar exposure testing of sputtered lubricants led to the use of a molybdenum disulfide - antimony trioxide mixture. Additional solid lubricants based on the results of Part I of this report were used. Self-lubricating composite rollers were fabricated from stretched carbon fiber reinforced thermid 600 containing the Ga/In/WSe_2 Westinghouse compact. Eleven hardcoated bearings were tested in a wide temperature range oscillatory mode with a peak rotational rate of 250° sec^{-1}.

Testing demonstrated that the best hardcoat-softcoat combination was obtained using DC magnetron sputtered NiCr/TiC-MoS_2/Sb_2O_3. Bearings so processed met all dynamic test requirements with significant torque improvements over the baseline grease at very low temperatures. Bearing failures were due to metal galling at the roller face - inner race shoulder interface.

Twenty-four hour corrosion tests using 100 percent humidity at 100°C were performed on several rollers from three bearings. Results of that test demonstrated significant flowering corrosion.

Examination of the used bearings revealed large scale delamination of the nichrome-titanium carbide hardcoat.

The results of this study (presented herein) indicate that simple DC magnetron sputtering of refractory ceramic hardcoats does not result in sufficient adhesion as to preventing significant spalling of the coating and subsequent loss of corrosion resistance. This spallation, however, does not appear to significantly affect bearing operation with respect to desirably low torque value in a wide temperature range, provided that a suitable solid lubricant is used in tandem.

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

FOREWORD

This report was prepared by Hughes Aircraft Company under an extension of AFML Contract No. F33615-76-C-5082, originally titled "Self-Lubricating Composite Bearing Materials," for AFML/MBT. The work was administered under the technical direction of the Fluids, Lubricants and Elastomers Branch, Non-metallic Materials Division of the Air Force Materials Laboratory, with Mr. B.D. McConnell acting as Project Engineer.

The principal investigatorship was shared between Messrs. M.N. Gardos and J.J. Bohner; the Program Manager was Mr. M.N. Gardos. Professional assistance is acknowledged from Messrs. A.A. Castillo, F.J. Melcher and W.J. Harney, Jr., of Hughes Aircraft Company; Mr. Richard P. Riegert of the Technology of Materials, Incorporated (Quad Group), Santa Barbara, California; Messrs. A. Garner, M. Wohltmann, L. Michon and M. O'Leary of Martin Marietta Aerospace, Orlando, Florida; and Capt. M. Thoms of ASD/YYMA, Wright-Patterson AFB, Ohio.

This report covers the research conducted in assessing state-of-the-art, corrosion preventive solid lubrication methods and selecting the ones most likely to succeed in the hydraulic actuator bearings of the ASALM missile. The document presented here deals with the experimental effort expended between 1 June 1978 and 15 May 1979 and is the Final Report on the additional work covered by the modified contract.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

SECTION I

INTRODUCTION

Most, if not all, of the current designs dealing with rolling element bearings are based on the use of conventional oils and greases. The traditional dependence on these lubricants in ball and roller bearings is now increasingly questioned because of the emergence of operational environments severe enough for oils and greases to fail in satisfying key performance requirements.

The demand for wider temperature range use [-54°C (-65°F) to +316°C (+600°F)] in an oxidative environment (air) after long dormancy requirements (up to 10 years) exposes the fundamental physical and chemical limitations of liquid and grease lubricants normally used for both corrosion protection and lubrication of high temperature, high load, rolling element bearings. Oils and greases tend to: (a) thicken or freeze at low temperatures; (b) vaporize and decompose at high temperatures; (c) migrate away from critical bearing surfaces; (d) collect dust and dirt; or (e) chemically degrade over long storage periods. The latter problem can cause further complications, because an oxidatively degraded product will not provide the same corrosion protection (if any) as the original, nondeteriorated version.

At the same time, there are no data that can reliably indicate the corrosion preventive capacity of oils and greases for 10+ years. The total lack of theoretical models which could approximate a prediction of corrosion preventative lifetime is presumably due to the complexity of the degradation modes.

The inherently greater chemical stability and wide temperature range usability of solid lubricants overcome most of the limitations of oils and greases. These lubricants offer the potential for high temperature performance (up to 537°C (1000°F) today and even higher in the future), with long storage stability and load-carrying capacity greater than that provided by conventional lubricating oils and greases. While the load carrying capacity and, in some cases, the chemical oxidative stability of greases may be increased somewhat by solid lubricant fillers (e.g., MIL-G-21164 greases compounded from MIL-G-23827 products and molybdenum disulfide), the rest of the use limiting factors remain objects of concern.

The urgent need for extreme environment lubrication techniques recently precipitated a long-range fundamental study of dry lubricated rolling element bearings (Reference 1). This ongoing work notwithstanding, the present necessities are immediate enough to seek an interim solution now, using whatever state-of-the-art solid lubrication methods are at our disposal.

AFML/MBT suggested the design upgrade of the Advanced Strategic Air-launched Missile (ASALM) hydraulic actuator bearings operating the missiles aerodynamic control surfaces by changing the currently used grease

(Reference 2), or antiseize paste lubrication (Reference 3) to an all-solid lubricant system. The primary role f the solid lubricant(s) is corrosion prevention of the bearings for a storage period of not less than 10 years. The secondary role is the capability to lubricate the bearings under high loads, in the oscillatory condition, at temperatures of [-54°C (-65°F) to +316°C (+600°F)] for at least 15 to 20 minutes. The subordinate position of this capability is due to the unusually short operational life of the hardware. Actually, some technologists believe that the bearings could survive the brief variable temperature flight time with no lubrication at all, in spite of the excessive bearing loads; the concomitant lack of corrosion prevention, however, is obviously unacceptable.

For better understanding of the bearing's mission, Figure 1 should be consulted. The full complement roller bearings indicated therein are exposed to aerodynamics-related peak loads significantly higher than their ultimate load rating (aircraft static capacity). This design, due to inherent packaging problems, did not allow larger size bearings to support and drive the fin shafts, resulting in damage to roller and race at these high loads and temperatures. Again, the short operational life-time of the assembly is the only factor significantly contributing to bearing survivability in this system.

Most solid lubricants that do exhibit high load-carrying capacity [e.g., molybdenum disulfide (MoS_2) and graphite] are not corrosion preventive compounds. Actually, in certain cases, they can induce or promote corrosion. Therefore, these lubricants are not useful by themselves in bearings needing long-term corrosion protection. It appeared that a multicomponent lubricant system was needed where one part offered protection from corrosion, while another provided the solid lubricant which remained functionally useful throughout the storage period. This combination must be applicable for use in precision (i.e., close tolerance) bearings and has to provide immediate and reliable operation of the actuator bearings after storage periods as long as 10 years, under any conceivable storage conditions.

Since Part I of this contract culminated in the successful development of high load, high temperature, polymeric self-lubricating composites (see Reference 4), it was our contention that recent advances in chemically inert, refractory hard coatings, coupled with the promising new developments in solid lubrication, may now offer a combination of corrosion protection and lubrication to the ASALM actuator bearings. Subsequently, this work was undertaken in Part II of the program and is described in the present report.

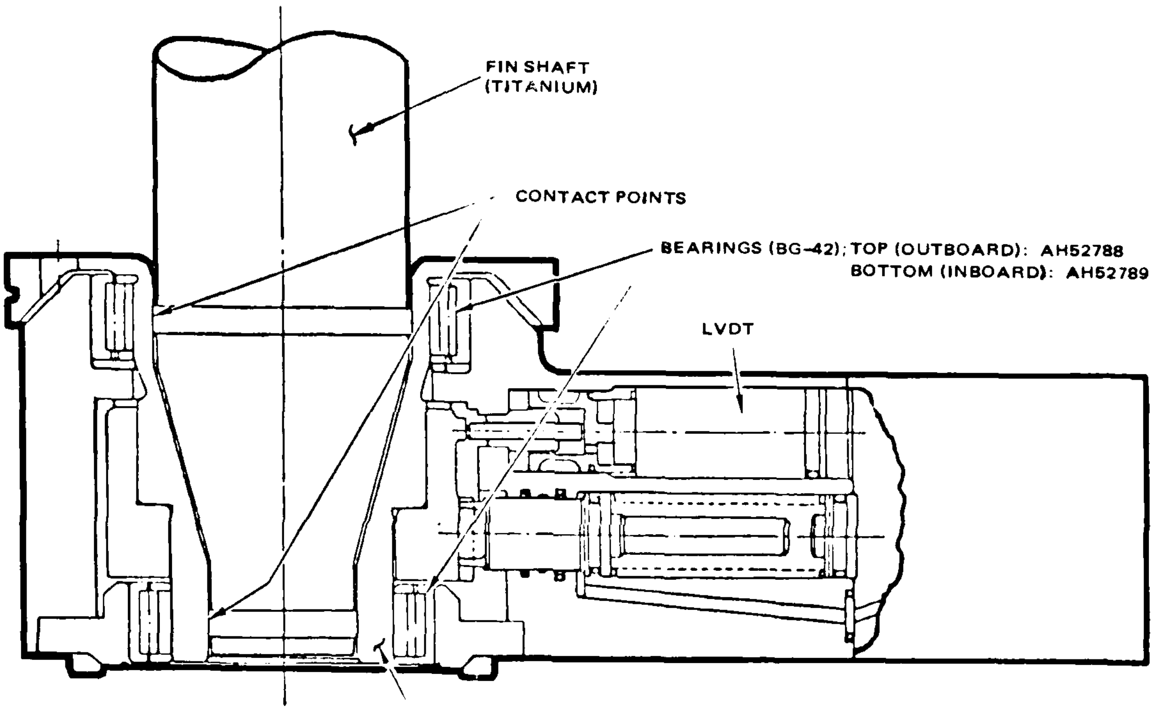


Figure 1. ASALM – Activator design-bearing detail (MDAC).

SECTION II

DESCRIPTION OF SOLID LUBRICATION TECHNIQUES

According to our plan, corrosion prevention and certain solid lubricant synergisms were to be provided by a thin (approximately 1200 Å) layer of sputtered hard ceramic coatings, such as titanium carbide (TiC) on the needles and on the races. The load-carrying capacity of the hard coating was to be enhanced by the most effective use of one of three likely solid lubrication techniques:

1. Transfer film formation by lipstick-type burnishing of the hard-coated races with a high temperature, powder-metallurgically prepared, self-lubricating compact and tumble burnishing the hardcoated rollers in the powder of the same compact;

2. Transfer film formation through the utilization of polymeric, self-lubricating composite rollers prepared according to the findings in Reference 4;

3. The application of a thin, sputtered layer of a solid lubricant top-coat, which exhibits satisfactory resistance to environmental degradation under storage, retaining its usefulness until and during missile flight.

The first two techniques may be considered as "field application" methods, i.e., those that are the least expensive and can be performed with a minimum of equipment, by field personnel with some training. The third technique, similar to the hardcoat application, is a laboratory prototype manufacturing kind. It requires specialized equipment, fully trained operators and it costs more.

A. SELECTION OF A CORROSION PREVENTIVE UNDERLAY FOR SOLID LUBRICANTS

1. Sputtering of Corrosion Preventive Hardcoat Candidates

The Hughes technical team selected the sputtered titanium carbide (TiC) coating (deposited in accordance with the Romelus process patented by Technology of Materials, Inc., Santa Barbara, California)* as the corrosion preventive, hardcoat underlay for the following reasons:

(a) This coating was successfully developed as a hard base coat, deposited on nickel-plated beryllium, covered by a thin, low shear strength MoS_2/Sb_2O_3 lubricant layer on top for effective gas (gyro) bearing operation (Reference 5). Despite the fact that the theoretical predictions brought forth recently in Reference 1 do not forecast success for satisfactory adhesion of physically vapor deposited, ion-sputtered hardcoats in high load, rolling element bearings, it seemed

*US Pat No. 4,124,472

reasonable to at least preliminarily investigate the performance of this well established process (reportedly effective under low load, high speed, moderate temperature, unidirectional sliding conditions) under high load, low oscillating speed, wide temperature range, rolling condition also.

(b) TiC itself is chemically inert. It is well known (Reference 6) that TiC is resistant to water at least to 649°C (1200°F), its oxidation in air does not become severe until approximately 1093°C (2000°F), and it does not react with nitrogen until the temperature increases above 1204°C (2200°F). Since Martin Marietta, Orlando, Florida (one of the ASALM system design integrators) now has data showing that the top flight temperature of the actuator bearings does not exceed 316°C (600°F) (see Reference 7), thermal oxidative/ hydrolytic (i.e., humidity) degradation of TiC is not an object of concern. There is, however, a greater need for a pinhole-free, non-porous coating, because the key to its effective corrosion protection is a well-adhering, dense and homogeneous nature. If these properties can be achieved, the film's corrosion protective ability during the -54°C(-65°F) to 71°C (160°F) temperature range, up to 95 percent relative humidity, 10 + years storage period is fully anticipated.

Therefore, all initial efforts focused on the corrosion preventive ability of sputtered TiC and other selected, rival hardcoat candidates and on attempts to prove the hardcoats' satisfactory adherence to the steel bearing substrates under rolling conditions, at loads far higher than those experienced in hemispherical gas bearings.

2. TMI Corrosion Test and Results

The MDAC*, St. Louis, Missouri, actuator bearings (see Figures 2 and 3) and the Martin Marietta versions (Reference 8) were to be fabricated from high strength, corrosion-resistant steels, such as 440C, BG-42 (a modified 440C) or M-50. If we were to use a steel similar in strength but marginal in corrosion resistance (such as 52100) for corrosion test specimens, the ability of any hardcoat to provide protection to the latter would imply overcapacity to protect the corrosion resistant versions.

TMI chose to evaluate DC magnetron-sputtered coatings of SiC, TiC, Fe_3O_4, HfC, TiB_2, B_4C and a proprietary mixture of $TaSi_2$, TaC and Al_2O_3 (TASO), using commercially available 52100 steel needle bearing rollers as test specimens (supplied by Hughes Aircraft Company). The test methods and the results are described in TMI's September Interim Report, Appendix A.

*McDonnell Douglas (MDAC) is the other potential supplier of the ASALM missile.

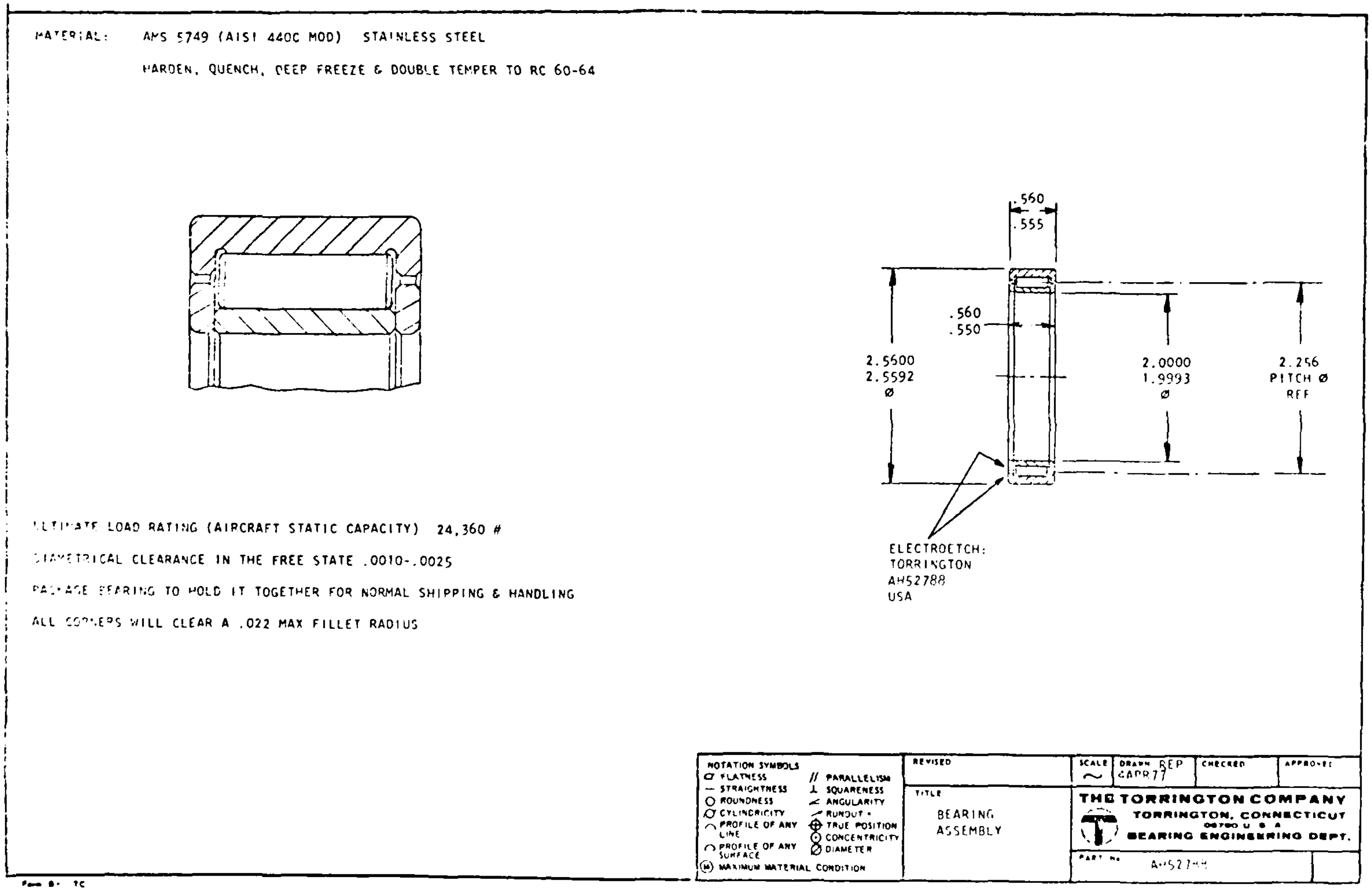


Figure 2. MDAC outboard actuator bearing.

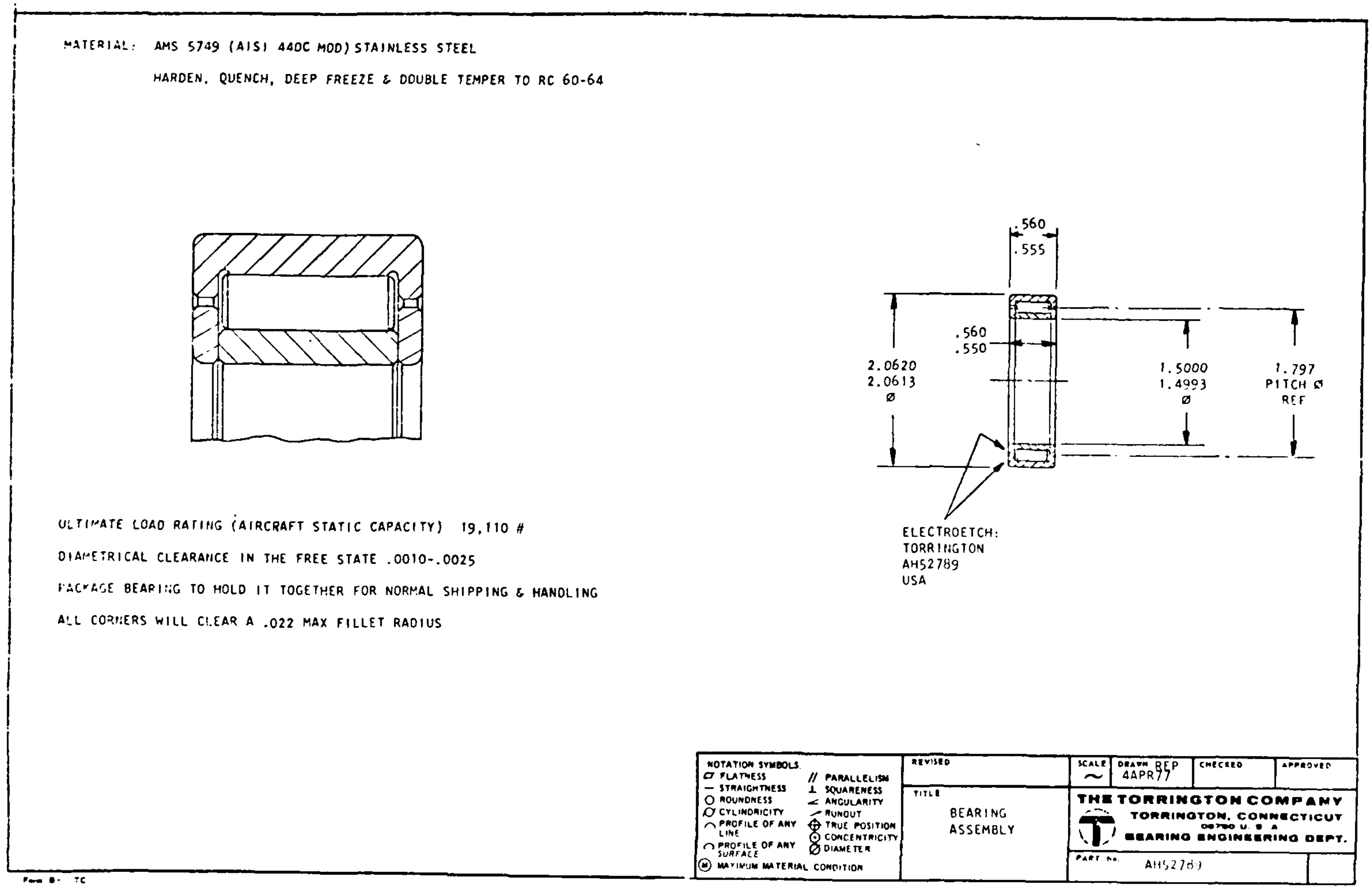


Figure 3. MDAC -- Inboard actuator bearing.

Essentially, hardcoated steel rollers were exposed to dilute HCl, dilute NaOH, a boiling, concentrated NaCl solution, boiling water and steam. The reason why such a variety of chemical environments were selected had been the lack of theoretical predictive methods capable of defining an accelerated corrosion method, representative of 10 years' worth of exposure to various storage conditions. We hoped to imply that overdesign may lead to survival during long-term storage.

The results indicated (see Table 1 of TMI's September Interim Report, Appendix A) that TiC and TASO were the most effective in providing the best (but not total) corrosion protection to the 52100 needles. Due to the more extensive experience with the former with respect to tribological use, the TiC coating was preferred for further work. Nevertheless, both hardcoats were subjected to friction and wear testing. Note that in the latter stages of work, TMI chose to impart complete corrosion protection to the bearing surfaces by a 1000Å thick, sputtered nichrome layer under the final hardcoat candidate film. This intermediate layer does not diminish (in fact, it may enhance) the adhesion of the TiC to the steel.

3. TMI Wear Tests and Results

(a) Disc-on-Disc Wear Tests - With respect to the tribological performance on any hardcoat in the present case, the main object of concern is its adherence to the rollers and the races under the extremely high Hertzian stresses of the actuator bearings at all test and operational temperatures.

Even though a low shear strength, compliant, top layer of a solid lubricant tends to reduce the maximum bearing stresses through a kind of cushion effect (see Reference 1), this reduction is estimated insufficient to alleviate the concern about possible coating delamination.

The tribological quality (judged according to TMI gas bearing lubrication standards) was determined on the TMI disc-on-disc spin wear tester, so as to yield a minimum of five minutes of operation at 10,000 rpm, at room temperature, under a 50-gram load applied onto identical contacting discs depicted in Figure 4. The starting torque had to be less than 0.8 and the running torque less than 0.6 on the TMI standard disc coating scales, at standard settings. Other particulars of the testing procedure are presented in TMI's December Interim Report, Appendix B.

The test results, also given in Appendix B, indicated that 440C flats coated with both hardcoats (no nichrome underlay) performed better than similarly processed 52100 specimens. Both hardcoats had similar wear resistance, although the TASO was more prone to be noisy. The 2500Å (10 μin.) films were considerably noisier than the 1250Å (5 μin.) ones. The sputtered, solid lubricant topcoat's presence, however, smoothed out all differences in noise as a function of hardcoat thickness.

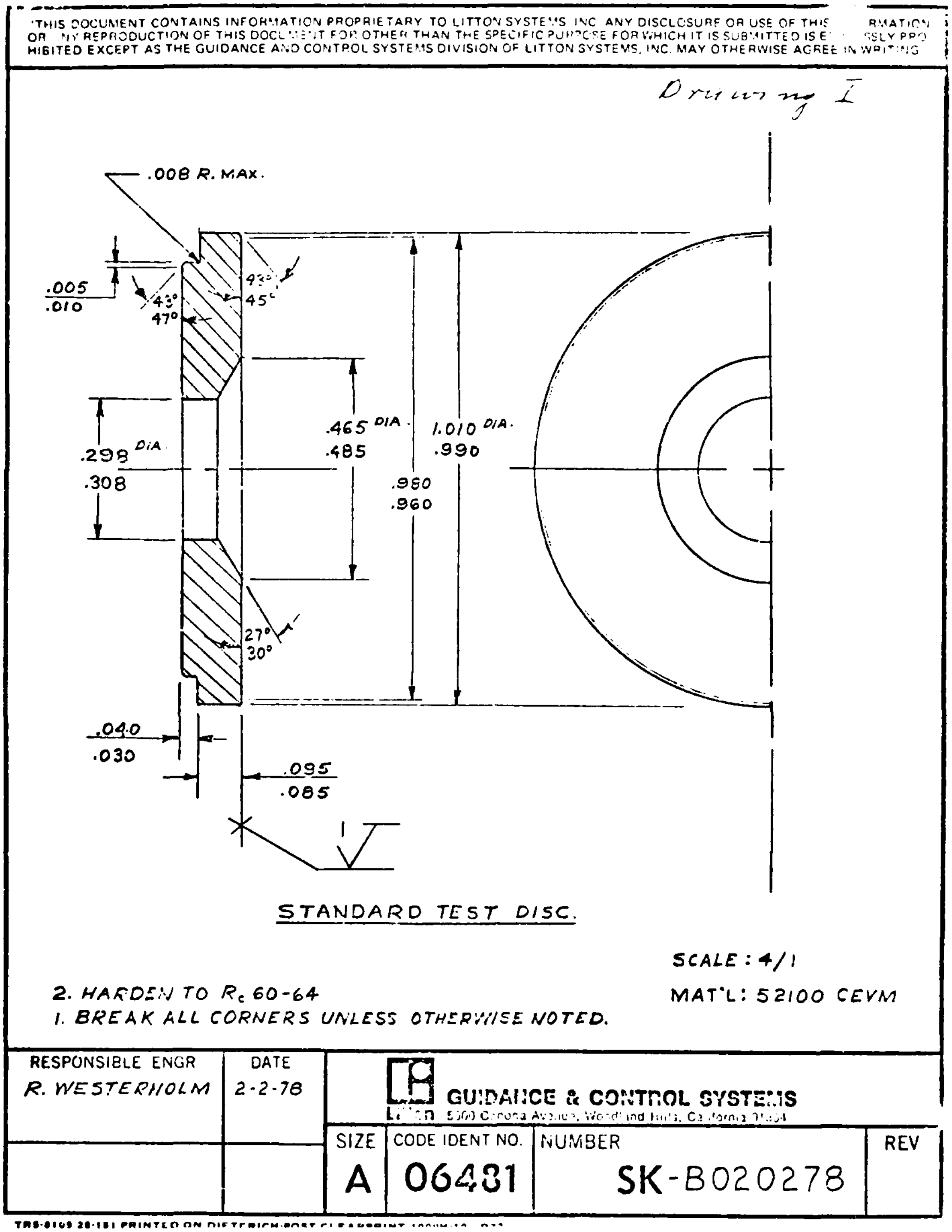


Figure 4. TMI standard test disc (courtesy of Litton Guidance and Control Systems, Woodland Hills, CA.)

(b) Needle Bearing Tests - It was recognized from the outset that the TMI disc spin wear test apparatus was not capable of testing the hardcoat candidates under the type and magnitude of loads the actuator bearings would see in service. In fact, the imposition of such high loads, especially in the manner occurring during flight, requires special test apparatus, beyond the means of either TMI or Hughes to have supplied within the scope of this program. While both Martin Marietta and MDAC seemed amenable to test the final, coated actuator bearings under more realistic conditions, there were two main obstacles to the application of low load, gas bearing coatings to high load, rolling element bearing use:

- TMI needed a test apparatus on its premises, which would have allowed testing and subsequent modification of the hardcoats to obtain the best attainable coating adhesion within the limitations of the sputtering process.

- The actuator bearings (see Figures 2 and 3 and Reference 8), fabricated from BG-42 and 440C respectively, were extremely expensive to be used as test specimens. The former costs ≅ \$1200 and the latter (due to lower sales prices associated with an early batch of 440C Patriot missile actuator bearings now serving to approximate the new ASALM design) is > \$400. Aggravating the cost problem is the procurement lead time, ranging anywhere from 6 months to 1 year.

As an interim, compromise solution, TMI agreeded to adapt an existing apparatus to accommodate testing of inexpensive (~\$6 ea.), off-the-shelf, 52100 needle bearings selected and supplied by Hughes Aircraft Company as surrogate specimens. These INA bearings were purchased from INA Bearing Co., Cheraw, So. Carolina, under INA Catalog No. NAO 35x50x17 (35mm ID, 50mm OD, 17mm width) with extra 2.5 mm diameter, ~17mm long rollers. Preliminary analysis indicated that these parts, supplied with a stamped, mild steel retainer permanently housing a complement of rollers, could be converted into a full-complement bearing by removing the retainer and replacing it with a full complement of specially purchased, INA Catalog No. NRB 2.5 X 16.8 G2 rollers. Note that these extra 52100 steel rollers also served as TMI corrosion test specimens.

By means of this simple bearing tester, TMI intended to correlate hardcoat (and later, the hardcoat/solid lubricant) performance between room temperature, low load sliding and room plus elevated temperature, higher load rolling conditions. The limited, 226 kg (500 lbs) maximum dead-weight test load was about an order of magnitude less than the aircraft static load capacity of the 52100 test bearing. Nevertheless, a cost/lead time analysis indicated that a more realistic load capacity had to be foregone in favor of simplicity and ready availability of TMI inhouse testing.

However, as indicated in TMI's December Interim and Special Reports, (Appendixes B and C), the 52100 bearings were unable to operate satisfactorily when assembled in the TMI low load, high/low temperature bearing tester. Repeated visits by Hughes bearing technologists and a concerted, cooperative effort failed to solve this problem, probably due to the inability of the off-the-shelf bearings to operate in the full complement mode. Recall that this bearing was designed to work with rollers operated and permanently retained by a stamped, mild steel cage. In view of the facts that (a) TMI had lost an invaluable tool in observing the low load, elevated temperature tribological behavior of the best sputtered hardcoats under rolling condition (which would have given at least an indication of a capacity to carry high loads), and (b) the scope of the work did not permit an extensive bearing tester/specimen development, the following changes had to be incorporated:

- The few of Martin Marietta's 440C steel Patriot actuator bearings that would have represented the final specimens slated for hardcoating and solid lubrication turned out to be the only viable test specimens remaining. This rendered the TMI bearing tester inoperable because it was designed for the smaller 52100 test bearings. Therefore, arrangements were made with Martin Marietta and the ASALM program office at the Wright-Patterson AFB to obtain additional Patriot bearings and have all of them tested at Martin Marietta Aerospace, Orlando, Florida.

- Consequently, TMI was asked to hardcoat all available Patriot bearings (a total of 11 ea) with TiC, per the Romelus process, but without the benefit of any development work the use of the TMI tester would have afforded.

B. SOLID LUBRICANT APPLICATIONS

The failure criterion of the actuator bearing is determined as a maximum allowable torque value estimated from a hydraulic pressure differential, measured between the imposed radial load on the bearings and the hydraulic device duplicating the effects of aerodynamic stress during flight. In order to keep this torque low, the traction (i.e., coefficient of friction) must be held as low as possible between the rolling/sliding surfaces. Although the rolling friction of solid lubricated bearings is inherently lower than that of oil or grease lubricated ones due to the lack of viscous shear losses, solid lubricants are extremely sensitive to edge loading and the characteristic misalignments in roller or needle bearings.

Therefore, conventional resin or inorganic bonded solid lubricants were not acceptable, because they exceed the dimensions of the available bearing clearances as applied and are also vulnerable to edge loading. Delamination and/or wear of these thick coatings results in large particles of debris which manifests themselves in excessively high torque.

Hughes bearing technologists measured the Patriot bearing clearance to be 62.5 μm (0.0025 in.). In view of hardcoat/softcoat covering of the outer race, roller and inner race surfaces, filling of the total radial clearance represents four times the deposited coating thickness. It follows that the total deposited hardcoat/softcoat film system thickness on any one bearing surface must not exceed 15.6 μm. Since TMI finally elected to deposit

1000 Å of nichrome (a corrosion prevention-assuring interface) onto the 440C surfaces, followed by 1250 Å of TiC, the resulting 2250 Å (0.225 μm) would allow the application of a more than 15 μm thick solid lubricant topcoat. Neither the Hughes field application methods nor the softcoat sputtering techniques of TMI apply coating thicknesses anywhere near that large in the present case.

1. Field Application Methods

(a) Stick-burnishing with the Ga/In/WSe_2 Westinghouse Compact - As described in Reference 4, this high temperature self-lubricating compact is an extremely effective solid lubricant, both at room and at elevated temperatures. One of the contributing factors to its success is the ready ability to form low friction transfer films, both in the sliding and the rolling mode. Another is its thermo-oxidative stability in air.

Early research by Jones and Gardos (Reference 9) showed that under simulated ball/ballpocket and ball/race contact conditions, the Westinghouse compact ballpocket debris can get transformed into a much more uniform transfer film on both the ball and the race than those generated from any other polymeric self-lubricating composite. This film-forming tendency is retained even when the compact is used as a pulverized solid lubricant additive in polymeric lubricant composites (Reference 4). In summary, the Westinghouse compact is singularly universal at low and high temperatures and load, under both sliding and rolling conditions.

As depicted in the SEM photomicrographs of the finely pulverized Westinghouse compact in Figure 5, its film-forming tendency appears to lie in its characteristic microfragmentation and the cohesiveness of the very small particle size powder. Under shear loads, the particles are consolidated into homogeneous films, both under single transfer (composite ball pocket to ball) or double transfer (ball pocket to ball and from there to race) conditions.

Therefore, one readily available and relatively simple solid lubrication method was the mechanical deposition of a thin Westinghouse compact layer on the hardcoated races by a special compact lipstick burnishing technique, developed by Hughes Aircraft Company (Reference 10). This process was originally established to pre-form a polymeric composite (single) transfer film on the races of bearings slated for operation with the same self-lubricating composite retainer. Normally, the rubbed film is formed as shown in Figure 6, under controlled load and speed conditions and in an inert atmosphere (due to the oxidative sensitivity of several solid lubricant pigment types). In the case of the Westinghouse compact, its oxidative stability precludes the need for an inert gas stick-burnishing environment.

Accordingly, the burnishing stick was fabricated from the Westinghouse compact, the races were inserted in a specially fabricated aluminum receptacle, which in turn was installed onto the shaft of a variable speed

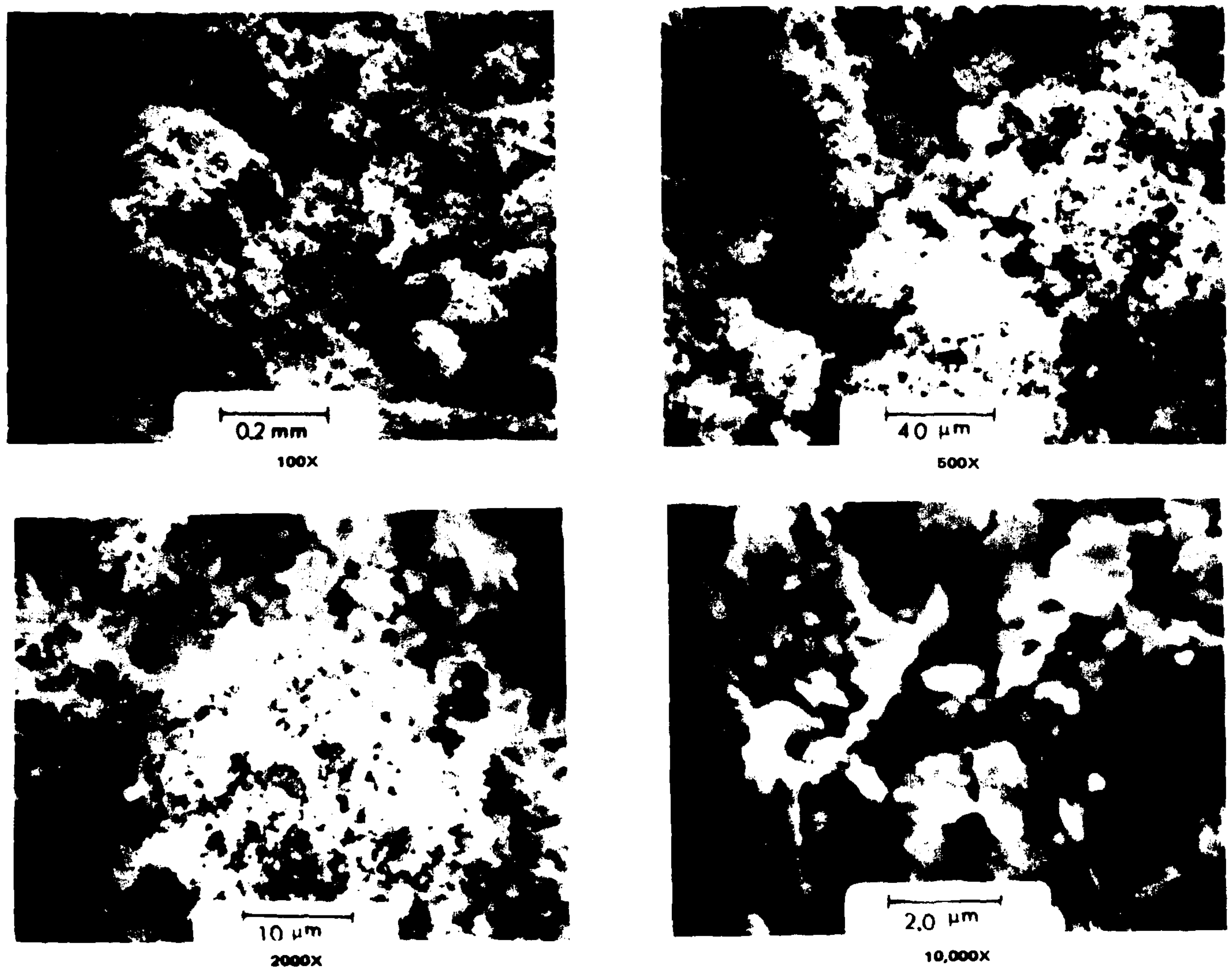


Figure 5. SEM photomicrographs of pulverized Westinghouse compact at 100X, 500X, 2000X and 10,000X magnifications.

Figure 6. Hughes Process (HP) 7-29 stick burnishing apparatus.

motor. By trial and error, it was found that 900 passes at a 120 rpm motor speed, under ~2.3 kg (5 pounds) of hand-applied load, visually complete transfer films were formed.

The hardcoated rollers were gently tumbled in a clean polyethylene bag, filled with a given charge of the Westinghouse compact powder, as specified by a special Hughes lubrication process (Reference 11). This process describes MoS_2 powder burnishing of rolling element bearing balls and races.

(b) Transfer Film Formation by Polymeric, Self-Lubricating Composite Rollers - As shown in the Part I Final Report of the present, unamended contract (see Reference 4), the Thermid 600 polyimide based, carbon/graphite fiber reinforced advanced self-lubricating composites readily form single transfer films under sliding conditions, at both low and high temperatures and loads. The film formation is enhanced by the fact the active solid lubricant pigment within these composites is the pulverized version of the Westinghouse compact.

Therefore, in order to fortify the rubbed-on, stick-burnished films of this compact (see description of the previous process), a scheme was devised to deposit a run-in, double transfer film on top of the single transfer compact layer, formed by one of our advanced, self-lubricating, polymeric composites.

Three major problems had to be overcome before this scheme became feasible:

- Transfer film formation on rolling element bearing components is achieved through the use of a self-lubricating compact or composite cage. The Patriot bearings are of the full complement kind, however, without a roller-separating cage. A partial complement of rollers fabricated from the composite could act as a pseudo-retainer, provided they could be specially designed for the application.

- The advanced Hughes composites are reinforced with a three-dimensional (3D), simple orthogonal weave of carbon or graphite fibers. These carbon/graphite preforms are woven in sheets of various thicknesses and are incorporated into the composite by special molding techniques to simultaneously align the majority of the fiber lay perpendicular to the applied bearing load and in the plane of sliding. This geometry provides the highest dynamic compressive strength to the composite, at the same time imparting acceptable tribological behavior. The machining of rollers from sheet-like slabs of orthogonal 3D weave reinforced composites would result in rollers of anisotropic strength in the radial direction and anisotropic surface morphology circumferentially. This could possibly affect the fragmentation characteristics of the roller surfaces and thus the homogeneity of the transfer film.

Therefore, the rollers were prepared by impregnation of a unidirectionally stretched HMS graphite fiber reinforcement structure with a Thermid 600 varnish, containing the powdered Westinghouse compact as the lubricant additive and dibasic ammonium phosphate as the adjuvant.

HMS, a Type I (high modulus, graphitic) fiber was selected, because its characteristic microfragmentation leads to greater composite wear and thicker transfer film formation (Reference 4). The unaxial structure, depicted in Figures 7 and 8, provides circumferentially isotropic properties in terms of strength and wear rate and, as a bonus, higher fiber tip wear at the roller face/race shoulder interface. This higher fiber wear results in the formation of a thicker transfer film there, the site of highest friction and greatest probability for surface damage.

- Ideally, alternating steel and self-lubricating composite rollers into a pseudo-retainer design should be applicable not only for run-in, but also for in-service operation of the actuator bearings. Unfortunately, these bearings are already overloaded in the full, steel roller complement mode during flight and cannot afford to contain rollers weaker than hardened steel. Therefore, the rollers are useful for transfer film formation only. Again, the HMS reinforcement structure lends itself better to this role, because it generally tends to provide a thicker film.

Consequently, a special fiber-stretching fixture was designed to work in conjunction with a special mold, which permitted the preparation of 2 each 7.62 cm x 1.9 cm x 0.76 cm composite moldings. These bars served as stock for the machining of 32 each 0.953 cm long x 0.635 cm dia. composite rollers by turning the bars into elongated cylinders and cutting the rods into rollers.

Every second steel roller in the bearing was replaced by a composite one. Each bearing was then rotated at 50 rpm under no load until the steel rollers became covered with an even transfer film whose appearance (i.e., color, reflectivity and visual consistency) did not change with further run-in. This run-in time (120 min) was recorded, the bearing disassembled and reassembled with the remaining hardcoated but still unlubricated rollers. The run-in was then repeated with the used composite rollers for the same length of time, in the same manner. Note that the hardcoated races and rollers were all previously stick-burnished and powder-tumbled with the Westinghouse compact to prevent damage early in the run-in process.

