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OPERATIONAL EVALUATION OF DRY THIN FILM LUBRICATED BEARINGS AND GEARS FOR USE IN AEROSPACE ENVIRONMENTAL CHAMBERS

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ARO, Inc.

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FOREWORD

The work reported herein was done for the General Electric Company under Program Element 62405334/8952, Task 895212.

The results of the tests presented were obtained by ARO, Inc. (a subsidiary of Sverdrup and Parcel, Inc.), contract operator of the Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), Arnold Air Force Station, Tennessee, under Contract AF 40(600)-1000. The tests were conducted from December 7, 1963 to January 27, 1964 under ARO Project No. SN 2215, and the report was submitted by the author on December 14, 1964.

Mr. R. E. Lee, Jr., General Electric Project Engineer, contributed greatly to the planning of the tests, and Mr. Jack Craig, General Electric Test Engineer, was extremely helpful in the execution of these tests.

This technical report has been reviewed and is approved.

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ABSTRACT

This report contains the results of a test program set up to determine the operational characteristics of dry, thin film lubricated bearings and gears, uniquely adapted for operation in a simulated space environment. Results indicate that dry, thin film lubricants and soft metal plating lubricants can be applied satisfactorily to certain types of bearings and gears and are capable of sustaining heavy loads at slow speeds in space environments.

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SECTION I
INTRODUCTION

Test chambers designed for simulation of the environmental extremes of space conditions, in which large space vehicles and space system components are tested, require a wide range of mechanical support mechanisms. Many of the various support mechanisms which make up a large testing complex must be designed to undergo the same hazardous conditions to which the test vehicles and components are subjected. These support systems are repeatedly used on test after test, and must be capable of many hours of reliable service.

Two of the most critical components in any mechanism, whether it be for atmospheric or space environment application, are bearings and gears.

Unlubricated metallic bearings and gears have a tendency to adhere to one another or "cold-weld" under high loads in the high vacuum conditions of space or simulated space. Conventional oils and greases are not adequate as lubricants because they evaporate, sublimate, decompose, or migrate under the conditions of high vacuum and radiation found in the space environment. Many lubricating materials which have satisfactory life expectancies in the natural space environment, however, are not adequate in a simulated space environment because of other considerations. Two of these extra considerations inherent in space simulators are the requirements that the materials used in support of a test (1) must not excessively contribute to the gas load of the system, and (2) must not contaminate the test article.

Prior research has been performed to adapt dry powders and dry, self-lubricating materials in ball bearings for use in ground test facility hoist motors operating in a pressure range of 1×10^{-6} to 1×10^{-9} torr and at temperatures in the range of -60 to 1000°F (Refs. 1 and 2). The speed and loading of the bearings were similar to that of a 2- to 7-hp motor. Following this, a program was initiated and is now continuing to study bearing and gear lubrication systems operating under heavy loads (Hertz stresses up to 150,000 psi) and low speeds (50 to 200 rpm) under the same pressure conditions and in the temperature range of 80 to 300°F (Refs. 3 and 4).

In this study, the contractor, General Electric Co., was asked to develop (1) manufacturing specifications for the dry, thin film lubricants and (2) special design criteria for bearings and gears utilizing these lubricants (Ref. 5). The purpose of this test was to evaluate the dry, thin film lubricants developed by the contractor in both bearing and gear

applications under vacuum conditions. Load and life factors were to be the two main criteria of evaluation.

SECTION II
LUBRICANTS

2.1 THEORY OF THIN FILM LUBRICATION

The theory behind thin film lubrication is indicated in Fig. 1. Friction and wear are the results of the welding together of surfaces at their points of contact and the subsequent tearing at the welded junctions when the surfaces are slid over one another. The frictional force F equals the product of the area of real contact A and the shear strength of the junction S. When a hard metal rides over a soft one, the hard body presses or plows into the soft one so that the area of contact is large; even though the shear strength of the junction may be relatively low, the frictional force is large because of the large area of real contact. The frictional force is also large when both bodies are hard since the strength of the junctions between them is large although the contact area is small. The ideal solution to the problem of friction is to use a thin film of material with low shear strength between two hard bodies so that the hard substrate supports the load and keeps the contact area small while the film reduces the junction strength. Together, these two factors reduce the frictional force.

Films of low shear strength can be provided by using laminar solids such as molybdenum disulfide (MoS_2) that have low shear strength along certain crystallographic planes, or by using soft metals, such as gold or silver, that have low shear strength in all directions. Boundary lubrication by oils, greases, and liquid metals accomplishes the same purpose, but the lubricants are not tightly bonded to the surface.

2.2 SOLID LUBRICANTS

Molybdenum disulfide appears to be the outstanding choice for moderate temperatures and heavy loads. Graphite is included in many compositions in amounts of from 0.5 to 15 percent. The graphite, although acting alone in vacuum has abrasive properties and is a detrimental agent, nevertheless, when combined with molybdenum disulfide, seems to have a synergistic effect, but no explanation for this has been given.

2.2.1 Binder

Molybdenum disulfide is most effective, and continues to be so for a greater length of time, if it is affixed to a surface with the aid of a binder or bonding agent. In general, however, bonding agents having good adhesive properties have very poor wear and frictional characteristics. It is necessary, therefore, to find the optimum ratio of solid lubricant to binder for each solid and resin type. With the commonly used MoS_2 + graphite solid lubricant and organic resin binders, the ratio usually runs in the neighborhood of 2:1, whereas with inorganic bonding agents it runs from 4:1 to 20:1 (Ref. 5).

2.2.2 Test Coatings

Molybdenum disulfide + graphite + sodium silicate binder was tested on both spherical and cylindrical roller bearings as well as on gears. This coating is available commercially under the name Alpha Molykote® X-15.

Molybdenum disulfide plus an epoxy binder was tested on both spherical and cylindrical roller bearings as well as on gears. This coating is available commercially under the name Alpha Molykote® X-106.

Molybdenum disulfide plus glass was tested on both spherical and cylindrical roller bearings. This coating was developed by General Electric Advance Technology Laboratories.

Approximate coating thicknesses on bearings and gears tested in this program were maintained between 0.0003 and 0.0005 in.

2.3 METAL FILMS

The physical properties of soft metals come close to those desired of a good solid lubricant. Soft metals have low shear strength, good lubricity, high thermal conductivity, and can be bonded strongly to the substrate metal as continuous films.

2.3.1 Test Platings

A 23-carat gold plating was tested on both spherical and cylindrical roller bearings as well as on gears.

Silver plating was tested on both spherical and cylindrical roller bearings as well as on gears.

A copper strike of an estimated 10^{-6} inch thickness was laid down on surfaces prior to both the above platings in order to improve adherence of the plating and to improve the corrosion resistance of the substrate.

Approximate plating thicknesses were maintained at 0.0001 to 0.0002 in. on bearings and gears tested in this program.

2.4 GREASE LUBRICATION IN VACUUM

Apiezon® "L," a low vapor pressure grease, was tested on four Norma Hoffmann ball bearings. Manufacturer's specifications show vapor pressure to be 10^{-3} torr at 300°C and 10^{-10} to 10^{-11} torr at room temperature. Special Buna-N elastomer shields were installed on the bearings which served to minimize lubricant creepage.

SECTION III
TEST APPARATUS

3.1 TEST CHAMBER

The Aerospace Research Chamber (7V), Fig. 2, was used to provide the environment for this test program. The inside of the chamber is 7 ft in diameter and measures 12 ft from door flange to door flange. Both ends are provided with doors which give full 7-ft access to the chamber, as shown in Figs. 3 and 4. The pumping system for this test consisted of one end cryopanel cooled to 77°K, liquid nitrogen temperature for pumping water vapor, and two 32-in. diffusion pumps, each in series with one of two 6-in. diffusion pumps backed by a single mechanical pump. Liquid nitrogen cooled baffles were employed to retard diffusion pump oil back-streaming.

3.2 GEAR TESTER AND TEST GEARS

The gear tests were conducted in a Four-Square Gear Tester, shown in Figs. 3 and 5. The tester consisted of two Boston Gear Reductor® units coupled together in a four-square arrangement. The tops of the Boston gear box housings were machined off to insure that the desired vacuum soaking of the test gears could be attained and, at the same time, to permit visual observation of the test gears during testing.

The gear ratio was 3.2:1 with 4-in.-pitch diameter (PD) large helical gears meshing with 1.25-in.-PD helical pinions. The tester support bearings were tapered roller bearings which were lubricated with Apiezon "L."

The large 4-in.-PD gears were joined together through a flexible coupling. The pinion gears were coupled by means of a specially designed torque coupling through which a torsional preload could be introduced into the four-square system. This load, when applied, could then be locked into the system and maintained throughout the test.

3.3 BEARING TESTER AND TEST BEARINGS

The test apparatus used to house and load the test bearings can be seen in Figs. 4 and 6. In Fig. 6, a section of the housing has been removed to expose the outer two test bearings. The center block can accommodate one or two test bearings, making four the maximum number of bearings that can be tested at one time.

The bearing tester consisted of a shaft supported at each end by relatively small support (non-test) bearings. These support bearings were lubricated with similar dry, thin film coatings. Suspended on this shaft between the two support bearings were a "yoke" and "loader" which housed the test bearings. Loading of the bearings was accomplished by means of a load bolt which passed through the yoke and threaded into the loader. Thus the bearings could be loaded in opposition to one another so that a known calibrated radial load could be introduced onto each test bearing.

Bearings tested were of three basic types:

1. Spherical roller bearings, as shown in Fig. 7, were made by S. K. F. Industries, Inc. of Philadelphia, Pennsylvania.
2. Cylindrical roller bearings, as shown in Fig. 8, were made by Rollway Bearing Company, Inc. of Syracuse, New York.
3. Ball bearings, as shown in Fig. 9, were made by Norma Hoffmann Bearings Corporation of Stamford, Connecticut.

All test bearings had a 100-mm bore with an outside diameter of 180 mm and were, therefore, all compatible with the one tester.

Table I lists test bearing types and material compositions.

3.4 DRIVE SYSTEMS

The gear tester was powered by a fractional horsepower ac motor through a gear reducer, as shown in Fig. 3. Input speed through the pinion gear shaft to the gear tester was 11.7 rpm (see Fig. 5). System protection was provided by placing a shear pin in the drive shaft coupling between the motor and gear reducer.

The bearing tester was powered by a 1-hp variable speed ac motor through a gear reducer, as shown in Fig. 4. Input speed to the bearing tester was 10 rpm in the first test and 8 rpm thereafter. A slip clutch was installed between the motor and the gear reducer to protect the system from damage caused by excessive torque.

Since it was necessary that the drive mechanisms be located outside the vacuum chamber, two vacuum-tight rotary seals were developed by AEDC. These seals, one for each drive system mentioned above, each used two guard vacuum cavities around the drive shaft, as shown in Fig. 10. The cavity nearest the drive motor was maintained at a reduced pressure between 1 and 10 microns by use of a mechanical pump. The cavity nearest the tester was maintained at pressures less than 1 micron by use of a 2-in. diffusion pump backed by a mechanical pump.

The performance of both seals was excellent. Chamber pressures as low as 8×10^{-9} torr were attained and maintained for significant periods of time.

3.5 INSTRUMENTATION

3.5.1 Gear Tests

Initial preload torque on the gear tester was read out through strain gages mounted on one of the shafts on the four-square tester. Running torque could not be read on the gear tester since provision was not made for slip ring signal transmission.

Copper-constantan thermocouples were used as sensors for representative temperatures of the tester and associated pieces of support hardware; temperatures were recorded each hour. Temperatures did not seem to be affected by increasing or decreasing torque. Tester temperatures ran around 85°F with ambient chamber temperatures about 55°F.

3.5.2 Bearing Tests

Strain gages were used to measure the bending-moment in the cantilever arm of the system torque translation arrangement of the G. E. bearing tester (See Fig. 6). This bending-moment strain gage signal was amplified and the output signal was recorded continuously. The instrumented cantilever arms were calibrated to measure torque in lb-in.

Radial load on the bearings was read out through strain gages mounted on the load bolt. Copper-constantan thermocouples were used as sensors for representative temperatures of the tester and associated pieces of support hardware; temperatures were recorded each hour.

3.5.3 Test Chamber

Chamber pressure was measured with two nude ionization gages. One gage was located near the bearing tester and the other near the gear tester. The pressure near the bearing tester, in the end of the chamber near the liquid nitrogen panel, was noted to be generally half an order of magnitude lower than the pressure near the gear tester.

A mass spectrometer was used prior to, during, and immediately after the test using Apiezon "L" low vapor pressure grease to determine whether or not the chamber background of hydrocarbons was affected by the use of the grease.

SECTION IV
PROCEDURE

4.1 GEAR TESTS

Test gears and pinions were weighed before and after testing. Weight changes are noted in Table II. Each gear and pinion was photographed before and after testing (See Figs. 11 through 16).

Support bearings were pressed onto shafts and the gears and pinions were assembled in the four-square housings. Axial movement of the gears was controlled by the use of shims placed in back of the support bearing outer race retainer. Each gear, shaft, and support bearing assembly was adjusted so that no axial movement was detected.

After assembly, the tester was run in with no preload for several minutes to check alignment.

Preloading was accomplished by means of the torque coupling located in line with and between the two pinion shafts as shown in Fig. 5. Attached to one half of the torque coupling was a 3/8-inch-diameter shaft with a full strain gage bridge. The strain gage indicator was calibrated in pound-inches of torque by hanging various known weights on a known length lever arm attached to one end of the strain gage shaft while holding the other end fixed. Then, after the indicator was calibrated, a known preloading torsional force could be introduced into the system by turning one face of the torque coupling with respect to the other face and at the desired reading, the preload could be locked into the system by tightening bolts in slotted heads of the torque coupling. After establishing the desired preload, the strain gage leads were disconnected from the indicator and secured around the strain gage shaft so that the gages would not be damaged by the rotation of the shaft during the test. No provision was made for indicating torque during the test. However, the preload torque was checked after each test and it was found that no appreciable decrease in torque had occurred.

4.2 BEARING TESTS

Four test bearings were installed in the G. E. bearing tester for each test. Two bearings were placed in the loader and two were located in the yoke, one on each side of the loader, as can be seen in Fig. 6. Axial spacers were necessary for both inner races and outer races to insure proper alignment of the bearing surfaces.

Radial load was applied in these tests by means of a load bolt through the yoke and into the loader. The amount of load was determined with a strain gage load cell located within the load bolt. An alternate method of measuring load in the event of strain gage failure was by use of a torque wrench with a 4:1 torque multiplier.

System torque is defined as the summation of all the torque exerted in all four test bearings minus the resisting torque in the two support bearings, which was considered, for all practical purposes, to be negligible. This system torque was indicated by restraining the yoke and loader bearing housing assembly from rotation using the load bolt as a torque arm or transmitter. Resisting this torque was one of two separate cantilever arms on which were mounted strain gages which in turn transmitted signals indicative of the amount of bending moment in the particular cantilever (depending on the direction of rotation of the bearings). This bending moment was directly proportional to the amount of torque in the test bearing system. Calibration was attained by hanging known weights on known length lever arms attached to the bearing housing and noting the indication on the strain gage readout. The strain gages were connected to an amplifier whose output was fed to a recorder, resulting in a continuous torque record during the entire bearing test operation.

Test bearings were run in with zero radial load for approximately three minutes (25-30 cycles) in air to check alignment. After run-in of bearings, the chamber was closed and pumpdown initiated. Upon reaching 1×10^{-7} torr, the bearing test was started and system torque recorded continuously.

Tests were terminated at the end of the scheduled test duration (at least 100 hours) or when slippage of the protective drive clutch occurred. This drive clutch was set to slip at 492 pound-inches which was considered to be low enough to protect the various links in the system from damage caused by excessive torque.

SECTION V
RESULTS

5.1 GEAR TESTS

Table III shows gear test conditions and results.

5.1.1 Gear Test 1

Gears tested in gear test 1 were lubricated with a coating of molybdenum disulfide (MoS_2) plus an epoxy binder. The test ran for 71.2 hours with a premature termination because of excessive torque which caused a pin to shear in the drive shaft coupling.

Post-test analysis revealed that one of the support bearings had failed, causing most of the excessive torque. However, the coating had begun to show signs of breaking down as was determined in the post-test analysis. Approximately 60 to 80 percent of the coating was intact over the gear tooth contact area. The remaining 20 to 40 percent ranged from marginal to depleted in appearance with base metal exposed in some places. It was therefore decided not to continue testing of these gears.

5.1.2 Gear Tests 2A and 2B

Gears tested in tests 2A and 2B were lubricated with 23-kt gold and silver plating. One gear and pinion were plated with 23-kt gold and the other gear and pinion were plated with silver. Assembly was such that a gold-plated gear was meshed with a silver-plated pinion and a silver-plated gear was meshed with a gold-plated pinion.

Test 2A ran for 136.6 hours, after which it was decided to resume testing at an increased load. Visual examiniation revealed that coatings

appeared intact over the gear tooth contact area. Test 2B was run with the preload carried on the opposite side to that in test 2A. This was done by applying the torsional force in the opposite direction. Test 2B ran satisfactorily for 108 hours without failure.

Post-test analysis revealed that the major portion of the coating was intact on the set which had the silver-plated gear meshing with the gold-plated pinion. Some areas were polished with moderate pitting over a small amount of the tooth contact area. On the set with gold-plated gear and silver-plated pinion, 50 percent of the coating was still intact with the other 50 percent of the tooth contact area extremely scored and galled.

5.1.3 Gear Tests 3A and 3B

The gears tested in tests 3A and 3B were lubricated with a coating of molybdenum disulfide + graphite + sodium silicate binder.

Test 3A ran satisfactorily for 117.3 hours. Coatings appeared intact and in excellent condition after visual examination at the conclusion of test 3A and, therefore, it was decided to subject the same gears to an additional test at an increased load of 160 lb-in. preload torque. As in test 2B, test 3B was run with the load applied to the opposite face of the gear teeth as those loaded in test 3A. Test 3B ran satisfactorily for 139.8 hours without failure.

Post-test analysis revealed that the major portion of the coating was still intact after testing; however, a very small area of the coating had broken down, exposing the base metal.

5.2 BEARING TESTS

Table IV shows bearing test conditions and results.

5.2.1 Evaluation Techniques

Before stating the results of the bearing tests, the basic evaluation technique or scaling factor should be explained. Bearing manufacturers rate their bearings along several parameters, one of the most basic of which is that of maximum static radial load capacity, C_r. By definition, static radial load capacity is the maximum radial load beyond which permanent deformation in excess of 0.0001 times the diameter of the rolling element occurs. Therefore, as can be seen in Table IV, we have rated the bearing tests on the basis of a C_r/P ratio, where P is the radial load introduced or preloaded upon the particular bearing. The lower the C_r/P

ratio, therefore, the more exacting is the test on the bearing. A C_r/P ratio between one and two is not uncommon for standard fluid-lubricated bearings in atmospheric applications.

5.2.2 Bearing Test 1

Test bearing configuration in Test 1 was as follows: two Rollway cylindrical roller bearings in the loader, and two SKF spherical roller bearings in the yoke. The lubricant for all four bearings was a MoS_2 + epoxy coating.

As can be seen in Fig. 17, system torque rose rapidly from 120 to 375 lb-in. during a 5-minute atmospheric run-in period after which the tester was shut off until chamber pressure reached the 10^{-7} torr range. During the first hour of operation under vacuum conditions, the system torque dropped rapidly from 200 to 70 lb-in. Failure occurred three times with two restarts. Each restart was made in the opposite direction of rotation from that which had just previously resulted in failure. This reversal did seem to alleviate conditions for possibly an hour; however, torque increase ultimately followed each restart. Test was terminated with a total test time of 26.6 hours.

Post-test analysis indicated that a combination of marginal lubrication, buildup of coating on parts, and cage wear was responsible for the premature bearing failure. The major portion of the trouble seemed to be in the spherical bearings located in the yoke positions.

5.2.3 Bearing Test 2

Because of the trouble with the SKF bearings located in the yoke positions in test 1, the test bearing configuration was reversed in test 2 with SKF spherical roller bearings located in the loader and Rollway cylindrical roller bearings located in the yoke. Molybdenum disulfide + epoxy was again used as the lubricating coating. The radial load, however, was decreased to 4,000 lb per bearing and the speed was reduced to 8 rpm. As in test 1, initial system torque was high, around 175 lb-in. during atmospheric run in as shown in Fig. 18. After 10 minutes of operation under vacuum conditions, however, system torque had dropped to 70 lb-in. Torque continued to drop until 25 lb-in. was reached. This torque was very smooth and steady over a period of 25 to 30 hours. After 54 hours of operation, a steady rise in system torque became evident. Test termination occurred after 95.8 hours when system torque exceeded 492 lb-in. and the drive clutch slipped.

It appeared, in post-test analysis, that, again, marginal lubrication and uneven buildup of coating on parts, along with cage wear, had combined

to cause the failure. Again, indications were that the spherical roller bearings had experienced more wear than the cylindrical roller bearings. However, it did seem that the new configuration with the spherical roller bearings in the loader, the point of maximum shaft deflection, was an improvement over the previous test setup because of the self-aligning features of the spherical roller bearings.

5.2.4 Bearing Test 3

The test bearing configuration was the same as that in test 2 with the SKF spherical roller bearings located in the loader and Rollway cylindrical roller bearings located in the yoke. These bearings were lubricated by a 23-kt gold plating. As can be seen in Fig. 19, initial system torque was 35 lb-in.; however, the immediate trend was an increase in torque. Test termination occurred after 4.4 hours when drive clutch torque was exceeded (492 lb-in.).

Post-test analysis revealed that even though failure of the system occurred early, the appearance of the bearing surfaces was excellent. A slight buildup of gold between the ID of the cages and the inner race of the SKF bearings was noted; however, no actual plating failure occurred over any of the four bearings.

5.2.5 Bearing Tests 4A and 4B

Four Rollway cylindrical roller bearings were tested using MoS_2 + graphite + sodium silicate lubricant. As can be seen in Fig. 20, initial system torque in test 4A was 120 lb-in., falling off rapidly and leveling off at 10 lb-in. Torque remained at around 10 to 15 lb-in. for the remainder of the test. Test was terminated after 130 hours in order to increase radial load for an additional test. Analysis was not made between test 4A and test 4B.

As can be seen in Fig. 21, initial system torque in test 4B was 60 lb-in., dropping off gradually during the first hour to 15 lb-in. Torque remained at around 15 lb-in. for 68 hours and then increased slightly to 30 to 40 lb-in. Test 4B was terminated at the end of 100 hours, making a total of 230 hours on the same four bearings.

Post-test analysis revealed that the coating was becoming marginal in some areas with a metallic appearance over a portion of the contact area indicative of coating depletion.

5.2.6 Bearing Test 5

Four Norma Hoffmann ball bearings lubricated with Apiezon "L," a low vapor pressure grease, were tested. As can be seen in Fig. 22, initial system torque was extremely low, 25 to 30 lb-in., and very steady. Test 5 was terminated after 80 hours, but could have gone on much longer

Post-test analysis revealed that all bearing components were in excellent condition.

5.2.7 Bearing Test 6

Four Rollway cylindrical roller bearings with silver plating as the lubricant were tested in test 6. As can be seen in Fig. 23, initial system torque was 70 lb-in. with a slight decrease after the first hour. Test termination occurred after 12.5 hours because of excessive torque.

Post-test analysis revealed that the major portion of the coating was intact, with the load zone considerably polished on the races. Some light buildup of silver on the outer race of one bearing in the loader was noted. Moderate wear of the snap ring surfaces was also an apparent contribution to premature failure.

5.2.8 Bearing Test 7

Test bearing configuration for test 7 was identical to that of test 3. Although different bearings were used, the bearing types and the lubricant were the same as those tested in bearing test 3. Loading conditions were also the same. As can be seen in Fig. 24, initial system torque was 35 lb-in. with failure occurring after 7 minutes. Rotation was reversed with failure occurring again in five minutes. Another restart resulted in 14 more minutes of operation before failure.

Post-test analysis revealed that some coating wear had occurred, with light buildup on the outer races of the spherical roller bearings. Again, indications were that the life of the system was limited by the SKF spherical bearings.

5.2.9 Bearing Test 8

Test bearing configuration was as follows: Two SKF spherical roller bearings were located in the yoke, and two Rollway cylindrical roller bearings were located in the loader. Lubricant tested on all four bearings was a MoS_2 + glass coating. As can be seen in Fig. 25, initial system

torque was 35 lb-in., dropping off to 15 lb-in. after two hours (960 cycles). Relatively low torque (30 to 50 lb-in.) was maintained for the next 50 hours with some momentary peaks of from 300 to 500 lb-in. Test termination occurred after 53.3 hours when torque exceeded 492 lb-in. and the clutch slipped.

Post-test analysis again reveals that trouble in the spherical roller bearings caused failure. Thrust seems to have caused coating depletion on one side of the inner race of each of the two spherical roller bearings, and on the portion of the outer races opposite the questionable inner races. Some cage wear was experienced in both SKF bearings also. The two Rollway bearings seemed to be in excellent condition.

SECTION VI
DISCUSSION OF RESULTS

6.1 GEAR TESTS

Of the four solid lubricants tested on gears, three performed satisfactorily. They were: MoS_2 + graphite + sodium silicate binder, 23-kt gold plate, and silver plate. The MoS_2 + epoxy binder was marginal and resulted in a premature test termination.

As can be seen in Figs. 11 through 16 (after test), there seemed to be some edge loading or misalignment in some cases, resulting in uneven wear on the gears. Some self-alignment features in design would probably improve operating conditions and extend coating life.