This run-in procedure provided a thicker transfer film than the original, stick-burnished one. Here, the film was definitely heavier on the races than on the rollers. Although there was

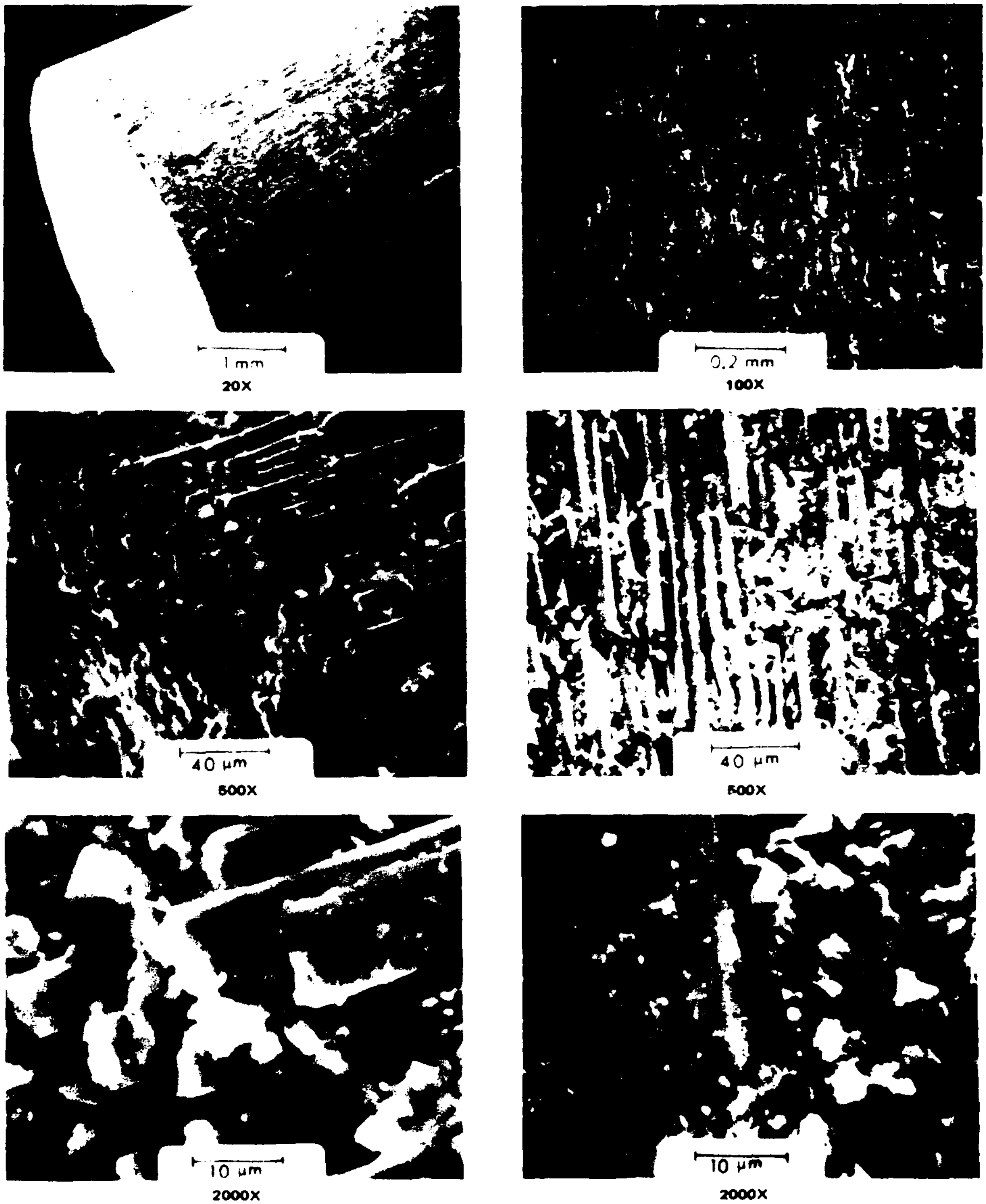


Figure 7. SEM photomicrographs of unused carbon fiber reinforced Thermid 600 composite rollers at 20X, 100X, 500X, and 2000X magnifications.

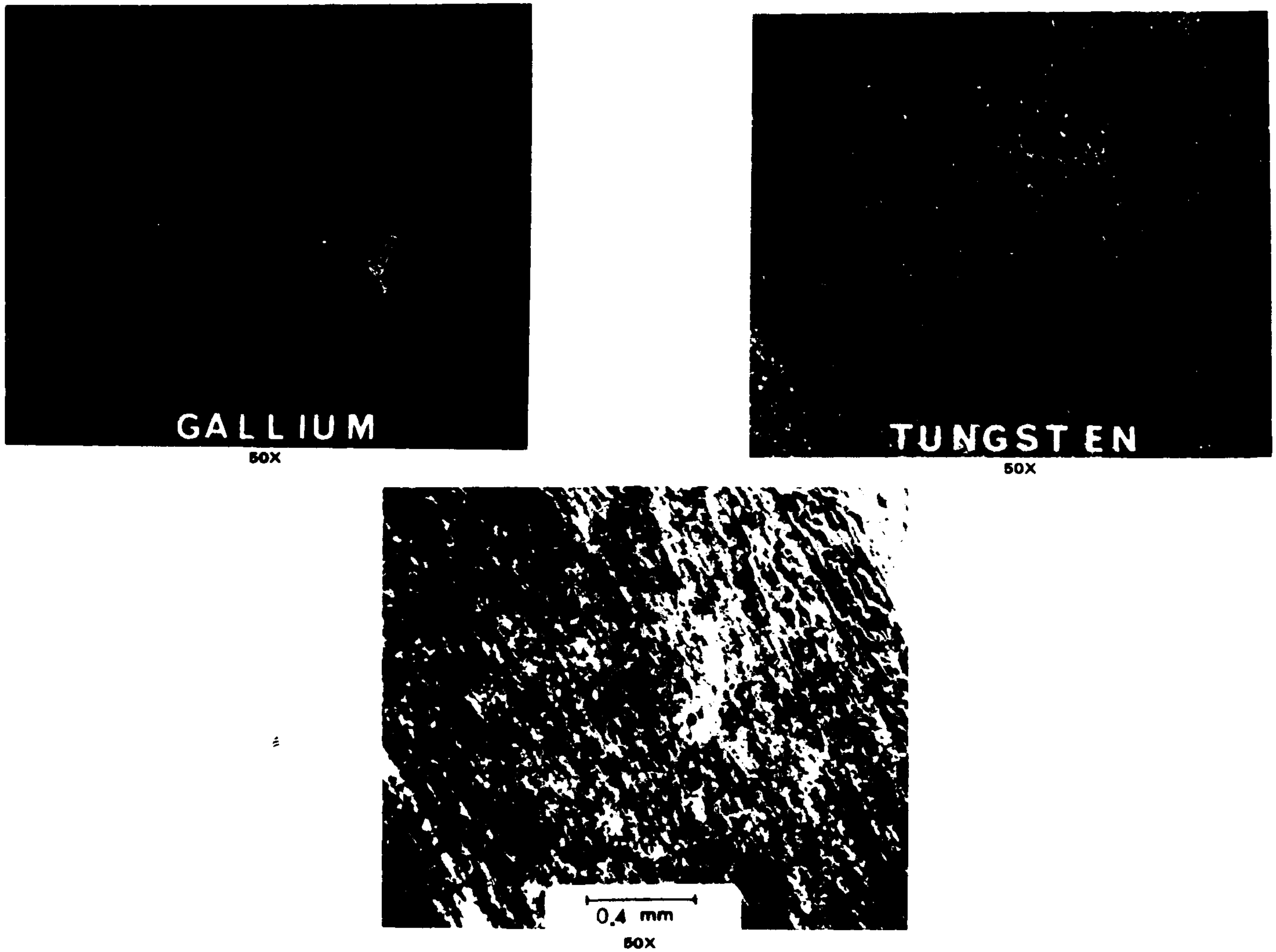


Figure 8. SEM photomicrograph and EDX dot maps of unused carbon fiber reinforced Thermid 600 composite rollers at 50X magnification.

no time for the accurate measurement of the respective coverages, it was estimated that the thickness on the rollers was less than 1 µm (40 µin) and on the races less than 2 µm (80 µin).

As shown in Figure 9, the used composite rollers retained their structural integrity during the run-in process and succeeded in depositing a complete transfer film. Only one roller cracked after 8 hours of use but it did not break into separate pieces.

2. Laboratory Prototype Application of Sputtered Softcoats

As previously mentioned, a MoS_2/Sb_2O_3 film, sputtered on the TiC hardcoat, was shown effective for gas (gyro) bearing operation. The respective processes were developed by TMI and are now licensed to selected inertial guidance equipment manufacturers.

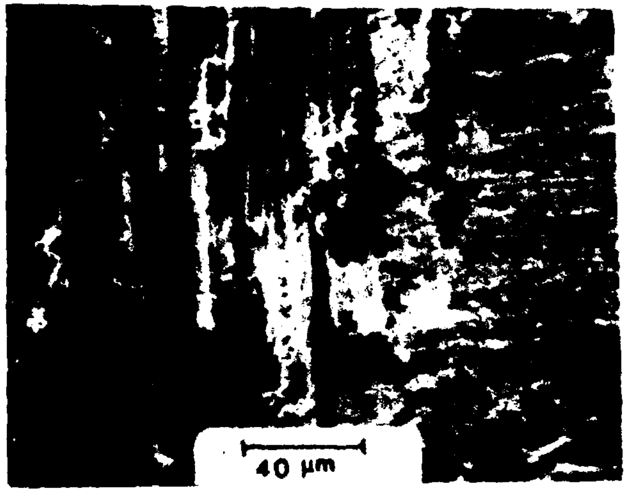
It still remained to be seen whether the MoS_2/Sb_2O_3 combination or any other rival softcoat could withstand the accelerated storage simulating, corrosive environments of HCl, NaOH and NaCl solutions and exposure to 100 percent humidity. TMI technologists selected boron nitride (BN) as one other likely solid lubricant candidate and both films were exposed to the above-listed environments. This was followed by friction and wear testing of the best softcoat on specimens previously hardcoated with TiC or TASO.

(a) TMI Corrosion Tests and Results - As described in TMI's Interim Report (Appendix B), BN and the MoS_2/Sb_2O_3 were deposited in three thickness ranges [750Å, 1250Å and 2500Å (3 µm, 5 µm and 10 µm)] and tested under the four environmental conditions. All BN compositions failed catastrophically on exposure to humidity and were eliminated from further consideration. The MoS_2/Sb_2O_3 was attacked by HCl, but gave reasonable resistance to the other test conditions.

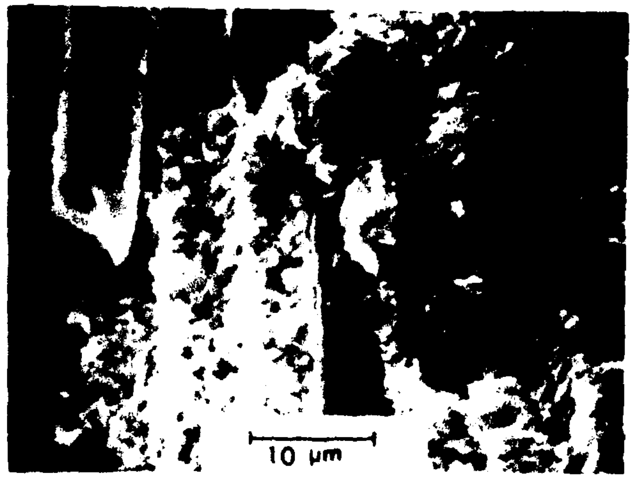
(b) TMI Wear Tests and Results - As shown in Appendix B, the 3000Å thick MoS_2/Sb_2O_3 topcoat imparted far better tribological properties to the hardcoated disc specimens than the hardcoats alone.

Some of the 52100 needle bearings were then coated with the TiC-MoS_2/Sb_2O_3 combination for the TMI bearing tests. However, as discussed before and more thoroughly explained in Appendix C, the needle bearing test efforts failed.

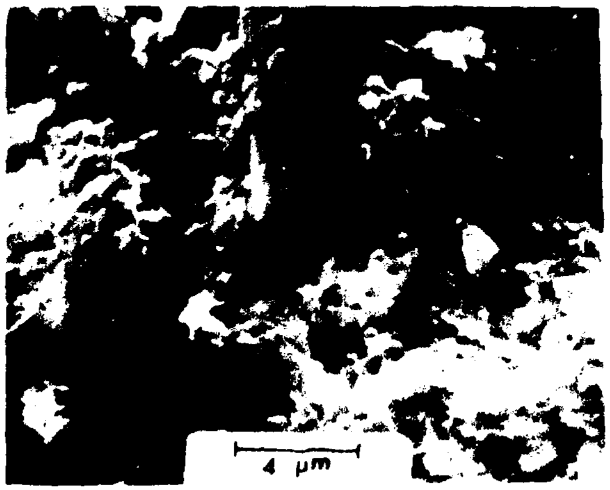
Therefore, the 11 each 440C Patriot missile actuator bearings were secured from Martin Marietta and coated with the sputtered nichrome (1000Å)/sputtered TiC (1250Å) hard layer combination, followed by the application of a 3000Å thick MoS_2/Sb_2O_3 film on all components of two of the 11 bearings.



500X



2000X



5000X

Figure 9. SEM photomicrographs of used carbon fiber reinforced Thermid 600 composite rollers at 500X, 2000X and 5000X magnification.

The races were fixtured in the sputtering chamber in the same manner as during hardcoating, to expose all areas (especially the frictional surfaces) evenly to the flux emanating from the target. Specifically, the races were fixtured as described in Reference 8, rotating in a double planetary mode. The rollers were rolled about randomly in a rotating, platelike trough, suspended under the magnetron target at an angle.

The coated but disassembled parts of each bearing were packaged separately and delivered to Hughes Aircraft Company for further disposition (see Figure 9).

Hughes technologists then completed solid lubrication of the hard-coated bearings using the "field application" methods, visually and photographically examined the components, disassembled all of the bearings and, finally, forwarded them to Martin Marietta for testing.

SECTION III

BEARING TESTS

A. INTERFACING WITH POTENTIAL ASALM MANUFACTURERS

Before any of the experimental work began on Part II of the program reported herein, we considered it prudent to contact not only the ASALM Program Office but also the potential missile manufacturers for their advice, suggestions and final concurrence on the soundness of the Hughes solid lubrication plan for the actuator bearings. The paragraphs below describe the chronology and substance of the interactions.

1. McDonnell Douglas Astronautics Company (MDAC) St. Louis, Missouri

Mr. Vern King, the Unit Chief in Design, was contacted by phone on 9 May 1978 to discuss the Hughes program and to solicit his suggestions and help in the forthcoming work. He did not encourage a Hughes visit to MDAC, since all actuator bearing work is subcontracted to an upstate New York firm (Moog). A copy of the Hughes proposal was forwarded to him for comments. On 14 July 1978, Mr. King called to provide feedback. The following key information was obtained during these two telephone calls:

(a) No extra BG-42 (modified 440C steel) Torrington bearings were available to our program for solid lubrication. The cost of each bearing would be approximately $1, 200 with machining lead time of about 6 months or more.

(b) The Braycote 3L-38RP grease (a Vydax 1000 telomer filled, organophilic bentonite and sodium nitrite thickened Fomblin Z perfluoroalkyl polyether compound manufactured by Bray Oil Co., El Monte, California) was jointly selected by Messrs. John Christian of AFML/MBT and the late Keith Demorest of the NASA Marshall Space Flight Center. Apparently the grease performed well during short term bearing tests. Mr. King was apprised of the marginal corrosion preventive capacity of fluorinated polyether greases, in spite of the presence of sodium nitrite. The effectiveness of this additive does not come into play until bearing operation.

It was also revealed that Mr. Christian himself offered a grease candidate, formulated at the AFML/MBT laboratories from a special, high temperature, fluorosilicone base oil.

(c) MDAC estimated 343°C (650°F) average and 382°C (720°F) peak actuator bearing temperatures. The actuator bearings are tested beyond their static load capacity. A pseudo duty cycle was designed, using a high temperature 382°C (720°F) maximum test bed, providing about ten different events. Each event lasts from a few seconds up to 200 seconds for the last event. The latter consists of two cycles per second, ±8 degree angle of oscillation at a kg (27,000) pound load (outboard bearing) and a kg (21,000) pound load (inboard bearing), both at the 382°C (720°F) top estimated bearing temperature. Note that MDAC was willing to do similar bearing tests for us if the need arose.

Since these phone conversations, MDAC decided to test two each of their actuator bearings in May 1979 on their own, operating with the TMI-applied nichrome/TiC coating system, lubricated with greases. These bearings failed, due to the development of high torques, as described later herein.

2. Martin Marietta Aerospace, Orlando, Florida

Mr. Earl Waters, the Systems Manager of the ASALM Program, was also contacted on 9 May 1978. Mr. Andrew Garner, a Senior Staff Engineer (then Principal Investigator for the actuator bearings) responded on 12 May 1978 and arranged a trip to Los Angeles for 19 May 1978. During that preliminary discussion at Hughes, we received assurances for Martin Marietta's full cooperation in the future. At that time, we planned another trip for Mr. Gardos (Hughes Program Manager) and Mr. William J. Harney, Jr. (Hughes bearing specialist) to visit the Martin Marietta plant and conduct detailed discussions on mutual technical information transfer.

The meeting at Martin was held on 25 May 1978. The minutes are described in ASALM TD&I Meeting Minutes, Appendix D. The discussions were fruitful; we received Martin Marietta's full cooperation in all endeavors. Their responses to the meeting is given in Martin Marietta's follow-up letter, Appendix E.

The key technical points of these interfaces may be summarized as follows:

(a) Martin Marietta agreed with the Hughes approach.

(b) Fel-Pro C-100, an antiseize paste consisting of MoS_2 and lead in a hydrocarbon oil carrier, is now used as a bearing lubricant. There seemed to be some bearing disassembly problems after high temperature tests due to evaporation of the fluid carrier and subsequent caking of the lubricant residue. As an additional problem, C-100 is not known as an effective, long-term corrosion preventive compound.

(c) Six each 440C Patriot missile actuator bearings (by SKF, Inc., Philadelphia, PA) were provided to the Hughes program (free of charge) for application of the best hardcoat/softcoat corrosion preventive solid lubricant coating combination and for subsequent testing in the Martin Marietta facility described in Reference 3. Note that the Torrington bearings, see Figures 2 and 3 are not used here.

(d) Martin Marietta technologists originally concurred with our selection of off-the-shelf 52100 bearings as the readily available, low cost test specimens for the preliminary screening tests. Since the hot hardness of 52100 is poor, we planned to select its top test temperature such that this temperature agrees with that corresponding to the hot (i.e., 316°C = 600°F) hardness of 440C. Some tests were to be run, however, with 52100 steel bearings up to 427°C (800°F) to see if this steel could perform at least adequately in short term, high temperature service. The derived advantages would have been obvious, both in terms of bearing cost and availability.

Since these initial contacts, Martin Marietta's evaluating role increased, due to the larger number of the Patriot test bearings. Under a separate contract, testing of the solid lubricated bearings and a few of the hardcoated ones operating with selected greases (the baseline specimens) commenced on 18 May 1979 (see forthcoming data and discussion).

Also, as previously mentioned, additional flight thermal data enabled the reduction of the top actuator bearing temperature in the Martin Marietta design from 371°C (700°F) to 316°C (600°F), (Reference 10).

Meanwhile, Mr. Garner was succeeded by Mr. M. Wohltmann and eventually by Mr. L. Michon as our liaison engineer at Martin Marietta.

B. THE MARTIN MARIETTA TEST APPARATUS AND PROCEDURE

1. Apparatus

Essentially, the test fixture consisted of a test shaft supported by two heavy-duty pillow blocks with the test bearing cantilevered outside the pillow blocks at one end of the test shaft. Radial loading was applied through the test bearing's outer race using a hydraulic jack rated at 2.22×10^5 N. (50,000 lbs). A second smaller hydraulic jack, acting through a 5.08×10^{-2} m (2 in.) bellcrank (pitman arm) at the opposite end of the test shaft, oscillated the test shaft and thus the bearing's inner race. A servo system utilizing a Linear Voltage Differential Transformer (LVDT) mounted on the smaller actuating hydraulic jack was used to control bearing rotation while a rotary potentiometer was employed to record shaft rotation. All test bearings were sinusoidally oscillated ±25° at 1.6 Hz, providing a peak rotational rate of 250° sec^{-1}. Shaft torque was obtained through a load cell located on the small hydraulic jack assembly. For a more detailed description of the apparatus and test procedure, see the Martin Marietta ASALM Bearing and Lubrication Final Test Report in Appendix D.

The preplanned thermal profiles were attained by conditioning the bearing at -540°C (-65°F) through Blowing LN_2 into the test chamber. The test assembly was thermally insulated by a loose fitting box surrounding the test bearing region, fabricated from fiberglass insulating board. At the start of the test, full current was applied to five Chromalox heaters which included one cartridge, one strip, one clamp and two ring heaters, all mounted in close proximity to the test bearing. Heating rates of approximately 0.407^{o} C sec^{-1} (44^{o}F/min) were sufficient to raise the bearing temperature from -54^{o}C (-65^{o}F) to the required 316^{o}C (600^{o}F) in 18 to 20 minutes.

2. Test Procedure

As discussed in the Bearing/Lubrication Test Plan (Martin Marietta Document TPL00900041-001) attached as Appendix G, the bearings were to be subjected to a two-step test procedure. However, early bearing failures revealing the cause of unusually rapid failure and speedy failure analysis led to modification of that test plan following a mid-program review by Hughes, Martin Marietta and AFML (HAC-MM-AFML) personnel. As a result of that review, a total of four separate test procedures were used. They are illustrated in Figures 1-3 and 3-1 of the Martin Marietta ASALM Bearing and Lubrication Final Test Report in Appendix F. Test Plans A& B constituted the original two-step procedure.

Under Test Plan A, as in all other plans, the bearing was oscillated briefly prior to the application of any radial loads in order to determine overall system frictional torque. At the beginning of each test the 31,581 N (7100 lb) radial load was applied and the heaters energized. The bearing was then operated until either failure occurred or 1000 seconds were accumulated. Failure was judged to have occurred when the bearing torque rose above 226 N-m (2000 inch pounds). During the 1000 seconds, the temperature rose from -54^{o}C to +316^{o}C (-65^{o}F to +600^{o}F).

Under Test Plan B, the bearing was tested first per Test Plan A. If the bearing was still operating at the end of 1000 seconds, the test was continued for another 1000 seconds with the temperature held at 315.5^{o}C (600^{o}F) and with the radial load doubled to 63,162 N (14,300 lbs). Here again the same failure criterion torque value was retained.

Under Test Plan C, the temperature was held at 316^{o}C (600^{o}F) and determine if thermal shock was the source of a given bearing failure. As in Test Plan A, the radial load of 31,581 N (7100 lbs) was applied for 1000 seconds.

According to Test Plan D, the radial load was reduced to 15,791 N (3550 lbs) while the temperature was driven from 54^{o}C (-65^{o}F) to 316^{o}C (600^{o}F). This was done to determine whether the high radial load was the source of early bearing failure, as would be expected if delamination of the TiC hardcoat was due to mechanical stresses encountered in testing.

C. TEST RESULTS

1. Overview

Of the 13 bearings available for testing, 12 were subjected to dynamic testing at Martin Marietta while one was retained at Hughes for corrosion resistance studies. Table 1 summarizes the results of the 12 bearing tests, lubricants used, hard coating (if any) and the Test Plans utilized. Individual test results [(Plan(s)] are presented both in tabular and graph format in the Martin Marietta ASALM Bearing and Lubrication Final Test Report (Appendix F). As can be seen from Table 1, seven bearings passed their respective tests with six of those meeting or exceeding the ASALM mission requirements.

2. Base Line Bearings

Four bearings were used to develop baseline performance data for the test bearings. Two bearings were grease lubricated with Fel-Pro C100 to test the effectiveness of this lubricant. This lubricant is considered by Martin Marietta as their best candidate and base line lubricant. One of these two grease lubricated was NiCr-TiC hard coated to see if the corrosion resistant hard coat could be used with the base line grease lubricant without performance degradation. Two other bearings were tested dry, one was unlubricated, the other stick burnished/tumble burnished with the Westinghouse compact. The first bearing tested was to evaluate the effectiveness of the bare hardcoat while the second was used to evaluate the effectiveness of the Westinghouse compact to lubricate a baseline-nonhardcoated bearing.

Both grease lubricated baseline bearings, numbers 6 and 12, easily passed Test Plan A. At the -53.9°C (-65°F) start temperature, however, the Fel-Pro C100 was a viscous semi-solid and that resulted in high bearing torques (247 N-m (2190 in-lb) and 134 N-m (1195 in-lb), respectively. The hardcoated bearing of that pair (No. 5, see Table 1) did have a higher initial torque than the base 440C bearing, but both had almost identical torques at the end of 16 minutes. This difference may be due to contamination and will be discussed in the used bearing analysis section of this report.

The dry baseline bearings showed mixed performance. Bearing No. 7, which was unlubricated, began to squeal after only 2 minutes under Test Plan A. By the end of 4 minutes of testing, the torque had been rising continuously, seizure was judged imminent and the test stopped in order to avoid damaging the test stand. Prior to the beginning of squeal, this bearing had exhibited a low torque, only 44 N-m (385 in-lbs). Adding a solid lubricant, as was done to bearing No. 13, significantly improves the bearing life. The testing of bearing No. 13 was stopped when the test stand failed 20 minutes into Test Plan B. Close examination of the torque curve, Figure A-12 in the

TABLE 1

SUMMARY OF MARTIN MARIETTA DYNAMIC BEARING TESTS AND RESULTS

Lubricant Application Method	Brg. No.	Hard Coating	Lubricant	Test Plan*	Test Results	Initial Torque	Final Torque	Comments
Dry Baseline	7	NiCr/TiC	None	A	Failed 4 min.	45.2 N-m (400 in-lbs)	196.6 N-m	Squeal at 3 min.
	13	None	Stick/tumble West. Comp. Burn.	B	Passed 20 min.	36.7 N-m (325 in-lbs)	68.4 N-m (605 in-lbs)	Test stand Fail prior to test Completion
Grease Baseline	5	NiCr/TiC	Fel-Pro C100	A	Passed 16 min.	247 N-M (2190 in-lbs)	32.7 N-m (290 in-lbs)	High -54°C torque
	12	None	Fel-Pro C100	A	Passed 16 min.	13.4 N-M (119 in-lbs)	68.4 N-m (605 in-lbs)	High -54°C torque
Stick/tumble Burnishing	1	NiCr/TiC	$Ga/IN/WSe_2$	A	Failed 4 min.	58.3 N-M (516 in-lbs)	192 N-m (1700 in-lbs)	Squeal 2 min. Seizure at 4 min
	2	NiCr/TiC	$Ga/In/WSe_2$	C	Failed 5 min.	31.2 N-m (276 in-lbs)	264 N-m (2340 in-lbs)	Low -54°C torque
Stick/tumble Burnish and Composite Roller Run-In	3	NiCr/TiC	$Ga/In/WSe_2$	A	Failed 4 min.	146.5 N-m (1296 in-lbs)	235 N-m (2080 in-lbs)	Stopped at high torque, no seizure
	4	NiCr/TiC	$Ga/In/WSe_2$	D	Passed 16 min.	38.4 N-m (340 in-lbs)	72.3 N-m (640 in-lbs)	Low -54°C torque
	8	NiCr/TiC	$Ga/In/WSe_2$	A	Passed 16 min.	65 N-m (575 in-lbs)	36.2 N-m (320 in-lbs)	TiC clean out f_K fell continuously
	9	NiCr/TiC	$Ga/In/WSe_2$	B	Failed 8 min.	38.9 N-m (345 in-lbs)	300.6 N-m (2660 in-lbs)	TiC clean out squeal at 6 min.
Sputtered	10	NiCr/TiC	MoS_2/Sb_2O_3	A	Passed 16 min.	57.6 N-m (510 in-lbs)	48.0 N-m (425 in-lbs)	Very smooth low torque
	11	NiCr/TiC	MoS_2/Sb_2O_3	B	Passed 32 min.	49.2 N-m (435 in-lbs)	83.6 N-m (740 in-lbs)	Very smooth low torque

*Test Plan

Conditions

- A -54°C (-65°F) to 316°C (600°F) in 1000 seconds at a constant load of 31,581 N-m (7100 lbs).
- B Test Plan A followed by 1000 seconds at 316°C (600°F) with load doubled to 63,162
- C +21°C (72°F) for 1000 seconds at a constant load of 31,581 N-m (7100 lb)
- D -54°C (-65°F) to 316°C (600°F) in 1000 seconds at a constant load of 15,790 N-m (3550 lbs).

Martin Marietta ASALM Bearing and Lubrication Final Test Report attached as Appendix F, shows that during the last 2 minutes the torque was beginning to increase rapidly. Had the test continued, it is probable that bearing No. 13 would have failed Test Plan B but it did pass the original mission requirements as set forth in Test Plan A (Test Plan A is the first 16 minutes of Plan B).

3. Stick-Burnished Bearings

The first "field application" lubrication method to be evaluated was the stick-burnishing method. Originally anticipated to be the least effective lubrication method, only two hardcoated bearings were prepared by this technique. The two hardcoated bearings were solid lubricated by stick burnishing the races with the Ga/In/WSe_2 compact and tumble burnishing the rollers in the powdered version of the same. The disassembled bearings were then shipped to Martin Marietta for testing.

Bearing No. 1 had an initial torque of 58.3 N-m (516 in-lbs) and operated smoothly for 2 minutes when it began to squeal. The torque rose at an increasing rate throughout the test until the bearing seized after only 4 minutes' operation. The last recorded torque was 192 N-m (1700 in-lbs). During dissassembly the outer race shoulder was chipped as shown in Figure 10.

Bearing Number 2 was tested later in the test series and therefore, as a result of the HAC-MM-AFML meeting, it was decided to test this bearing per Test Plan C, i.e., cycle at the normal load of 31,581 N-m but remove the thermal shock by running at a constant temperature of 21°C (70°F). The initial torque was 31.2 N-m (276 in-lbs) and rose at an increasing rate during the test to 264.4 N-m (2340 in-lbs) at failure which occurred 5 minutes 20 seconds after starting. This was accompanied with bearing squeal for the last 3 minutes.

4. Transfer Film Lubricated Bearings

A total of four hardcoated bearings received lubrication using the self-lubricating composite rollers. Also it is this group of bearings which were affected most by the decisions reached at the HAC-MM-AFML meeting. At the time of that meeting, bearings No. 1, 3 and 7 had already failed in Test Plan A, with bearing No. 5 being the only one to complete the test procedure successfully. Also, preliminary analysis (to be presented later in this report) had begun to shed light on failure mechanisms. For instance, it was learned that the hardcoated bearings which were not sputter coated with MoS_2-Sb_2O_3 were coated with a fine dust of TiC particles approximately 4.0×10^{-7} to 3.6×10^{-6} meters (1.6×10^{-5} to 1.4×10^{-4} in) in diameter (see Figure 11). This was discovered in time to clean bearings 8 and 9 only. All other bearings were already tested in this loose TiC debris-coated condition. Bearings 8 and 9 were cleaned by assembling the

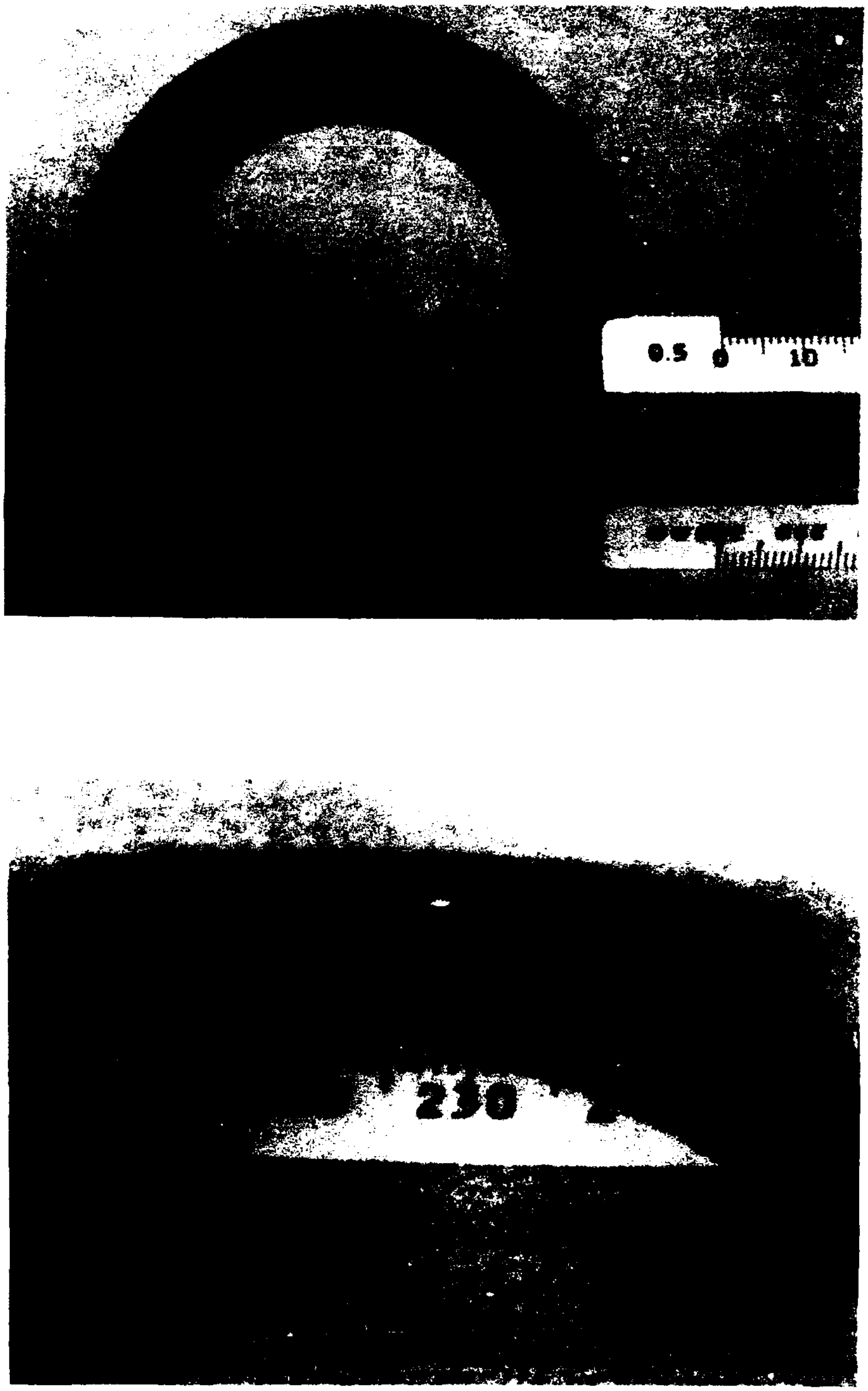


Figure 10. Stick burnished bearing number 1 showing chipped outer race shoulder.

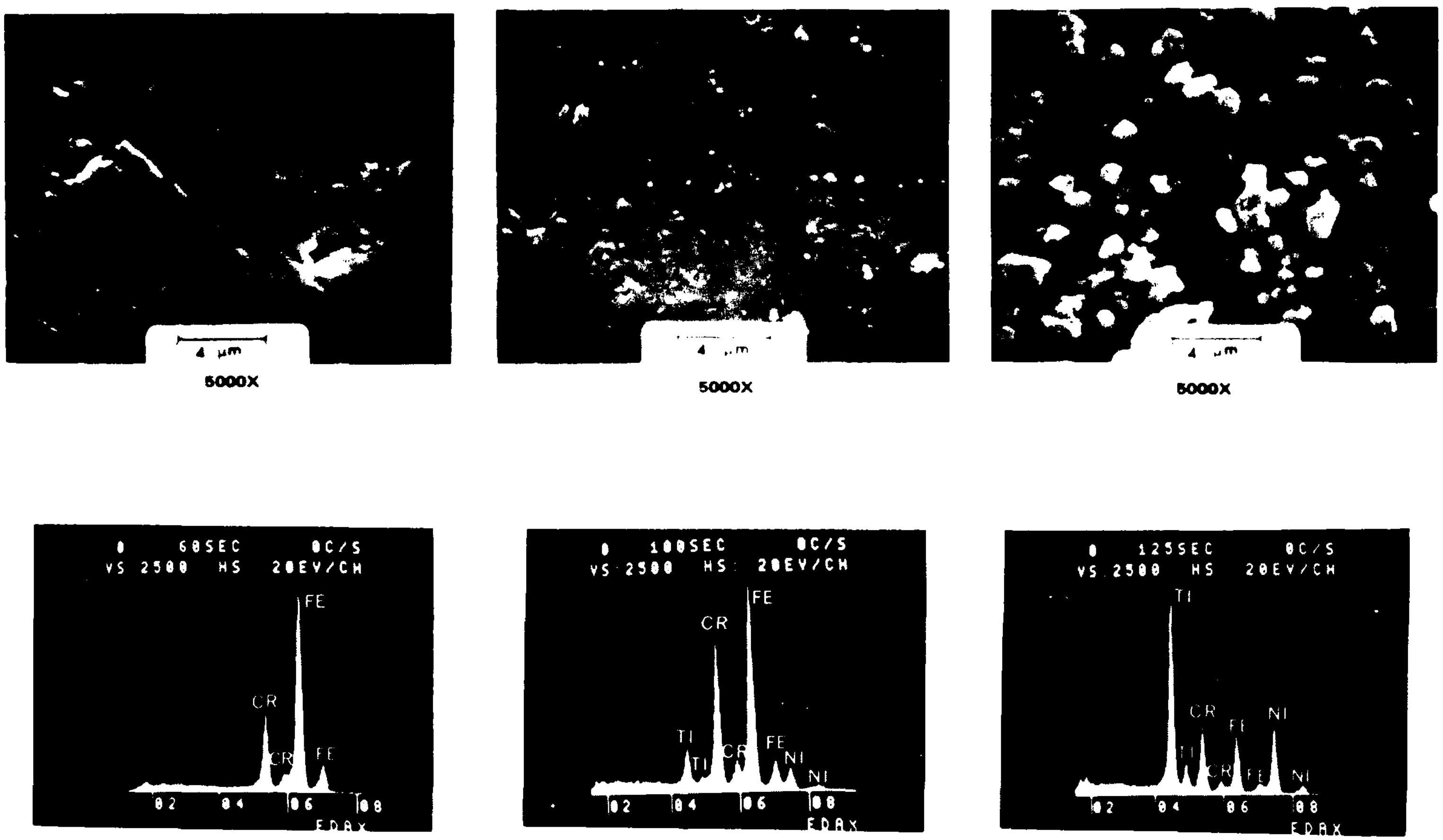


Figure 11. SEM photomicrographs and EDX spectra of bare 440 C and TiC particle coated roller surfaces at 5000X magnification.

bearing and rotating it by hand while rinsing the roller-race contact zone with Freon TF.* The loose debris was collected on a 0.45 μm Millipore filter depicted in Figure 12. Figure 13, showing the appearance of the outer races of bearings No. 8 and 9 following that wash, indicates that more than just loose debris is removed under this lightly loaded, benign rotation. The photographs indicate significant spallation, totally unexpected of a hard, tightly adhering TiC coating.

The races of bearing No. 3 were burnished with the Ga/In/WSe_2 compact and then assembled with alternating NiCr-TiC coated and self-lubricating composite rollers. The bearing was run-in as described earlier in this report.

Bearings No. 4, 8 and 9 were processed similarly, except that in order to protect the bare hardcoated rollers during the first stages of run-in, they were separately tumble burnished in the powdered Westinghouse compact prior to run-in with the composite rollers.

Bearing Number 3 had a high initial torque of 14.7 N-m (1296 in-lbs) which decreased for the first 7 minutes of Test Plan A. Therefore, it began to increase until a failure was declared after 9 minutes, 14 seconds when the torque exceed 235 N-m (2080 in-lbs). It was decided at the HAC-MM-AFML meeting to test bearing No. 4 per Test Plan D, i.e., the temperature profile remained the same as in Test Plan A but the load was reduced by one-half to 15,791 N-m (3,550 lbs). This was done to determine if it was the high load that was leading to bearing failure. The bearings torque was low initially and remained roughly constant for the first 5 minutes; then it rose rapidly for 1 minute. From the sixth minute to the successful completion of the test at 16 minutes, the torque rose slowly to a maximum of 72.3 N-m (640 in-lbs).

Bearing No. 8 successfully completed Test Plan A but an identically prepared bearing, No. 9, failed Test Plan A after 9 minutes and 50 seconds. Both bearings had low initial torques typical of the solid lubricated bearings.