6.2 BEARING TESTS

Of all the solid lubricants tested on bearings, only one - MoS_2 + graphite + sodium silicate binder - met the 100-hour test requirements. The MoS_2 + epoxy binder, the 23-kt gold plating and silver plating, and the MoS_2 + glass lubricants all fell short of the 100-hour test requirements.

The MoS_2 + epoxy coated bearings experienced increasing torque during atmospheric operation. After 10 to 60 minutes of operation under vacuum conditions, however, the torque dropped off to a reasonable level. New coatings of bonded MoS_2 films slough off slightly when first subjected to rubbing or sliding. For example, particles of the film flake off and contaminate bearings so that the torque of freshly coated bearings is very high during the first few minutes of run in (Ref. 8). The MoS_2 + graphite + sodium silicate coated bearings showed no increase in torque during atmospheric

operation but did show a steady decrease in torque during the first hour of operation under vacuum conditions. The favorable operation at atmospheric conditions is to be expected of the MoS_2 + graphite + silicate coating because of the contribution of the graphite with adsorbed water vapor.

The cylindrical roller bearing appeared to suffer less coating damage, in general, then its spherical roller counterpart. The simpler design of the cylindrical bearing compared with that of the double row spherical bearing would appear to be a major contributing factor since the former must contend only with radial loads, whereas the spherical must also contend with some thrust loading caused by bearing misalignments.

It appears that a small-to-moderate material transfer can result in a substantial increase in torque when solid film lubricants are used. This is particularly true for thin metal platings. Furthermore, test results indicated that thin platings are more sensitive than MoS_2 coatings to misalignment and load variations.

In some cases, MoS_2 + graphite + silicate coatings would wear to a point of marginality before bearing torque increased substantially. However, on the other hand, thin platings needed only a small amount of coating transfer to create a rapid rise in bearing torque (Ref. 1).

The Apiezon "L" grease tested in the Norma Hoffmann ball bearings performed extremely well with no measurable hydrocarbon contribution to the chamber environment.

SECTION VII
CONCLUSIONS

The following conclusions resulted from this test program:

1. The MoS_2 + graphite + sodium silicate binder coating functioned well on both bearings and gears. Other coatings had only moderate success on bearings and gears.
2. Gold and silver platings functioned reasonably well on gears, but not on bearings.
3. Apiezon "L" grease functioned extremely well on the ball bearings tested.
4. Good alignment, with as many self-alignment features as possible, is critical when thin, dry film lubrication is being used.

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TABLE I
TEST BEARING TYPES AND MATERIAL COMPOSITIONS

Manufacturer	Bearing No.	Type	Type of Loads	Bore Size (mm)	Element Material					Internal Clearance, in.
					Outer Race	Inner Race	Rolling Element	Cage	End Ring	
SKF	22220 CY	Spherical Roller	Radial & Thrust	100	52100	52100	52100	Bronze	---	0.0030-0.0041
Norma Hoffmann	6220 ZF	Ball	Radial	100	52100	52100	52100	SAE 1010	Special[1] Shields	(2)
Rollway	E5220 B	Cylindrical Roller	Radial	100	52100	52100	52100	Segments SAE 1110 Ends SAE 1020	Snap Ring SAE 1060	0.0035-0.0051
Rollway	E 1206-B[3]	Cylindrical	Radial	30	52100	52100	8620 Case Hard-ened	SAE 1010	---	(2)
Norma Hoffmann	22206 HL[3]	Spherical	Radial & Thrust	30	52100	52100	52100	Brass	---	0.0018-0.0024

(1)The special shield consisted of an SAE 1010 ring to which was molded Buna-N elastomer.

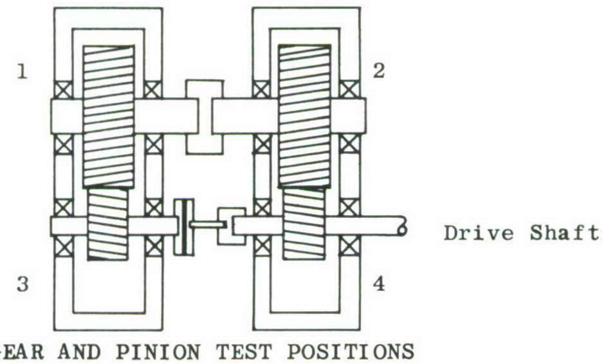
(2)The clearance specified for these bearings was C-3, or larger, which required a radial bearing clearance greater than the normal clearance stipulated by the bearing manufacturer. This is necessary where solid film lubricants are to be applied.

(3)These bearings were used for support bearings in the bearing tester.

TABLE II
GEAR TEST RESULTS

Gear Weight Changes

Test No.	Gear or Pinion Designation	Test Position	Weight Before Test, g	Weight After Test, g	Weight Change
1	3G	1	2122.120	2122.070	-0.050
	4G	2	2122.400	2112.32	-0.080
	Single Shafted 3P	3	397.210	397.125	-0.085
	Double Shafted 4P	4	443.720	443.625	-0.095
2 A & B	1G	1	2122.64	2122.60	-0.04
	2G	2	2122.47	2122.15	-0.32
	Single Shafted 1P	3	397.370	396.775	-0.595
	Double Shafted 2P	4	442.514	442.532	+0.018
3 A & B	5G	1	2121.52	2121.40	-0.12
	6G	2	2119.71	2119.50	-0.21
	Single Shafted 5P	3	396.545	396.450	-0.095
	Double Shafted 6P	4	443.521	443.391	-0.130



GEAR AND PINION TEST POSITIONS

TABLE III
GEAR TEST CONDITIONS AND RESULTS

Test No.	Lubricant	Preload Torque, lb-in.	Tangential Tooth Load, lb	Surface Compressive Stress, psi	Test Duration, hr	Total Cycles	Pinion Speed, rpm
1	MoS_2 + Epoxy	80	128	47,900	71.2	52,140	11.7
2-A and 2-B	23-kt Gold Meshing with Silver	80 160	128 256	47,900 67,700	136.6 108.0 244.6	95,712 75,816 171,528	11.7 11.7
3-A and 3-B	MoS_2 + Graphite + Sodium Silicate Binder	80 160	128 256	47,900 67,700	117.3 139.8 257.1	82,932 98,697 181,629	11.7 11.7

Note: Gear Materials

Helical shaft pinion: 1-1/4-in. P.D., AISI 4140 Steel; Rockwell C 37-29
Helical large gear: 4-in. P.D., AISI 4340 Steel; Rockwell C 40-42

Environment

Gear tests were conducted at pressures ranging from approximately 1.0×10^{-7} to 1.0×10^{-8} torr.

TABLE IV

BEARING TEST CONDITIONS AND RESULTS

Test No.	Type of Bearing	No. of Bearings	Applied Radial Load, lb/bearing (P), lb	Bearing Static Radial Load Capacity (C_r), lb	C_r/P Ratio(1)	Lubricant	Speed, rpm	Total Cycles	Test (4) Duration, hr
1	SKF Spherical Rollway Cylindrical	2 2	5,600 5,600	56,000 71,000	10.0 12.7	MoS_2 + Epoxy Binder	10	14,101	26.6 (failure)
2	Rollway Cylindrical SKF Spherical	2 2	4,000 4,000	71,000 56,000	17.7 14.0	MoS_2 + Epoxy Binder	8	45,205	95.8 (failure)
3	SKF Spherical Rollway Cylindrical	2 2	4,000 4,000	56,000 71,000	14.0 17.7	23-kt Gold	8	1,938	4.4 (failure)
4A	Rollway Cylindrical	4	5,100	71,000	14.0	MoS_2 + Graphite + Sodium Silicate	8	62,640	130.0
4B	Rollway Cylindrical	4	7,100	71,000	10.0	MoS_2 + Graphite + Sodium Silicate	8	46,859 109,499	100.0 230.0 (2)
5	Norma Hoffman Ball	4	2,850	28,500	10.0	Apiezon "L" Grease	8	36,302	80.0 (3)
6	Rollway Cylindrical	4	5,100	71,000	14.0	Silver	8	6,239	12.5 (failure)
7	SKF Spherical Rollway Cylindrical	2 2	4,000 4,000	56,000 71,000	14.0 17.7	23-kt Gold	8	217	0.5 (failure)
8	Rollway Cylindrical SKF Spherical	2 2	4,000 4,000	71,000 56,000	17.7 14.0	MoS_2 + Glass Binder	8	24,765	53.3 (failure)

(1) C_r/P is the ratio of the static radial load capacity (C_r) of the bearing to the applied radial load (P); C_r is defined as the radial load which will result in a permanent deformation of the bearing elements in excess of 10^{-4} times the diameter of the rolling element.

(2) This set of bearings was subjected to two test periods of 100 hours or more each. The second test period was at an increased load over the previous test load.

(3) This test was installed in the chamber for a weekend period only and although only 80 hours were compiled on the bearings, this does not mean that failure occurred. To the contrary, the results of this test were extremely good.

(4) Bearing tests were conducted at pressures ranging from approximately 1.0×10^{-7} to 1.0×10^{-8} torr.