5. Sputtered MoS_2-Sb_2O_3 Lubricated Bearings

The solid lubricant system which was anticipated to have the best performance was sputtered MoS_2-Sb_2O_3. Two NiCr-TiC hardcoated bearings, No. 10 and 11, were sputtered at TMI with a 7.6×10^{-8} m (3×10^{-6} in.) thick soft top coat of MoS_2-Sb_2O_3 solid lubricant, per the previously described procedure. Following this lubrication step, the bearings were assembled and run-in for 4 hours. Figure 14 shows these sputtered and run-in bearings with bearings No. 3 and 4 following their run-in using the self-lubricating composite rollers. Visible in the figure is evidence of chatter or

*Trichlorotrifluoroethane solvent, by DuPont

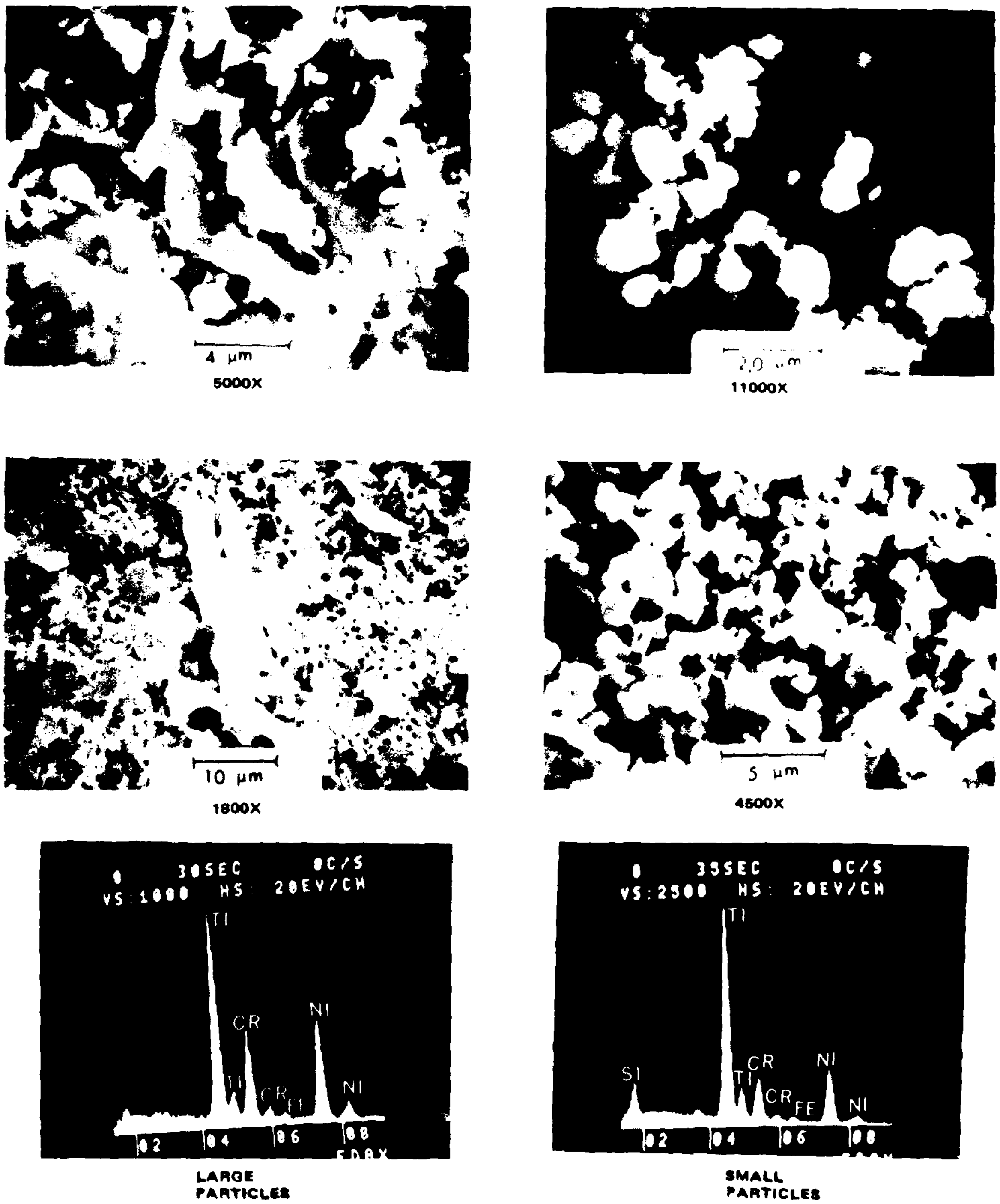
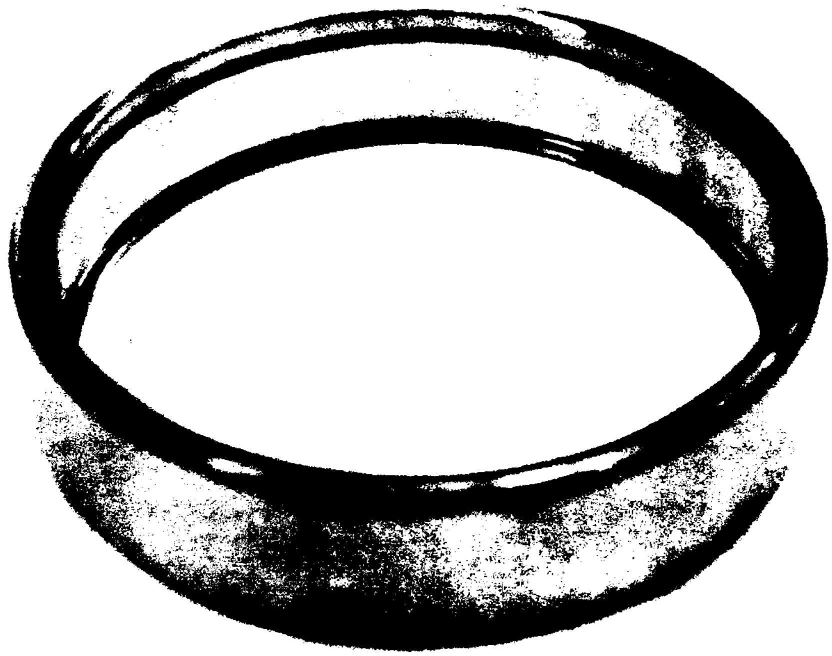
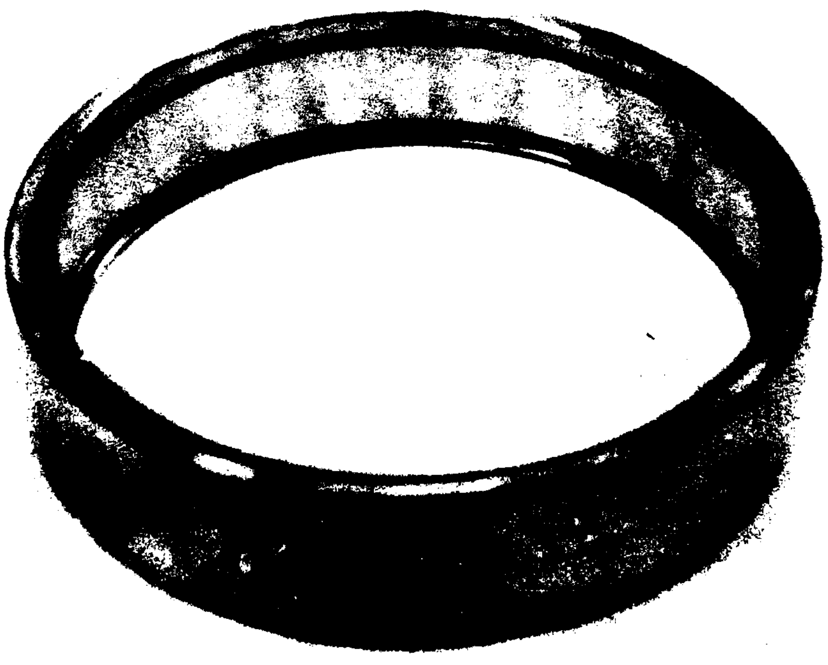


Figure 12. SEM photomicrographs and EDX spectra of TiC/NiCr debris washed from unused bearing 8 and details of roller surface damage following cleaning at 1800X, 4500X, 5000X and 11,000X magnifications.

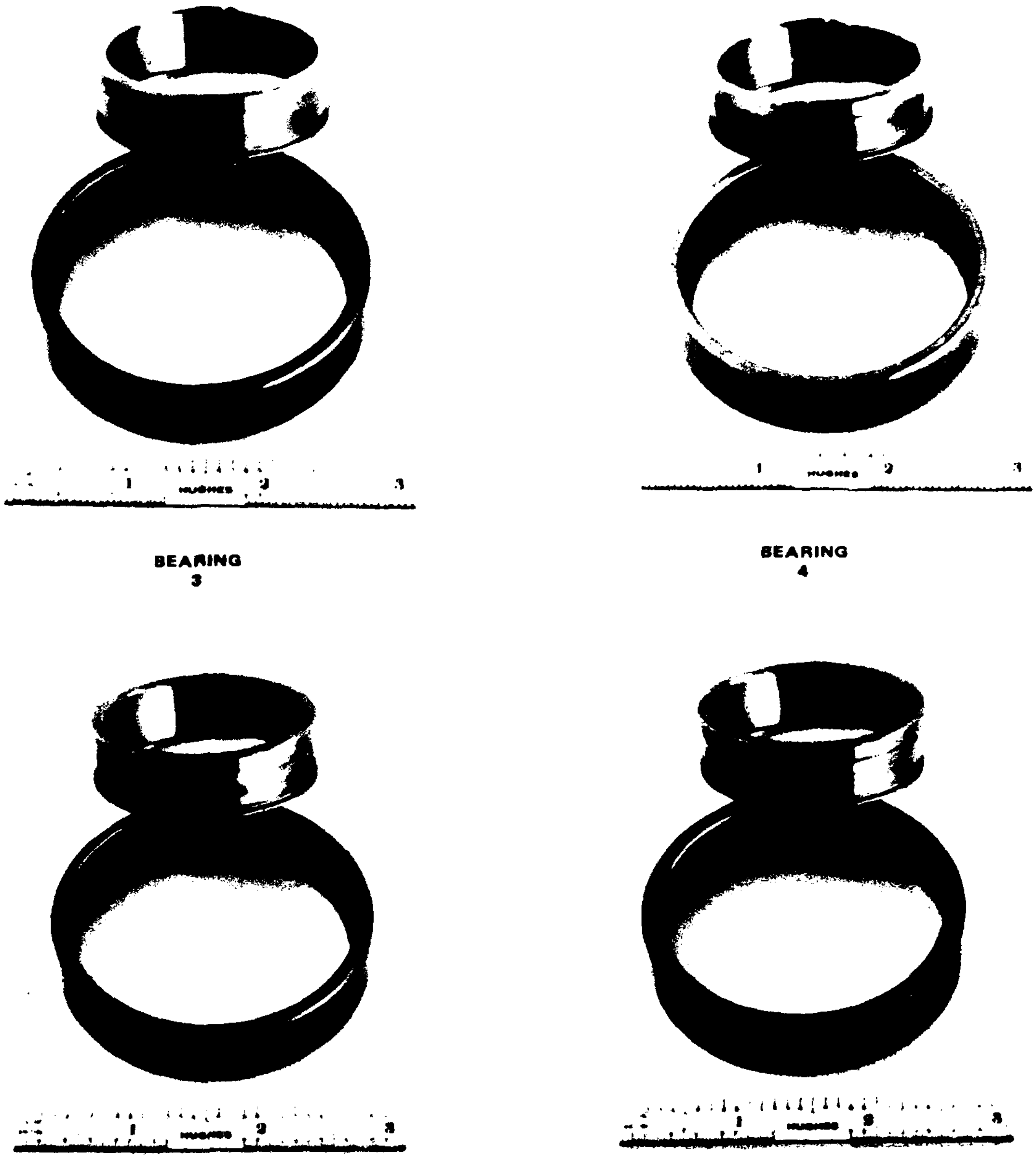


NUMBER 8



NUMBER 9

Figure 13. Appearance of unused bearings numbered 8 and 9 following cleaning in Freon TF, showing significant spallation of the TiC coating.



BEARING
3

BEARING
4

BEARING
10

BEARING
11

BRG 4	=	4R-54288
5	=	4R-54289
10	=	4R-54290-1
11	=	4R-54290

Figure 14. Appearance of run-in bearings 3, 4, 10 and 11 prior to high load - high temperature testing.

spallation similar to that seen in the cleaned bearings (Numbers 8 and 9 in Figure 13). There is no operational need to run-in the sputter lubricated bearings here, as there is no transfer film to establish and any torque reduction on run-in is inconsequential. They were run-in here only to provide a pre-test conditioning equivalent to that of the composite roller run-in bearings. Such run-in may in fact reduce the operational life time of solid lubricant sputtered bearings, especially in the case of marginal adhesion of the solid lube film.

Bearing No. 10 was tested per Test Plan A and had an initial torque of 57.6 N-m (510 in-lb). The torque increased for 2 minutes, then decreased for 5 more minutes. From 7 minutes to the successful end of the test, the torque remained constant at 46.9 N-m (514 in-lb). The test stand operator noted some brief squeaking at about the 4-minute mark.

Bearing No. 11 had an initial torque of 49.2 N-m (435 in-lbs). Tested per Test Plan B, the torque decreased slowly for the first 16 minutes to a value of 32.2 N-m (285 in-lbs). At that point the load was doubled and the torque rose to 69.5 N-m (615 in-lbs) increasing slightly through the next 16 minutes to 83.6 N-m (740 in-lbs). No noise was heard at any time during this test.

Except for the grease lubricated baseline bearings, only these sputter lubricated bearings met ASALM mission requirements consistently as defined by the criteria in Test Plan A. In view of the remarkably better, i.e., lower, low temperature initial torque, (not unexpected, due to the lack of viscous shear of the low-temperature thickened grease) and the lack of the elevated temperature caking tendency of the grease, the sputtered MoS_2-Sb_2O_3 soft coating seems to be the lubricant of choice for this application.

SECTION IV

ANALYSIS OF USED BEARINGS

A. ANALYTICAL APPARATUS AND PROCEDURE

All used bearings were shipped to Hughes in the as-tested, and assembled condition. Upon receipt, the bearings were visually inspected and then photographed at various stages of disassembly. Following disassembly, selected rollers and inner races were examined by Scanning Electron Microscopy (SEM) and Energy Dispersive X-ray analysis (EDX). The outer races were not studied by SEM or EDX because they were too large to fit into the SEM cavity without sectioning. Also, the inner races are predicted to have experienced higher stress levels than the outer races (Reference 12). Therefore it was expected that the inner races would be more severely damaged and thus were of greater interest.

Because time was of the essence, the analysis of used bearings was brief and not all components of all bearings were examined. Analysis may continue at a later date and the results may then be reported under separate cover.

1. Baseline Bearings

Both TiC hardcoated baseline bearings were available for study. Number 5 was grease lubricated and successfully completed Test Plan A while No. 7 was unlubricated and failed after only 4 minutes of Test Plan A.

Bearing No. 5 is shown in Figure 15. Visible is the dried grease which was found caked along the ends of the rollers. Upon receipt the bearing could be turned by hand but displayed very uneven torque in doing so. Following disassembly, the components were cleaned of the caked Fel-Pro C100 residue. Examined visually, the races were observed to have scuffed regions spaced at approximately the same interval as the rollers were. This scuffing first shown here in Figure 13 seems to be typical of most of the bearings. No other damage was noted to the races, but there was a slight burnish to the roller faces. Rollers from this bearing were used in a corrosion test, the results of which will be presented later in this report.

Bearing No. 7 failed Test Plan A. Failure was caused by metal-to-metal contact at the roller face - inner race shoulder interface and shaving of the roller edge against the edge of the race shoulder. Figure 16, showing the used bearing prior to disassembly, clearly reveals the resultant roller edge damage. This mode of failure was found to be prevalent in all of the failed bearings. Unfortunately, bearings No. 12 and 13 were not available for analysis within the time constraints of this contract.

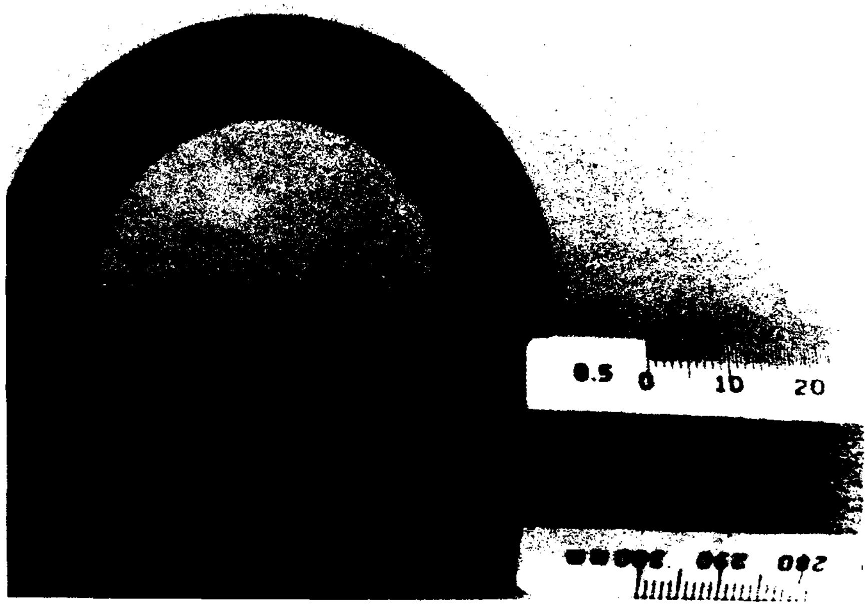


Figure 15. Appearance of grease lubricated bearing number 5 following testing showing heavy caking of lubricant.

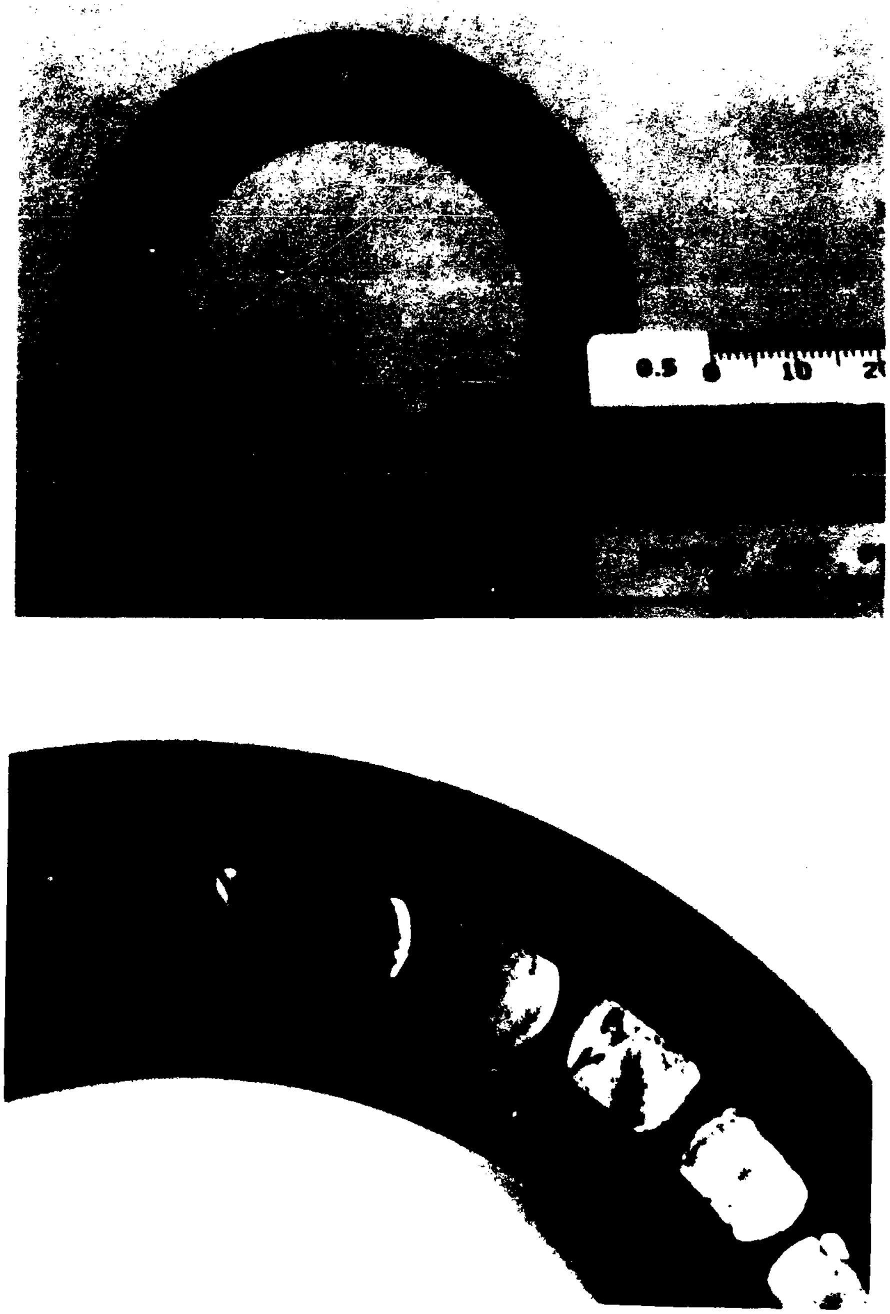


Figure 16. Unlubricated, hardcoated bearing following testing showing severe roller face damage.

2. Stick-Burnished Bearings

Bearing No. 1 was the first bearing to be tested and returned to Hughes. Because it was the only bearing allowed to seize, it was damaged more than any other bearing tested. Figure 17 shows bearing No. 1 at various stages of disassembly. Clearly evident is the damage resulting from the roller face-inner race shoulder sliding contact. The damage is on both ends of the affected rollers; thus it appears that neither test shaft flexing nor bearing misalignment are responsible for the observed damage. Figure 18 clearly shows the large scars resulting from edge galling. Roller damage was confined to approximately 15 rollers which operated in the region of highest load. Clearly, the least effective lubrication procedure failed to adequately protect the roller faces under the high apparent shear loading conditions. The roller face-race shoulder interface is recognized as the region most likely to experience lubrication difficulties and subsequent metal-to-metal contact (References 13 and 14). The shear forces there are high and are increased by increasing the roller skew angle and on thrust loading. Examination of the Patriot bearing reveals a maximum roller skew angle of approximately 2.5°. Thrust loading can originate by test shaft flexing as well as several other sources. It therefore appears to be a typical roller bearing failure mode which affected the stick-burnished bearings. The study of the roller's cylindrical load surface is deferred to bearing No. 3's analysis, which follows.

3. Transfer Film Lubricated Bearings

In accordance with the second "field lubrication" technique, the self-lubricating composite rollers described previously in this report were used. The new rollers are shown in Figure 7. This method was anticipated to yield better results than stick burnishing. Following a total of 8 hours of use in developing the transfer film the rollers developed a smooth burnish along the circumference of the roller face, as shown in Figure 19. The EDX spectra in Figure 19 are representative of the worn and unworn areas of the roller face. What is noteworthy is that the soft lubricating composite has collected Ti/Cr/Fe/Ni debris in the run-in process. The presence of iron indicates that in addition to the loosely adhering TiC particles described previously, some 440C bearing material was removed in this mild run-in procedure.

In order to demonstrate the maximum load carrying capacity of the unidirectional stretched graphite fiber reinforced self-lubricating composite rollers, seven of the used rollers were subjected to axial and radial compressive crush strength testing. The results given in Table 2 show much higher strength in the direction of fiber lay rather than perpendicular to it, as expected. There is now proof that the low strength of the rollers perpendicular to their axes precludes their use as a load bearing, sacrificially lubricating element of the ASALM bearings.

Visual examination of bearing number 3 again revealed that the rollers had scraped against the inner race shoulders just as bearing number 1 had.

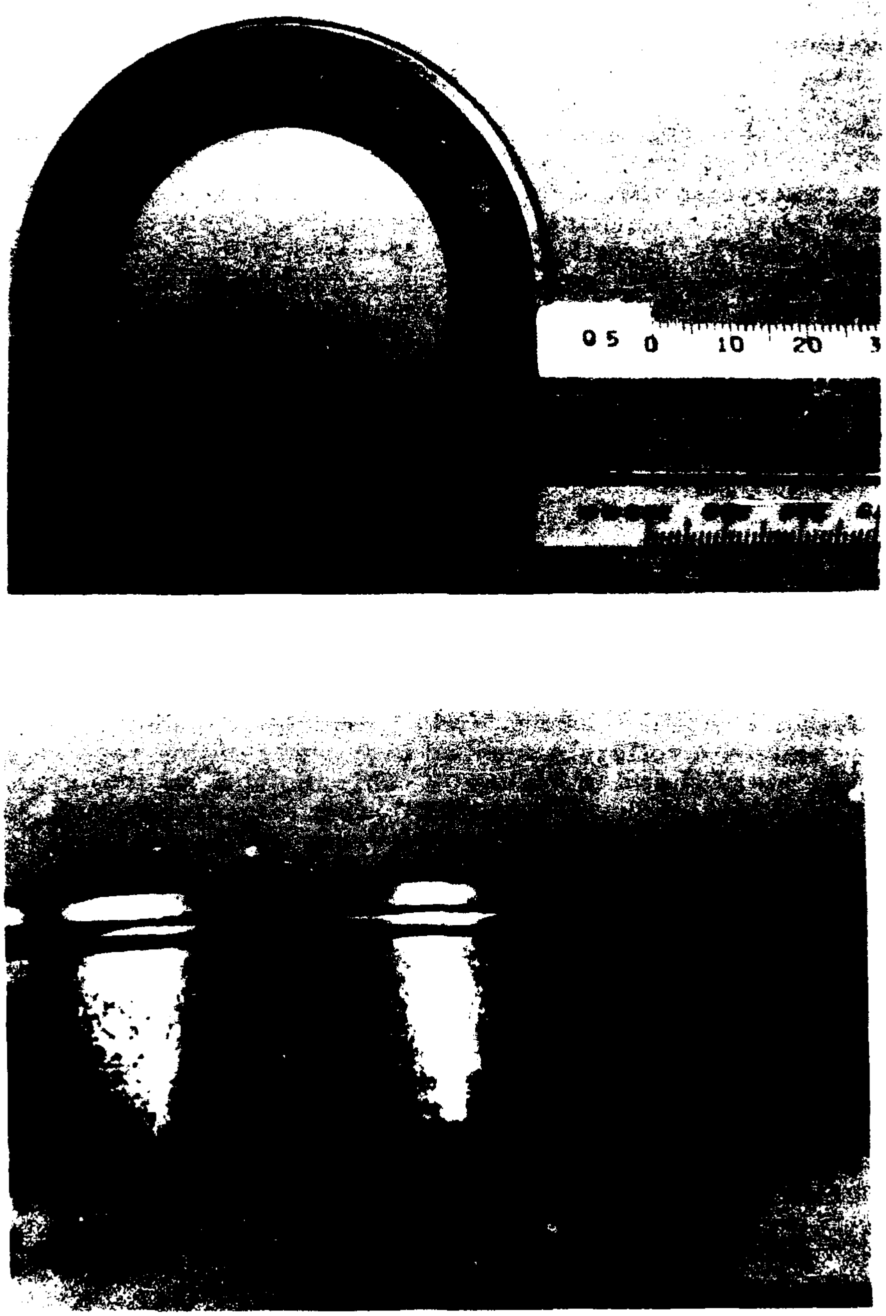


Figure 17. Stick burnished bearing number 1 showing severe roller and inner race damage.

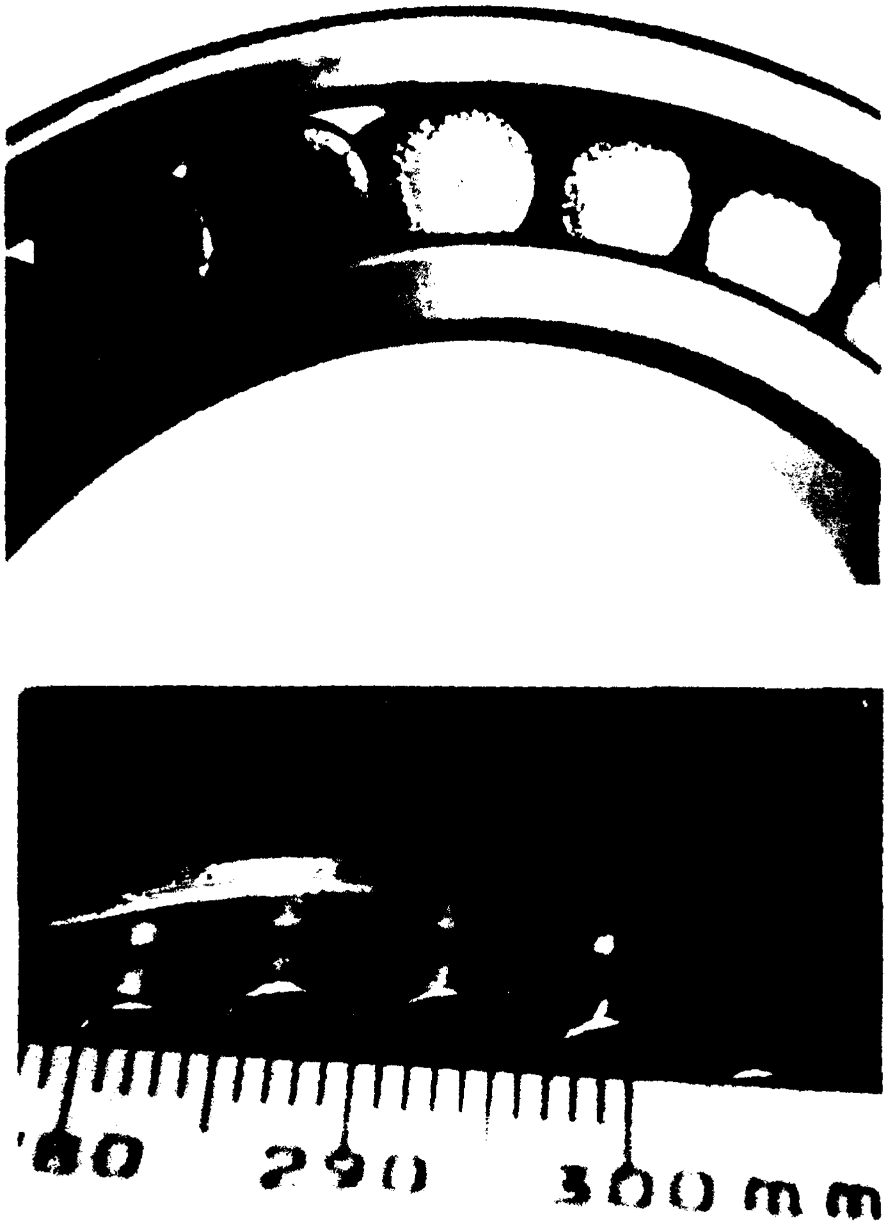


Figure 18. Severe roller damage in bearing number 1 which leads to bearing seizure.

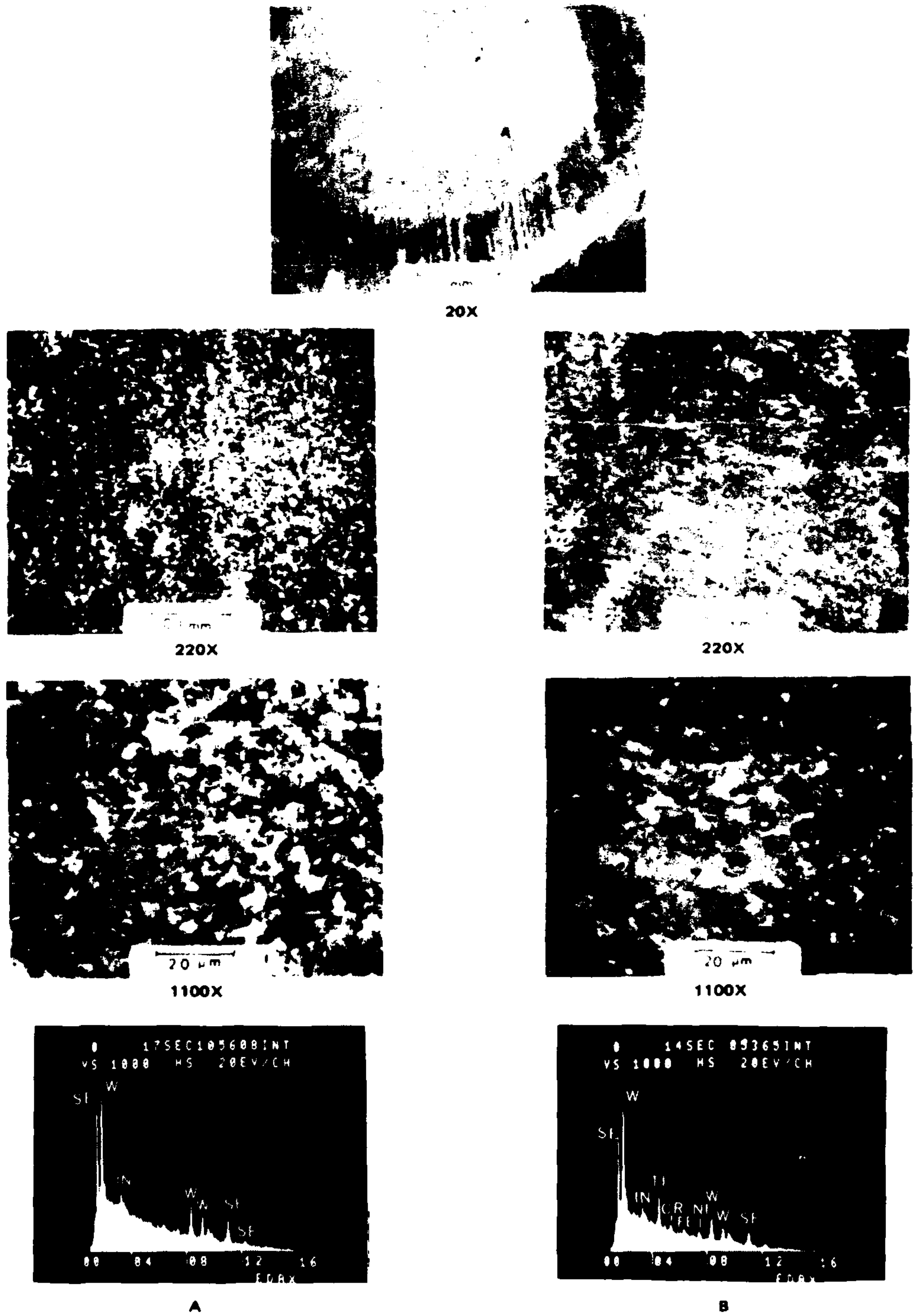


Figure 19. SEM photomicrographs and EDX spectra of (A) new and (B) used areas of carbon fiber reinforced Thermid 600 composite rollers following 8 hours use at 20X, 220X and 1100X magnifications.

TABLE 2.

COMPRESSIVE STRENGTH OF USED SELF-LUBRICATING COMPOSITE ROLLERS AT ROOM TEMPERATURE

Loading Directions	
Parallel To Fiber	Perpendicular to Fiber*
1.9×10^8 Pascals (27.7 KSI)	355 N (80 lbs)
2.6×10^8 Pascals (37.2 KSI)	222 N (50 lbs)
2.7×10^8 Pascals (38.3 KSI)	1067 N (240 lbs)
	1357 N (305 lbs)
*Cylindrial geometry and method of loading do not permit unit load values	

The damage observed in this bearing was less than that in number 1 and the roller faces had large areas of smooth burnish of the lubricant. Apparently, the wear of the self-lubricating composite rollers against the race shoulder had provided a thicker and more effective lubricant film than was applied by stick burnishing alone.

Scanning Electron Microscopy of the steel rollers after testing revealed that in some regions, the Westinghouse compact was almost nonexistent. In these unlubricated areas no loose debris was found and the NiCr-TiC hard-coat was removed. In other areas, the Westinghouse compact had accumulated along with considerable quantities of TiC particles. This difference in appearance is illustrated in Figure 20.

The minute size of the TiC debris seen here, approximately 1.5×10^{-6} m (6.0×10^{-5} in), is typical of the debris observed in other bearings tested. It is generally accepted by ceramics experts that 1×10^{-6} m is the smallest diameter to which ceramics can be reduced using such techniques as ball milling.

In fact, the roller bearings used here closely approximate the action of a ball mill in crushing any TiC coating that fails to remain adherent to the bearing surfaces. The observed flash coating, of TiC particles were as large as 3.6×10^{-6} meters (1.4×10^{-4} in) in diameter particles. Thus they too would be reduced in size by the milling action. The higher initial torque of the TiC coated greased bearing (No. 5) versus the bare bearing (No. 12) can thus be understood in terms of grinding down the initial flash coating to the "equilibrium" size.

As mentioned earlier, the measured clearance of the fully hardcoated bearing is 6.16×10^{-5} m (2.4×10^{-3} in), much more than was needed to tolerate the 1.0×10^{-6} m TiC debris. Thus even if all of the coating were to be removed, the debris would not by itself be able to jam the bearing.

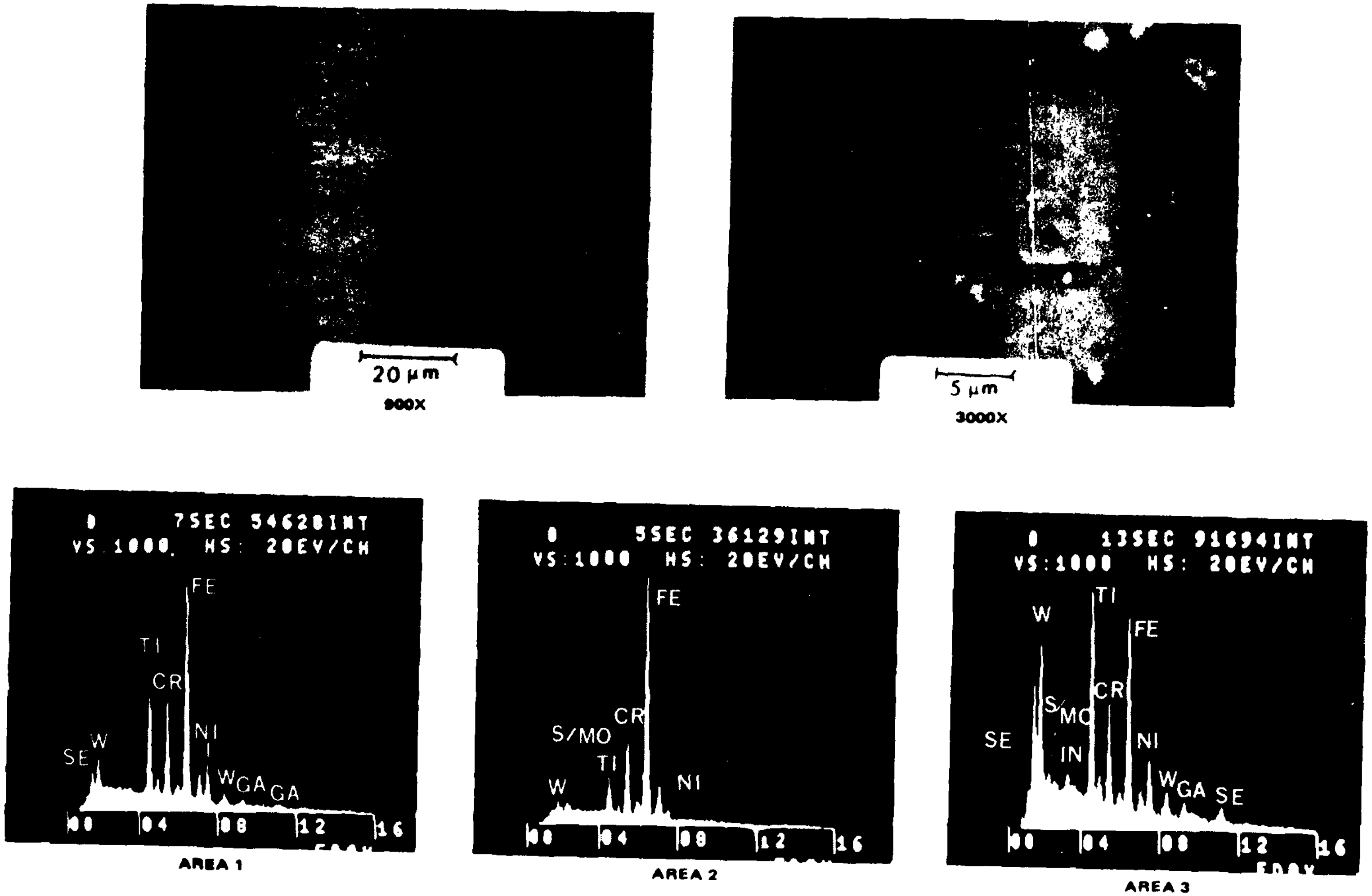


Figure 20. SEM photomicrographs and EDX spectra of worn areas of bearing number 3 showing local variations in lubricant/TiC debris concentration at 900X and 3000X magnifications.

The only bearing to complete its Test Plan successfully, No. 8, was not available for study within the time frame allowed by the contract.

4. MoS_2/Sb_2O_3 Lubricated Bearings

The sputtering of a tightly adhering, dense MoS_2/Sb_2O_3 lubricating soft-coat over the TiC hardcoat was originally anticipated to provide the best corrosion-preventive, lubricating system for the Patriot bearings. In view of the test results, it appears that this original conjecture was valid.

Figure 21 shows the face of a roller removed from bearing No. 11. Visible is the typical burnish along the circumference. EDX analysis could detect no measurable change in material composition in this burnished region when compared with the unworn central portion of the roller face, thus indicating that the TiC hardcoat was not removed in this high load region.

Although in view of the corrosion test results it is likely that the NiCr-TiC hardcoat is severely fractured in this region, with any spalled and fragmented coating being held in place by the lubricant. This "macadam road" like consistency has been seen in other solid lubricated systems where debris becomes entrained in the lubricant.

The sputtered MoS_2/Sb_2O_3 appears to have protected this area of the bearing sufficiently well to avoid scraping the roller faces and thereby prevented bearing failure. SEM and EDX examination of the inner race revealed that despite the presence of chatter-like marks, as shown in Figure 22, the TiC is still present in approximately equal amounts in both areas and is in fact present to nearly the same extent as in the new surfaces. There were regions with some TiC removed, but they were less extensive and of less severity than that seen in any other bearing examined. In general the MoS_2/Sb_2O_3 lubricated bearings demonstrated a greater ability to retain the NiCr-TiC hardcoat than did any other bearings.

5. Corrosion Testing

Bearing Nos. 5, 6 and 10 were used to test the effectiveness of the corrosion resistant hardcoat. Following dynamic testing, rollers from bearings 5 and 10 were stripped of lubricant, as were rollers from the untested bearing, No. 6. The rollers were exposed to 100 percent relative humidity at 100°C (212°F) for 24 hours. After optical examination, they were exposed for a second 24-hour interval and re-examined.

Flowering corrosion was observed on rollers removed from bearings 6 and 10 but not on rollers removed from No. 5 as illustrated in Figure 23. The corrosion was always found along the roller face circumference, the same area as to where the metal-to-metal contact occurs, resulting in roller damage. Evidence of corrosion on bearing 6 is disturbing as that bearing had not been subjected to testing under load; even the degreasing procedure was

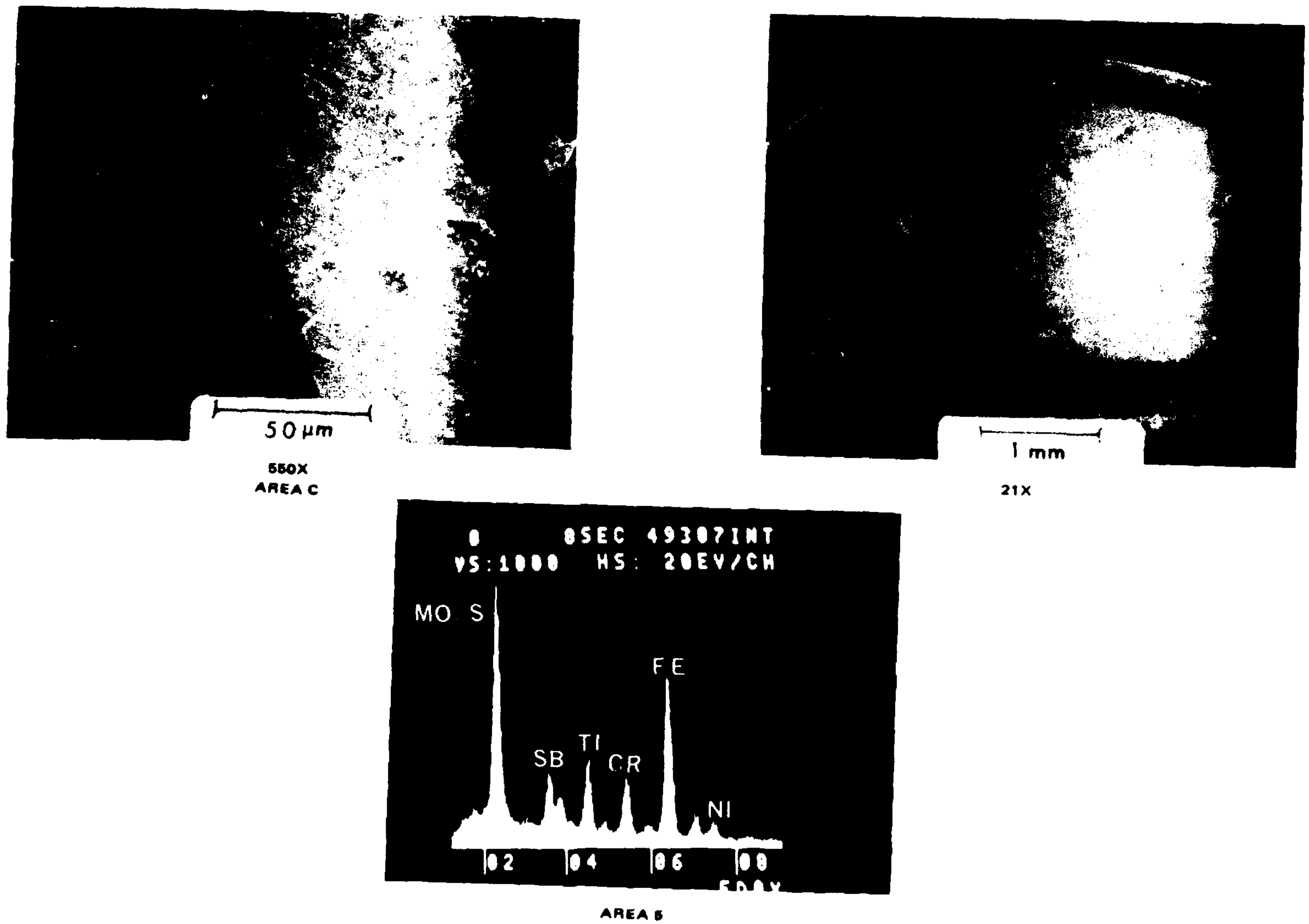


Figure 21. SEM photomicrographs at 21X and 550X magnifications and EDX spectra of a roller from bearing number 11, showing retention of the NiCr–TiC hardcoat.