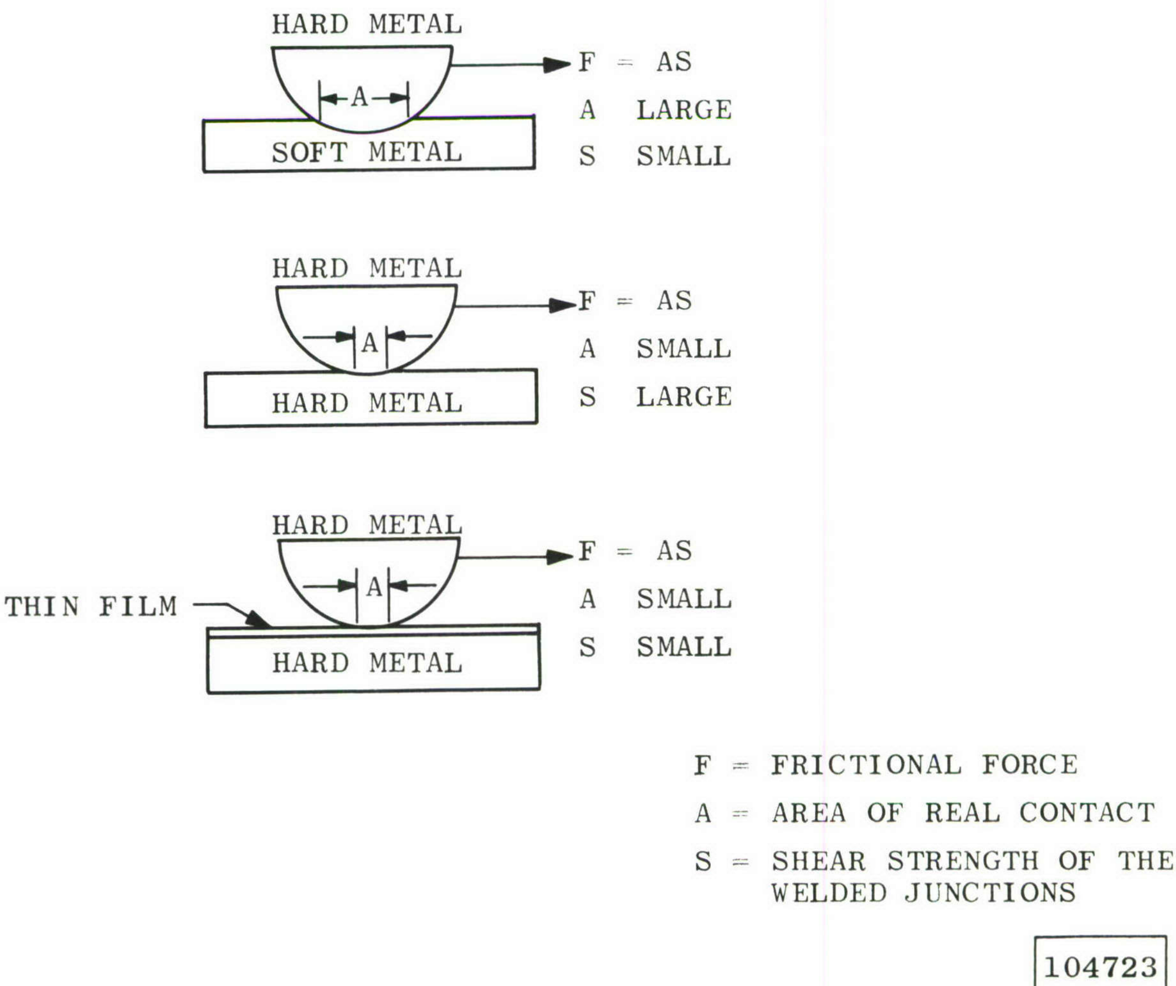


Fig. 1 Lubrication of Solids by Thin Films

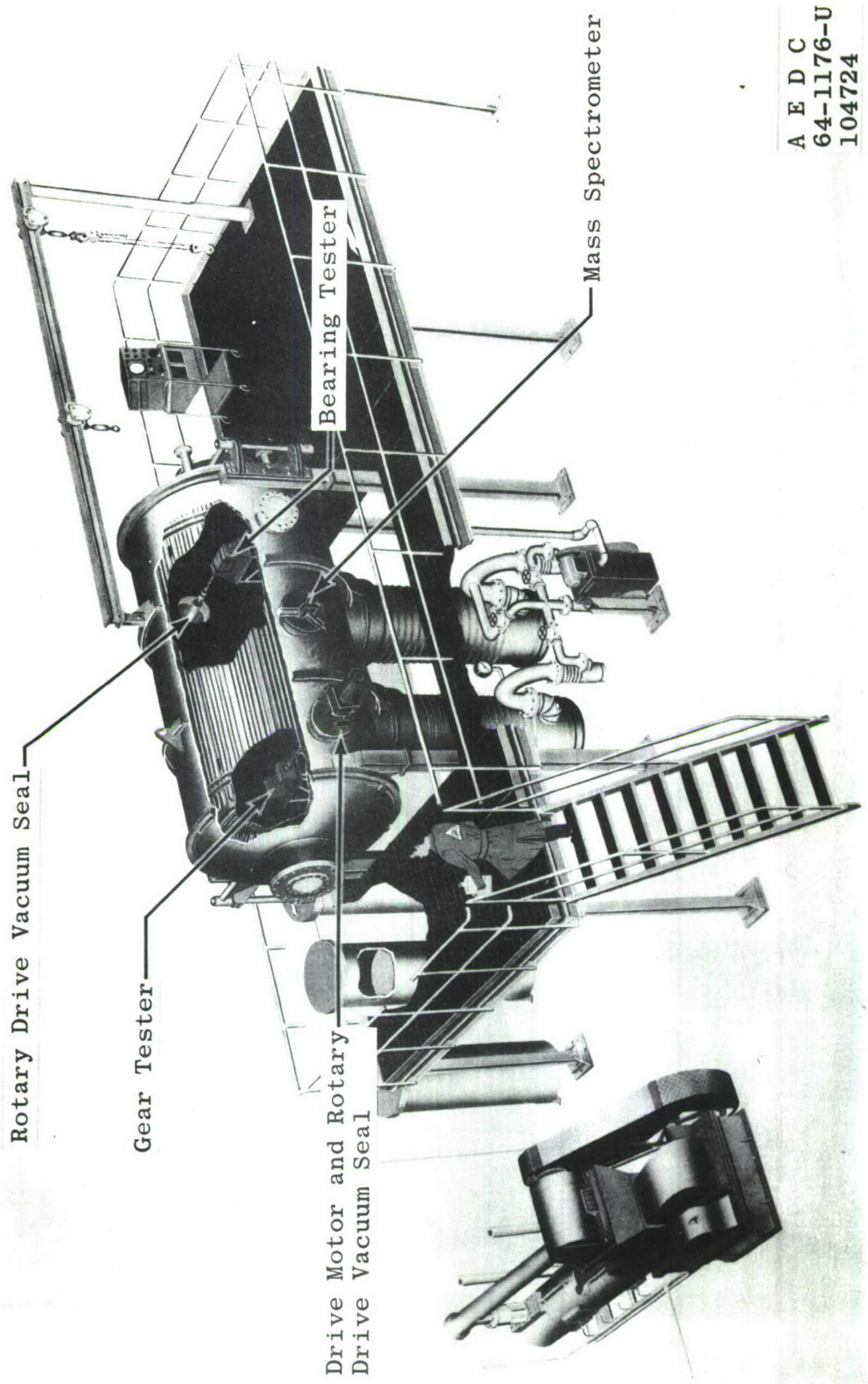


Fig. 2 Aerospace Research Chamber (7V)

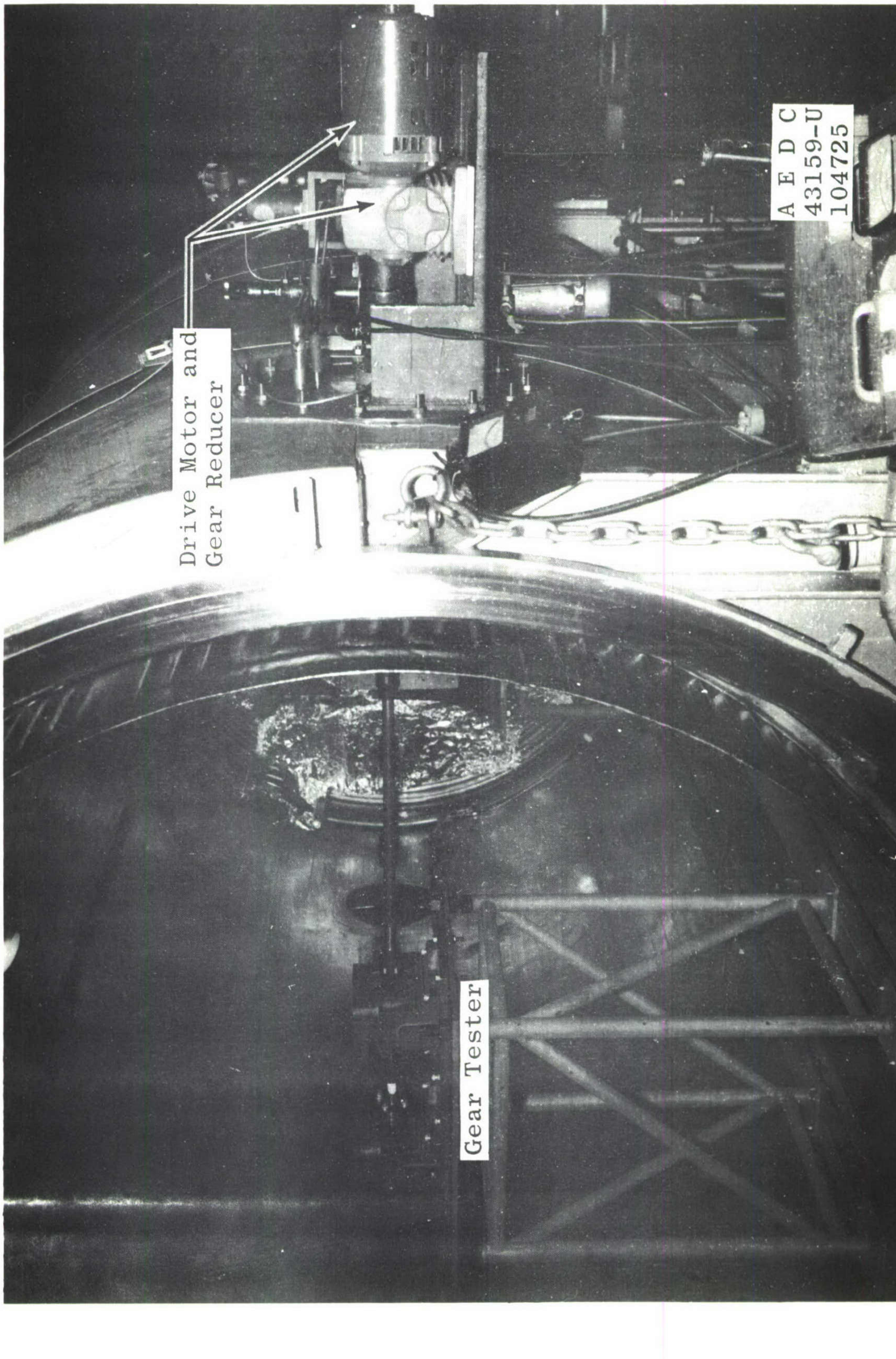


Fig. 3 Four-Square Gear Tester in ARC (7V)

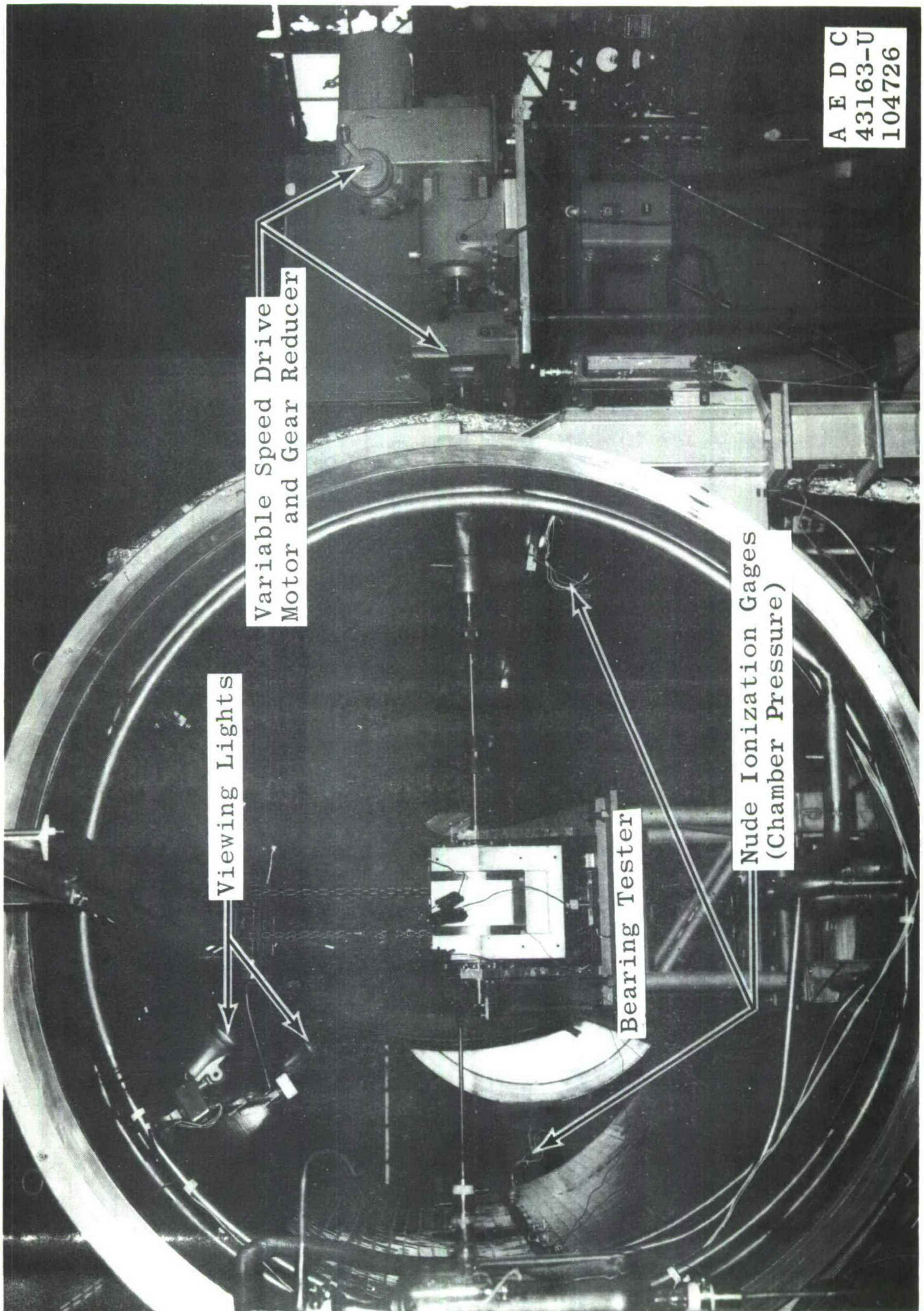


Fig. 4 Bearing Tester in ARC (7V)