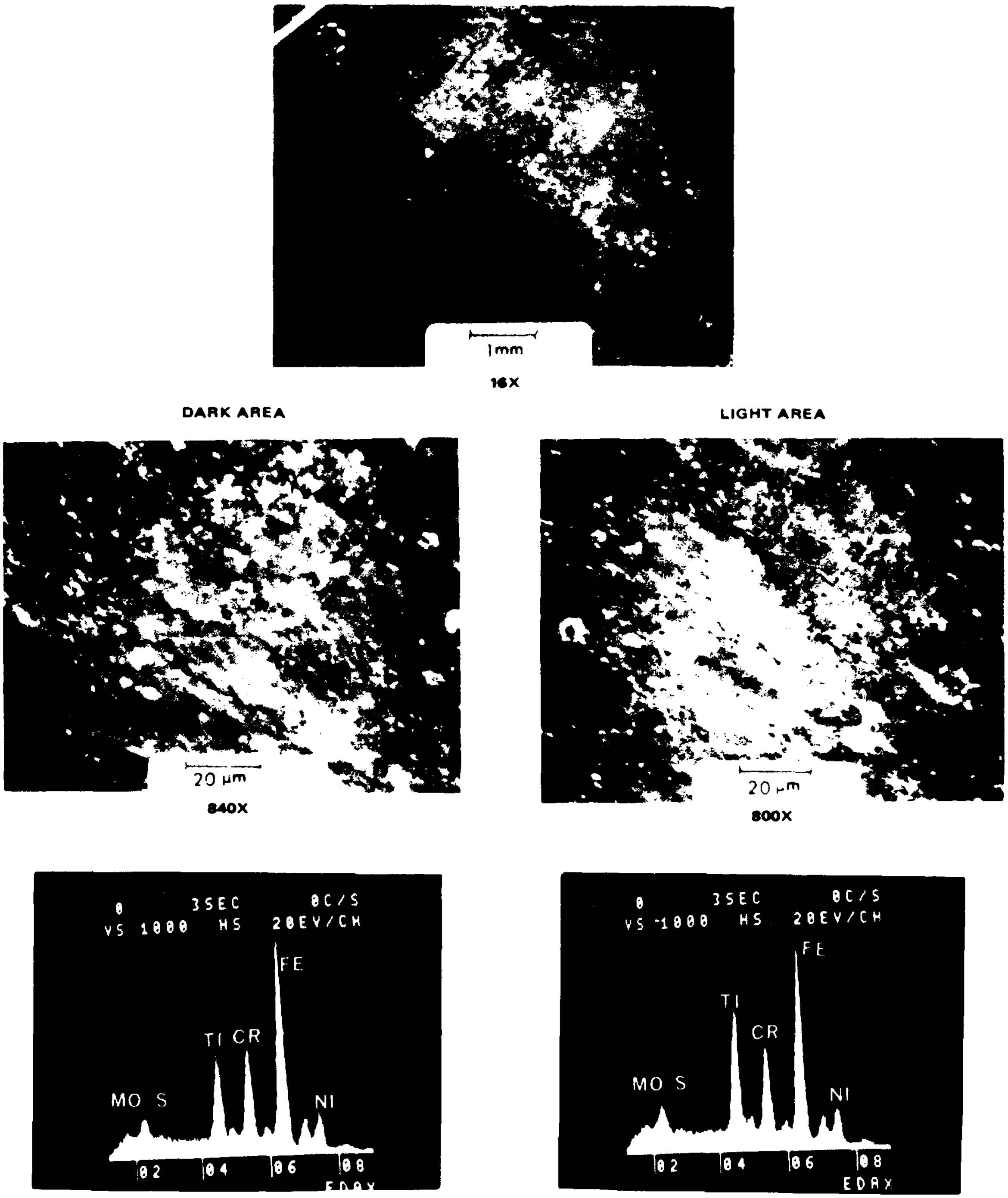
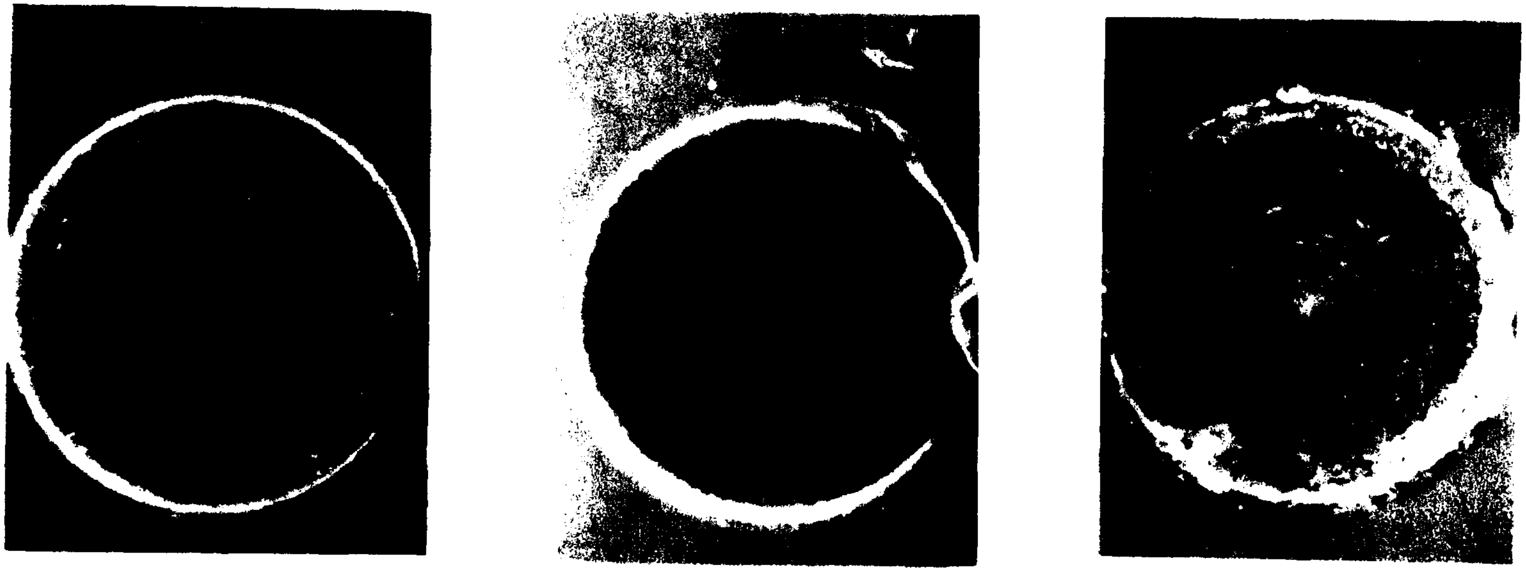


Figure 22. SEM photomicrographs at 16X, 800X and 840X magnifications and EDX spectra of scuffing of bearing 11's inner race with only a slight change in TiC coating coverage.



BEARING 5 | BEARING 6 | BEARING 10

Figure 23. Appearance of used rollers following 24 hour 100% relative humidity testing, 15X magnification.

done with the bearing disassembled. Thus either the NiCr-TiC handcoat is of itself porous and/or fractured prior to testing or it was easily removed by rotating the bearing by hand with no load, in the grease-lubricated condition. In view of the large amounts of NiCr-TiC removed from bearings 8 and 9 during cleaning, the latter case is not unreasonable.

The large amount of oxidation present on the MoS_2/Sb_2O_3 lubricated bearing was unexpected. Apparently the bearing did sustain extensive damage to the roller faces, thereby allowing the corrosion to occur. It should be noted that bearing No. 10 did experience distress during testing as demonstrated by the squealing heard at the 4-minute point during testing. It is likely that a temporary lubricant failure at that time resulted in a cracked or spalled NiCr-TiC corrosion resistant hardcoat and subsequent flowering corrosion during the humidity test.

The lack of corrosion on the baseline (greased) bearing No. 5 is puzzling. Why should a bearing which has undergone the full Test Plan A cycling exhibit less corrosion than an untested bearing, No. 2. A possible explanation is that the Fel Pro C 100 antiseize compound contains additive(s) which may transform into boundary layers with some corrosion protective properties, especially on exposure to elevated temperatures. Furthermore, while the stationary bearing is slowly cooling from the 316°C (600°F) final test temperature, any corrosion preventing compound could act upon the 440 C steel exposed either through cracks in the NiCr-TiC coating or at places where the coating had been lost altogether through spallation.

6. Comparison With Other Tests

Parallel to the Hughes effort to develop a corrosion resistant lubricating softcoat system for the ASALM actuator assembly, the AFML/MBT pursued further ASALM component high temperature stability tests, both in-house (Reference 15) and through McDonnell Douglas (MDAC) as mentioned earlier. MDAC sent to Technology of Materials Inc., Santa Barbara, CA., one set of ASALM actuator bearings of their design. These bearings were reportedly processed to obtain the same NiCr-TiC hardcoat thicknesses as the bearings prepared for Hughes and were then sputter lubricated with MoS_2/Sb_2O_3. The bearings were then tested by Moog Inc., East Aurora, NY. Their final report, titled "Advanced Lubricant Evaluation Test Report for Moog Model 17E319 Servoactuator MDAC PO Y6E262", is attached herein as Appendix H. The Moog tests were much more rigorous in that bearing stresses were varied to simulate anticipated flight conditions. Also, the loads were higher and included bending moments. Moog evaluated three different lubricants: the previously discussed Braycote 3L-38RP grease, the AFML supplied MCG-70852542 (fluorosilicone-type) grease and the sputtered MoS_2/Sb_2O_3 solid lubricant. MDAC issued a summary report on the Moog tests of the ASALM control surface actuator. The report is attached here as Appendix I.

Only the solid lubricated bearing was TiC hardcoated. The Moog test results indicate that neither of the advanced lubricants performed as well as the original Braycote Grease. During the high temperature duty cycle, Braycote 3L-38RP resulted in a bearing torque of 51.98 N-m (400 in-lbs) at the end of 33-1/3 minutes. The Air Force grease resulted in a torque of 87.0 N-m (770 in-lbs) at the end of 20 minutes. The NiCr-TiC - MoS_2/Sb_2O_3 hardcoat softcoat system performed worst with a bearing torque of 187.6 N-m (1660 in-lbs) after only 22 minutes. It should be noted that for the next 11 minutes, the solid lubed bearing continuously experienced a decreasing torque level due to an accidental hydraulic leak which in effect lubricated the outboard bearing. Although Scanning Electron Microscope photographs of the used solid lubricated bearings were made available to Hughes, detailed failure analysis was not possible and no conclusions are offered. The nebulous nature of the high magnification photomicrography and the lack of opportunity to inspect the actual bearings preclude our ability to conduct a fair failure analysis.

Comparison of the Moog test results to those obtained by Hughes in this report is difficult because of the large differences in bearing design, test stand fixturing, loading history and applied temperatures. It is anticipated that the Braycote 3L-38 lubricated Moog bearings would have exhibited higher torques at -54°C (-65°F) but not as much as was encountered using Fel Pro-C100 as was done by Hughes. It is curious that after the high temperature Moog Test using the solid lubricant the bearing showed relatively low torque despite the high torques experienced during the test. This phenomena was also noted with the bearings tested at Martin Marietta when bearing number 1 seized in test but was free turning upon dissassembly from the test stand.

SECTION V

CONCLUSIONS

A 14-month engineering study was concluded with the testing and evaluation of 440C alloy steel roller bearings which were coated with a ceramic, corrosion resistant hardcoat and lubricated with one of a variety of solid lubricant systems.

The primary objective of this study was to provide corrosion resistance for a storage period of not less than 10 years for the Advanced Strategic Air-Launched Missile (ASALM) hydraulic control actuator bearings. The urgent need for extreme environment lubrication techniques necessitated the use of currently available state-of-the-art coating technology. Therefore, a base layer of nichrome 1000Å thick (4.0×10^{-6} in.) and a final hardcoat of titanium carbide 1250Å thick (5.0×10^{-6} in.) was DC magnetron sputtered in a downward direction onto the bearing components. The two materials were selected on the basis of corrosion tests results. Those tests used a variety of aggressive media to attack 52100 alloy steel which had been ceramic coated. Results of those tests were encouraging in that the coating markedly reduced corrosion, although pinholes in the titanium carbide coating did force the addition of a nichrome underlay to avoid direct exposure of the final 440C alloy components to the corrosive environment.

The secondary task of this study was to provide the hardcoated bearing with solid lubrication that could meet ASALM missile requirements. This goal was approached using three solid lubricant systems;

- Transfer film formation via burnishing with Westinghouse compact,
- Transfer film formation via burnishing with Westinghouse compact and run-in using polymeric, self-lubricating composite rollers. The rollers consisted of a Thermid 600 polyimide base with carbon/graphite fiber reinforcement to which the powdered Westinghouse compact is added,
- Magnetron sputtering of MoS_2/Sb_2O_3 solid lubricant film. The lubricated bearings were the dynamically tested under conditions approximating those anticipated during the ASALM mission:
  - Flight duration = 1000 seconds,
  - Aircraft Static Load = 31,581 N (7100 lbs),
  - Temperature Profile = -54°C to 316°C (-65°F to 600°F).

The data indicate a considerable variation of bearing performance level with lubrication mode. The more significant discoveries of this study and ideas for further work stemming from these findings are summarized below:

- Titanium carbide was found to be the most corrosion resistant ceramic in tests which by the use of aggressive chemical media it was hoped to lead to "overdesign". This approach was necessary in the absence of any theoretical models whereby short term accelerated tests can be correlated to long term storage.

- DC magnetron sputtering of ceramic hardcoats currently must be augmented by the use of a second coating of a nichrome underlay. This initial layer of nichrome serves two purposes. First, it reduces the likelihood that pinholes or cracks in the ceramic extend to the base metal. Second, it is anticipated that it will be shown to be an adhesion aid for the ceramic hardcoat. The result of the first effect is to further reduce corrosion and the second effect is to help maintain coating integrity during bearing operation.

- The sputtered nichrome/titanium carbide coating system demonstrated very little adhesive strength to the base 440C steel. Adhesion of the brittle coatings was so low that unloaded rotation of unlubricated bearings by hand resulted in significant spallation of the coatings.

- The sputtered nichrome-titanium carbide corrosion resistant hardcoat is reported by TMI to have substantial corrosion resistance when tested on unused rollers. Exercising the assembled bearing, however, results in such significant spallation of the nichrome-titanium carbide coating that corrosion resistance cannot be assured for ten years. In fact, one hand rotated bearing evidenced significant flowering corrosion after only a single 24 hour 100 percent relative humidity test.

- The ability of the nichrome-titanium carbide hardcoats to resist spallation and or cracking is improved in varying degrees by the lubricant system employed. The least effective lubrication mode was to burnish the races with a special compact lipstick comprised of Westinghouse compact, while the rollers were tumble burnished in a powder of the same compact. The second most effective lubrication mode was from a transfer film by first burnishing as in the first lubrication mode followed by running in the bearing using alternating steel and self lubricating composite rollers. These rollers comprised of a Thermid 600 polyimide reinforced with uniazially stretched carbon/graphite fibers and with a Westinghouse compact lubricating pigment. The most effective lubricant was a sputtered 3000Å (1.2×10^{-5} in.) thick film of MoS_2/Sb_2O_3. These three lubrication methods representing two "field application" methods and one laboratory method performed as anticipated.

Apparently the ability of lubricant to remain on the roller face in the presence of high shearing (sliding) loads which tend to displace the lubricant out of the contact zone is the primary factor affecting bearing performance in the study. This also indicated that certain hardcoat/softcoat systems that are effective under sliding condition at low unit load may not necessarily be equally effective under rolling conditions, at higher Hertzian stresses.

- Of the bearings evaluated in this study which failed, all showed a common failure mechanism. Rollers in the high load region of the bearing experienced high shear stresses and in some cases suffered shearing of metal at the circumference of the roller face. The friction developed between the roller face and the inner race shoulder because of lubricant failure.

- Despite the failure of the corrosion resistant hardcoats to remain fully adherent, the spalled debris was crushed to such a fine particle size that it was unable to jam the bearing and prevent rotation in view of the generous clearances of the test bearings. The small particles of coating became mixed and entrained in the lubricant film, forming a "macadam road", composite solid lubricant film.

- All the solid lubricant systems employed in this study resulted in significantly lower bearing torques at the -54°C (-65°F) test start than the base line grease. The sputtered MoS_2/Sb_2O_3 was the best, resulting in very smooth operation at all temperatures.

Advanced techniques to apply ceramic surfaces to bearing materials are being developed as part of the ongoing DARPA/AFML/Hughes Solid Lubricated Rolling Element Bearings Program. Preliminary results of that program indicate that physical vapor deposition by ion sputtering will not provide the hardcoat adhesion needed, and reactive sputtering or chemical vapor deposition are necessary to form ceramic metal interfacial diffusion layers to obtain the coating adhesion currently unattainable.

It should be remembered that this study was an attempt to utilize available state-of-the-art techniques to solve a difficult and urgent problem. Finally, the techniques applied in this program have been evaluated using only one type of rolling element bearing with the results described above.

REFERENCES

1. Gardos, M.N., "Solid Lubricated Rolling Element Bearings", Semiannual Status Report No. 1, DARPA Order No. 3576, USAF Contract No. F33615-78-C-5196, Hughes Aircraft Company Report No. FR-79-76-595, 15 February 1979.

2. Personal communications with Mr. Vern King, MDAC (also see Paragraph IIIA1 of this report).

3. Johnston, J.G., O'Leary, M. and Gardner, A.M., "High Temperature Bearing/Lubricant Test Report," Martin Marietta Aerospace Report OR 12,858, December 1973.

4. Gardos, M.N. and Castillo, A.A., "Self-Lubricating Composite Bearing Materials — Part I: Development of a High Load, High Temperature Self-Lubricating Composite," Final Report, AFML-TR-79-4091, Part I, August 1979.

5. Riegert, R.P., "Sputtered Coatings for Gas Bearing Applications," Final Report AFML-TR-78-192, Technology of Materials, Incorporated, Santa Barbara, California, December 1978.

6. Shaffer, P.T.B., "Handbook of High Temperature Materials, No. 1; Materials Index," Plenum Press, NY, 1964, pp 121-122.

7. Martin Marietta cover letter (Doc. No. 3236-79-0069) by Angiulli, F.J. to Bohner, J.J. of Hughes Aircraft Company (and others), with Martin Marietta internal memorandum attached, dealing with the Bearing/Lube Test task thermal analysis; letter dated 22 March 1979.

8. Martin Marietta Drawing No. 10251122, "Bearing, Roller, Cylindrical — Control Actuation System", 12 May 1969.

9. Jones, J.R. and Gardos, M.N., "Transfer Film Formation by Lubricative Composites," Proc. First Int. Conf. on Solid Lubrication, 24-27 August, 1971, Denver, Colorado, ASLE SP-3, pp 185-197.

10. Hughes Process Specification (HP) 7-29,

11. Hughes Process Specification (HP) 7-24, "

12. Harris, Tedric A., "Rolling Bearing Analysis," John Wiley and Sons Inc., New York, 1966.

13. Morrison, F.R., Pirvics, J., and Crecelius, W.J., "A Functional Evaluation of a Thrust Carrying Cylindrical Roller Bearing Design," Journal of Lubrication Technology, Vol. 101, No. 2, pp 164-170, April 1979.

14. Zanker, Adam, "Wear Life of Roller Bearings Can Be Estimated Quickly", Product Engineering, February, 1978.

15. Snyder, C. E. Jr., Tamborski, C., Gopal, H., and Svisco, C.A., "Synthesis and Development of Improved High-Temperature Additives for Polyperfluoroalkylether Lubricants and Hydraulic Fluid", Journal American Society of Lubrication Engineers, Vol. 35, No. 8, August 1979.

APPENDIX A

INTERIM REPORT ON THE STUDY OF THE ESTABLISHMENT OF CORROSION-RESISTANT TRIBOLOGICAL COATINGS

September 29, 1978

Hughes Aircraft Co.
Attn: G. S. Pomatto
Bldg. 17 M/S J129
Centinela & Teale Sts.
Culver City, CA 90230

Re: P. O. 04-492-902-FAC
Solid Lubrication of ASALM Missile Actuator Bearings

Subject: Interim Report on the Study of the Establishment of Corrosion-Resistant Tribological Coatings

Gentlemen:

An initial evaluation of corrosion resistance of a variety of selected tribological coatings involved the coating of one hundred and forty (140) 52-100 alloy bearing rollers supplied by Hughes Aircraft with materials known to have reasonable adherence to 52-100 and good-to-intermediate tribological characteristics. Since the overriding factor of this study is to develop chemical inertness as a primary objective and wear resistance as a secondary objective, some materials of known good chemical resistance but of unknown tribological value were included.

The materials initially studied were SiC, TiC, Fe_3O_4, HfC, TiB_2, B_4C, and a proprietary mixture of $TaSi_2$, TaC and Al_2O_3. All materials were sputtered in a downward direction from a 3-inch Sloan sputtergun. Monitoring was by quartz crystal monitor and the initial work was completed using estimated frequency shifts on the crystal monitor as a guide to thickness. Thickness was subsequently confirmed on witness plates run simultaneously with the rollers. Thickness, in all cases, was between 3 and 5 microinches.

Initial chemical evaluation was made by 24-hour exposure of multiple samples of each of the seven coatings in dilute HCl, dilute NaOH, boiling in concentrated NaCl, boiling in water, and exposure to steam. From these initial studies TiB_2 coatings were found to be seriously attacked by both NaCl and HCl and all samples were attached by exposure to HCl.

A second series of samples was prepared at 10-microinch coating thickness of test samples using TiC, B_4C, HfC, Fe_3O_4, SiC and the complex oxide-silicide. In addition to the as-coated test, a second series of the same compositions were heat-treated for two hours in air at 400°C so as to appraise the effects on properties.

Hughes Aircraft Co.
Culver City, CA

September 29, 1978

The results of this second series of tests are summarized in Table 1 in which the test surfaces are classified from best to poorest in resistance to corrosion by a designation of merit by numbers 1 through 6. These comparisons were made for each of four chemical conditions on both heat-treated and as-deposited coatings.

TABLE 1

MERIT OF CORROSION RESISTANCE OF
SIX COATING MATERIALS
OF 10 MICROINCH THICKNESS

Corrosion Reagent	B_4C (HT)	TiC (HT)	HfC (HT)	TASO (HT)	SiC (HT)	Fe_3O_4 (HT)
HCl	1	2	3	4	5	6
NaCl	4	3	5	1	2	6
NaOH	6	3	4	1	2	5
100% Humidity	2	1	6	3	4	5
Fig. of Merit:	13	9	18	9	13	22

(HT signifies "heat-treated")

Corrosion Reagent	B_4C	TiC	HfC	TASO	SiC	Fe_3O_4
HCl	4	3	2	1	5	6
NaCl	4	3	5	2	1	6
NaOH	6	2	3	1	4	5
100% Humidity	3	1	2	4	5	6
Fig. of Merit:	17	9	12	8	15	23

A figure of merit was established by taking the simple sum of classification numbers for each coating under the various corrosion conditions. As can be seem in Figure 1, TiC and TASO were the most effective in corrosion resistance by reference to this figure of merit. In addition, these two were either first or second in corrosion resistance in more than half of the test conditions. In light of this these two materials were selected for further study. All others will

Hughes Aircraft Co.
Culver City, CA

September 29, 1978

be dropped from active consideration unless some deficiency is encountered in future testing. It might be noted that TiC is the material most commonly used in our patented Romelus Process for tribological coatings. Since this material has been widely studied in the past, it will be the preferred composition of the future unless the tantalum, aluminum, oxide-silicide (TASO) shows superior oxidation resistance.

In addition to the above initial evaluation study, a bearing tester was constructed for the testing of full roller bearings at temperatures up to 600°F under loads up to 500 pounds with cyclical rotation through an arc of plus or minus 10°. No work beyond the initial construction was completed as no test bearings have yet been supplied by Hughes Aircraft for simulated testing of this equipment.

<u>Work to be Accomplished During Next Report Period</u>
Work to be accomplished during the coming report period includes the balance of chemical resistance and thickness of the two hard coatings selected for further studies, initial studies on chemical resistance of lubrication coatings and initial testing of tribological characteristics of most promising chemically resistant hard coating.

Respectfully submitted,

TECHNOLOGY OF MATERIALS

Richard P. Riegert

Richard P. Riegert

RPR/aj

APPENDIX B

INTERIM REPORT ON THE STUDY OF THE ESTABLISHMENT OF CORROSION-RESISTANT TRIBOLOGICAL COATINGS

December 1, 1978

Hughes Aircraft Co.
Attn: G. S. Pomatto
Bldg. 17 M/S J129
Centinela & Teale Sts.
Culver City, CA 90230

Re: P.O. 04-492-902-FAC
Solid Lubrication of ASALM Missile Actuator Bearings

Subject: Interim Report on the Study of the Establishment of Corrosion-Resistant Tribological Coatings

Gentlemen:

This is a bi-monthly report on the testing of ASALM missile-actuated bearings of 52-100 alloy composition in preparation for ultimate application to 440-C type needle bearings. Since all detailed testing was at various interim stages at the beginning of November, no interim report was submitted at that time. Extensive sample preparation and the conclusions of corrosion testing was underway at that time. No definitive results could be reported other than that work was underway.

Corrosion work on lubrication films was completed during the period. 120 samples of two different lubricant compositions (boron nitride and MoS_2/Sb_2O_3) were deposited in three thickness ranges (3, 5, and 10 microinches) and tested under four environmental conditions. The environments included: aging in HCl solution, sodium hydroxide solution, sodium chloride solution, and exposure to 100% humidity conditions. All boron nitride compositions failed catastrophically on exposure to 100% humidity and as such were eliminated as prospective candidates for lubrication of ASALM bearings. The MoS_2/Sb_2O_3 compositions were susceptible to attack by HCl and gave reasonable resistance to the other test conditions. The chemical resistance was greater with the thicker films. As such, MoS_2/Sb_2O_3 has been selected as the only candidate for thin film lubrication of hard coated bearings, even though it is anticipated that its oxidation resistance may be inadequate for the application. Only subsequent testing can identify whether the oxidation resistance is adequate under the conditions prescribed under this contract.

Substrate	Hard Coat	Lube Coat	Torque Range	Noise Level	Load-to-Stall	Appearance After Test
52-100	5 m/in TiC	0	.2 -.4	Inter.	120 gms.	Poor
52-100	10 m/in TiC	0	.2 -.4	Noisy	170 gms.	Poor
52-100	5 m/in TASO	0	.2 -.4	Inter.	90 gms.	Poor
52-100	10 m/in TASO	0	.2 -.5	Very noisy	220 gms.	Poor
440-C	0	0	.3 -.65	Very noisy	crash	Excellent
440-C	5 m/in TiC	0	.16 -.3	X-lo noise	" @ 330 rpm	Excellent
440-C	10 m/in TiC	0	.16 -.6	Noisy	100 gms.	Excellent
52-100	5 m/in TiC	$MoS_2Sb_2O_3$	.14 -.16	V. low	No stall @ 500 gms	Fair
52-100	5 m/in TiC	$MoS_2Sb_2O_3$	.12 -.15	V. low	No stall @ 500 gms.	Fair
52-100	5 m/in TASO	$MoS_2Sb_2O_3$	.12 -.14	V. low	No stall @ 500 gms.	Fair
52-100	5 m/in TASO	$MoS_2Sb_2O_3$	.08 -.10	V. low	No stall @ 500 gms.	Fair
52-100	0	$MoS_2Sb_2O_3$	.13 -.16	V. low	No stall @ 500 gms.	Fair
52-100	0	0	NA	V.noisy	Crashed at start	
52-100	0	0	.2 -.8	V.noisy	Crashed at 150 rpm	
440-C	0	0	.3 -.65	V.noisy	Ran irregularly.	

Hughes Aircraft Co.
Culver City, CA

December 1, 1978

Spin Wear Testing

36 samples of spin wear test flats were coated with various thicknesses with previously qualified hard coatings of TiC and tantalum silicide/tantalu, oxide/aluminum oxide composition (TASO), in five micro- and 10 microinch thicknesses both with and without lubrication coats of MoS_2/Sb_2O_3. A summary of wear test data is shown in Figure 1.

The above test involves the rotation of a 1-inch torroidal-shaped test flat against a static test flat in face-to-face contact under 50 gm. load. The test procedure involves rotation at 100 rpm for five minutes, 330 rpm, for five minutes, and 1000 rpm for five minutes, under this minimum load. Subsequent to the speed test, a progressively increasing load is applied to the static test flat until failure of the test or until the maximum condition of the instrument which is 500 gm.-load is reached. The efficacy of the wear surface is related to the load supported without substantial increase in torque. Torque measurements are continuously recorded along with the recording of six segments of the acoustic spectrum. As can be seen in Figure 1, the quality of 440-C material is uniformly superior to tests run on coated 52-100. Secondly, all lubricated surfaces were substantially superior in wear resistance to hard coated and uncoated material. Finally, it is apparent that TiC and TASO have similar wear resistance, although TASO is somewhat more prone to be noisy in the test surface. 10 microinch films are substantially noisier than five microinch films on 52-100 material, although lubricated hard coats appear to have similar ranges of noise level regardless of hard coating thickness.

52-100 Bearing Coating

8 needle bearing sets were coated with 5 microinch and 10 microinch coating of TiC and TASO. Half of such coated bearing sets were overcoated with 3 microinches of MoS_2/Sb_2O_3. All test sets have been completed and initial quantities have been assembled for testing.

High Temperature 500 Pound Load Bearing Tester

The test assembly described in the previous letter report has been brought to completion and operated with oil lubricated retainer-separated needle bearing sets of 52-100 material supplied by Hughes Aircraft Company. Temperature calibration is presently underway and tests of the completed bearing sets will begin shortly and final test data should

Hughes Aircraft Co.
Culver City, CA

December 1, 1978

be available before the end of the next report period.

Conclusions

A summary of conclusions from the above and previous work are as follows:

1. TiC and TASO are roughly equivalent in spin wear test wear characteristics.
2. 10 microinch hard coatings are more noise susceptible than are 5 microinch hard coatings.
3. 10 microinch hard coatings are somewhat superior in chemical resistance than are 5 microinch hard coatings.
4. Boron nitride is hydroscopic and inappropriate as a lubricant for ASALM bearings.
5. MoS_2/Sb_2O_3 is an excellent lubricant for ASALM bearings and has intermediate resistance to severe chemical conditions. Its oxidation resistance, under conditions prescribed in the specifications, is at present unknown and will be deduced in final 52-100 bearing testing.
6. Lubricated hard coats give substantially superior wear resistance on disk testing than hard coats alone. In addition, noise levels in lubricated films are substantially lower than unlubricated films.

Work To Be Accomplished Next Report Period

The only work to be completed on this contract is the high temperature high load testing of 52-100 bearings already coated and assembled, to reconfirm efficacy and reproducibility of these results and prepare final coatings for 440-C ASALM needle bearings for delivery at the end of this contract.

Respectfully submitted,

TECHNOLOGY OF MATERIALS

Richard Riegert

Richard P. Riegert

RPR/mr

APPENDIX C

SPECIAL INTERIM REPORT: DEFICIENCY OF 52-100 ASALM BEARINGS

December 18, 1978

Hughes Aircraft Co.
Attn: G. S. Pomatto
Bldg. 17 M/S J129
Centinela & Teale Sts.
Culver City, CA 90230

Re: P.O. 04-492-902-FAC
Solid Lubrication of ASALM Missile Actuator Bearings

Subject: Special Interim Report: Deficiency of 52-100 ASALM Bearings

Gentlemen:

Testing of coated ASALM bearings was delayed for one week awaiting a torque measuring device promised to be delivered in late November. The device has as yet not been received but testing was begun.

Testing of TiC coated 52-100 needle bearings arranged in full complement array was initiated in the second week of December. Room temperature operation of the bearings under load, however, indicated that a "lock-up" condition of the rollers periodically occurred. When this condition occurred, extreme forces were generated between the needles and the races. The cause of the condition is off-axis orientation of the needles with respect to the races. In this event, the needles bridged across the race radius.

First, assuming that the condition was the result of TiC coating, attempts were made to operate MoS_2 lubricated bearings, since very low friction coefficients were measured on such surfaces. Lock-up was again displayed.

A full complement array of needles was installed in an oil lubricated and uncoated bearing set and the same effect was displayed. Apparently, the 52-100 bearings cannot be operated without conventional retainer sets.

It is not known if the same condition exists for 440-C bearings, but unlike retainer separated bearing sets, no full complement set of 52-100 bearings is usable without lock-up.

As of this moment, no further testing can proceed without a resolution of the above problem.

Hughes Aircraft Co.
Culver City, CA

December 18, 1978

Technology of Materials requests immediate consideration of this matter by the technical staff of Hughes Aircraft Company.

Very truly yours,

TECHNOLOGY OF MATERIALS

Richard P. Riegert

Richard P. Riegert

RPR/mr

APPENDIX D

ASALM TD&I MEETING MINUTES

ASALM TD&I MEETING MINUTES

AF8-015

Martin Marietta Corporation
Orlando Division

SUBJECT: ASALM Actuator Bearing Lubrication and Corrosion Protection

LOCATION: Martin Marietta Corporation, Orlando, Florida

DATE: 25 May 1978

ATTENDEES: Bill Harney, Hughes Aircraft
Mike Gardos, Hughes Aircraft
Andrew Garner, MMC
William Harkness, MMC
James Johnston, MMC
Alan Mortensen, MMC
George Myers, MMC
Michael O'Leary, MMC

BACKGROUND:

Mr. Gardos submitted a proposal to the Air Force Materials Laboratory which describes a program to develop a corrosion resistant, lubricating coating for ASALM actuator bearings[1]. The core of this work is evaluating the performance of TiC (titanium carbide) sputtered coatings for its capability to maintain a "pinhole free" coverage after exposure to typical bearing environments. Additionally, the suitability of three solid film lubricant techniques will be evaluated.

This meeting was arranged with the assistance of Mr. McConnell of AFML to help Mr. Gardos determine what the actual ASALM environments and loads are expected to be, and to solicit recommendations for the test item and the test procedure.

ACTION ITEMS:

1. Prepare a data and hardware package for Mr. Gardos which will contain the following items:

   a) One copy of OR 12,858, "Bearing Test Report."

   b) PTV rack and pinion actuator cutaway views.

   c) PTV rack and pinion actuator shaft and bearing housing details.

   d) Baseline requirements for FSD bearings.

e) Four new or serviceable Patriot roller bearings (P/N 10251122).

f) Recommendations to be considered for inclusion in Hughes Aircraft Company's plan for testing the coated bearings.

Action: a) M. O'Leary
b) J. Johnston
c) J. Johnston
d) W. Harkness
e) W. Harkness
f) W. Harkness and M. O'Leary

Package Assembly: M. O'Leary, 6/16/78

DECISIONS AND/OR AGREEMENTS REACHED:

1. Martin Marietta will furnish approximately four Patriot roller bearings for use by Hughes Aircraft in their corrosion protection development work.

2. Martin Marietta will compile data and recommendations for Hughes Aircraft's use and consideration in developing test plans and procedures.

SUMMARY OF DISCUSSIONS:

1. Mr. A. Garner presented an overview of the ASALM program. Environments and loads of the actuator bearings were outlined.

2. Mr. M. Gardos discussed the background for the work to be accomplished for AFML and described the sequence and anticipated results of the contract effort (Reference 1).

3. A group discussion of ASALM peculiar requirements closed the meeting.

Michael D. O'Leary

References:

1. "Technical Proposal for Additional Effort on Self-Lubricating Composite Bearing Materials," Hughes Aircraft Corporation, Ref. No. D-6532-002, 6 April 1978.

2. "High Temperature Bearing/Lubricant Test.Report," J. G. Johnston and M. D. O'Leary, Martin Marietta Corp., OR 12,858, December 1973.

APPENDIX E

MARTIN MARIETTA ASALM BEARING DATA

MARTIN MARIETTA AEROSPACE

ORLANDO DIVISION
POST OFFICE BOX 5837
ORLANDO, FLORIDA 32805

Document No.: 3236-78-0235
Mail Point: 491

Date: 21 June 1978

Hughes Aircraft Company
Michael N. Gardos
Technology Group Leader, Technology Support Division
Culver City, California 90230

Subject: Martin Marietta ASALM Bearing Data

Reference: (a) Meeting, 25 May 1978, Martin Marietta Corporation, Orlando, Florida

Dear Sir:

The material enclosed is in response to agreements reached at the above referenced meeting. Note that the proprietary classification of the test report has been removed.

Enclosed data includes the following items:

1. One copy of OR 12,858, "High Temperature Bearing/Lubricant Test Report"
2. PTV Rack and Pinion Actuator Cut-Away Views (drawing no. 6392315, sheets 1 and 2)
3. PTV Rack and Pinion Actuator Shaft and Bearing Housing Details (drawing no. 63923152, sheets 1 and 4)
4. Baseline Requirements for FSD Bearings
5. Recommendations to be considered for inclusion in Hughes Aircraft Company's plan for bearing tests
6. One copy ASALM TD&I Meeting Minutes No. AF8-015.

Action Item no. 1(d) in item 6 above will be shipped under separate cover at a later date. Any inquires on this subject should be directed to Mr. Andrew Garner, (305) 352-4139 or Mr. Michael O'Leary, (305) 352-4663.

Very truly yours,

MARTIN MARIETTA CORPORATION

Frank J. Angiulli
Frank J. Angiulli
Contracts Manager
ASALM Programs

MOL/skp
Distribution:
Capt. M. Thoms, YYMA, w/o encls.
D.B. McConnell, AFML/F, w/o encls.

Enclosure # (4)

BASELINE REQUIREMENTS
VALIDATION/FSD BEARINGS

The current level of defintion for Validation/FSD bearing loads is such that the PTV requirements may be used. See Martin Marietta drawing 63923108.

Enclosure # (5)

PHASE IV TEST RECOMMENDATIONS

1. Expose coated bearings to combined thermal cycling, transportation vibration, and captive carry vibration followed by humidity testing. Purpose: Determine the integrity of the corrosion protection system.

2. The bearing duty cycle is comprised of approximately 1000 seconds with axial loads and small displacement oscillation. Maximum loading is in a transverse direction. Testing the bearings with the loads applied in the correct directions is recommended.

3. Correlation of bearing test results with paste anti-seize lubrication can be achieved by supplying Martin Marietta with coated Patriot bearings for testing in the facility described in OR 12,858.

APPENDIX F

ASALM BEARING AND LUBRICATION
FINAL TEST REPORT – TASK II-C

ASALM BEARING AND LUBRICATION

FINAL TEST REPORT - TASK II-C

Contract F33615-74-C-5128

(CDRL Data Item A00N)

OR 15,666 July 1979

Martin Marietta Aerospace
Orlando Division
P.O. Box 5837
Orlando, Florida 32855

FOREWORD

Martin Marietta Aerospace submits this document to the U.S. Air Force ASD/YYMA, Wright-Patterson AFB, Ohio, in response to CDRL Data Item No. A00N of Contract F33615-74-C-5128.

CONTENTS

SUMMARY

Martin Marietta has completed a bearing and lubrication test in support of a Hughes Aircraft Company (HAC) and U.S. Air Force Materials Laboratory (AFML) bearing materials development program. Thirteen bearings, similar to those proposed for use in the Advanced Strategic Air-Launched Missile (ASALM) control actuation system, were coated with varying combinations of corrosion resistant hardcoating and lubricant. The purpose of these coatings is to protect the bearings, typically made of heat treated 440C or M-50 steel, from long term corrosion occuring during the storage life duration of missiles. An additional sidelight to the hardcoat investigation is the development of a lubrication scheme that would enable the hardcoated bearings to provide the equivalent life-load capability of non-hardcoated bearings and also be stable under the long term storage conditions.

HAC had selected, as the most promising candidate, a hardcoating scheme consisting of a 1000.0A° Nichrome film followed by a 5.0 microinch coating of titanium carbide, (TiC) applied by a sputtering process. Martin Marietta was tasked under the ASALM Technology Development and Integration contract to evaluate the life-load capability of bearings, when subjected to ASALM environmental conditions, using the HAC-developed hardcoating scheme.

Martin Marietta's screening process resulted in the conclusion that the sputtered TiC/Nichrome hardcoat followed by a sputtered $MoS_2 \cdot Sb_2O_3$ dry lubricant coating is the most promising scheme for long term corrosion protection of anti-friction bearings. It is recommended that a larger sample size be further tested to determine applicability for use on ASALM.

1.0 TEST OBJECTIVES AND APPROACH

The objective of the Martin Marietta bearing and lubrication test was to provide HAC with the frictional and load carrying characteristics of candidate hardcoat/lubrication systems and participate in the evaluation and selection of the best candidate(s) for the ASALM CAS application.

Roller bearings used in the Patriot missile hydraulic CAS, conforming to the requirements of Martin Marietta drawing 10251122-1, Rev. B, were selected as the baseline test specimen configuration. The bearing selection was based on similar load carrying requirements for both CAS applications. The bearings are fabricated of 440C grade steel and were readily available from bearing suppliers as a consequence of the previous production runs made in support of Patriot procurement. Martin Marietta procured 12 bearings from the Split Ball Bearing Division of MPB Corporation and shipped these test specimens to HAC for hardcoat and lubrication coating. An additional pair of bearings were withheld from the corrosion resistant hardcoating process to be used as a baseline for performance comparisons with the hardcoat/lubrication candidate.

The Patriot CAS bearings were all sputtercoated with the baseline Nichrome/TiC hardcoat and, as originally planned, were divided into five pairs of different lubrication configurations.

Each lubrication pair was to be subjected to the Air Force approved two-level test discussed in the Bearing/Lubrication Test Plan (Martin Marietta document TPL 00920041-001) as summarized in Table 1-I. In the first test, the bearing was placed in the Martin Marietta-designed test fixture, shown in Figures 1-1 and -2, and refrigerated to -65°F. At the start of test, five electrical heaters of 2.1 kW capacity were excited and a radial load of 7,150 pounds (the bearing rated static load capacity) was applied. At the same time, a ± 25 degree sinusoidal motion of the bearing inner race was initiated at 1.6 Hz, providing a peak rotational rate of 250 deg/s. The electrical heaters are sized to increase the bearing temperature from -65°F to +600°F within 1000 seconds, simulating the ASALM flight environment. In the event that the first bearing of a pair passed this test, the second bearing was subjected to a duplicate test, except that at the end of 1000 seconds, the radial load was increased to 14,300 pounds (twice the bearing rated static load) and the test continued for an additional 1000 seconds with the heaters modulated to maintain the temperature constant at 600°F. The additional 1000 seconds at twice the rated load was designed to force failure of the bearings and to permit use of the run time before failure as a measure of "goodness" of the hardcoat/lubrication combination. The radial load and bearing temperature time profiles for the flight simulation and limit load tests are shown in Figure 1-3.