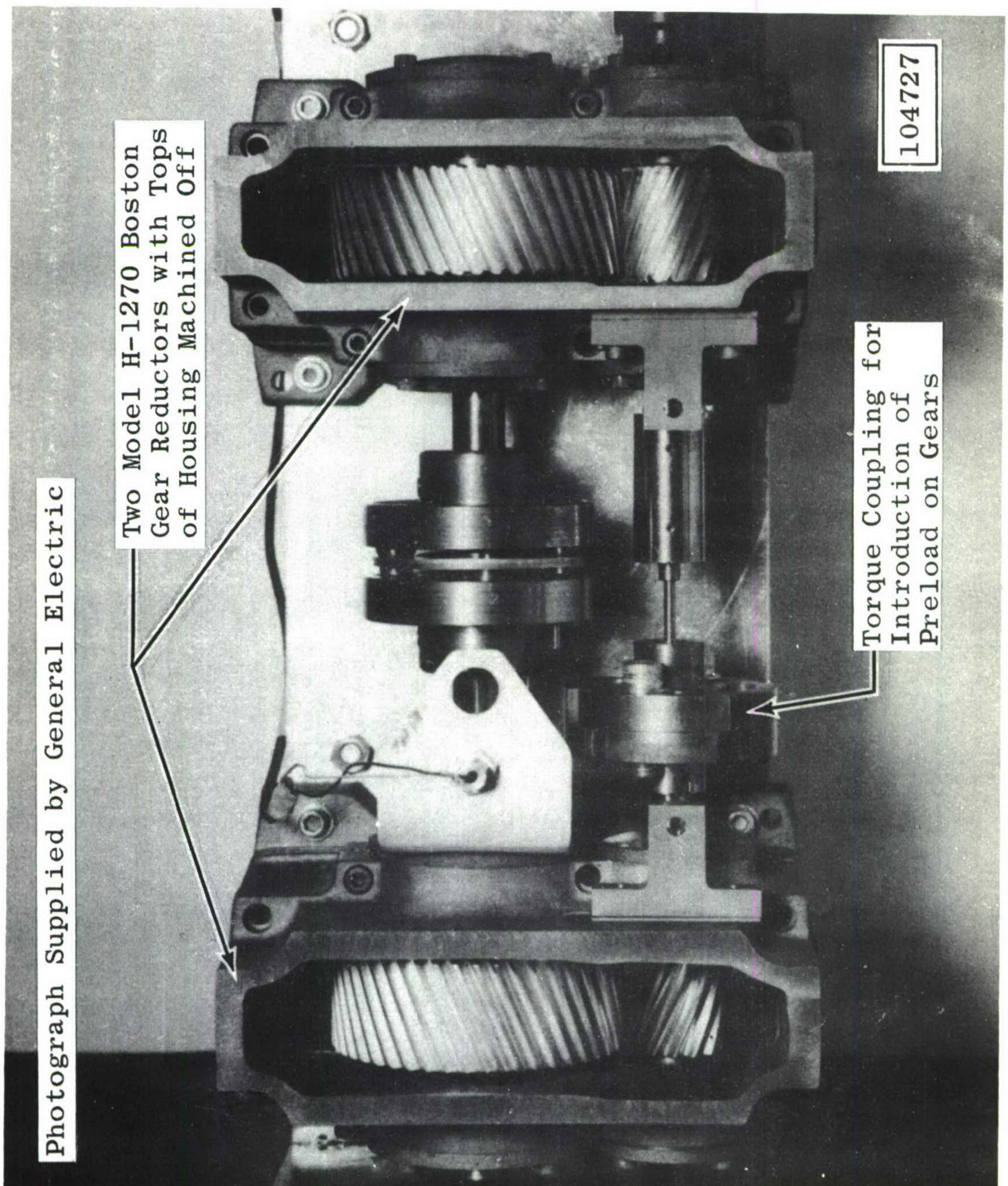


Fig. 5 Four-Square Gear Tester

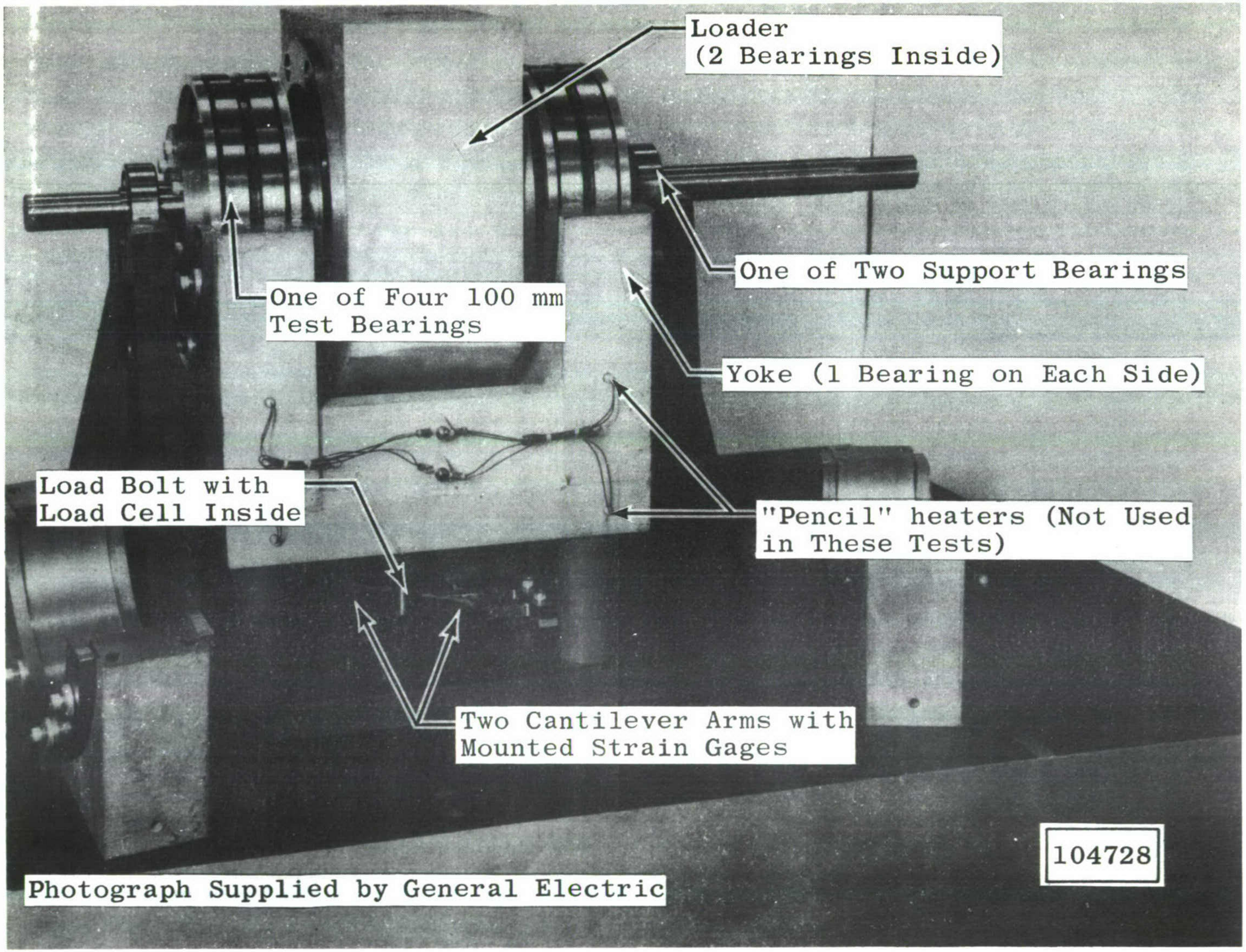


Fig. 6 Bearing Tester

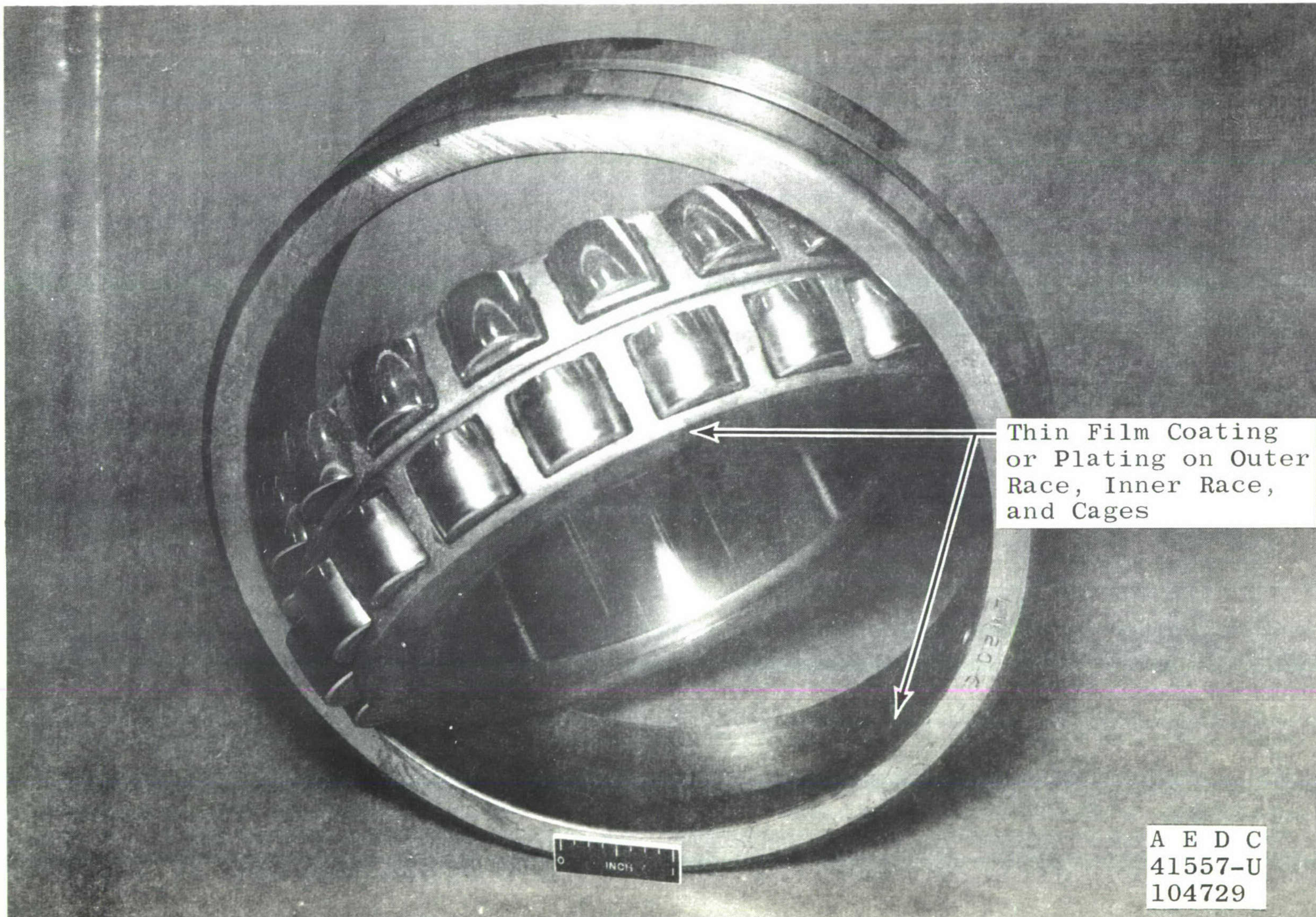


Fig. 7 Spherical Roller Bearing

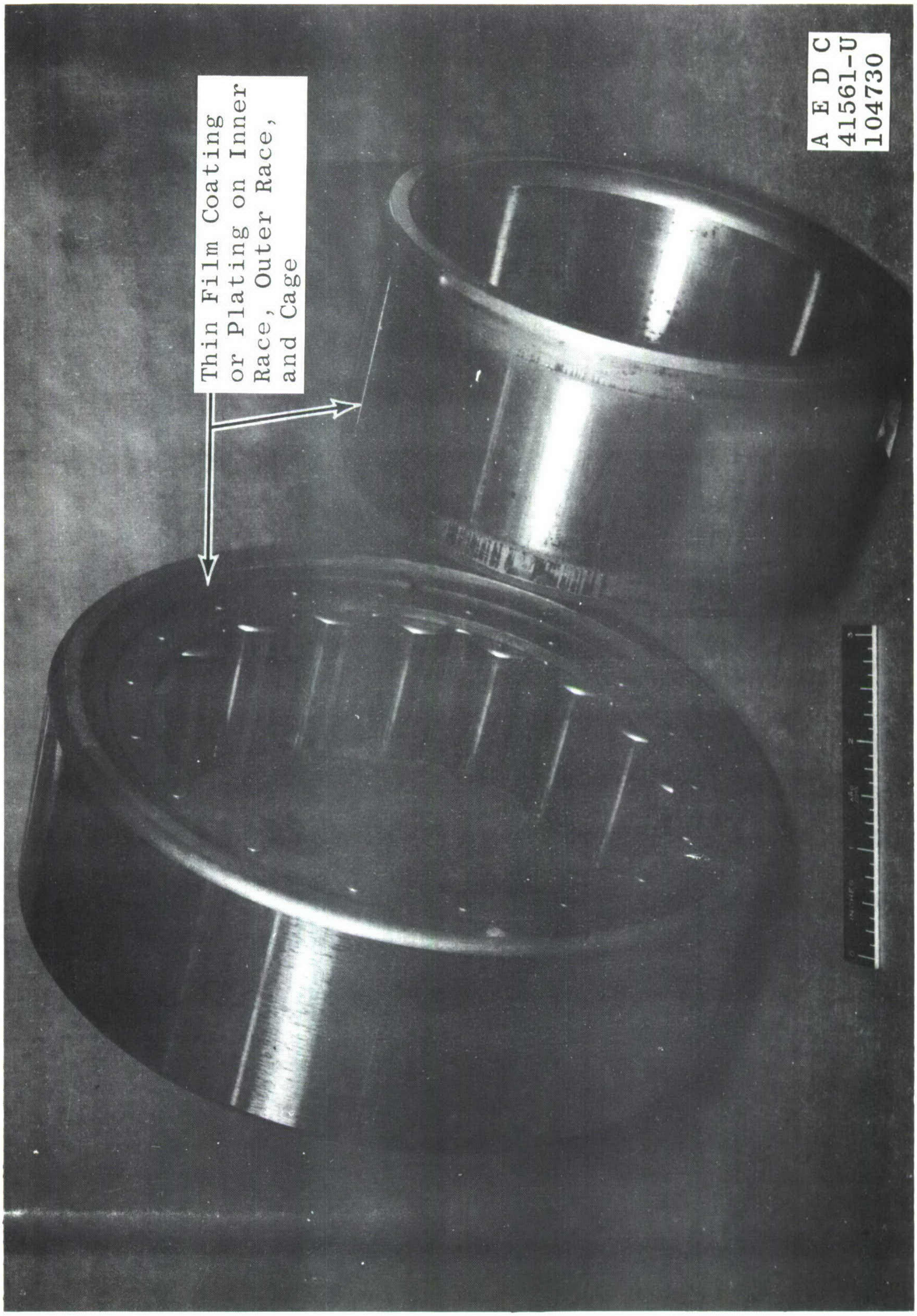