TABLE 1-I

Test Matrix for Bearing and Lube Test

		Test Conditions		
		Parameter	Flight Simulation Test	Limit Load Test
Bearing No.	Configuration	Time	1000 seconds	1000 seconds
		Temperature	-65 to +600°F	+600°F
		Radial Load	7150 pounds	14,300 pounds
1	TiC + no lube		*	
1A	TiC + no lube		*	*
2	TiC + MoS_2 + SbO_3		*	
2A	TiC + MoS_2 + SbO_3		*	*
3	TiC + Westinghouse Composite		*	
3A	TiC + Westinghouse Composite		*	*
4	TiC + HAC Composite		*	
4A	TiC + HAC Composite		*	*
5	TiC + Fel-Pro C-100		*	
5A	TiC + Fel-Pro C-100		*	*

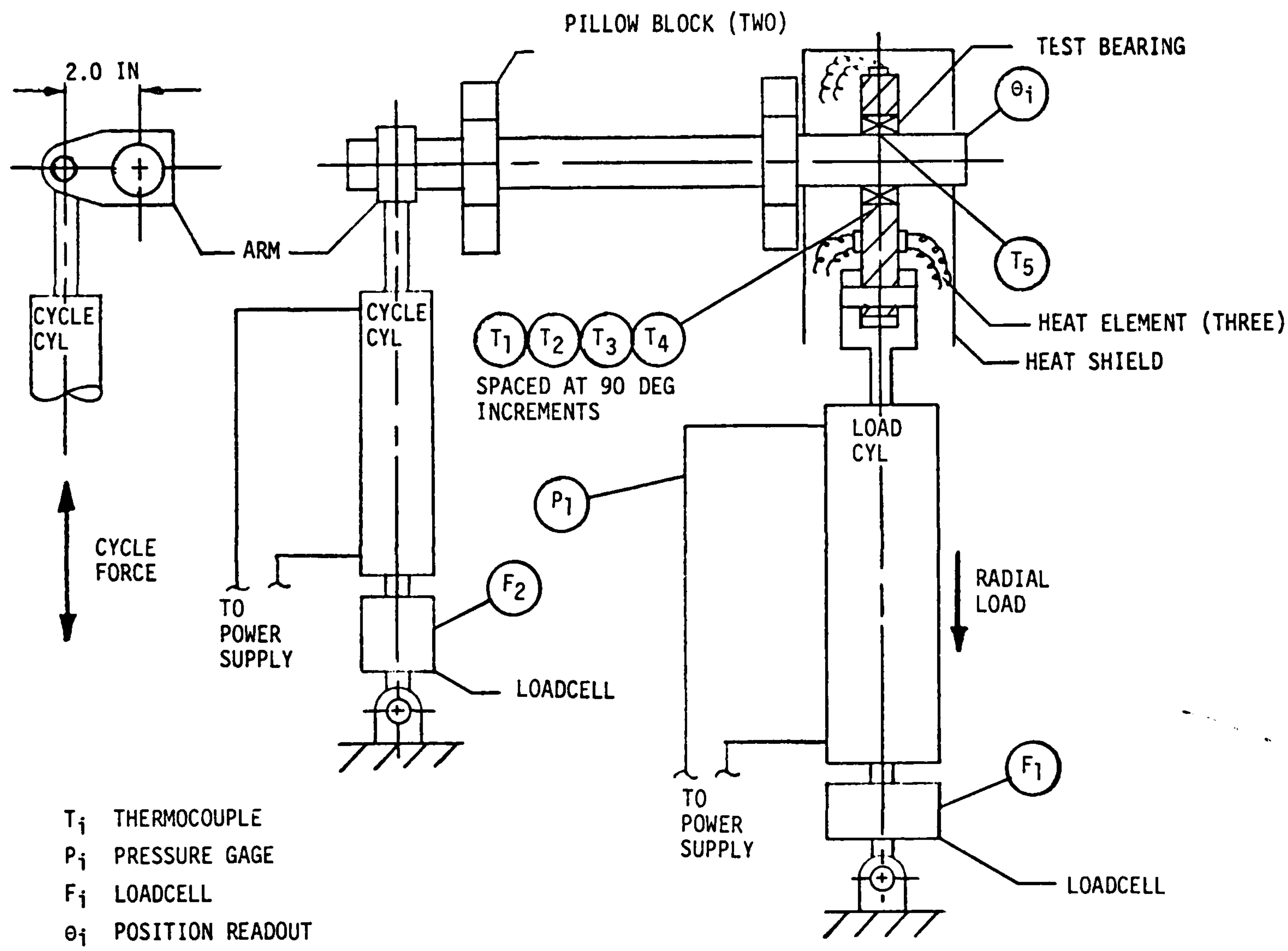


Figure 1-1. Bearing/Lubrication Test Set-up Schematic

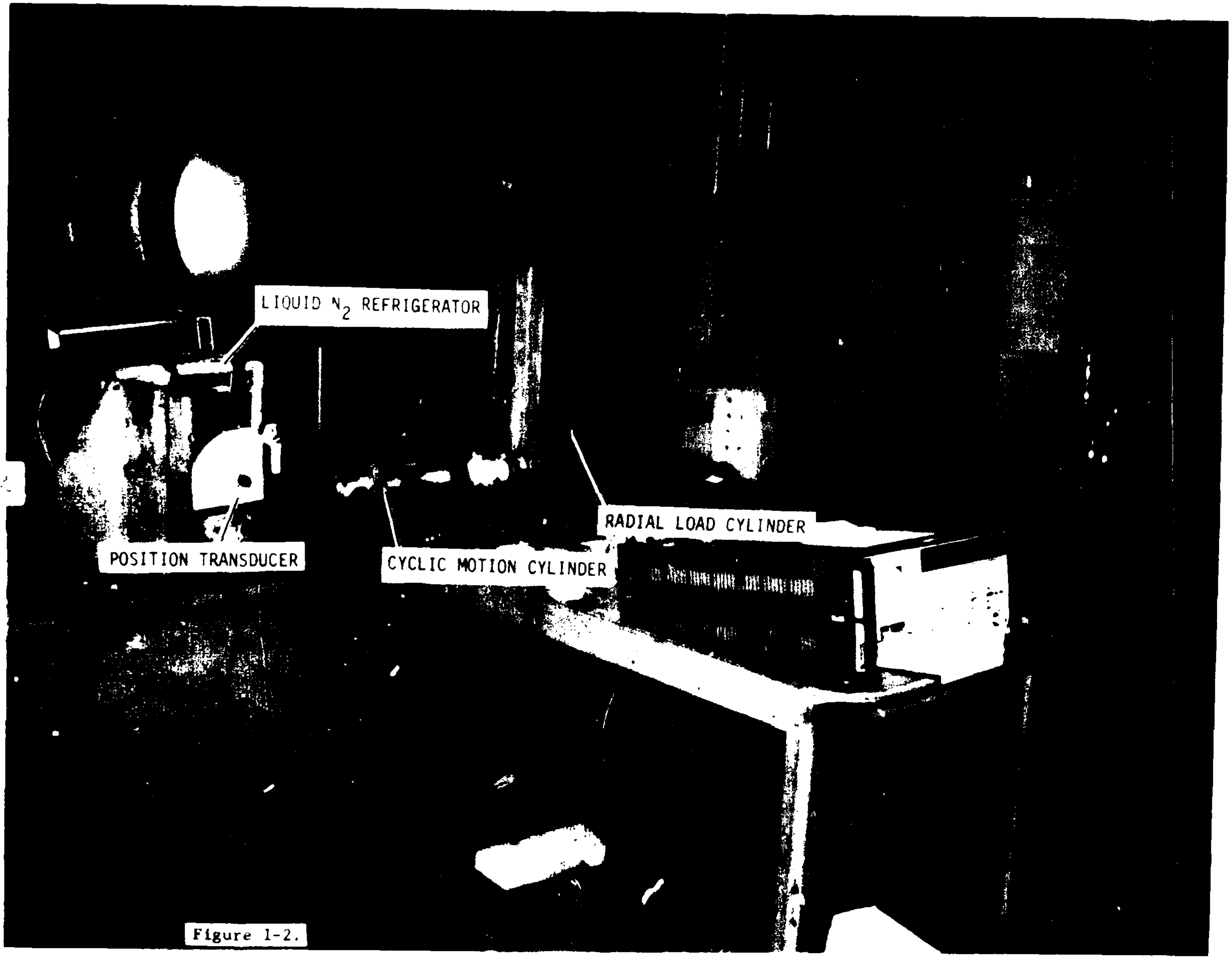

LIQUID N_2 REFRIGERATOR
POSITION TRANSDUCER
CYCLIC MOTION CYLINDER
RADIAL LOAD CYLINDER

Figure 1-2.

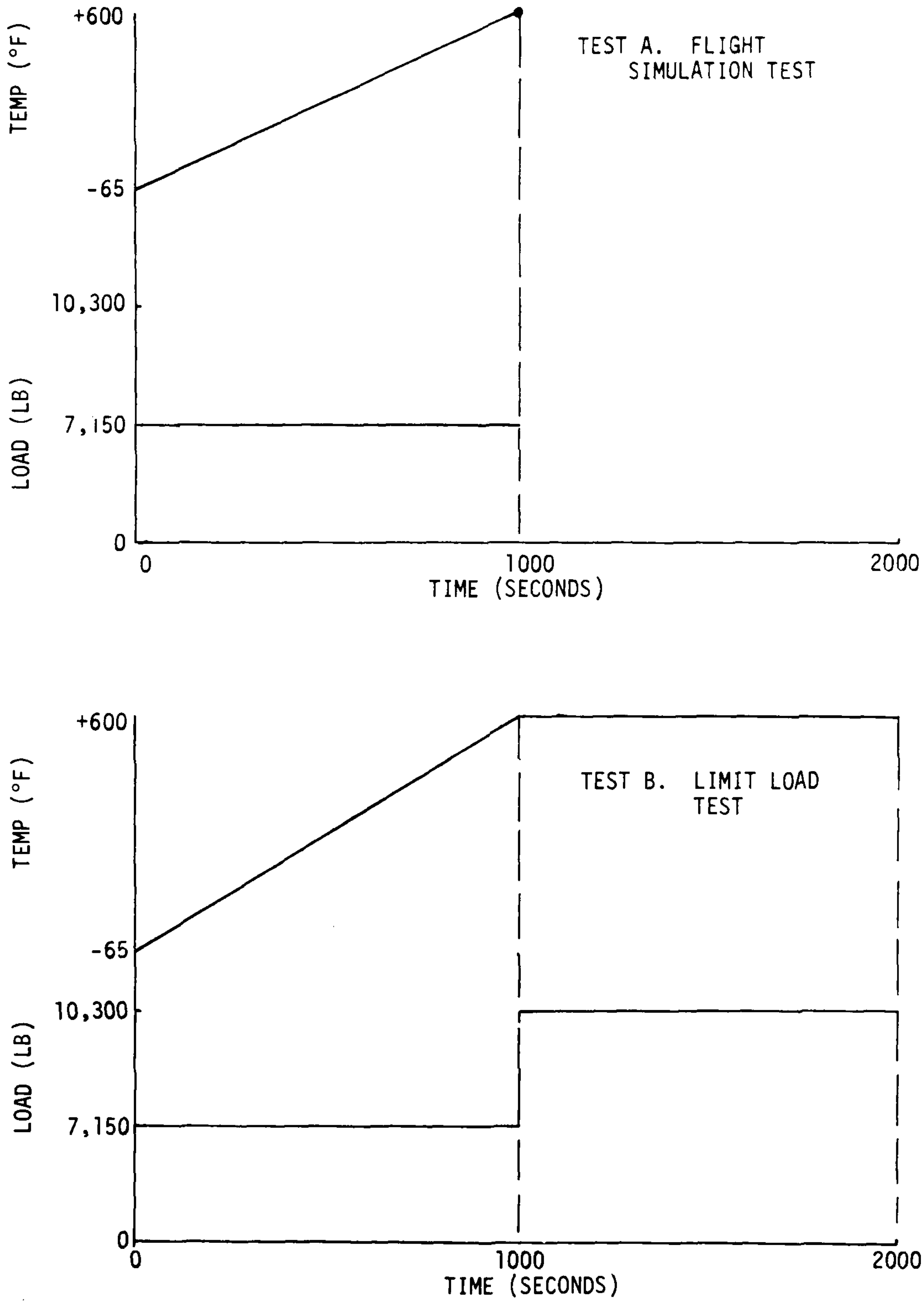


Figure 1-3. Temperature and Load Profiles for Tests A and B

2.0 HARDCOAT/LUBRICATION CONFIGURATIONS

The original bearing/lubrication test plan considered the test of ten bearings hardcoated and lubricated as shown in Table 2-I. After the test program started, an additional pair of bearings were located in-house at Martin Marietta and HAC provided another bearing. It was decided to use these additional three bearings in tests to develop a baseline performance guideline for comparison with the original ten bearings.

TABLE 2-I

Hardcoat/Lubrication Configurations

Bearing Number	Coating	Lubricant
1	1000.0A° Nichrome followed by 5.0 microinch TiC	Stick burnished races with Ga/In/WSe_2 composite (Westinghouse compact)
2	Same as 1	Same as 1
3	Same as 1	Stick burnished races with Ga/In/WSe_2 composite followed by powder burnishing of rollers and then 4-hour run-in using alternate rollers of Hughes Self-Lubricating Composite
4	Same as 1	Stick burnished races with Ga/In/WSe_2 composite followed by 4-hour run-in using alternate rollers of Hughes Self-Lubricating Composite
5	Same as 1	Fel-Pro C100
6	Same as 1	Same as 6
7	Same as 1	None
8	Same as 1	None
9	Withheld at HAC	
10	Same as 1	3.0 microinch sputtered coating of $MoS_2 \cdot Sb_2O_3$ run-in for 4 hours
11	Same as 1	Same as 10

3.0 TEST RESULTS

The screening of the 13 hardcoat/lubricant candidates was conducted between 17 May and 1 June 1979. A bearing test results matrix is given in Table 3-I, showing the configuration, applied test, and results of the 13 bearing tests conducted. Time plots of the coefficient of friction and test specimen average temperature are presented in Appendix A. Data sheets tabulating the reduced friction force, friction torque, coefficient of friction and average temperature are also presented in Appendix A.

Problems with data gathering systems of the test resulted in the loss of temperature data for tests of bearings 1 and 3, and loss of temperature and friction data for bearing 6. The temperature curve for bearing 4 will closely approximate the gradients experienced by bearings 1, 3, and 6, and can be used to estimate the actual temperatures experienced.

The first bearing of each of four pairs was subjected to the flight simulation test, Test A (see Figure 1-3). The results of Test A for bearings 1, 3, 5, and 7 are shown in Table 3-II. Of the four bearings tested, only bearing No. 5 passed flight simulation Test A. This bearing was lubricated with Fel Pro C100, a molybudenum disulfide based grease used on the ASALM-PTV hydraulic CAS bearings. This lubrication is not considered a true candidate for the ASALM application because of the unknown and suspect long term storage characteristics and was incorporated into the test program only for comparison with the dry lubricant candidates. The mate to No. 5 (bearing 6) was subjected to Test B, the limit load test, and successfully completed the 2000 second test cycle.

TABLE 3-II

Initial Bearing Test Results

Bearing No.	Failed/Passed	Time at Failure
1	Failed	4 min
3	Failed	9 min
5	Passed	-
7	Failed	4 min

TABLE 3-I

Bearing Test Results

Bearing No.	Hard-coat	Lubri-cant	Test A -65°F to 600°F Rated Load (16 min)	Test B -65°F to 600°F 2x Rated Load (32 min)	Test C Room Temp (Load Full)	Test D -65°F to 600°F (1/2 Load)	Re-Test of Test A or Test B	Remarks
1	Yes	1	Failed at 4 min	-	-	-	-	Scrubbing noted on roller shoulders
2	Yes	1	-	No test	Failed at 5 min	-	-	
3	Yes	2	Failed at 9 min	-	-	-	-	Same as 1
4	Yes	3	-	No test	-	Good; 16 min +	-	
5	Yes	FC 100	Good; 16 min +	-	-	-	-	
6	Yes	FC 100	-	Good; 32 min +	-	-	-	
7	Yes	None	Failed at 4 min	-	-	-	-	Same as 1
8	Yes	3	-	-	-	-	Good No. 1 = 16 min +	Bearing originally non-lubricated, subsequently cleaned to remove TiC flash and coated with lubricant No. 3
9	Yes	3	-	-	-	-	Failed at 10 min	
10	Yes	4	Good; 16 min +	-	-	-	-	
11	Yes	4	-	Good; 32 min +	-	-	-	
12	No	FC 100	-	Good; 32 min +	-	-	-	Reference bearing
13	No	3	-	Test aborted at 20 min	-	-	-	Friction data indicates bearing failure was imminent

Lubricant 1: Stick burnished races with $Ga/In/WSe_2$ composite (Westinghouse compact)

Lubricant 2: Stick burnished races with $Ga/In/WSe_2$ composite followed by powder burnishing of rollers and then 4-hour run-in using alternate rollers of Hughes Self-Lubricating Composite

Lubricant 3: Stick burnished races with $Ga/In/WSe_2$ composite followed by 4-hour run-in using alternate rollers of Hughes Self-Lubricating Composite

Lubricant 4: 3.0 microinch sputtered coating of $MoS_2 \cdot Sb_2O_3$ run-in for 4 hours

The poor performance of the dry lubricant candidates, 1 and 3, and of the non-lubricated candidate, No. 7, resulted in a review of the program plan by HAC, Martin Marietta, and AFML personnel. Post-test examination of the failed bearings indicated that the bearing surfaces were covered with a "flash" of extremely fine TiC particles that became intimately mixed with the dry lubricant during application of the lubricant film. Wear evidence indicated that the TiC/lubricant combination contributed to the poor load-life performance of the bearings. Since no simple method of dry lubricant removal is known, it was decided that the remaining burnished dry lubricant candidates, No. 2 and No. 4, would be tested to determine if the failures were caused by thermal shock or by a reduced load carrying capacity. Candidate 3 was subjected to Test C and 4 to Test D, as indicated in Figure 3-1. In Test C, the specimen was subjected to the full rated load but the temperature was room ambient, thus eliminating temperature shock considerations. The bearing failed after approximately 5 minutes of operation. Bearing 4, subjected to the thermal gradient (but only half the rated load), survived the full 16 minutes test cycle indicating that reduced load carrying capacity due to hardcoating, and not thermal shock, was contributory to the failure of the burnished dry lubricated candidates.

In an attempt to determine if cleaning of the TiC flash from hardcoated bearings would increase the load carrying capability, the original non-lubricated hardcoated candidates, Nos. 8 and 9 were cleaned to remove the TiC particles, burnish dry lubricated with the lubricant used in bearing 4, and reassembled. The first bearing of the pair, No. 8, successfully passed the flight simulation test, Test A. The second bearing, No. 9, was to be subjected to the limit load test, Test B, but failed after 10 minutes, during the portion of the test duplicating the flight simulation test (load = 1 x bearing rated load and temperature = -65 to +600°F). The failure of the hardcoated, cleaned, and burnished dry lubricated bearing indicates that the lubricant scheme inadequately protects the bearing hardcoat.

The remaining hardcoated candidates, Nos. 10 and 11, which used a sputtered $MoS_2 \cdot Sb_2O_3$ dry lubrication scheme, showed promise of meeting the bearing performance requirements. This estimate was based on microscopic inspection results that indicated that the TiC flash evidently was removed during the lubricant sputtering process. The resulting bearing operation felt much smoother. Bearing 10 passed the flight simulation test and No. 11 passed the limit load test (2000 seconds at up to 2 x bearing rated). The performance of the $MoS_2 \cdot Sb_2O_3$ sputtered dry lubricated candidate indicates that this scheme warrants further investigation as a hardcoated bearing lubricant.

As a final baseline check on lubricant performance, the two additional bearings, designated 12 and 13, while not hardcoated, were tested to determine if the dry lubricant was acceptable even on a non-hardcoated bearing. The first candidate, No. 12, was coated with the PTV hydraulic CAS baseline grease (Fel Pro C100) and passed the limit load test, Test B (2000 seconds at up to 2 x rated load). The second candidate, No. 13, was burnish dry lubricated with the same lubricant used in candidates 3 and 4. The bearing limit load test, Test B, was aborted at approximately 20 minutes due to a

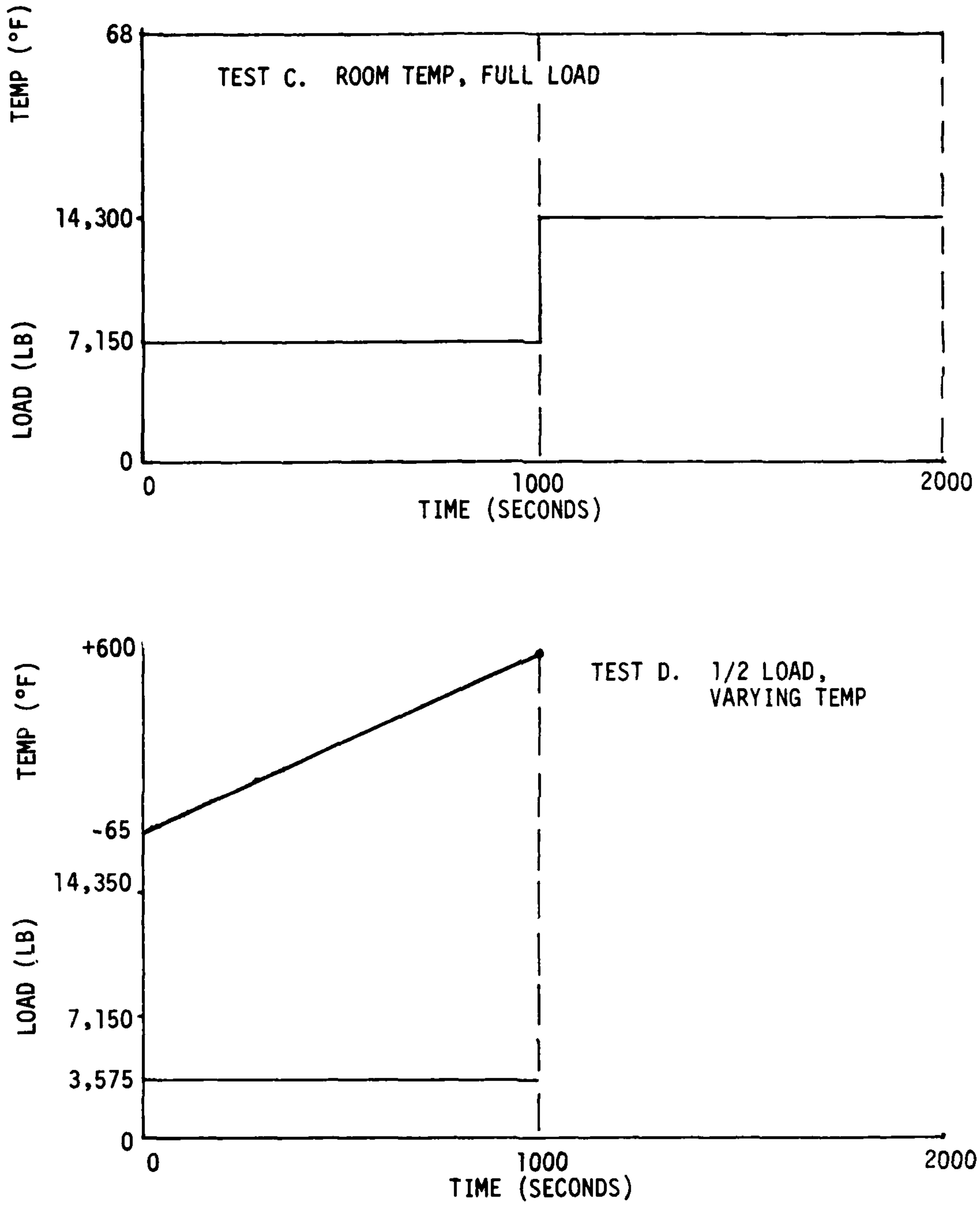


Figure 3-1. **Temperature and Load Profiles for Tests C and D**

test fixture failure. Repeated cooldown to -65°F (with attendant frost buildup) and heating to 600°F deteriorated the insulation in the Chromalox heaters used to provide the temperature gradients and the resultant arcing forced shutdown of the test.

Oscillograph traces of the friction force versus time data did not indicate any problem with the test bearing. Initial quick-look inspection indicated that the bearing should have successfully completed the entire test cycle. However, the plot of the reduced coefficient of friction data, Figure A-12, shows a steep gradient between 18 and 20 minutes at failure, indicating doubt whether this bearing would have successfully completed the 2000 second test cycle. Since the data indicates a gradual reduction of friction coefficient prior to 1000 seconds when the load was doubled, it was concluded that the dry lubricant's ability to protect the non-hardcoated bearing under overload conditions is questionable.

An important characteristic of any bearing/hardcoat/lubricant combination is the friction torque required to rotate the bearing. The friction torque required at the start of a flight simulation test (-65°F) and at the end of test, 1000 seconds (≈600°F), is tabulated in Table 3-III for the candidates that successfully passed a 1000 second duty cycle. The Fel Pro C100 grease provides acceptable bearing load-life performance, however, its suspect long term storability and high friction at low temperature precludes consideration as a viable lubrication candidate.

The sputtered $MoS_2 \cdot Sb_2O_3$ lubricated candidate is the only scheme that provides acceptable bearing load-life performance without excessive friction at low temperatures.

TABLE 3-III

Bearing Coefficient of Friction Comparison

Bearing No.	Hardcoat	Lubricant	Coefficient of Friction (μ) at -65°F	Coefficient of Friction (μ) at ≈600°F t = 1000s
5	Yes	Fel Pro C100	0.278	0.037
6	Yes	Fel Pro C100	N/A***	N/A***
8	Yes	*	0.073	0.041
10	Yes	Sputtered $MoS_2 \cdot Sb_2O_3$	0.065	0.054
11	Yes	Sputtered $MoS_2 \cdot Sb_2O_3$	0.055	0.036
12	No	Fel Pro C100	0.151	0.041
13	No	**	0.041	0.035

*Stick burnished races with $Ga/In/WSe_2$ composite followed by powder burnishing of rollers then 4-hour run-in using alternate rollers of Hughes Self-Lubricating Composite

**Similar to * except that rollers were tumble burnished and run-in without composite rollers

***Test data not available due to instrumentation problems

4.0 CONCLUSIONS AND RECOMMENDATIONS

Based on a relatively small sample size of two entries per candidate, the results of the bearing tests conducted by Martin Marietta clearly indicate that the sputtered $MoS_2 \cdot Sb_2O_3$ dry lubricant coating is superior to the two burnished-applied dry lubricant candidates investigated. Fel Pro C100 grease provides acceptable bearing life-lubrication performance but exhibits large coefficient of friction changes with temperature and has questionable long term storage characteristics. The bearing tests also indicate that the ability to withstand the flight load-duration requirements is not influenced by temperature shock, but is primarily a function of the hardcoat/lubricant capability to withstand the stress induced by the radial load on the bearing.

To fully validate the $MoS_2 \cdot Sb_2O_3$ approach, two steps must be taken. First, the sample size must be increased to ensure that the program to date has not been subjected to random successes. Secondly, the actual bearings intended for use in the ASALM actuator need to be tested. Patriot CAS bearings were selected for this task because of the significantly reduced cost compared to the TD&I or PTV CAS bearings, and also because of an excessive lead time required to procure the TD&I bearings. Present estimates indicate that 70 weeks lead time is required before delivery by Split Ball Bearing Division of MPC Corporation.

Martin Marietta recommends that a follow-on development program be pursued that would entail the test of a minimum of 10 hardcoated and lubricated bearings of the type required in the ASALM CAS. As part of the added test program, bearing performance sensitivities to humidity variations and angular rate would be determined.

APPENDIX TO MARTIN MARIETTA REPORT

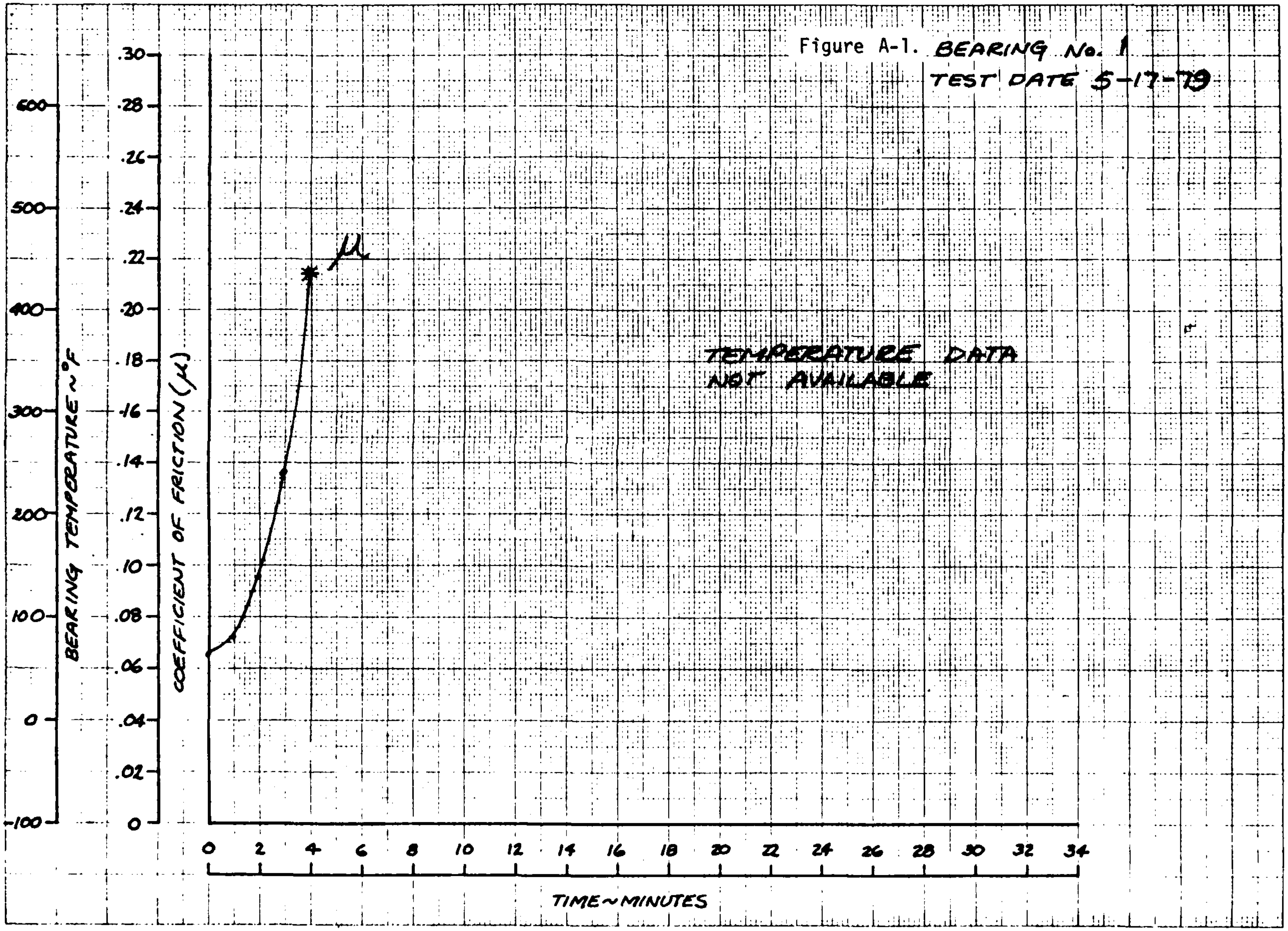


Figure A-1. BEARING No. 1
TEST DATE 5-17-79

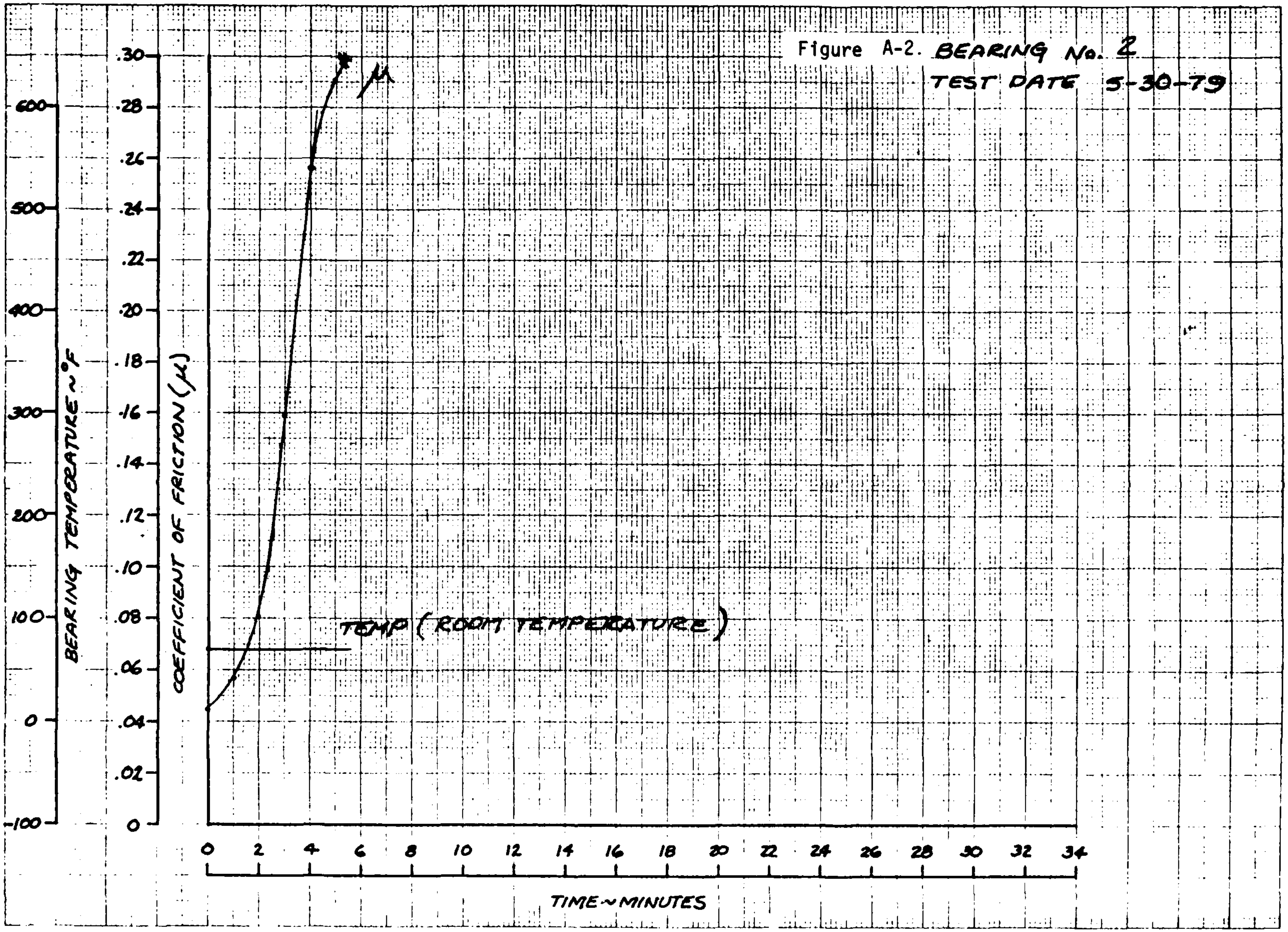


Figure A-2. BEARING No. 2
TEST DATE 5-30-79

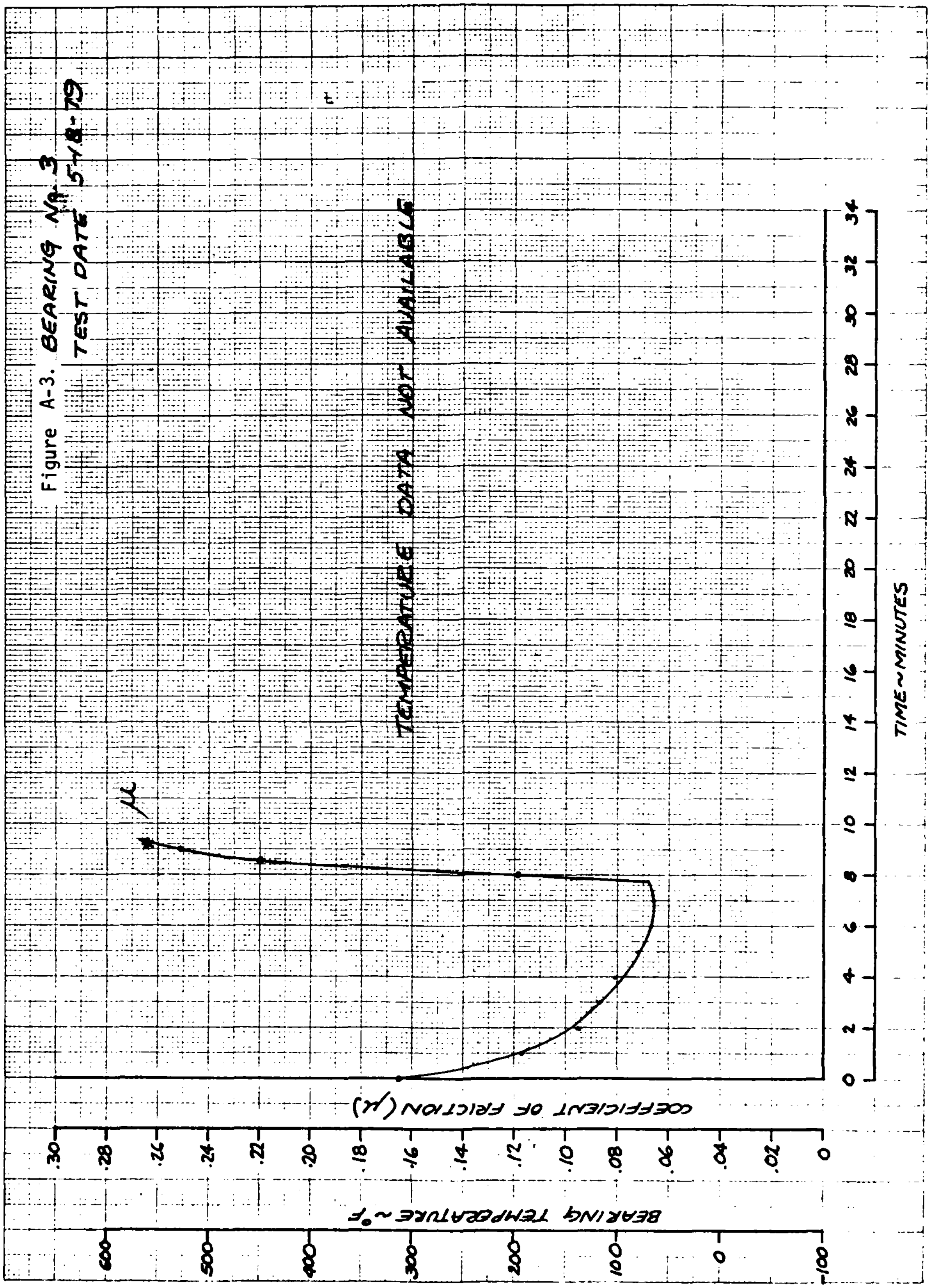

BEARING No. 3
TEST DATE 5-18-79
TEMPERATURE DATA NOT AVAILABLE
μ
TIME ~ MINUTES
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
COEFFICIENT OF FRICTION (μ)
.30
.28
.26
.24
.22
.20
.18
.16
.14
.12
.10
.08
.06
.04
.02
0
BEARING TEMPERATURE ~ °F
600
500
400
300
200
100
0
-100

Figure A-3.

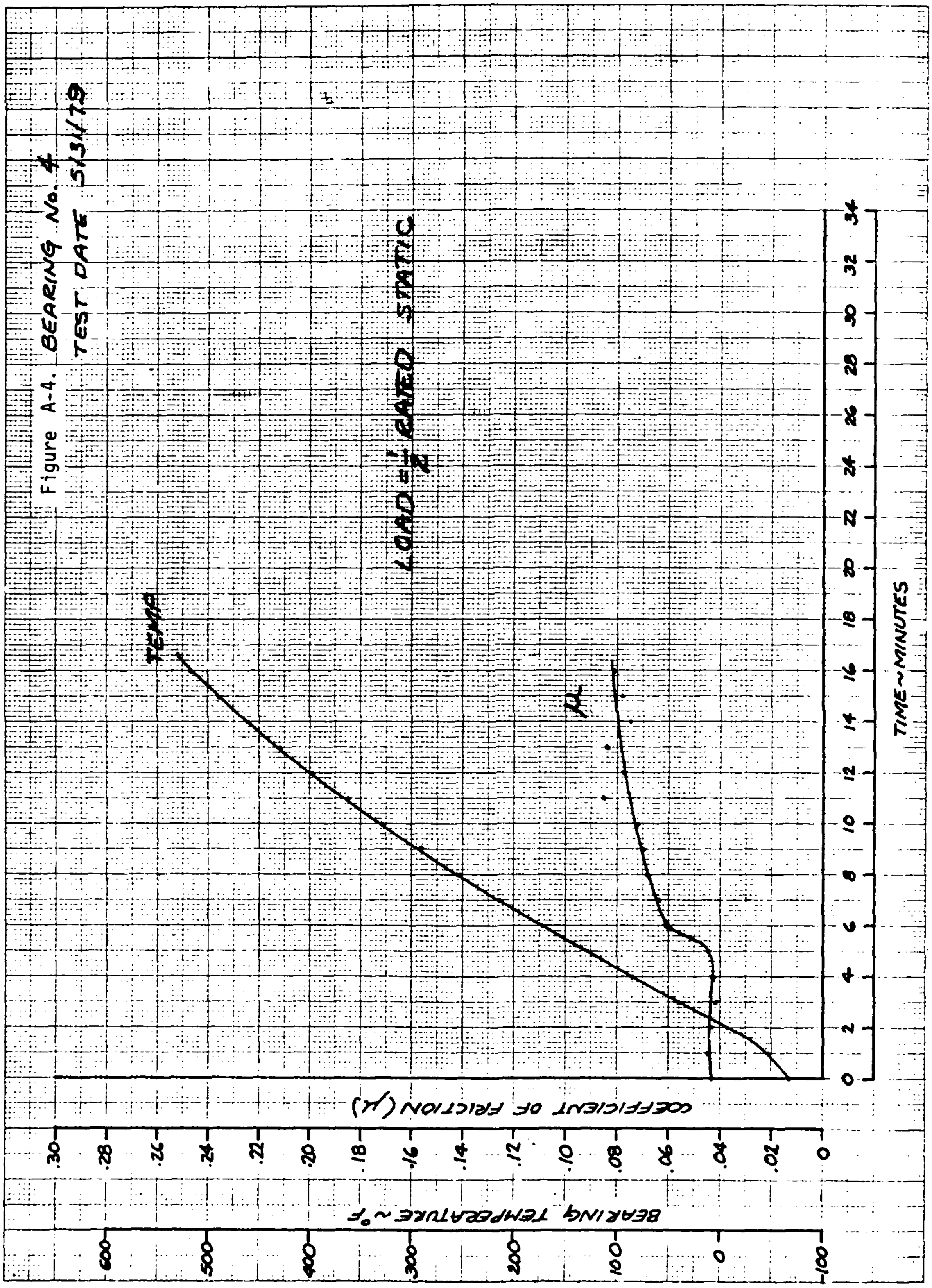


Figure A-4. BEARING No. 4
TEST DATE 5/31/79

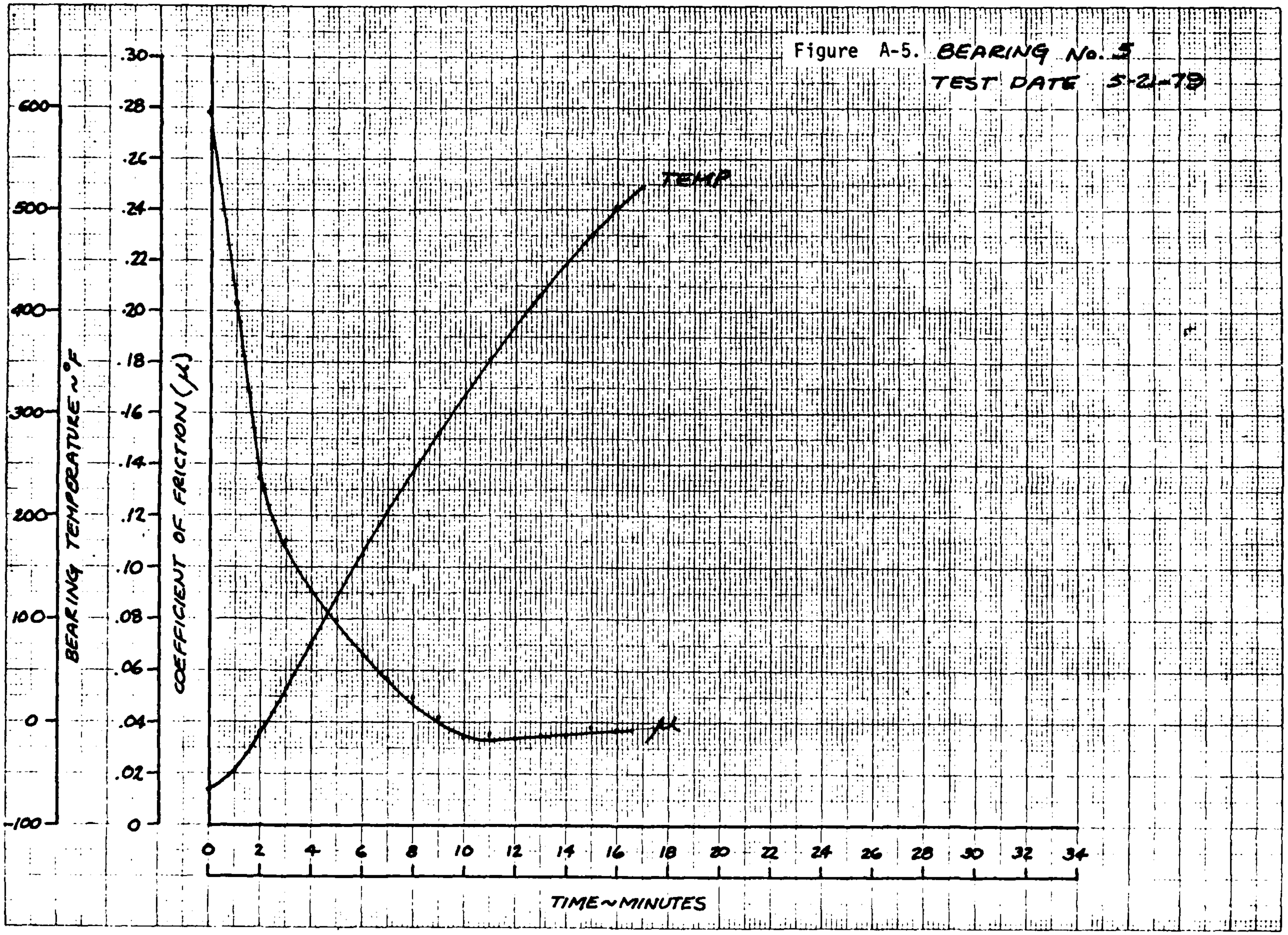


Figure A-5. BEARING No. 5
TEST DATE 5-21-79

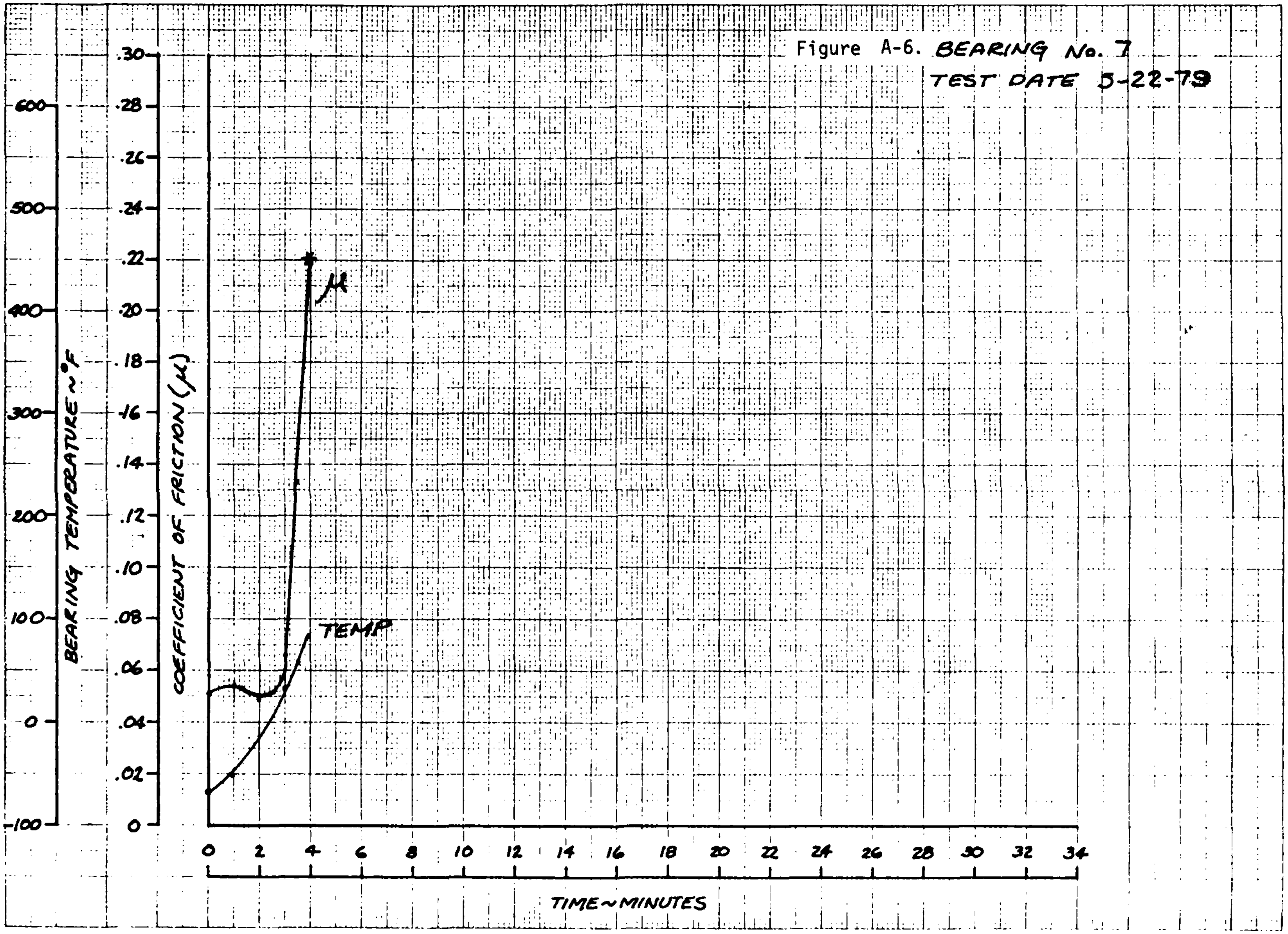


Figure A-6. BEARING No. 7
TEST DATE 5-22-79

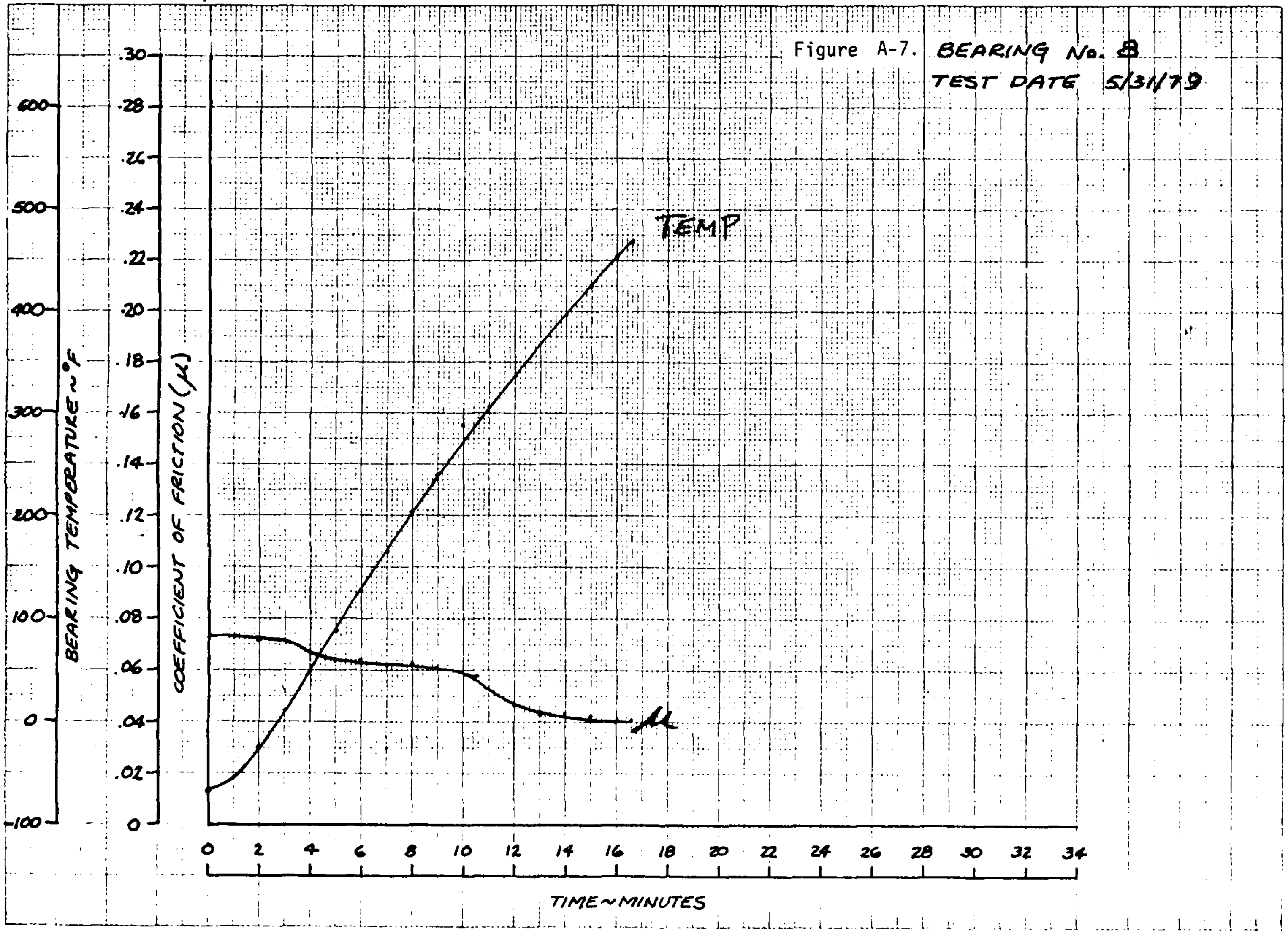
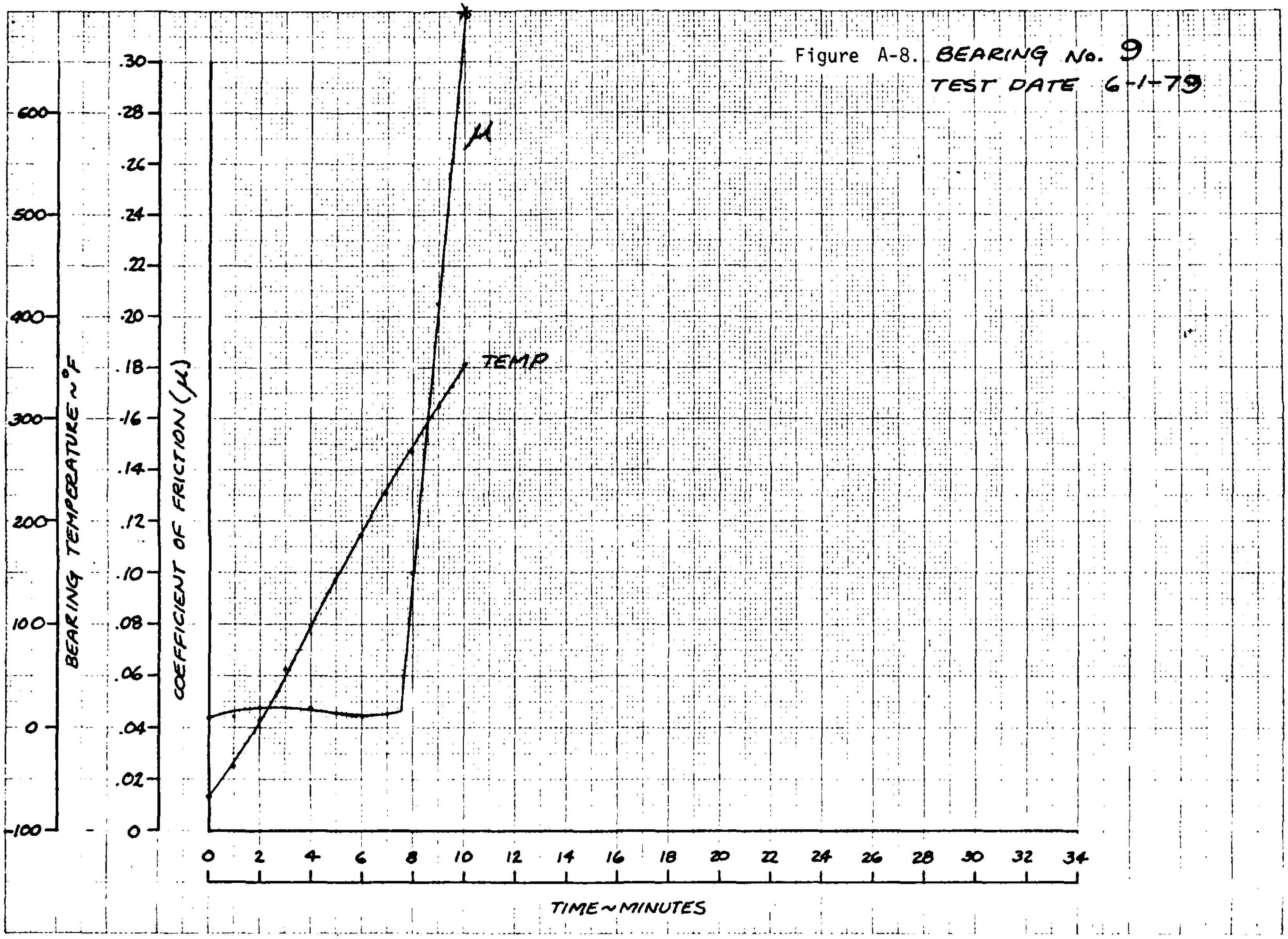


Figure A-7. BEARING No. 8
TEST DATE 5/31/79

Figure A-8. BEARING No. 9
TEST DATE 6-1-79



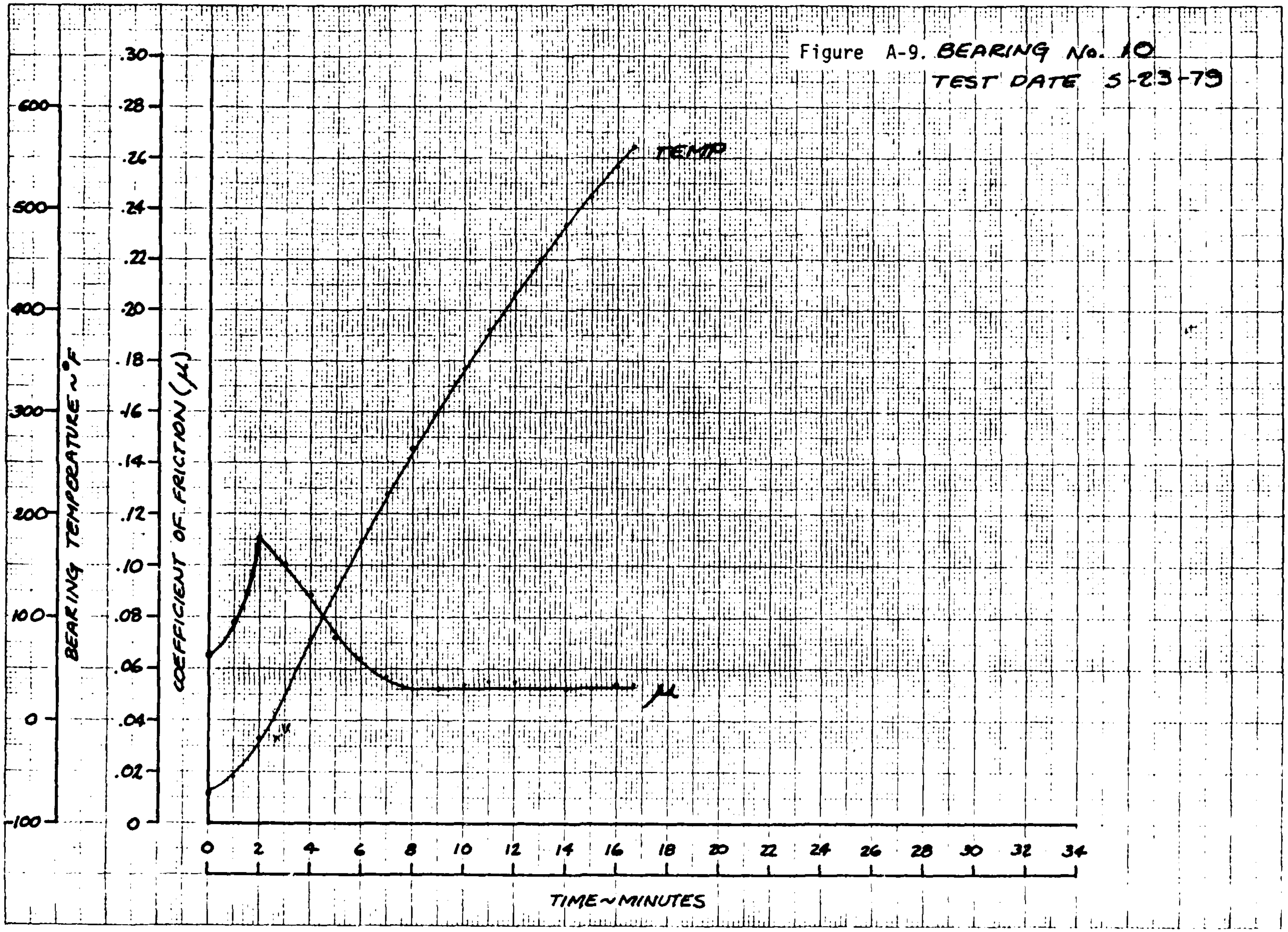


Figure A-9. BEARING No. 10
TEST DATE 5-23-79

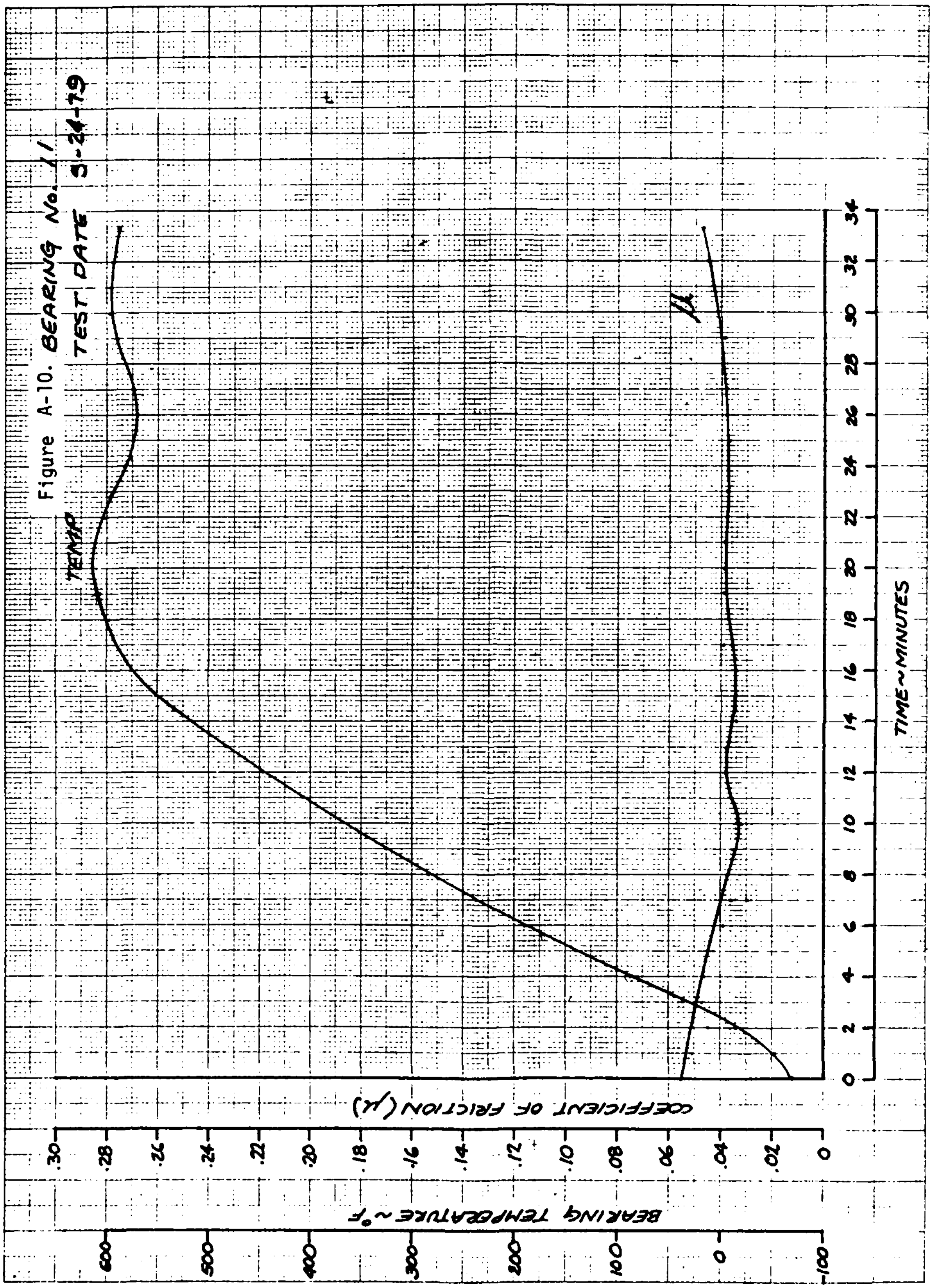


Figure A-10. BEARING No. 11
TEST DATE 5-24-79

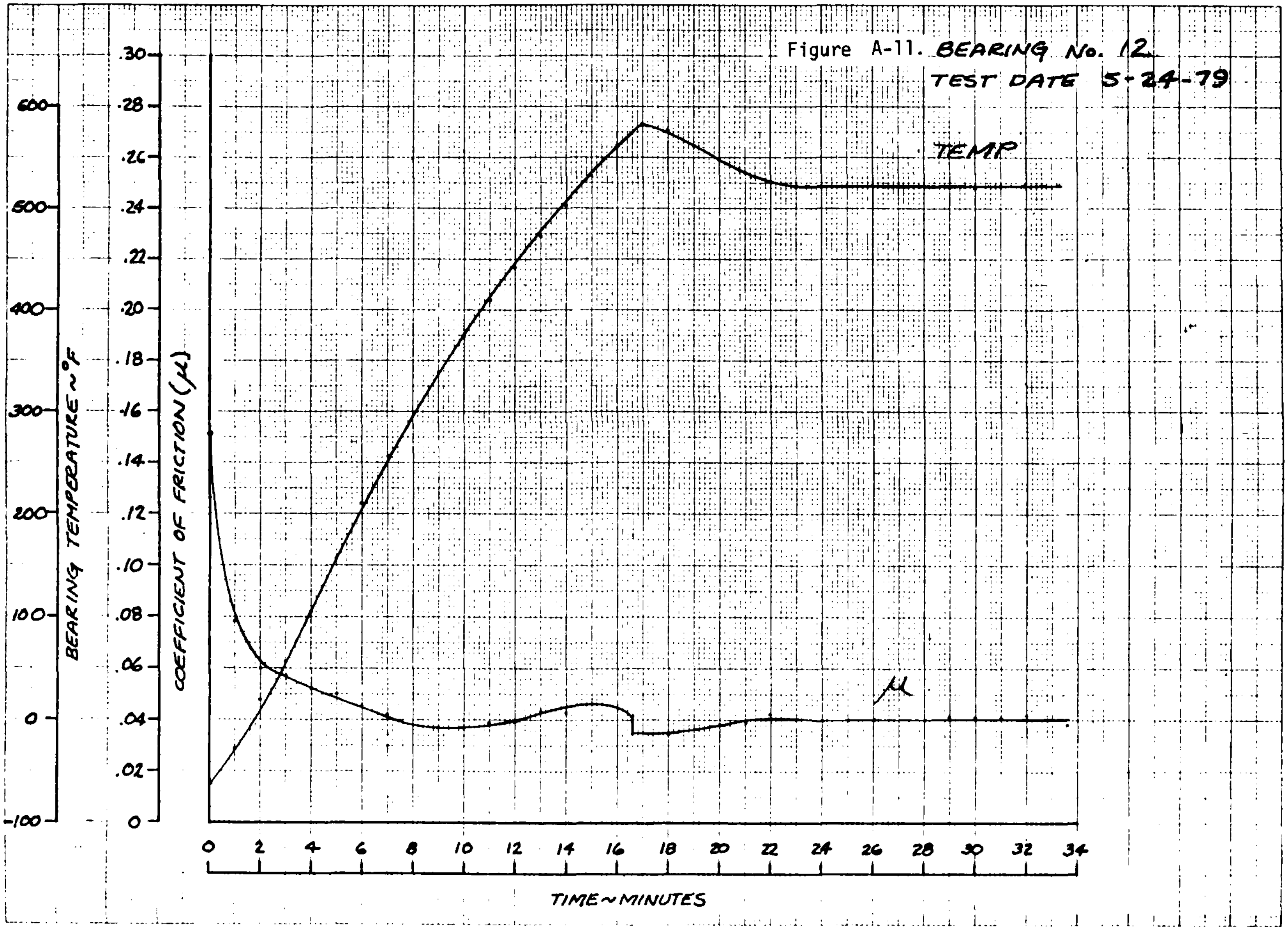


Figure A-11. BEARING No. 12
TEST DATE 5-24-79

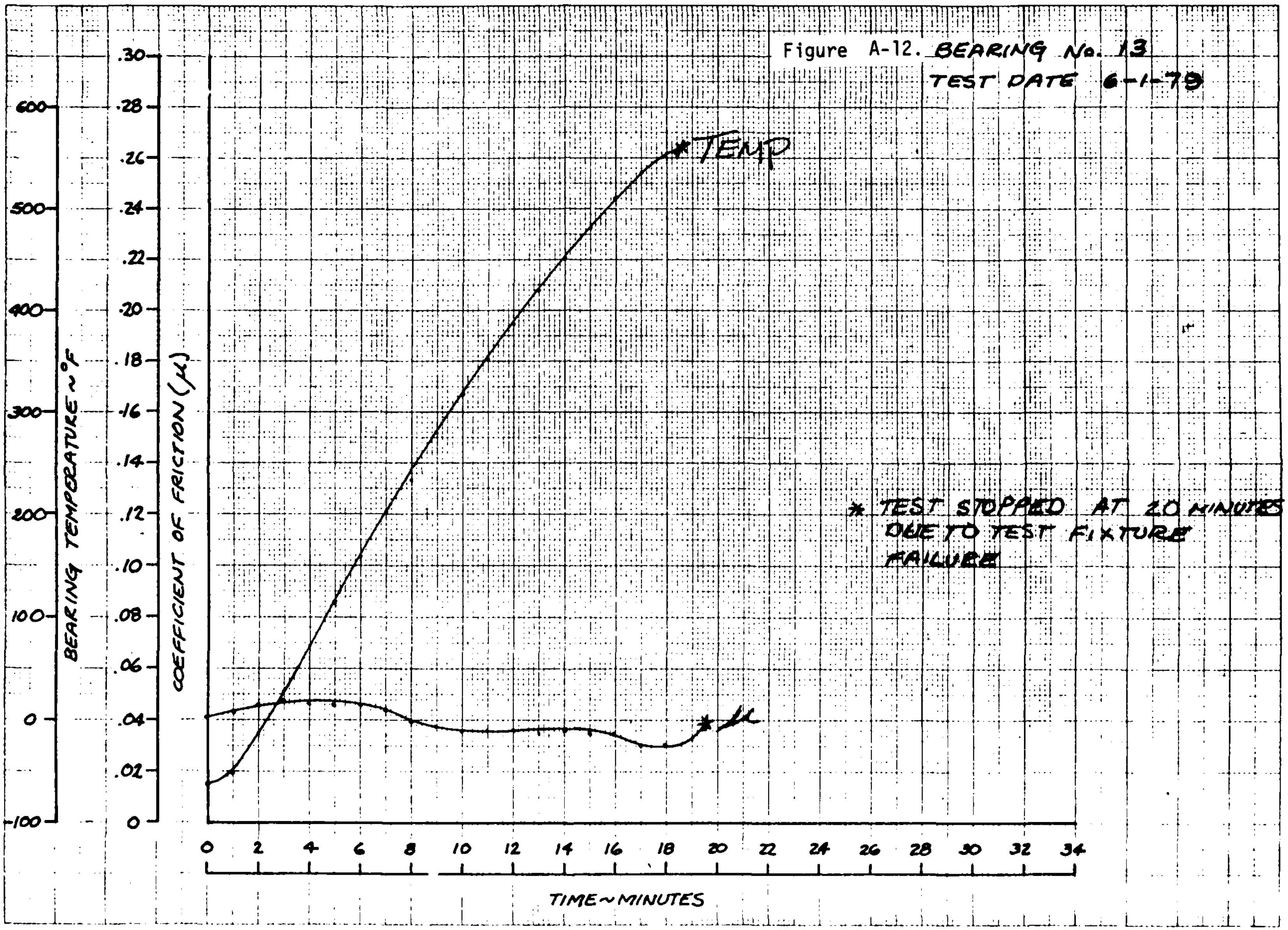


Figure A-12. BEARING No. 13
TEST DATE 6-1-79

TABLE A-I. BEARING No. 1

TEST DATE 5-17-79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	258	516	.0657	*
1	283	564	.0717	
2	376	752	.0956	
3	535	1070	.1360	
4	850	1700	.2161	
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
16:40				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

* TEMPERATURE DATA NOT AVAILABLE

START TEST @ -65°F WITH 2.1KW HEATERS ON

TABLE A-II. BEARING No. 2

TEST DATE 5/30/79 (ROOM TEMP, RATED STATIC LOAD)

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	138	276	.035	70°F
1	225	450	.0572	
2	318	636	.0809	
3	625	1250	.1589	
4	1008	2016	.2563	
5	1145	2290	.2912	
~~6~~ 5:20	1170	2340	.2975	
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
16:40				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-III. BEARING No. 3

TEST DATE 5/18/79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	648	1296	.1648	*
1	460	920	.1170	
2	373	746	.0949	
3	343	686	.0872	
4	320	640	.0814	
5	283	566	.0720	
6	265	530	.0674	
7	260	520	.0661	
8	468	936	.1190	
8.5	865	1730	.2200	
9.0	988	1976	.2512	
9.25	1040	2080	.2645	
12				
13				
14				
15				
16				
16:40				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

* TEMPERATURE DATA NOT AVAILABLE. TEST STARTED AT -65°F WITH 2 1 KW HEATERS EXCITED.

TABLE A-IV. BEARING No. 4

TEST DATE 5/31/79 (½ RATED STATIC LOAD)

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	170	340	.0432	-68.4
1	178	356	.0453	-47.9
2	168	336	.0427	+6.8
3	165	330	.0420	39.3
4	170	340	.0432	85.1
5	175	350	.0445	129.1
6	240	480	.0610	172.4
7	253	506	.0643	213.9
8	268	536	.0682	253.8
9	275	550	.0699	292.2
10	283	566	.0720	328.9
11	333	666	.0847	364.0
12	305	610	.0776	397.5
13	330	660	.0839	429.1
14	295	590	.0750	459.6
15	308	616	.0783	488.9
16	320	640	.0814	517.5
16:40				529.8
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-V. BEARING No. 5

TEST DATE 3/21/79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	1095	2190	.2784	-66.4
1	800	1600	.2034	-47.9
2	530	1060	.1348	-9.6
3	433	865	.110	+32.7
4	358	715	.0909	76.2
5	310	620	.0788	119.5
6	258	515	.0655	162.0
7	223	445	.0566	203.2
8	195	390	.0496	242.5
9	165	330	.0420	280.1
10	135	270	.0343	316.4
11	138	275	.0350	351.3
12	135	270	.0343	384.3
13	136	273	.0347	415.7
14	138	275	.0350	445.8
15	148	295	.0375	474.5
16	145	290	.0369	501.9
16:40	145	290	.0369	521.6
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-VI. BEARING No. 7

TEST DATE 5/22/79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	200	400	.0509	-66.7
1	213	425	.0540	-50.8
2	193	385	.0490	-13.0
3	260	520	.0661	+32.8
~~4~~ 3.52	523	1045	.1329	57.1
~~5~~ 3.92	870	1740	.2212	83.4
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
16:40				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-VII. BEARING NO. 8

TEST DATE 5/31/79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	288	575	.0731	-66.3
1	288	575	.0731	-54.6
2	283	565	.0718	-24.8
3	280	560	.0712	+10.8
4	265	530	.0674	49.0
5	253	505	.0642	88.1
6	253	505	.0642	127.0
7	243	485	.0617	165.3
8	245	490	.0623	202.2
9	240	480	.0610	237.9
10.5	228	455	.0579	288.5
11	—	—	—	304.7
12	183	365	.0464	336.4
13	170	340	.0432	367.1
14	168	335	.0426	396.8
15	165	330	.0420	425.4
16	158	315	.0401	453.1
16:40	160	320	.0407	466.6
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-VIII. BEARING No. 9

TEST DATE 6-1-79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	173	345	.0439	-66.2
1	175	350	.0445	-36.4
2	188	375	.0477	+8.7
3	—	—	—	56.0
4	188	375	.0477	99.1
5	178	355	.0451	142.1
6	175	350	.0445	185.6
7	178	355	.0451	227.6
8	393	785	.0998	268.7
9	805	1610	.2047	311.2
9.50	1028	2055	.2613	332.1
10	1330	2660	.3382	353.2
11				
12				
13				
14				
15				
16				
16:40				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-IX. BEARING NO. 10

TEST DATE 5/23/79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	255	510	.0648	-72.3
1	305	610	.0776	-56.8
2	438	875	.1113	-17.2
3	395	790	.1004	—
4	345	690	.0877	78.9
5	283	565	.0718	127.1
6	250	500	.0636	172.3
7	225	450	.0572	219.2
8	203	405	.0515	263.7
9	205	410	.0521	300.9
10	213	425	.0540	341.9
11	218	435	.0553	380.3
12	218	435	.0553	416.3
13	208	415	.0528	450.2
14	203	405	.0515	482.4
15	205	410	.0521	513.2
16	213	425	.0540	542.9
16:40	213	425	.0540	559.6
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

TABLE A-X. BEARING NO. 11

TEST DATE 5-24-79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	218	435	.0553	-69.8
1	200	400	.0509	-53.3
2	175	350	.0445	-14.5
3	185	370	.0470	35.2
4	183	365	.0464	87.6
5	168	335	.0426	139.6
6	155	310	.0394	189.7
7	143	285	.0362	237.2
8	145	290	.0369	282.6
9	138	275	.035	325.8
10	125	250	.0318	367.1
11	143	285	.0362	406.6
12	148	295	.0375	445.0
13	148	295	.0375	481.9
14	133	265	.0337	516.9
15	133	265	.0337	551.2
16	133	265	.0337	574.5
16:40	143 308	285 615	.0362 .0391	585.2
17	—	—	—	
18	288	575	.0366	597.4
19	308	615	.0391	607.4
20	308	615	.0391	613.5
21	308	615	.0391	610.6
22	295	590	.0375	602.7
23	300	600	.0381	592.5
24	—	—	—	582.2
25	288	575	.0366	574.0
26	295	590	.0375	570.1
27	300	600	.0381	574.0
28	—	—	—	580.9
29	308	615	.0391	589.0
30	323	645	.0410	596.4
31	335	670	.0426	596.6
32	355	710	.0451	593.6
33:20	370	740	.0470	587.8

TABLE A-XI. BEARING NO. 12

TEST DATE 5-24-79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	593	1185	.1507	-62.8
1	308	615	.0782	-28.9
2	238	475	.0604	+19.3
3	223	445	.0566	57.3
4	205	410	.0521	105.1
5	195	390	.0496	155.7
6	178	355	.0451	210.2
7	165	330	.0420	256.4
8	148	295	.0375	236.1
9	145	290	.0369	336.9
10	145	290	.0369	373.9
11	153	305	.0388	409.3
12	153	305	.0388	442.3
13	168	335	.0426	473.8
14	168	335	.0426	503.7
15	180	360	.0458	531.6
16	163	325	.0413	559.2
16:40	275	550	.0350	
17				583.1
18	270	540	.0343	576
19	288	575	.0366	561.3
20	300	600	.0381	547.9
21	305	610	.0388	536.1
22	330	660	.0420	525.5
23	318	635	.0404	521.6
24	310	620	.0394	521.3
25	310	620	.0394	520.9
26	308	615	.0391	521.3
27				521.2
28	313	625	.0397	521.5
29	320	640	.0407	521.8
30	325	650	.0413	522.2
31	323	645	.0410	522.6
32	328	655	.0416	523.3
33:20	303	605	.0385	524.3

TABLE A-XII. BEARING No. 13

TEST DATE 6-1-79

TIME (MINUTES)	FRICT. FORCE (POUNDS)	FRICT. TORQUE (INCH-POUNDS)	COEFF. FRICTION (μ)	BEARING TEMP (°F)
0	163	325	.0413	-66.3
1	168	335	.0426	-51.0
2	183	365	.0464	-13.4
3	188	375	.0477	+28.5
4	183	365	.0464	71.8
5	183	365	.0464	115.4
6	183	365	.0464	158.4
7	173	345	.0439	200.7
8	158	315	.0401	233.3
9	148	295	.0375	279.9
10	143	285	.0362	317.0
11	140	280	.0356	352.8
12	140	280	.0356	387.2
13	138	275	.0350	419.7
14	140	280	.0356	451.1
15	138	275	.0350	481.1
16	138	275	.0350	509.6
16:40				
17	233	465	.0296	536.7
18	238	475	.0302	555.4
19	263	525	.0333	559.2
19.5	303	605	.0385	
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33:20				

APPENDIX G

BEARING/LUBE TEST PLAN

DOCUMENT TYPE: Test Plan	TITLE: Bearing/Lube Test Plan	TPL 00920041-001
PROGRAM: SAL		

Initial Distribution:

*W. H. Appich
*A. A. Boza
*T. V. Harvey
*V. D. Hewitt
*L. F. Michon (12)
*M. D. O'Leary
*M. Sikes
*W. Spyckaboer
*E. W. Waters

Distribution Limits:

Unlimited

Distribution Authorized By: T. V. Harvey
T. V. Harvey

* Names coded by asterisk will receive complete copy.

PREPARATION DRAFT	CHECK	PROJECT	ISSUE DATE	REVISION	PAGE	
		T. V. Harvey	FEB 27 1979	®	1 OF 10	MARTIN MARIETTA CORPORATION Orlando, Florida 32855
ENGR.		SYSTEM INTEG.	REFERENCE NO.		CODE IDENT. NO.	DOCUMENT NO.
ENGR. V. D. Hewitt			F33615-74-C-5128		04939	TPL 00920041-001

Form D-78 Sep 78

Summary

1. Objective

The objective of this test plan is to set forth the procedures for the performance of the bearing/lube test conducted under Task IIC of the ASALM TD&I contract, (Contract No. F33615-74-C-5128)

2. Reason

The test is to be performed to determine the capability of corrosion protective coatings and high temperature lubricants developed by Hughes Aircraft Company under contract to AFML can withstand the ASALM temperature environments.

3. Approach

Five bearing/lube candidates will be placed in a Martin Marietta Corporation designed test fixture and subjected to radial loads and 25° oscillatory motion. During the test, the specimen temperature will be controlled to meet ASALM flight temperature profiles.

4. Governing ASALM Requirement

The motion and temperature profiles used in this test are based on actuator requirements given in drawing 63900555. These levels are based on analytical studies and wind tunnel tests conducted under Martin Marietta Corporation Technology Development and Integration contract.

5. Conclusions

N/A

1.0 Purpose and Objectives

1.1 Introduction

The Hughes Aircraft Company (HAC) is under contract (USAF Contract Number F33615-76-C-5082) to the Air Force Materials Laboratory (AFML) to study self-lubricating composite bearing materials. The purpose of the composite coating is to provide long term corrosion protection for the bearing elements. One requirement of the HAC/AFML contract is to develop a composite lubricated roller bearing that might be used in an ASALM flight vehicle control actuation system. Martin Marietta Corporation (MMC) is under contract (USAF Contract Number F33615-74-C-5128) to the Air Force Aeronautical Systems Division (ASD) to provide Technology Development and Integration for the ASALM program. Under Task IIC of the MMC/ASD contract MMC is to provide HAC with bearing requirements and to test candidate bearing combinations under simulated flight conditions.

1.2 Purpose

Roller bearings conforming to drawing number 10251122-1 Rev. B have been supplied to HAC for use in applying varying lubrication and anticorrosion coatings. This test plan sets forth the procedures, instrumentation and environments to be applied during testing of the bearing configuration candidates.

1.3 Objective

The testing of the various bearing/lube candidates is intended to provide frictional and load carrying characteristics for evaluation by HAC and to assess the ability of these candidates to meet ASALM bearing lubrication requirements.