Fig. 8 Cylindrical Roller Bearing

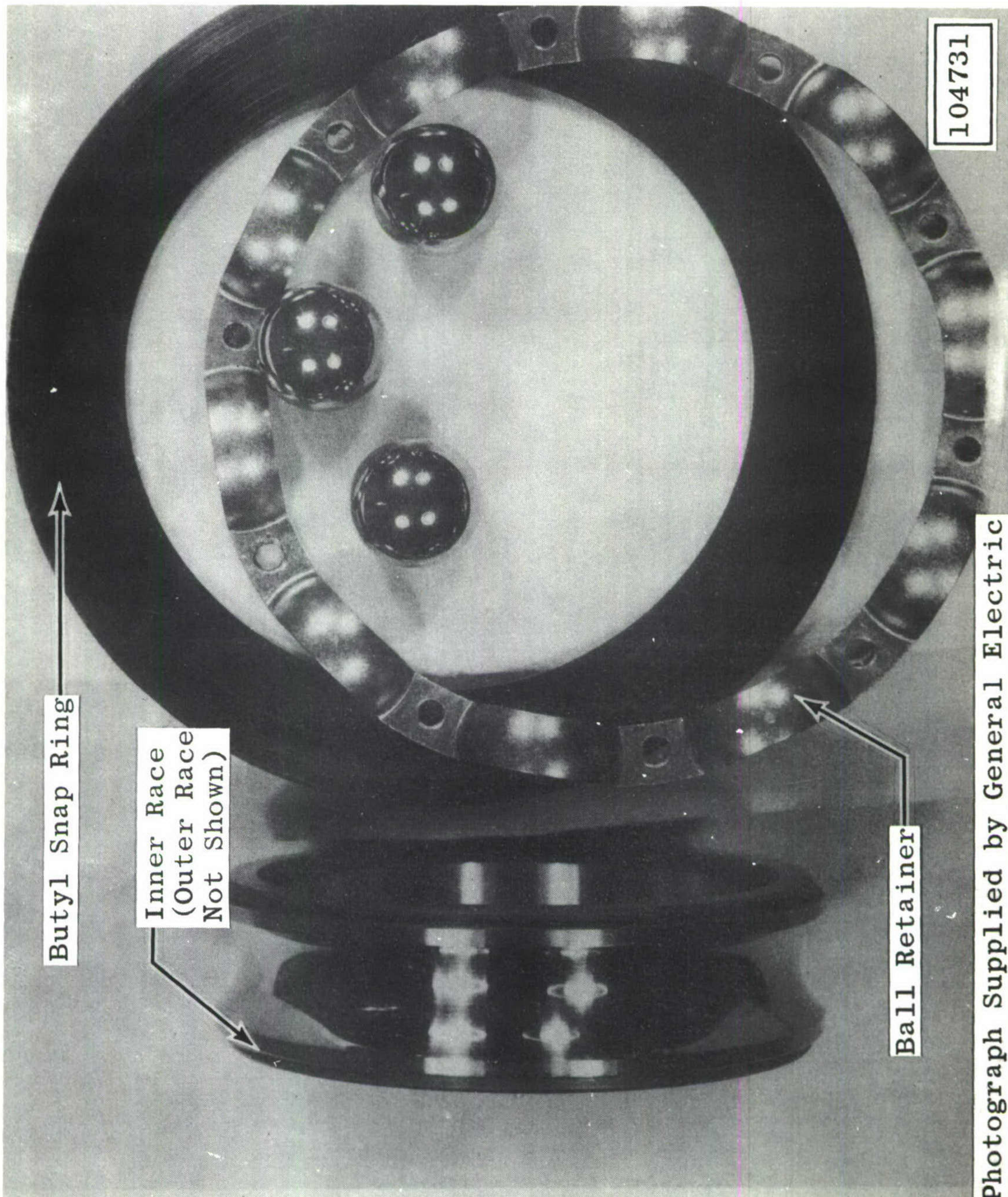


Fig. 9 Ball Bearing

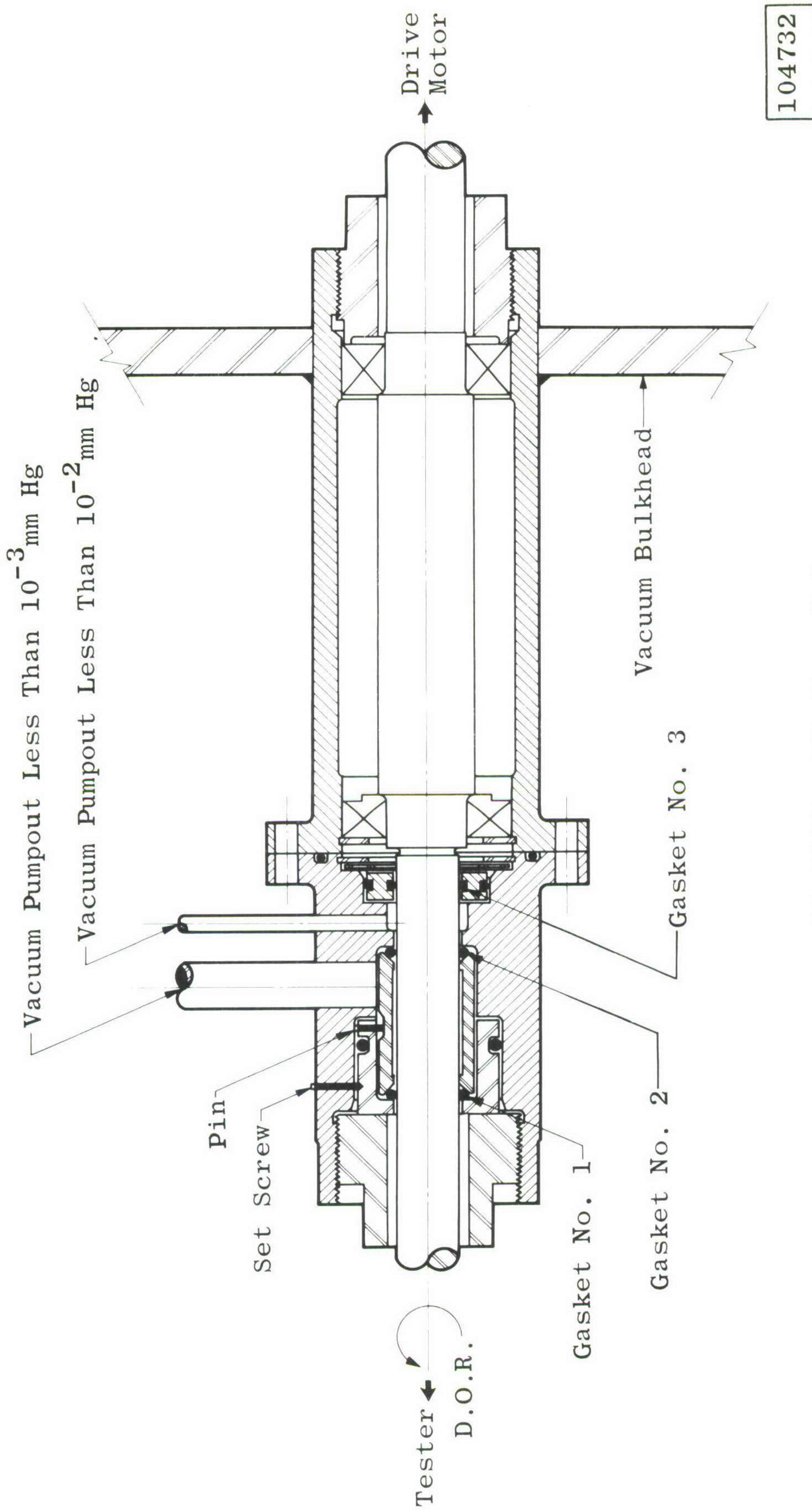
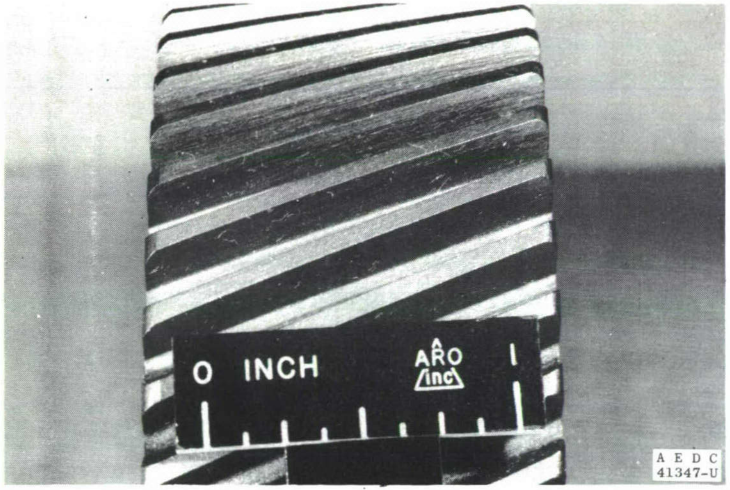
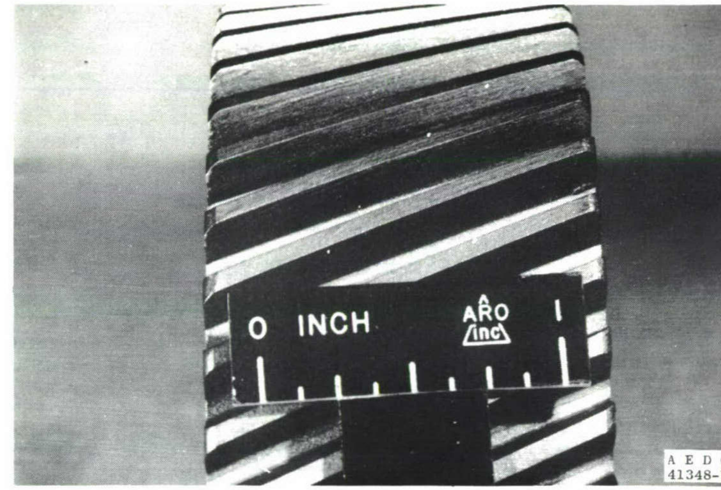


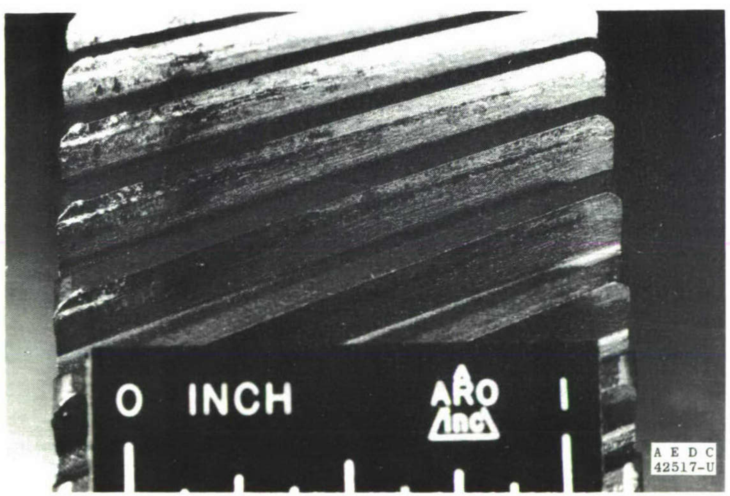
Fig. 10 Rotary Drive Vacuum Seal



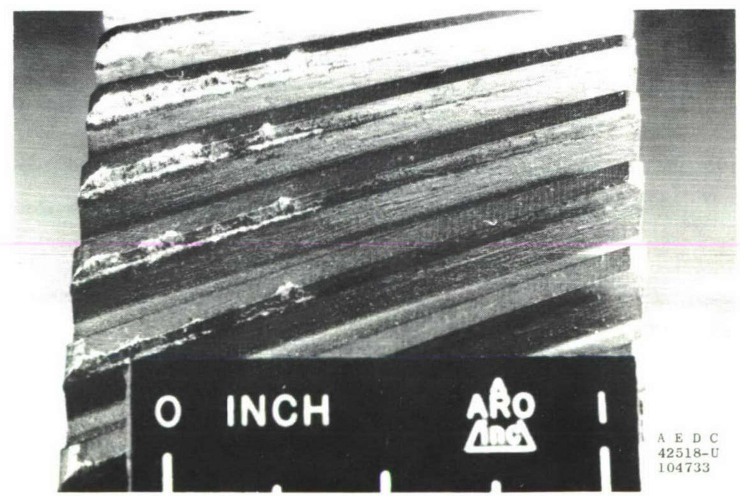
Gear No. 3 Before Test 1



Gear No. 4 Before Test 1

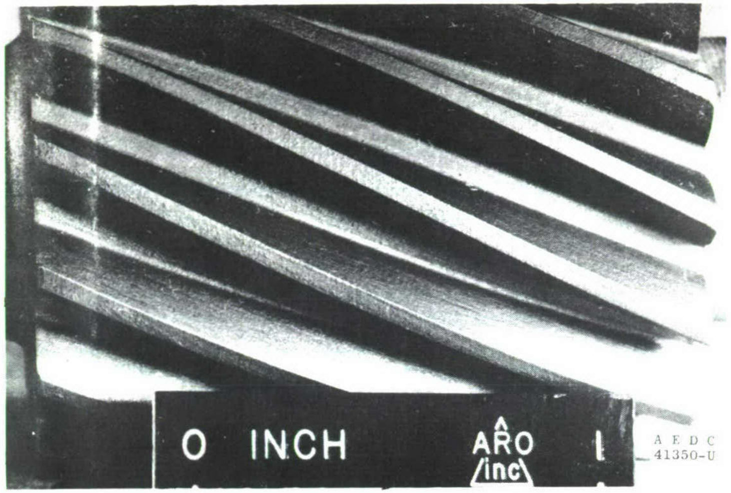


Gear No. 3 After Test 1

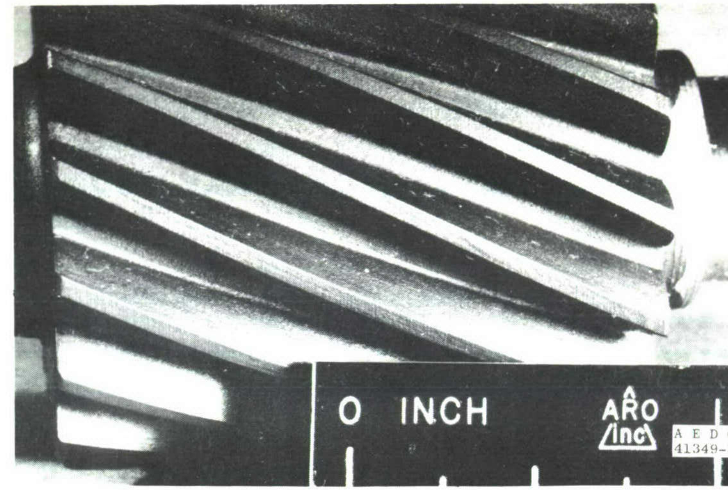


Gear No. 4 After Test 1

Fig. 11 Gears Before and After Test 1



Pinion No. 3 Before Test 1



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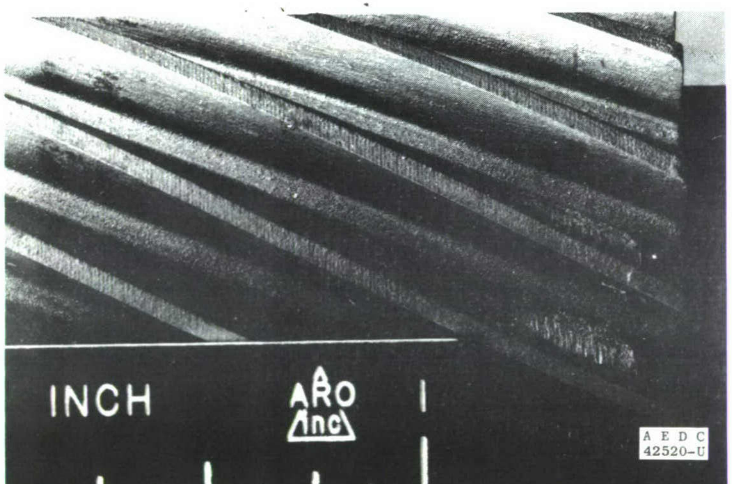
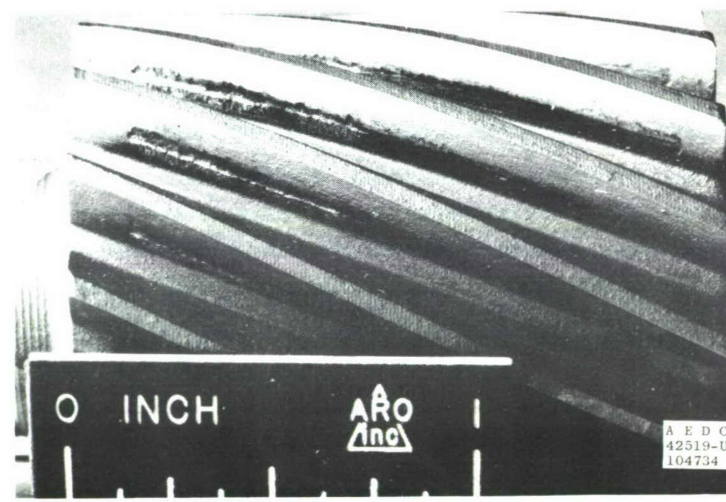
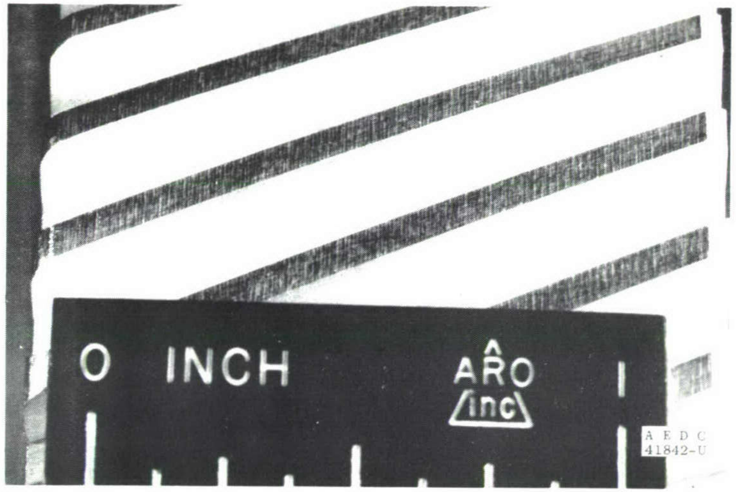


Pinion No. 3 After Test 1

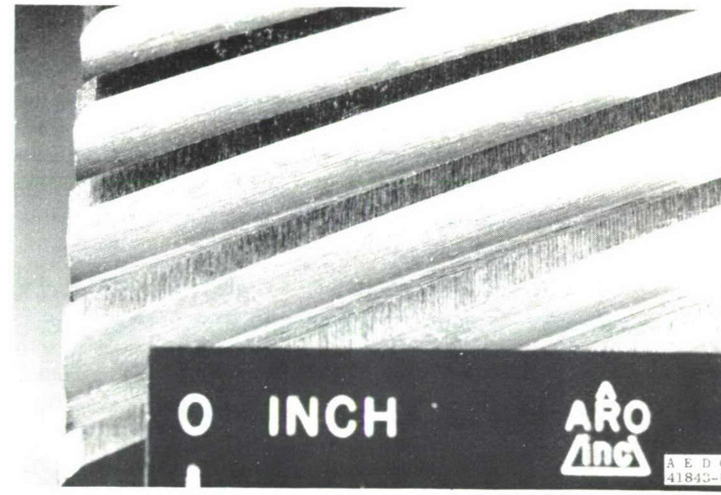


Pinion No. 4 After Test 1

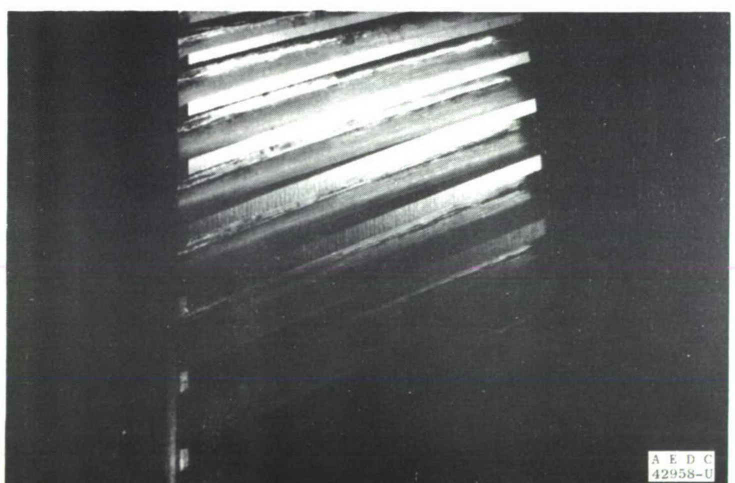
Fig. 12 Pinions Before and After Test 1



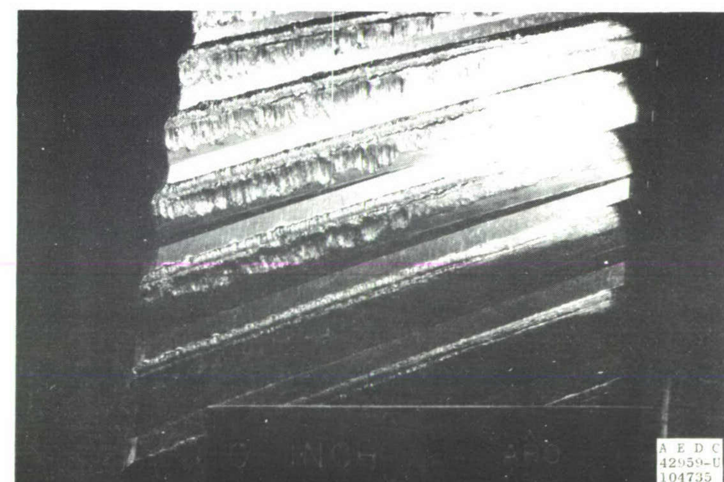
Gear No. 1 Before Test 2A



Gear No. 2 Before Test 2A



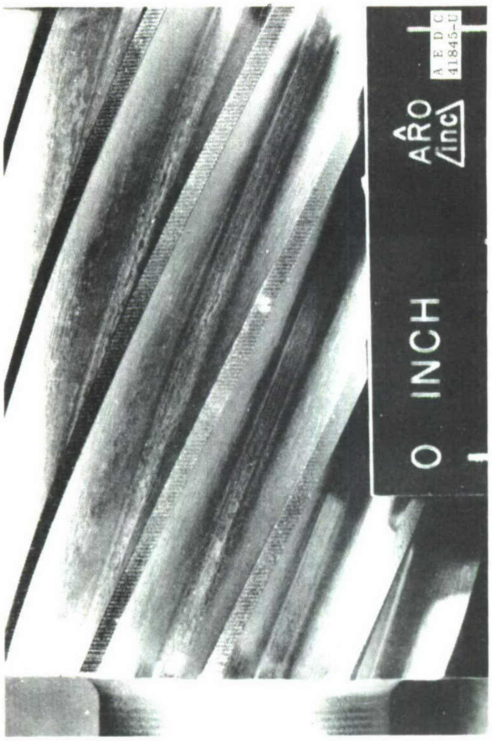
Gear No. 1 After Test 2B



Gear No. 2 After Test 2B

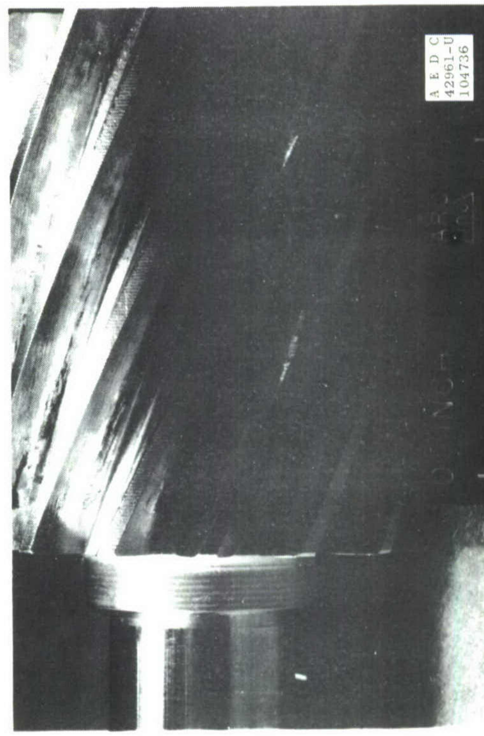
Fig. 13 Gears Before and After Tests 2A and 2B