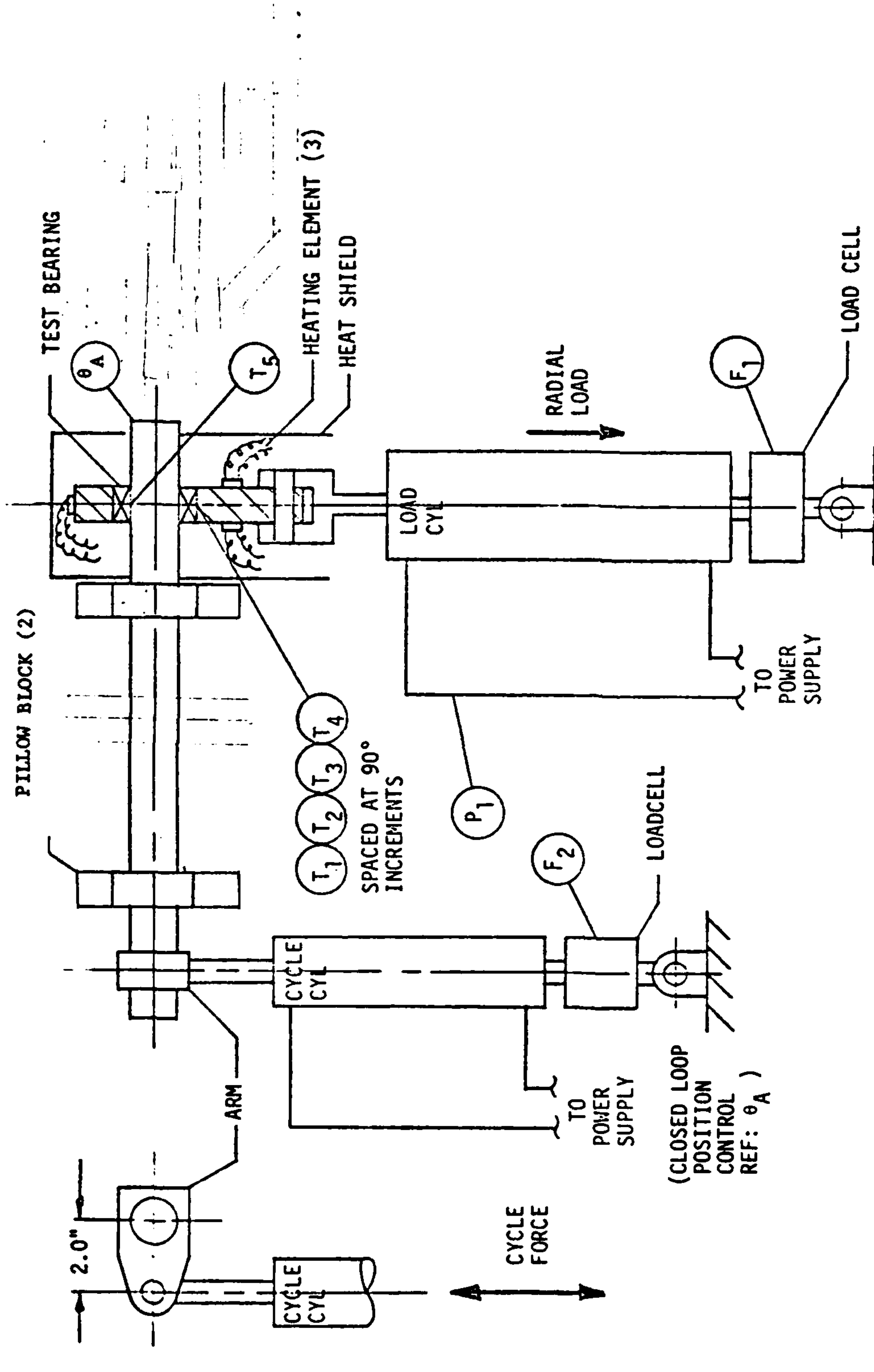
2.0 Configuration

The bearings to be tested conform to the requirements of MMC drawing number 10251122-1 Rev. B and are to be tested in the test fixture shown schematically in Figure 2.0-1.

3.0 Description of Testing

3.1 Location

All tests covered by this test plan will be conducted in the Climatic Test Laboratory of MMC located in Orlando, Florida.



TEST SET UP SCHEMATIC

FIGURE 2.0-1

3.2 Type of Test

Two bearings each of five different hardcoat/composite lubricant configurations will be tested. The first of each pair will be subjected to a simulated flight environment test. In the flight environment test, a constant (bearing static capacity) radial load will be applied while the inner race is rotated in oscillatory motion. During the 1000 second duration, the temperature of the test specimen will be adjusted to simulate a nominal long range cruise flight temperature profile. The remaining bearing of each of the five candidates will be also subjected to the flight simulation test. Upon successful completion of the flight simulation portion of test, the radial load will be increased to twice the bearing static capacity and the test continued for an additional 1000 seconds at a constant temperature of 600°F.

A summary of the bearing configurations and test conditions is given in Table 3.2-I.

3.3 Radial Loads

The rated static load of the test bearings is 7,150 lbs. This load will be applied and held constant during the flight simulation portion of the test. During the limit load tests, the radial load will be increased to 14,300 lbs.

3.4 Rates

The inner race of the bearing is attached to the test fixture and the assembly rotated in a $\pm$ 25° sinusoidal motion. The frequency of the sinusoidal motion will be controlled to provide a peak rotational rate of 250°/sec minimum.

3.5 Thermal Conditioning

The bearing temperatures shall be controlled to provide the thermal profile given in Figure 3.5-1 (TBDL). Minimum requirements indicated in the HAC SOW indicate an initial temperature (soak) of -65°F and a final temperature of 600°F and a total test duration of 1000 seconds.

TABLE 3.2-I. Test Condition/Bearing Configuration Matrix

Bearing No.	Configuration	Test Conditions		
		Parameter	Flight Simulation Test	Limit Load Test
		Time	1000 sec	1000 sec
		Temperature	-65 to +600°F	+600°F
		Radial Load	7150 lb	14,300 lb
1	TiC Hardcoat + no lube		*	*
1A	"		*	
2	TiC Hardcoat + MoS_2 + SbO_3		*	*
2A	"		*	
3	TiC Hardcoat + Westinghouse Composite		*	*
3A	"		*	
4	TiC Hardcoat + HAC Composite		*	*
4A	"		*	
5	TiC Hardcoat + Fel-Pro C-100		*	*
5A	"		*	

3.6 Failure Criteria

The failure of a bearing/lube candidate to sustain the applied load for the duration of the test shall constitute a failure. In addition, a sudden increase in the bearing coefficient of friction of greater than 100% shall indicate that the candidate is suspect and shall cause that particular test to cease to allow visual inspection of the bearing.

4.0 Test Procedure and Limits

The bearing/lube candidate will be inserted in the bearing test fixture and the fixture will be bolted to a support and load fixture. The support and load fixture is a massive beam arrangement which provides the dual functions of holding the test fixture and providing the structural support for the radial load application. Hydraulic cylinders attached to nearby structural columns will be used to provide the shaft rotary motion and bearing radial loads.

The test fixture containing the test specimen will be soaked to -65°F for four hours prior to initiation of the test. The loads given in para. 3.4, the rates given in para. 3.5 and the thermal profiles given in para. 3.6 will be applied during the test. The test will be repeated for the 10 specimens that will be provided by HAC.

6.0 Instrumentation

The bearing and bearing test fixture shall be instrumented for temperature, pressure, shaft position and load as shown in Figure 2.0-1. The purpose of each measurement channel is given in Table 6.0-I.

6.1 Data Collection

Data will be recorded on a direct write oscillograph and a fourteen channel magnetic tape. This will provide redundancy for critical measurements and allow quick-look evaluation of the test. Correlation of the time base for the oscillograph and magnetic tape will be provided for by a timing signal which will appear on both records.

6.2 Instrumentation Calibration

6.2.1 Position Transducer

The rotational position of the test fixture shaft shall be measured using a rotary potentiometer or RVDT (Rotary Variable Differential Transformer). Calibration of the transducer and signal conditioning electronics shall be based on manufacturer's data.

TABLE

INSTRUMENTATION LIST

DESCRIPTION	SYMBOL	UNITS	INSTRUMENT RANGE	TOLERANCE
TEMPERATURE - BEARING OUTER RACE	T1	°F	0 - 1000°F	±10°F
TEMPERATURE -BEARING OUTER RACE	T2		0 - 1000°F	±10°F
BEARING OUTER RACE	T3		0 - 1000°F	±10°F
BEARING OUTER RACE	T4		0 - 1000°F	±10°F
BEARING INNER RACE	T5		0 - 1000°F	±10°F
LOAD CELL - LOAD CYLINDER	F1	LBS	0 - 30000LB	±2%°F PT.
CYCLE CYLINDER	F2	LBS	0 - 500LB	±2%°F PT.
LOAD CYLINDER PRESSURE	P_1	PSIG	0 - 5000 PSI	±2%
SHAFT CYCLE AMPLITUDE	θ_A	DEG.	±30°	±1/2°

6.2.2 Thermocouples

Thermocouples will be used to determine critical temperatures of the test fixture and bearing test candidate. The thermocouples shall be calibrated against an ice point reference.

6.2.3 Forces

The load and cycle hydraulic cylinder forces will be measured using strain gage instrumented force dynamometers (load cells). Calibration of the load cells shall be based on manufacturer's data.

6.2.4 Load Cylinder Pressure

The load cylinder hydraulic pressure will be measured using a strain gage instrumented diaphragm pressure transducer. Calibration of the pressure transducer shall be based on manufacturer's data.

7.0 Test Installation

The bearing test fixture will be mounted with the shaft in a horizontal position. The fixture will be mounted in a massive beam frame to prevent rotation of the fixture and to react the loads imposed by the hydraulic load cylinders. The test will be conducted in the MMC Climatic Test Laboratory.

8.0 Data Reduction

The data which is obtained in performing these tests will be analyzed to determine that the bearing/lube candidate is performing as desired. The Test Instrumentation section of this test plan indicates the parameters to be measured, methods used, and storage of the test data.

The data which has been displayed on the oscillograph recorder will be analyzed to determine that proper temperature, position and load profiles (time histories) had been experienced. A "quick-look" capability will afford minimum data analysis time for appropriate corrective action to be taken.

The data, reference section 6.0, which is displayed on the oscillograph record is also stored on 14 channel magnetic tape. At the completion of the test run, the magnetic tape may be fed, at a reduced speed, into an X-Y plotter. In this manner, plots of the following variables may be presented:

1. Bearing temperature vs. time
2. Bearing load vs. time
3. Bearing friction load vs. time

9.0 Safety

9.1 Safety Factor

The test fixture used in this test will be designed with a safety factor of 400% on limit load.

9.2 Safety Procedures

Procedures related to personnel safety will be developed by Engineering and approved by Martin Marietta Safety. These procedures will ensure that no hazard is created by test of the EMAS control actuator.

APPENDIX H

ADVANCED LUBRICANT EVALUATION TEST REPORT FOR MOOG MODEL 17E319 SERVOACTUATOR

ADVANCED LUBRICANT EVALUATION

TEST REPORT

FOR

MOOG MODEL 17E319 SERVOACTUATOR

MDAC PO Y6E262

Moog Report No. MR T-2649

May 29, 1979

MOOG INC., EAST AURORA, NEW YORK 14052 716/652-2000 TWX-710 264 1442 TELEX 91-399

ADVANCED LUBRICANT EVALUATION

TEST REPORT

FOR

MOOG MODEL 17E319 SERVOACTUATOR

Moog Report No. MR T-2649

May 29, 1979

Prepared by: G. L. Chenault
G.L. Chenault
Sr. Project Engineer

TABLE OF CONTENTS

1.0 INTRODUCTION & SUMMARY

A high temperature test program was conducted by Moog Inc. to evaluate the performance of the shaft bearings in the Moog Model 17E319 Servoactuator using two different advanced lubricants. A load simulator was used to generate shaft root bending moments and torsional hinge moments. Electrical heaters were used to heat the shaft and actuator mounting pads.

The original bearing grease, used during the original bearing development tests of April, 1978, was Braycote 3L-38RP. The first advanced lubricant tested was a high temperature grease provided by the Air Force Materials Laboratory and designated MCG-70852542. This grease is referred to hereafter as A.F. Grease. The second advanced lubricant tested was a Dry Film Lubricant applied to the bearings by MDAC - St. Louis.

Neither of the advanced lubricants performed as well as the original Braycote Grease. During the high temperature duty cycle, the maximum flight actuator friction with the Braycote Grease was 460 in-lb at the end of the 33 1/3 minute test. The flight actuator friction reached 770 in-lb at 20 minutes with the A.F. Grease under the same test conditions. During the high temperature duty cycle with the Dry Film Lubricant, flight actuator friction reached 1660 in-lb at 22 minutes.

2.0 TEST SETUP

The test setup was identical to that shown in Moog Report MR T-2546 "Bearing Development Test Report" except for some minor changes in the temperature channels and oscillograph scaling. The data channels are identified in Tables 2-1 and 2-2.

TABLE 2-1

OSCILLOGRAPH TRACE IDENTIFICATION

ITEM NO.	DESCRIPTION	CALIBRATION	ZERO POSITION
1	Root Bend Actuator ΔP	500 psi/in	1
2	Hinge Moment Actuator ΔP	1000 psi/in	4
3	Flight Actuator ΔP	1000 psi/in	3
4	Actuator LVDT	11.93°/in	4
5	Shaft Position (Master Pot)	10°/in	5
6	Command	11.93°/in	4
7	Valve Current (Error Signal)	20 ma/in	6½
8	Supply Pressure	1000 psi/in	1
9	Return Pressure	200 psi/in	1
10	Flight Actuator ΔP Minus Hinge Moment Actuator ΔP	500 psi/in	6

TABLE 2-2

TEMPERATURE RECORD IDENTIFICATION

CHANNEL NO.	DESCRIPTION
1	Forward Mounting Pad (Control)
2	Forward Mounting Pad (Limiter)
3	Forward Mounting Pad
4	Aft Mounting Pad (Control)
5	Aft Mounting Pad (Limiter)
6	Aft Mounting Pad
7	Ambient
9	Outboard Bearing - Inner Race
10	Shaft Limiter
12	Inboard Bearing - Outer Race
14	Outboard Bearing - Outer Race
15	Shaft (at Heater Lead End)
22	Hydraulic Fluid Supply
23	Hydraulic Fluid Return

3.0 FAILURE CRITERIA

The flight actuator friction limit of 400 in-lb allows for a 3 percent safety margin on meeting the power point of 3100 in-lb at 150 degrees/second. The actual measurement of this friction level was complicated by the test fixture friction. Extensive friction tests, at 40,000 in-lb root bending moment, indicated 180 in-lb of friction within the hinge moment actuator. Thus, this number was added to the 400 in-lb for a total of 580 in-lb as the basic friction limit.

Under the above load conditions with either of the new bearing lubricants, friction was already about 81 percent of 580 in-lb at room temperature.

For example:

Braycote Grease	320 in-lb
A.F. Grease	475 in-lb
Dry Film Lube	460 in-lb

It was decided to continue the temperature test until the 400 in-lb limit tripled. This would give a stopping point of:

$$(3 \times 400) + 180 = 1380 \text{ in-lb}$$

The flight actuator ΔP and hinge moment actuator ΔP were subtracted electrically and displayed on an oscilloscope. This eliminated the effects of pressure transients in the hinge moment actuator servoloop. This pressure trace was also recorded on the oscillograph. This trace was read with the actuator passing through null in each direction for an accurate reading of friction. Readings at the $\pm$ 8 degree position would contain some geometrical error due to shortening of the moment arm and high readings at actuator turn-around due to burned grease having a "ramp" effect on the bearing rollers. For example: The A.F. Grease test was stopped when the oscilloscope showed peak-to-peak readings of 1600 in-lb. The friction through null, determined by later analysis of the oscillograph record, showed 950 in-lb.

4.0 TEST PROCEDURE

Prior to each of the two new-lubricant tests, a pressure decay test was performed to check for external leakage. The actuator was filled with M2-V fluid and pressurized to 100 psig with the return port blocked. The time to decay to 90 psig was then measured. The test was repeated without the actuator attached to account for any leakage in the hydraulic supply circuit. The pressure decay from 100 psig to 90 psig was found to be equivalent to 5 drops of hydraulic fluid, by actual test.

The Bray Grease test was run with actuator S/N 1. The A.F. Grease and Dry Film Lube tests were run with actuator S/N 3. The actuator was installed in the load simulator and subjected to the duty cycle of Table 5-2. The temperature of the shaft sleeve, just under the outboard bearing inner race, was used as the reference point to establish the required temperature profile (Channel 9 on the temperature record).

The test plan called for a linear temperature rise from 410°F to 720°F in 1200 seconds. Temperature was then stabilized to 720°F + 25°F - 50°F for an additional 800 seconds while operating at Event 10 of the duty cycle.

The flight actuator ΔP and hinge moment ΔP were subtracted electrically and monitored continuously on an oscilloscope.

5.0 RESULTS & CONCLUSIONS

5.1 BRAYCOTE GREASE (3L-38RP)

The results, of the original bearing test using Braycote Grease, are presented in Tables 5-1 through 5-3 and Figures 5-1 through 5-4.

Table 5-1 lists the friction torque measured prior to the high temperature test. As stated in Section 3.0, 180 in-lb of friction was established as the friction level assignable to the hinge moment actuator with 40,000 in-lb root bending moment applied. Thus, the flight actuator friction (for Data Reference 216) is: 320 - 180 = 140 in-lb.

Figure 5-1 is a sample of the oscillograph records taken throughout the.high temperature test. This record was taken during Event 9 of the duty cycle.

Table 5-2 shows the actual duty cycle achieved along with friction levels and bearing race temperature. The friction levels were read at the end of each duty cycle event so that temperatures would be at their highest value for each case. The flight actuator friction values at 41,560 in-lb, were calculated by subtracting 180 in-lb from the ΔP values listed in the previous column of Table 5-2. At the end of the flight duty cycle, the flight actuator friction was 355 in-lb compared to 400 in-lb maximum allowable. At the end of the extended cycling test, the flight actuator friction was 460 in-lb compared to 540 in-lb allowable.

The temperatures of all four bearing races are plotted versus test time in Figure 5-2. The shaft temperature started at 1000°F, reached 1265°F in 19 minutes and was held near this level for the remainder of the test. Shaft heater input power was 1278 watts during the ramp and 765 watts during the extended test. Both actuator mounting pads were held constant at about 300°F.

Table 5-3 lists the friction levels measured after the hot test. Compared to Table 5-1, no significant increase in friction was evident. The results were within the repeatability limits normally seen during these tests. (Approximately 4 hours of high load friction tests were run on these bearings, at room temperature, trouble-shooting the actuator and test fixture during the 5 week period preceding the hot test.)

Following the friction tests the actuator was disassembled and the bearing races inspected. Very minor burnish marks from the rollers were evident on the bearing races. Figure 5-3 and 5-4 show profile plots of the inner race of the outboard and inboard bearings respectively. The large deviations are merely surface runout. The small deviations are roller indentations. These appear to be a little greater on the inboard bearing and are approximately 10 to 15 microinches. (A more sensitive plotter scale could not be used because of surface runout.)

Standard long-life roller bearing design criteria states that indentations greater than 0.0001 times the roller diameter will cause eventual failure. For this bearing design, the limit would be 13 microinches. Thus, this bearing will not last indefinitely under load. However, in view of its endurance record to date, the bearing design is more than adequate to meet the 20 minute duty cycle.

TABLE 5-1

FRICTION LEVELS BEFORE HOT TEST

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	160	202
10,000	0	0	183	203
20,000	0	0	180	204
30,000	0	0	195	205
40,000	0	0	215	206
0	0	3000	257	207
10,000	0	3000	243	208
20,000	0	3000	275	209
30,000	0	3000	295	210
40,000	0	3000	330	211
0	3100	3000	240	212
10,000	3100	3000	250	213
20,000	3100	3000	280	214
30,000	3100	3000	290	215
40,000	3100	3000	320	216

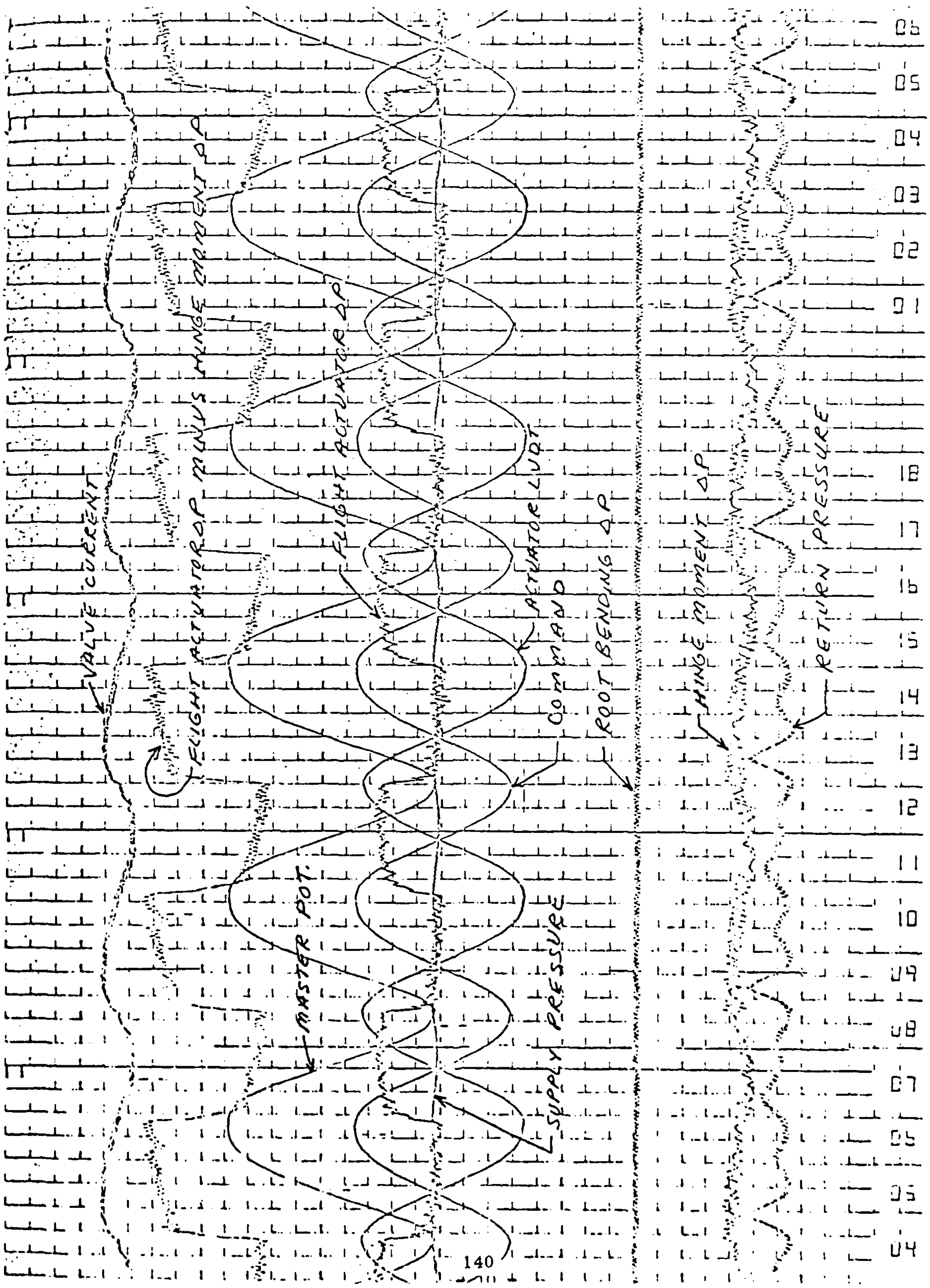

VALVE CURRENT
FLIGHT ACTUATOR ΔP MINUS HINGE MOMENT ΔP
FLIGHT ACTUATOR ΔP
ACTUATOR LVDT
COMMAND
ROOT BENDING ΔP
HINGE MOMENT ΔP
RETURN PRESSURE
MASTER POT.
SUPPLY PRESSURE

TABLE 5-2

ACTUAL DUTY CYCLE & FRICTION LEVELS DURING HOT TEST

	Event	Total Time (Minutes)	Time Increment (Seconds)	Cycling Frequency (Hz)	Cycling Amplitude (degrees)	Hinge Moment (in-lb)	Root Bending Moment (in-lb)	Flight Actuator ΔP Minus Hinge Actuator ΔP (psid)	Flight Actuator Friction at 41,560in-lb (in-lb)	Bearing Race Te (Channe #) (°F)
Flight Duty Cycle	1	.033	2	3	± 14.65	2040	20,110	360		412
	2	.100	4	30	± 1.15	-	10,055	-		413
	3	.150	3	2	± 19.8	1550	25,025	355		415
	4	1.483	80	2	± 1.85	950	9,831	315		435
	5	1.533	3	1	± 19.8	1020	10,055	300		435
	6	13.367	710	1	± 1.10	106	4,022	340		635
	7	13.417	3	1	± 19.8	990	10,055	320		636
	8	14.083	40	1	± 1.10	1930	10,055	320		645
	9	16.667	155	1	± 8.4	2450	29,941	370		677
	10	17		2	± 8.4	3090	41,560	450	270	680
Extended T		18						490	310	691
		20	200					535	355	710
		22						550	370	728
		24						540	360	745
		26						580	400	725
		28						610	430	725
		30						590	410	740
		32						650	470	750
		331/3	800					640	460	757

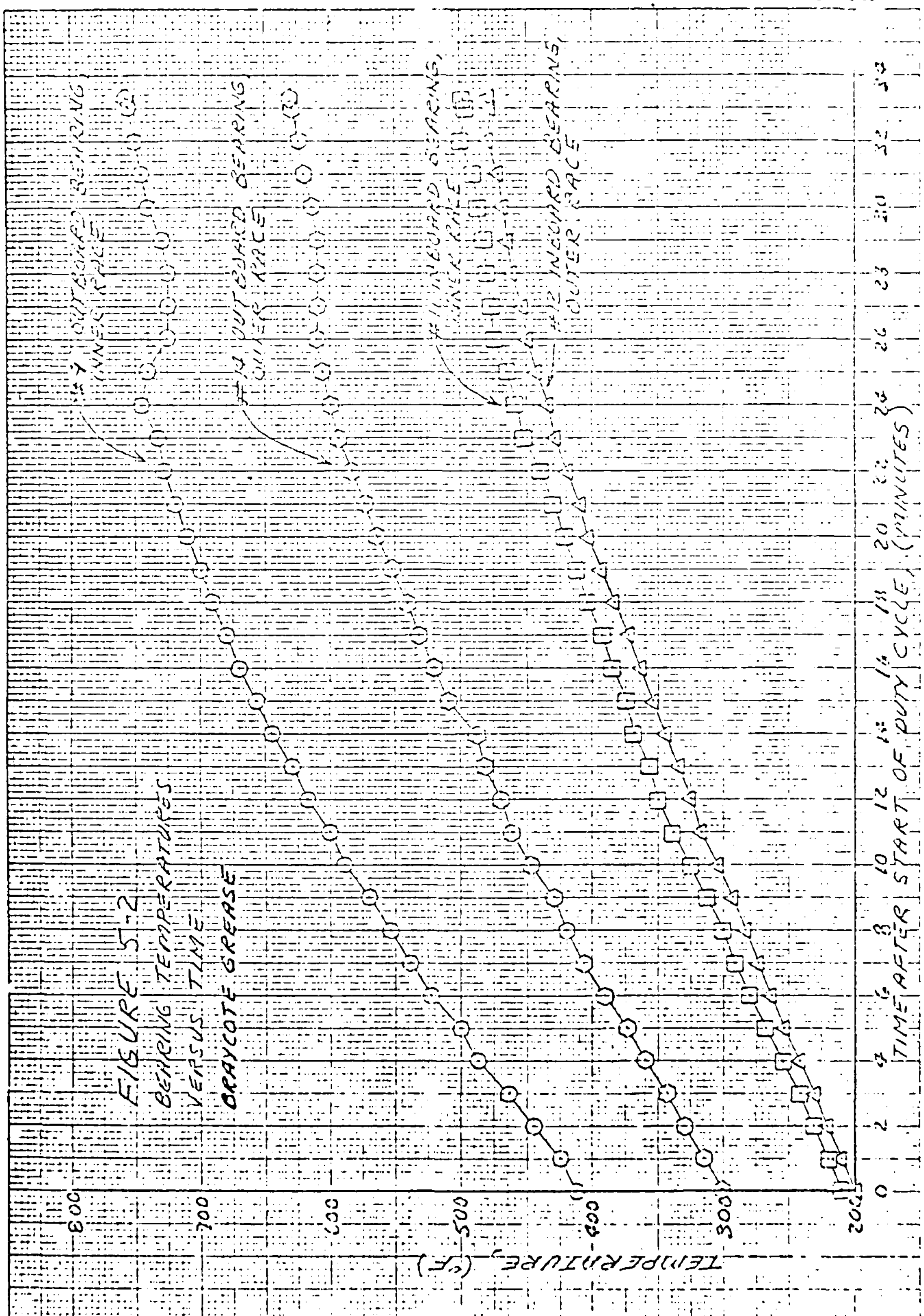


FIGURE 5-2
BEARING TEMPERATURES
VERSUS TIME
GRAYCOTE GREASE

TABLE 5-3

FRICTION LEVELS AFTER HOT TEST

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference	% Change From Table 5-
0	0	0	125	217	-22
10,000	0	0	150	218	-18
20,000	0	0	150	219	-16.7
30,000	0	0	187	220	-4.1
40,000	0	0	240	221	11.6
0	0	3000	273	222	6.2
10,000	0	3000	280	223	15.2
20,000	0	3000	285	224	3.6
30,000	0	3000	320	225	8.5
40,000	0	3000	348	226	5.5
0	3100	3000	270	227	12.5
10,000	3100	3000	260	228	4.0
20,000	3100	3000	290	229	4.6
30,000	3100	3000	310	230	6.9
40,000	3100	3000	340	231	6.3

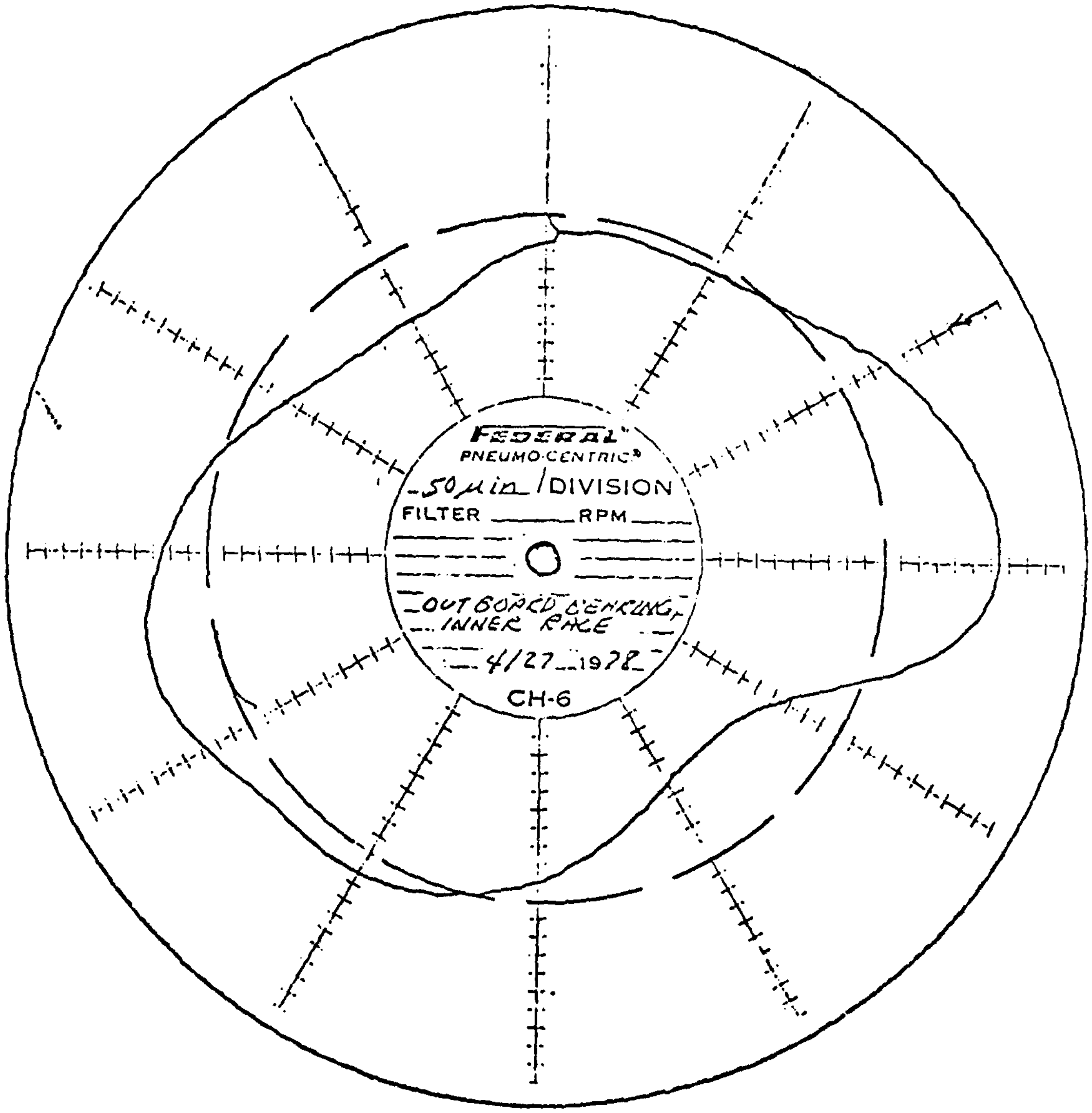

FEDERAL
PNEUMO-CENTRIC
50μin /DIVISION
FILTER
RPM
OUTBOARD BEARING,
INNER RACE
4/27 1978
CH-6

FIGURE 5-3

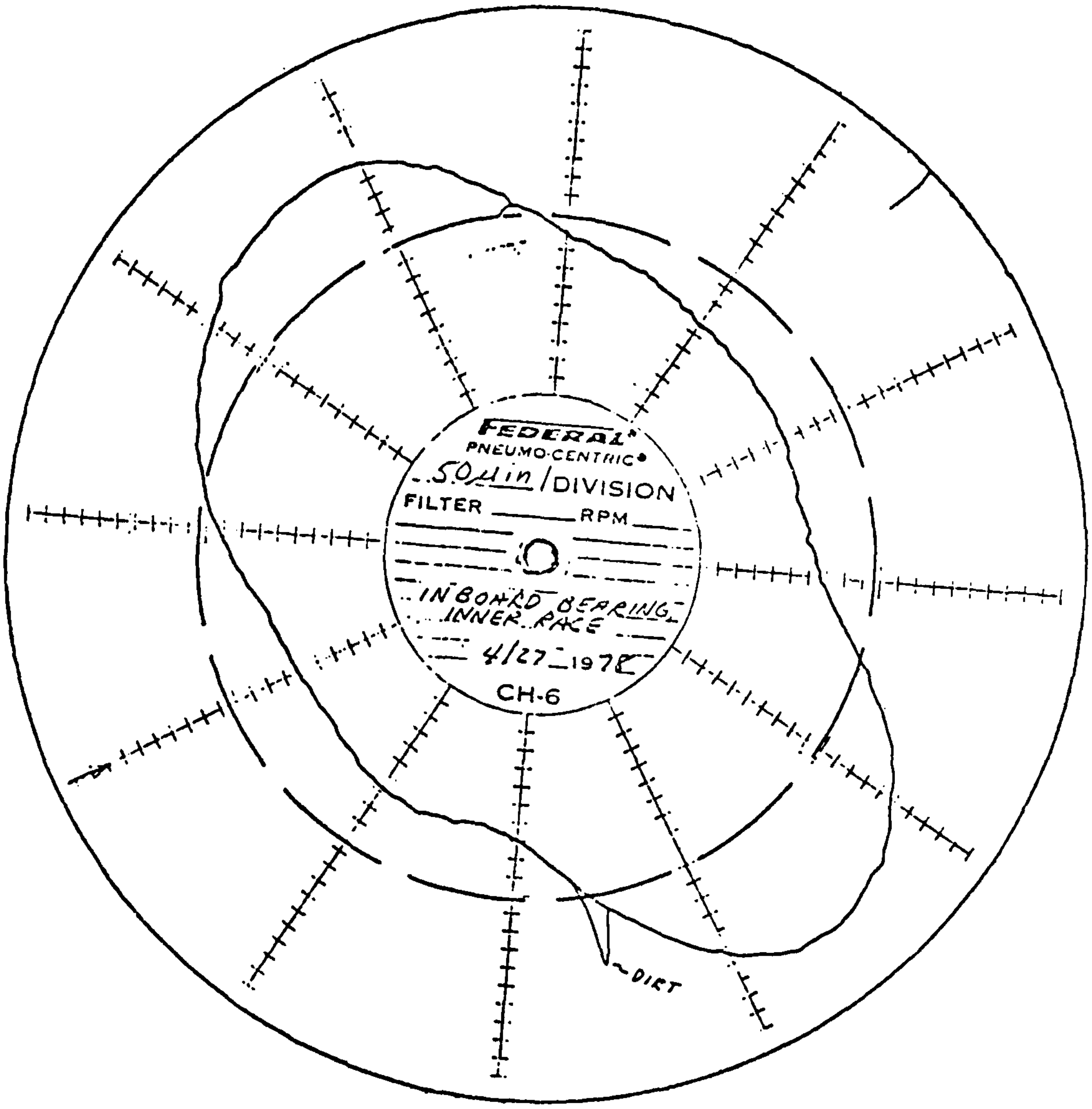

FEDERAL
PNEUMO-CENTRIC
.50μin /DIVISION
FILTER
RPM
INBOARD BEARING
INNER RACE
4/27 1978
CH-6
DIRT

FIGURE 5-4

5.2 A.F. GREASE (MCG 70852542)

When S/N 3 actuator was first tested at room temperature, the full load friction was about 600 in-lb. Considerable test time was spent trying to isolate this problem. As careful measurements of the test fixture showed no abnormalities, it was decided to test the difference between the Braycote Grease and the A.F. Grease at room temperature. The original development test bearings from S/N 1 actuator were cleaned up, filled with Braycote grease, and installed in S/N 3 actuator. The friction results are shown in Table 5-4 (and include the 180 in-lb of hinge moment actuator friction). Friction at full load was 360 in-lb (Data Ref. 81). The S/N 1 bearings were then cleaned, filled with A.F. Grease, and re-tested in S/N 3 actuator. These friction results are shown in Table 5-5. Full load friction had increased to 590 in-lb (Data Ref. 96).

The new bearings with A.F. Grease were installed in S/N 3 actuator. The actuator assembly was then tested for external leakage as described in Para. 4.0. Leakage was 4.97 drops per hour. The friction torque is presented in Table 5-6. Full load friction was 475 in-lb (Data Ref. 112).

Table 5-7 shows the actual duty cycle achieved along with friction levels and bearing race temperature At the end of the normal flight duty cycle (20 minutes), total friction reached 950 in-lb. As stated in Para. 3.0, the peak-to-peak reading on the oscilloscope indicated 1600 in-lb at this time. This was an abnormal condition not seen before. For fear of damaging the actuator/test fixture, the root bending load was removed at 22 minutes. Figure 5-5 shows the bearing temperatures versus time. The inboard bearing inner race thermocouple (Channel 11) was damaged and, therefore, is not shown. It is safe to assume that this temperature was about the same difference from Channel 12 as shown in Figure 5-2.

Table 5-8 lists the friction levels measured after the hot test. Full load friction measured to 800 in-lb. This is a 68% increase over the pre-test friction at full load.

Upon disassembly of the actuator and bearings, it was found that the grease had pretty well burned out and left behind a charred residue. The residue had a tendency to jam the roller bearings thus contributing to the high friction levels. The grease residue was sent to the Air Force for analysis.

Profile plots of the bearing inner races are shown in Figures 5-6 and 5-7. No significant differences in roller indentation were evident when compared to Figures 5-3 and 5-4 of the original bearing test.

TABLE 5-4

FRICTION LEVELS BEFORE HOT TEST

S/N 1 BEARINGS, BRAYCOTE GREASE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	175	67
10,000	0	0	168	68
20,000	0	0	182	69
30,000	0	0	207	70
40,000	0	0	260	71
0	0	3000	265	73
10,000	0	3000	280	74
20,000	0	3000	305	75
30,000	0	3000	360	76
40,000	0	3000	383	77
0	3100	3000	270	77a
10,000	3100	3000	290	78
20,000	3100	3000	320	79
30,000	3100	3000	340	80
40,000	3100	3000	360	81

TABLE 5-5

FRICTION LEVELS BEFORE HOT TEST

S/N 1 BEARINGS, A.F. GREASE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	175	82
10,000	0	0	245	83
20,000	0	0	275	84
30,000	0	0	320	85
40,000	0	0	400	86
0	0	3000	368	87
10,000	0	3000	376	88
20,000	0	3000	396	89
30,000	0	3000	450	90
40,000	0	3000	495	91
0	3100	3000	340	92
10,000	3100	3000	400	93
20,000	3100	3000	430	94
30,000	3100	3000	480	95
40,000	3100	3000	590	96

TABLE 5-6

FRICTION LEVELS BEFORE HOT TEST

S/N 3 BEARINGS, A.F. GREASE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	218	98
10,000	0	0	225	99
20,000	0	0	267	100
30,000	0	0	340	101
40,000	0	0	400	102
0	0	3000	383	103
10,000	0	3000	387	104
20,000	0	3000	444	105
30,000	0	3000	466	106
40,000	0	3000	550	107
0	3100	3000	300	108
10,000	3100	3000	370	109
20,000	3100	3000	450	110
30,000	3100	3000	450	111
40,000	3100	3000	475	112

TABLE 5-7

ACTUAL DUTY CYCLE & FRICTION LEVELS DURING HOT TEST - A.F. GREASE

Flight Duty Cycle

Event	Total Time (Minutes)	Time Increment (Seconds)	Cycling Frequency (Hz)	Cycling Amplitude (degrees)	Hinge Moment (in-lb)	Root Bending Moment (in-lb)	Flight Actuator ΔP Minus Hinge Actuator ΔP (psid)	Flight Actuator Friction at 40,442in-lb (in-lb)	Bearing Race Temp (Channel #9) (°F)
1	.033	2	3	± 15	1930	19,216	440		435
2	.100	4	30	± 1.0	-	9,384	-		436
3	.150	3	2	± 20	1530	24,131	480		437
4	1.483	80	2	± 1.9	960	9,384	325		455
5	1.533	3	1	± 20	1120	9,384	430		455
6	13.367	710	1	± 1.0	120	3,575	340		611
7	13.417	3	1	± 20	1160	9.384	470		612
8	14.083	40	1	± 1.0	1900	9,384	345		620
9	16.667	155	1	± 7.8	2440	29,047	615		661
10	17		2	± 7.8	3120	40,442	710	530	667
	18						810	630	685
	20	200					950	770	716
	22						950	770	738

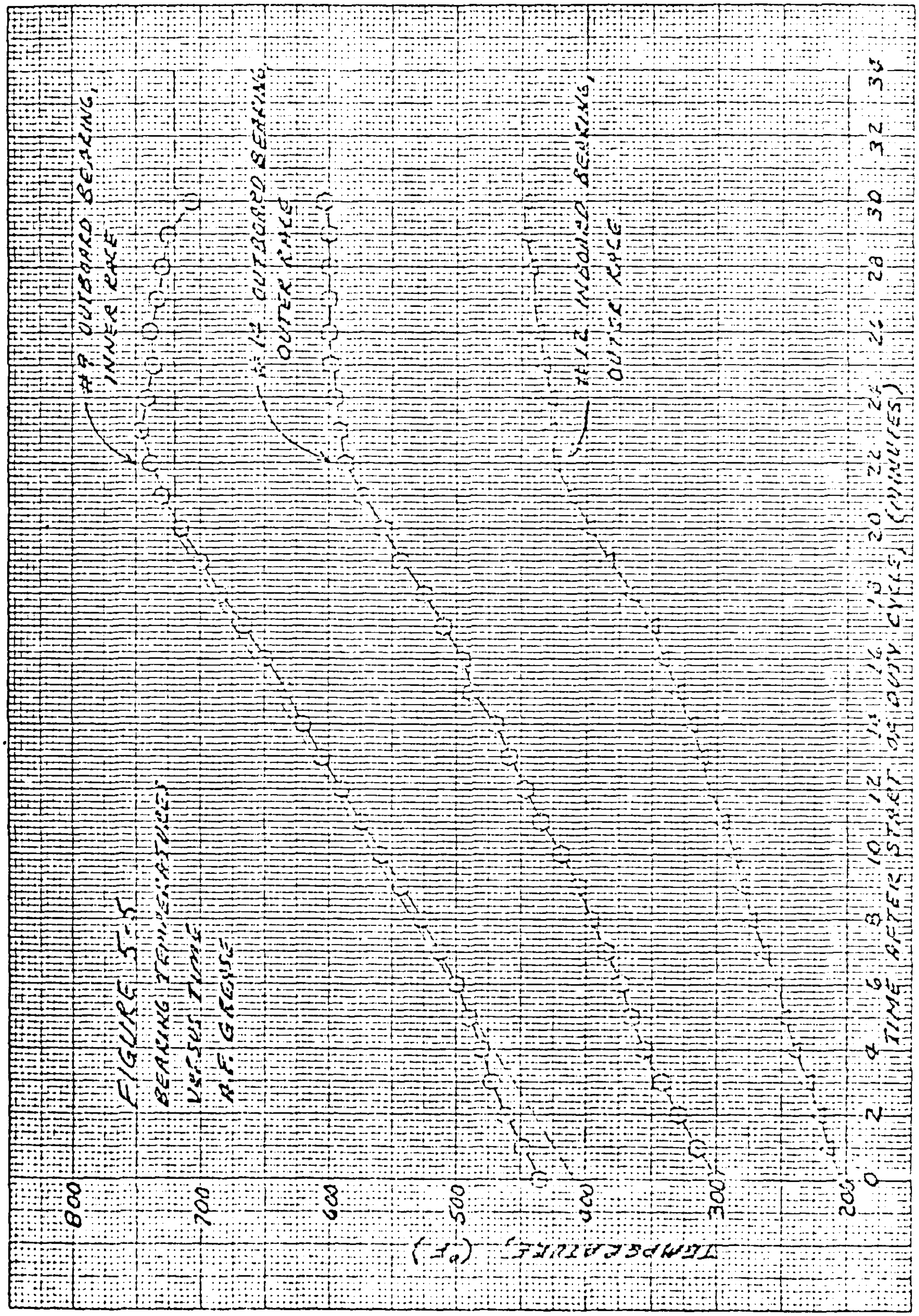


FIGURE 5-5
BEARING TEMPERATURES
VERSUS TIME
A.F. GREASE
#9 OUTBOARD BEARING,
INNER RACE
#12 OUTBOARD BEARING,
OUTER RACE
#12 INBOARD BEARING,
OUTER RACE
TEMPERATURE (°F)
800
700
600
500
400
300
200
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
TIME AFTER START OF DUTY CYCLE, (MINUTES)

TABLE 5-8

FRICTION LEVELS POST HOT TEST

S/N 3 BEARINGS, A.F. GREASE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	180	113
10,000	0	0	200	114
20,000	0	0	257	115
30,000	0	0	323	116
40,000	0	0	430	117
0	0	3000	312	118
10,000	0	3000	396	119
20,000	0	3000	490	120
30,000	0	3000	575	121
40,000	0	3000	735	122
0	3100	3000	390	123
10,000	3100	3000	450	124
20,000	3100	3000	570	125
30,000	3100	3000	680	126
40,000	3100	3000	800	127

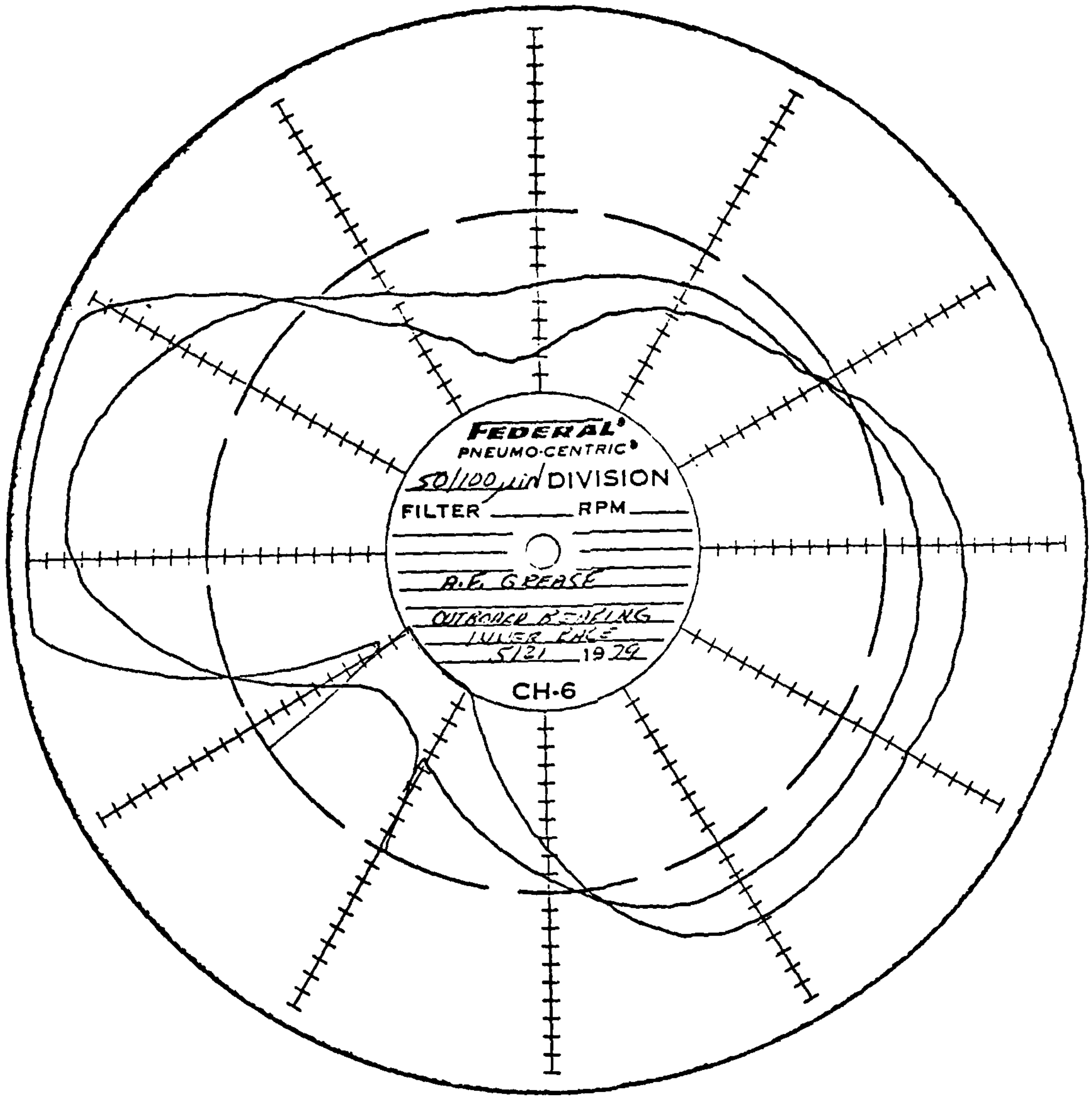

FEDERAL
PNEUMO-CENTRIC
50/100 μin DIVISION
FILTER RPM
R.E. GREASE
CH-6

FIGURE 5-6

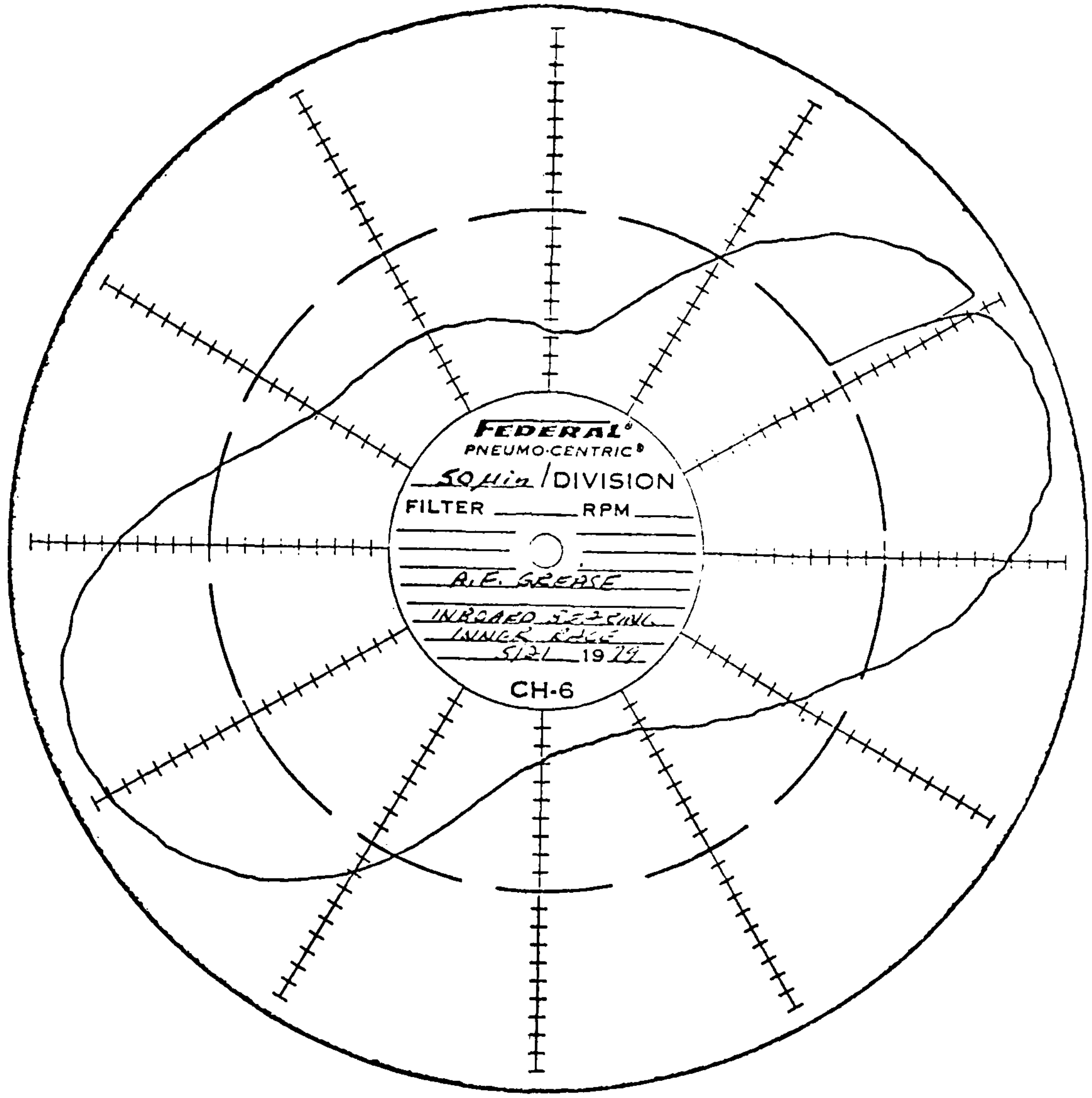

FEDERAL
PNEUMO-CENTRIC
50 μin /DIVISION
FILTER
RPM
A.E. GREASE
INBOARD BEARING
INNER RACE
5/21 1979
CH-6

FIGURE 5-7

5.3 DRY FILM LUBE

Upon disassembly after the A.F. Grease tests, the actuator pistons were lightly scored at the external dynamic sealing surface. This was caused by the 17-4 PH pistons rubbing on the bare Titanium actuator bores. The actuator bores were originally Tiodized to prevent this problem but the Tiodize coating had been worn away during the Demonstration Test Program. It was decided to hard chrome plate the piston surface for the Dry Film Lube test. After actuator assembly, the pressure decay leakage test showed a leakage rate of 3.0 drops per hour as compared to 4.97 drops per hour for the previous test.

The friction levels before the hot test are shown in Table 5-9. Full load friction was 460 in-lb (Data Ref. 19).

As this was to be the final high temperature test, it was decided to let the test run the full 2000 seconds regardless of friction levels. Table 5-10 shows the actual duty cycle achieved along with friction levels and bearing race temperature. Total friction reached 1840 in-lb at 22 minutes and had declined to 1270 in-lb at 33 1/3 minutes. This friction decrease is attributed to M2-V fluid leakage into the outboard bearing. The actuator was mounted in the test fixture with the shaft end down. This placed the outboard roller bearing at the lowest location in the actuator. Any fluid leakage, past the actuator piston dynamic seals during cycling, would eventually run through the outboard bearing and provide an automatic (though unwanted) oiling.

Figure 5-8 shows the bearing temperature versus time. The outboard bearing temperature (Channel 9) reached 790°F at 26 minutes. This is 70°F greater than the 720°F nominal specified. Of particular interest is the sudden rise of the inboard bearing temperature (Channel 12) at 16 minutes. This is attributed to a sudden increase in inboard bearing friction caused by wearing away of the Dry Film lube. (Subsequent disassembly showed the presence of dry film particles on the crankshaft next to the inboard bearing).

The friction levels after the hot test are listed in Table 5-11. Full load friction increased by 28% to 590 in-lb. The increase would probably have been higher if the outboard bearing had not been exposed to M2-V hydraulic fluid.

No abnormal conditions were found upon actuator disassembly other than the dry film particles noted above. Profile plots of the bearing inner races are shown in Figures 5-9 and 5-10. No significant differences in roller indentation were evident when compared to Figures 5-3 and 5-4 of the original bearing test with Braycote grease.

TABLE 5-9

FRICTION LEVELS BEFORE HOT TEST

DRY FILM LUBE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	188	5
10,000	0	0	200	6
20,000	0	0	245	7
30,000	0	0	290	8
40,000	0	0	380	9
0	0	3000	385	10
10,000	0	3000	363	11
20,000	0	3000	396	12
30,000	0	3000	463	13
40,000	0	3000	10	14
0	3100	3000	300	15
10,000	3100	3000	300	16
20,000	3100	3000	350	17
30,000	3100	3000	430	18
40,000	3100	3000	460	19

TABLE 5-10

ACTUAL DUTY CYCLE & FRICTION LEVELS DURING HOT TEST - DRY FILM LUBE

	Event	Total Time (Minutes)	Time Increment (Seconds)	Cycling Frequency (Hz)	Cycling Amplitude (degrees)	Hinge Moment (in-lb)	Root Bending Moment (in-lb)	Flight Actuator ΔP Minus Hinge Actuator ΔP (psid)	Flight Actuator Friction at 40,666in-lb (in-lb)	Bearing Race Temp (Channel #9) (°F)
Flight Duty Cycle	1	.033	2	3	$\pm$ 14.8	1930	18,769	435		414
	2	.100	4	30	$\pm$ 1.0	---	9,384	-		414
	3	.150	3	2	$\pm$ 20	1490	23,684	490		415
	4	1.483	80	2	$\pm$ 1.9	970	8,938	370		422
	5	1.533	3	1	$\pm$ 19.5	1030	8,938	410		422
	6	13.367	710	1	$\pm$ 1.0	100	3,128	460		615
	7	13.417	3	1	$\pm$ 19.8	1000	9,384	440		616
	8	14.083	40	1	$\pm$ 1.0	1940	8,938	440		620
	9	16.667	155	1	$\pm$ 7.8	2420	28,153	1010		658
	10	17		2	$\pm$ 7.8	3060	40,666	1240	1060	664
		18						1505	1325	678
		20	200					1730	1550	710
Extended Test		22						1840	1660	743
		24						1780	1600	775
		26						1520	1340	790
		28						1305	1125	766
		30						1310	1130	748
		32						1265	1085	743
		33 1/3		800				1270	1090	740

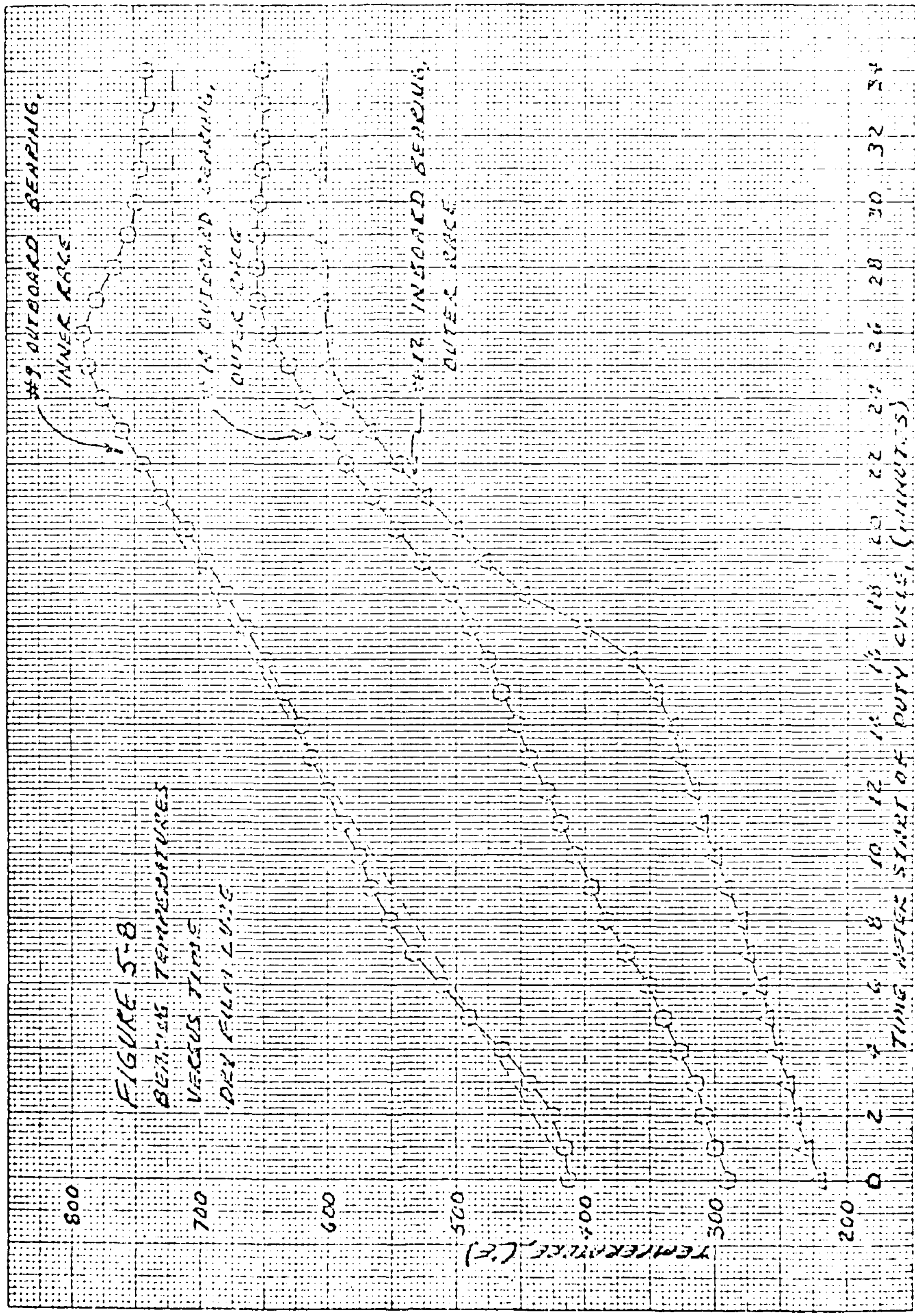


FIGURE 5-8
BEARING TEMPERATURES
VERSUS TIME
DRY FILM LUBE

TABLE 5-11

FRICTION LEVELS AFTER HOT TEST

DRY FILM LUBE

Root Bending Moment (in-lb)	Hinge Moment (in-lb)	Hinge Moment Actuator Pressure (psig)	Friction Torque (in-lb)	Data Reference
0	0	0	205	24
10,000	0	0	225	25
20,000	0	0	230	26
30,000	0	0	333	27
40,000	0	0	347	28
0	0	3000	317	29
10,000	0	3000	340	30
20,000	0	3000	405	32
30,000	0	3000	506	33
40,000	0	3000	628	34
0	3100	3000	355	35
10,000	3100	3000	360	36
20,000	3100	3000	385	37
30,000	3100	3000	520	38
40,000	3100	3000	590	39

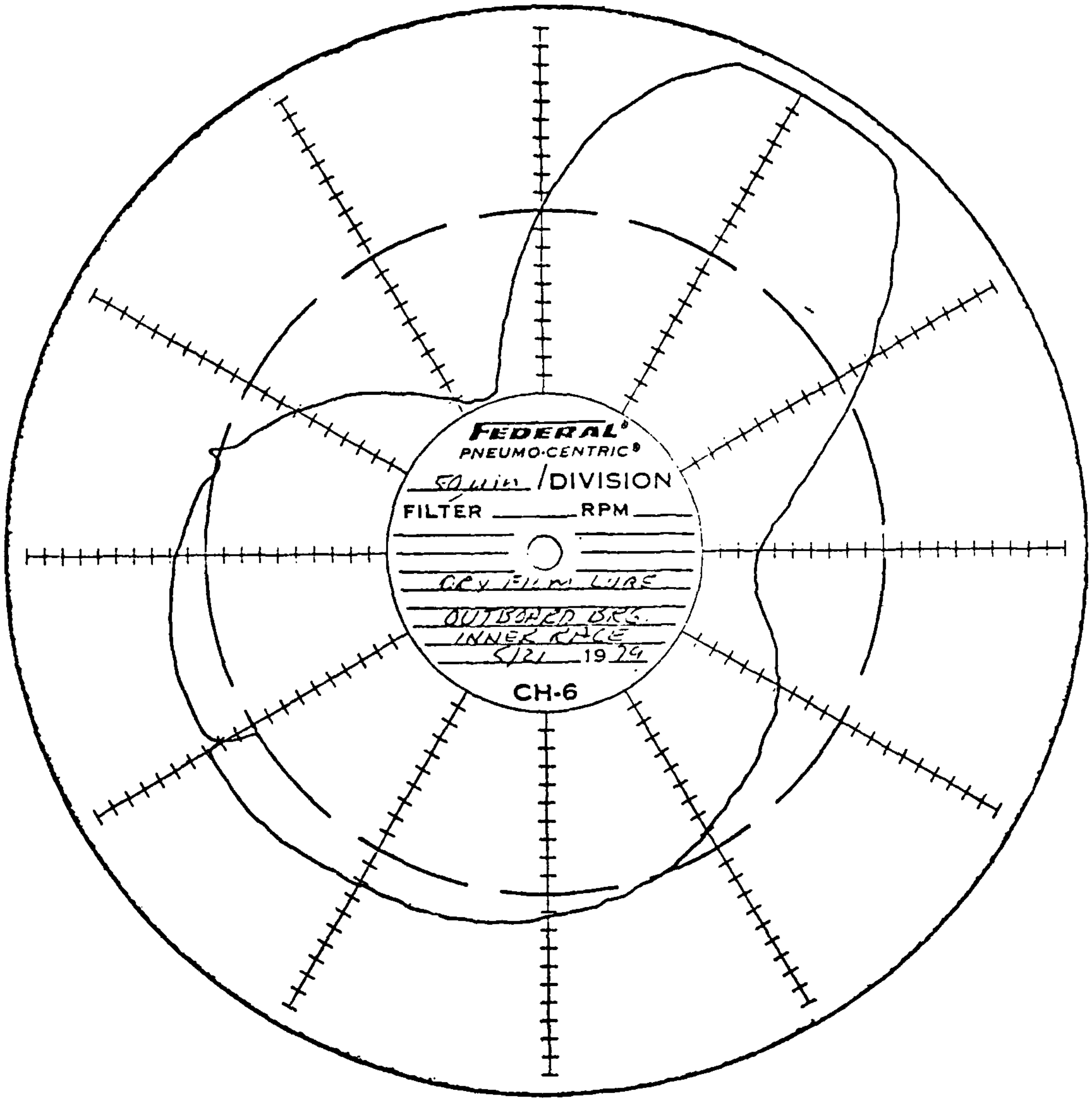

FEDERAL
PNEUMO·CENTRIC
50 μin /DIVISION
FILTER RPM
DRY FILM LUBE
OUTBOARD BRG.
INNER RACE
5/21 1979
CH-6

FIGURE 5-9

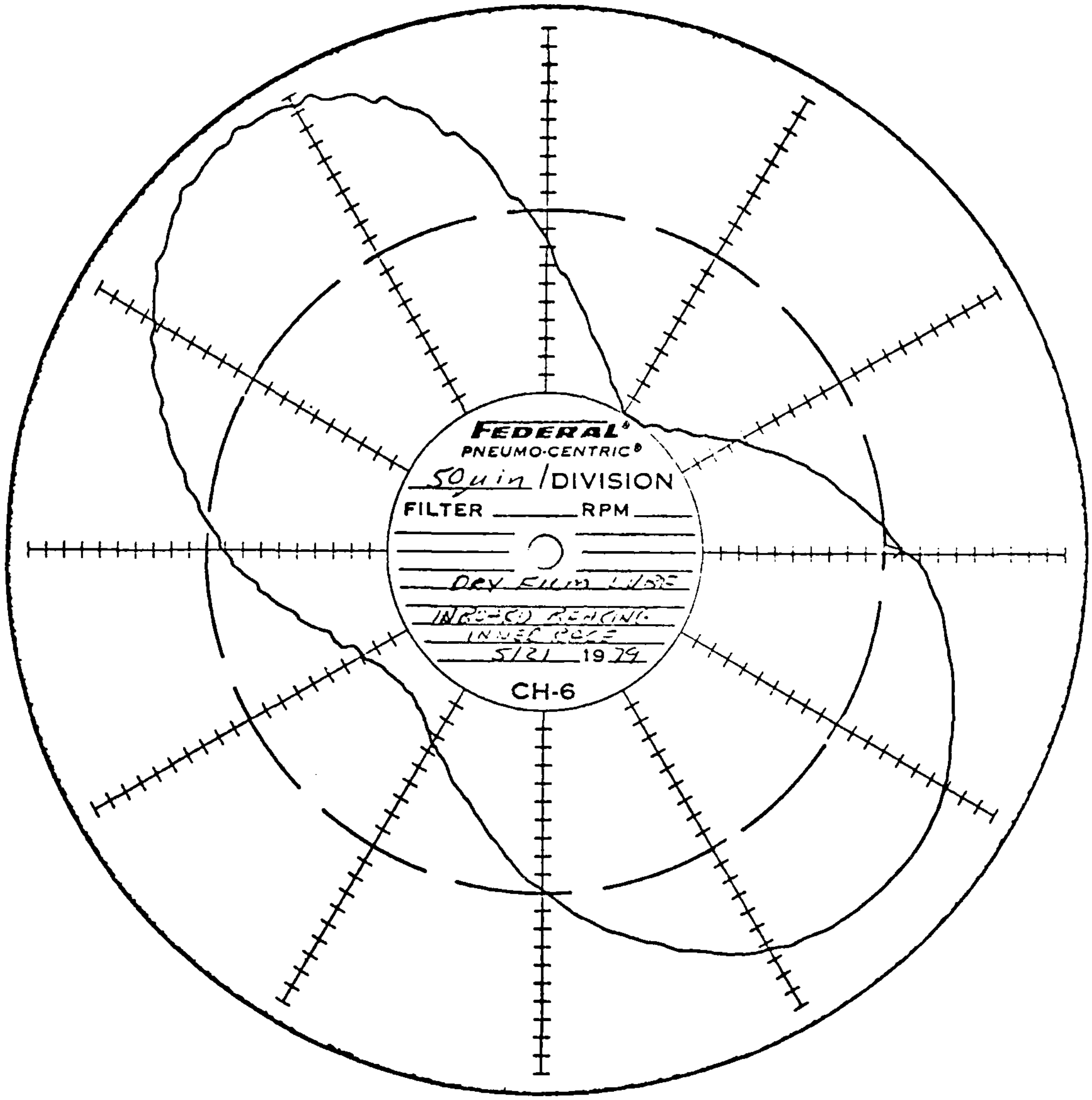

FEDERAL
PNEUMO-CENTRIC
50μin /DIVISION
FILTER RPM
DRY FILM LUBE
5/21 1979
CH-6

FIGURE 5-10

APPENDIX I

SUMMARY - BEARING LUBRICANT TESTS
ASALM CONTROL SURFACE ACTUATOR

SUMMARY - BEARING LUBRICANT TESTS

ASALM CONTROL SURFACE ACTUATOR

25 JUNE 1979

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY-ST. LOUIS
P. O. BOX 516
ST. LOUIS, MISSOURI 63166

SUMMARY - BEARING LUBRICANT TESTS
ASALM CONTROL SURFACE ACTUATOR

SCOPE

This summary compares the test results obtained from the ASALM Control Surface Actuator with two newly developed AFAML lubricants to the baseline tests using Braycote 3L-38RP lubricant. Its intent is to supplement the information presented in Moog Report, MRT-2649.

ACTUATOR DESCRIPTION

Figures 1 and 2 depict the ASALM actuator configuration. The actuator uses high pressure hydraulic fluid to power linear pistons which in turn provide rotary output to the output shaft. The output shaft is supported by specially designed BG-42 vacuum melted steel bearings capable of carrying large loads at high temperature. It is desired to provide additional corrosion resistance to these bearings through the development of lubricants tailored to this purpose.

TEST SET-UP & DESCRIPTION

All tests were conducted in the Moog, Incorporated set-up designed expressly for this purpose. Details of the test set-up and the tests using Braycote lube were reported in Final Test Report, Control Surface Actuator dated 12 February 1979. Briefly the set-up has the capability of providing external loads of zero to 3100 in-lb hinge moment and zero to 40,000 in-lb root bending moment on the fin shaft in parallel with bearing temperatures of up to 750°F. The tests with the two AFML developed lubricants are reported in Moog report MRT-2649 and a summary of all the test results included herein. The Braycote tests were conducted with Moog Model 17E319 servoactuator, S/N 1 and the A/F lubes tested in an actuator of the same model number, S/N 3. The original test actuator, S/N 1, had experienced enough wear that it was judged to be inferior to S/N 3 actuator at the time the A/F lube tests were conducted. All lube tests used the same load and temperature profiles. New bearings were utilized for each test.

LUBRICANT DEFINITION

The baseline lubricant, Braycote 3L-38RP, was selected because it was considered to possess the best corrosion resistance properties of the available high temperature lubricants. The Air Force Materials Laboratory (AFML) developed two lubricants as potential replacements for the Braycote grease. The first, also a grease, is defined as MCG-70852542 and provided by AFML. The second, a combination sputtered titanium carbide hard coat and molybdenum disulfide (MoS_2) dry film lubricant was applied per AFML procedures by Technology of Materials, Incorporated of Santa Barbara, CA. The AFML designation is AF SL-30.

FRICTION DETERMINATION

Early in the actuator test program, extensive testing was done to determine the friction levels of the various test components. The levels determined at design operating loads were 180 in-lb for the hinge moment actuator and 140 to 180 in-lb for the flight actuator. The design of the flight actuator is such that the required slew rate could be achieved with a total internal friction of ∿400 in-lb, thus allowing a considerable margin for change due to temperature, etc. The 400 in - lb margin was thus established as the allowable friction level at the end of the design duty cycle.

As an added information item, the actuator with Braycote lubricated bearings was cycled beyond the design duty cycle time for an additional 800 seconds. Had the friction during this added time exceeded 540 in-lb, the test would have been terminated. This friction level, determined arbitrarily, was intended to preclude damage to the test fixture which would influence subsequent testing. Since the testing with the AFML lubricants was the last identified use for the fixture, these tests were allowed to continue at higher friction levels. The AFML grease lubricated bearing test was terminated at ~760 in-lb but the AF-30 lubricated bearings were tested through the added 800 seconds regardless of friction since it was the last test.

TEST DATA SUMMARY

Figures 3 and 4 reflect the data taken for all three lubricant systems under a variety of conditions. The pre-test data indicates that the AFML developed lubricants tend to show higher friction levels than the baseline Braycote lubricant. Comparing the pre-test and post-test data for the Braycote indicates essentially no change due to having been subjected to the test duty cycle. However the AFML lubricants both indicate an increase in friction following the test duty cycle.

CONCLUSIONS

No significant damage was noted in the bearings with any of the three lubricants. The baseline lube, Braycote 3L-38RP, demonstrated friction levels compatible with the specification rate requirements. However, both of the AFML developed lubricants appear to contribute to friction levels which would preclude operating at specification rates.

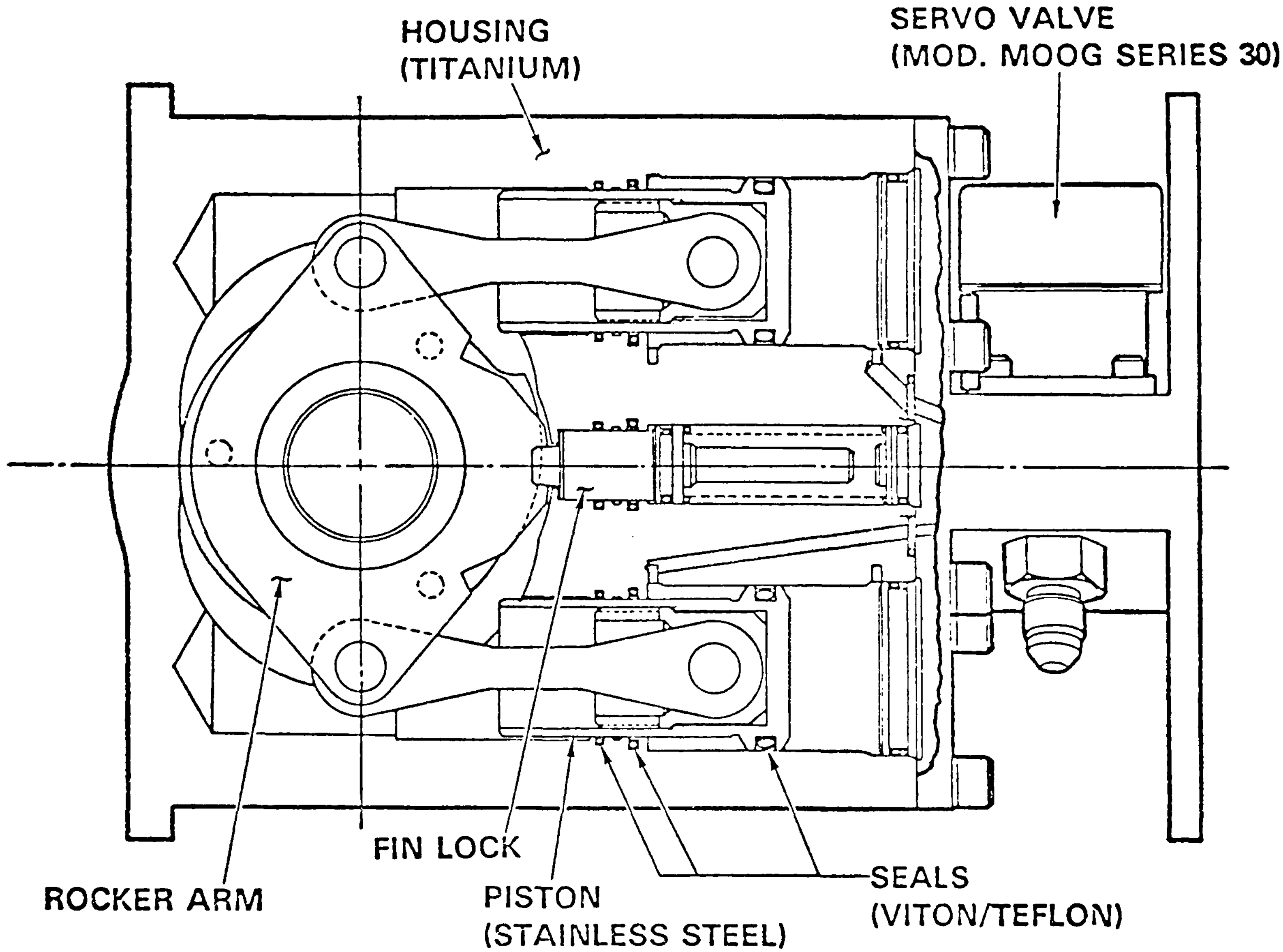


FIGURE 1 ASALM CONTROL SURFACE ACTUATOR DESIGN

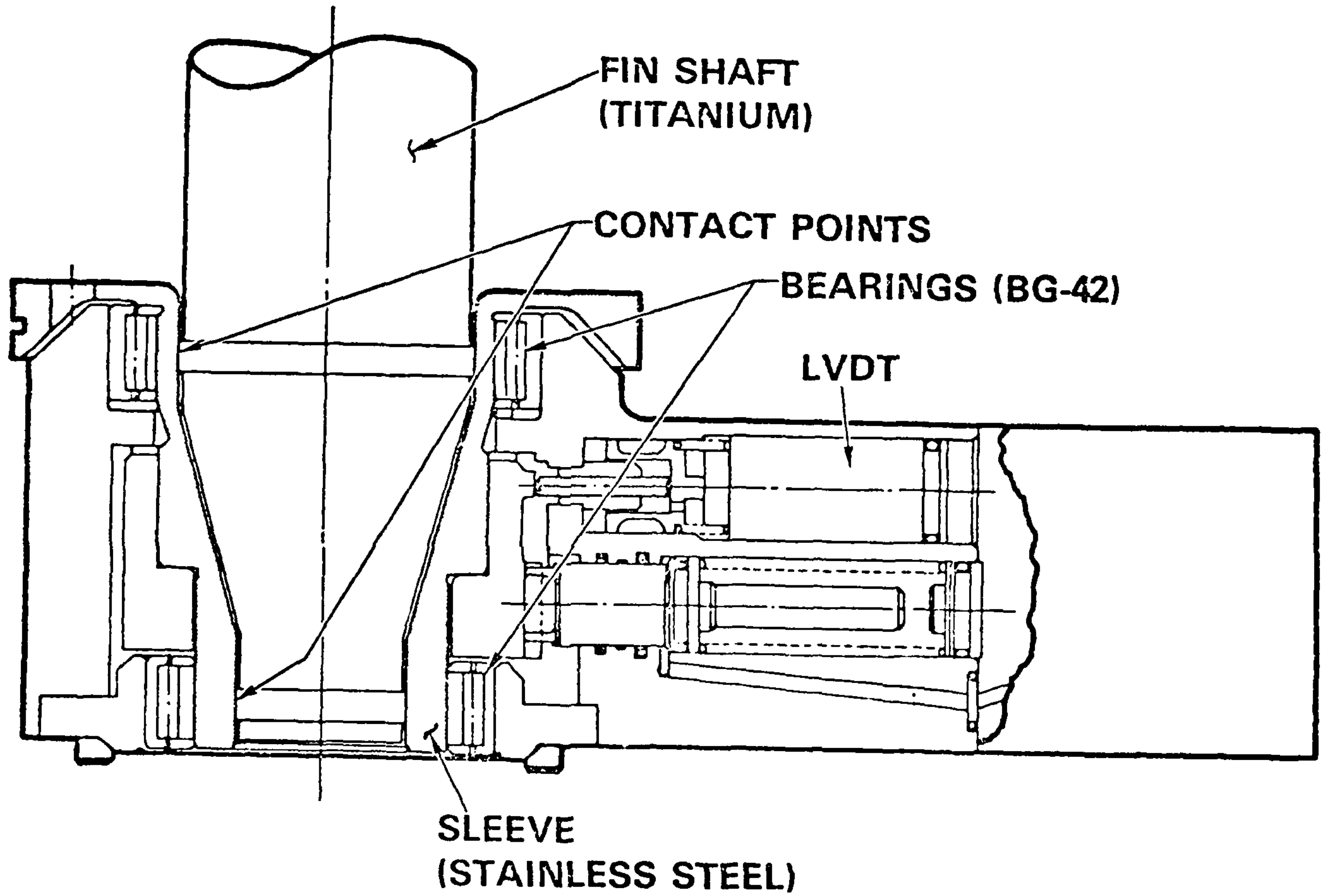


FIGURE 2 ASALM ACTUATOR DESIGN-BEARING DETAIL

TEST CONDITION	BRAYCOTE 3L-38RP		A/F GREASE MCG 70852542		A/F HARDCOAT & MoS_2, SL-30	
	PRE-TEST	POST-TEST	PRE-TEST	POST-TEST	PRE-TEST	POST-TEST
A	215	240	400	430	380	347
B	330	348	550	735	540	628
C	320	340	475	800	460	590

1. ALL VALUES ARE FRICTION IN IN-LB
2. ALL TEST CONDITIONS WITH 40,000 IN-LB ROOT BENDING LOAD ON ACTUATOR

FIGURE 3. AMBIENT TEMPERATURE FRICTION LEVELS AT SELECTED CONDITIONS

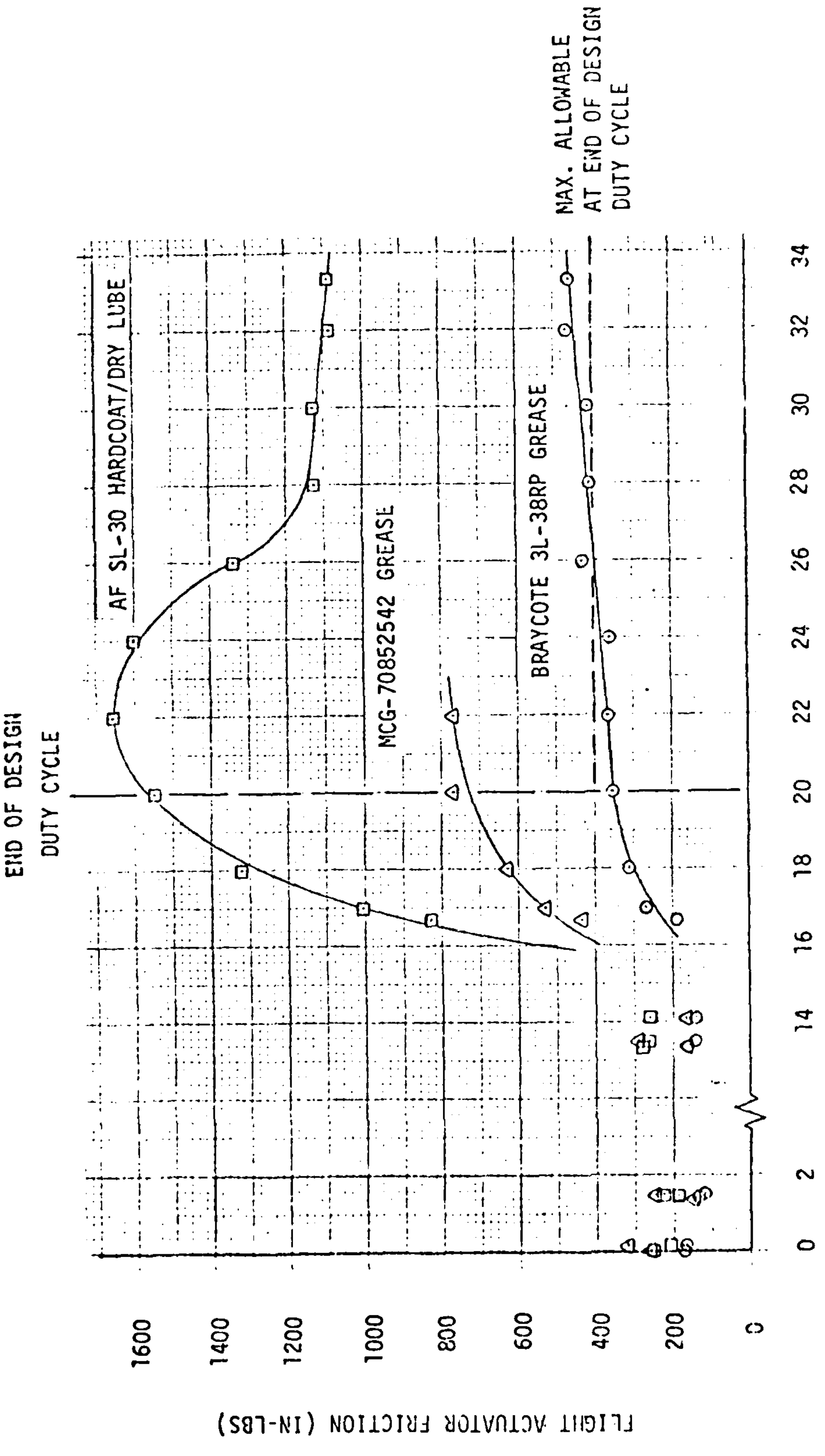


FIGURE 4 LUBRICANT FRICTION COMPARISON DURING DESIGN DUTY CYCLE

☆U.S.Government Printing Office: 1980 — 657-084/587