Pinion No. 1 Before Test 2A



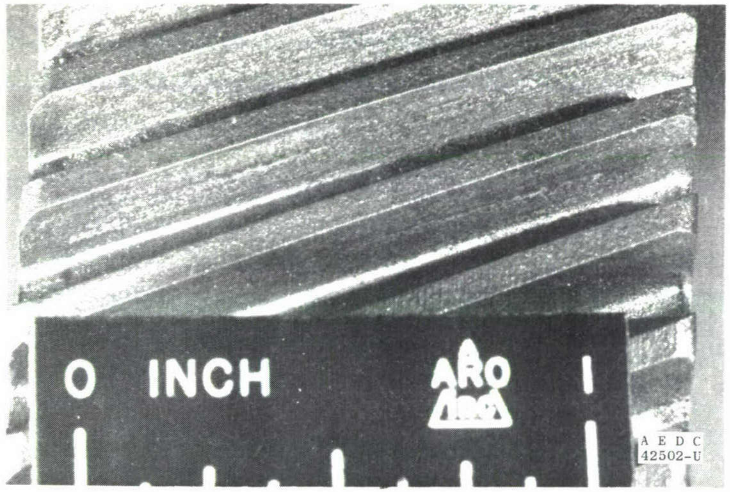
Pinion No. 2 Before Test 2A

Pinion No. 1 After Test 2B

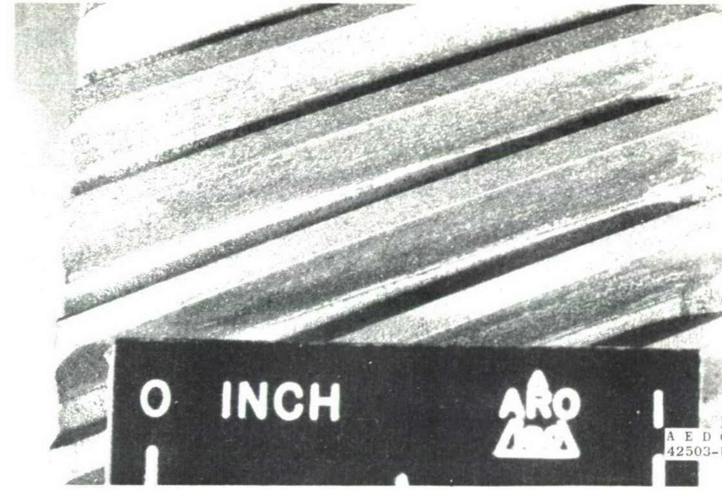


Pinion No. 2 After Test 2B

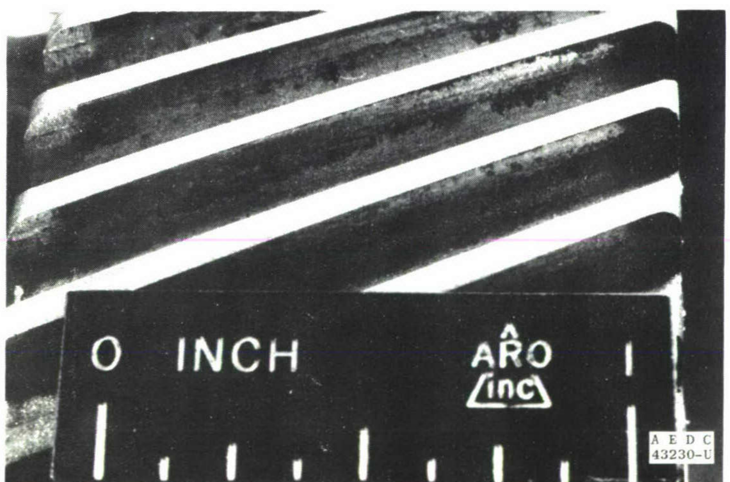
Fig. 14 Pinions Before and After Tests 2A and 2B



Gear No. 5 Before Test 3A



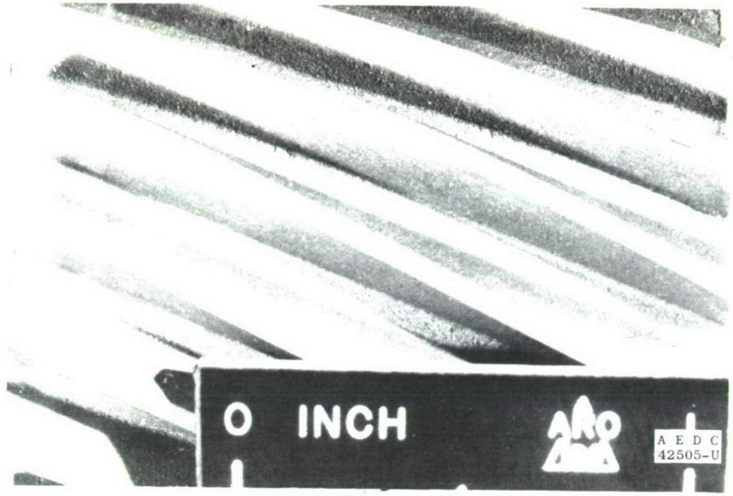
Gear No. 6 Before Test 3A



Gear No. 5 After Test 3B

Gear No. 6 After Test 3B

Fig. 15 Gears Before and After Tests 3A and 3B



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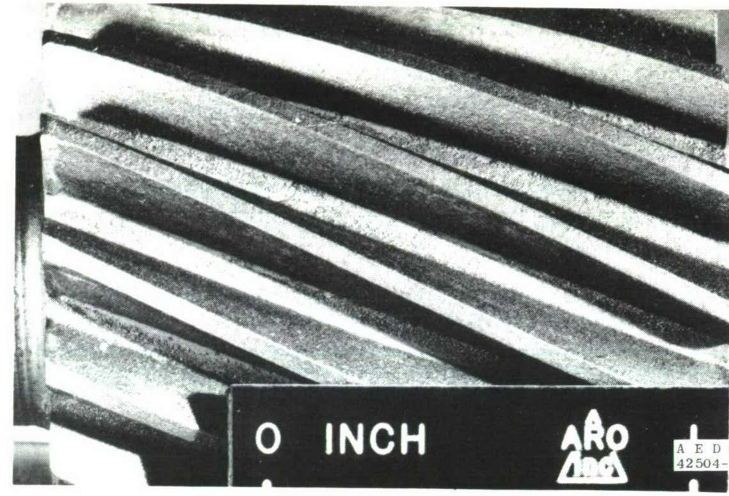
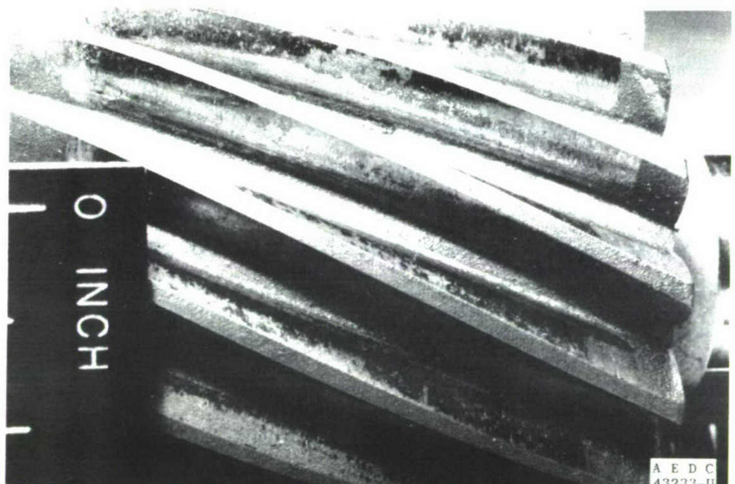
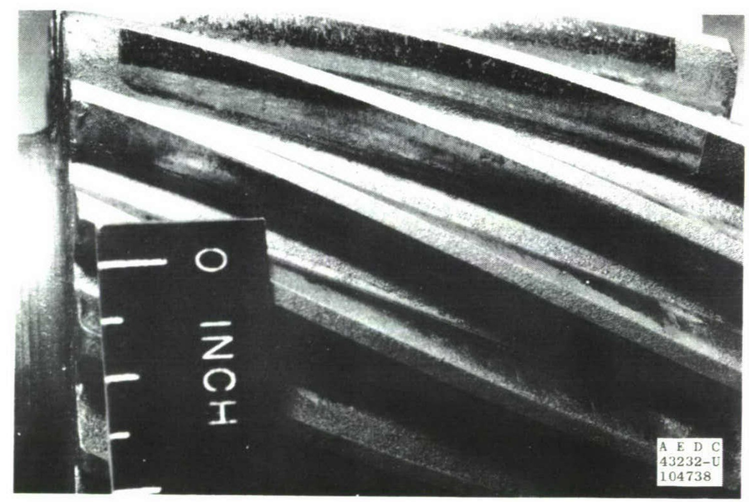


Pinion No. 6 Before Test 3A



Pinion No. 5 After Test 3B



Pinion No. 6 After Test 3B

Fig. 16 Pinions Before and After Tests 3A and 3B

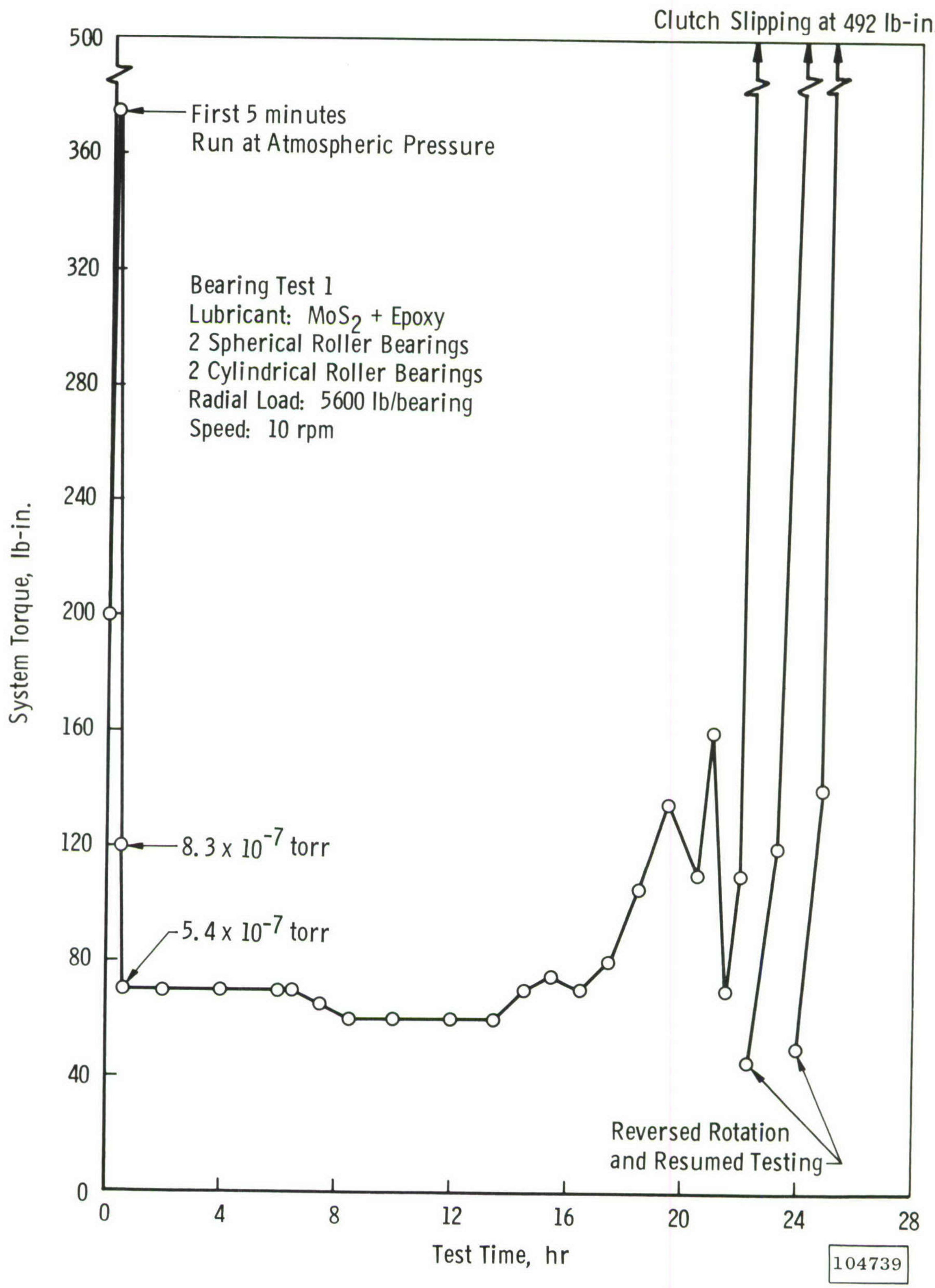


Fig. 17 Torque Versus Time for Bearing Test 1

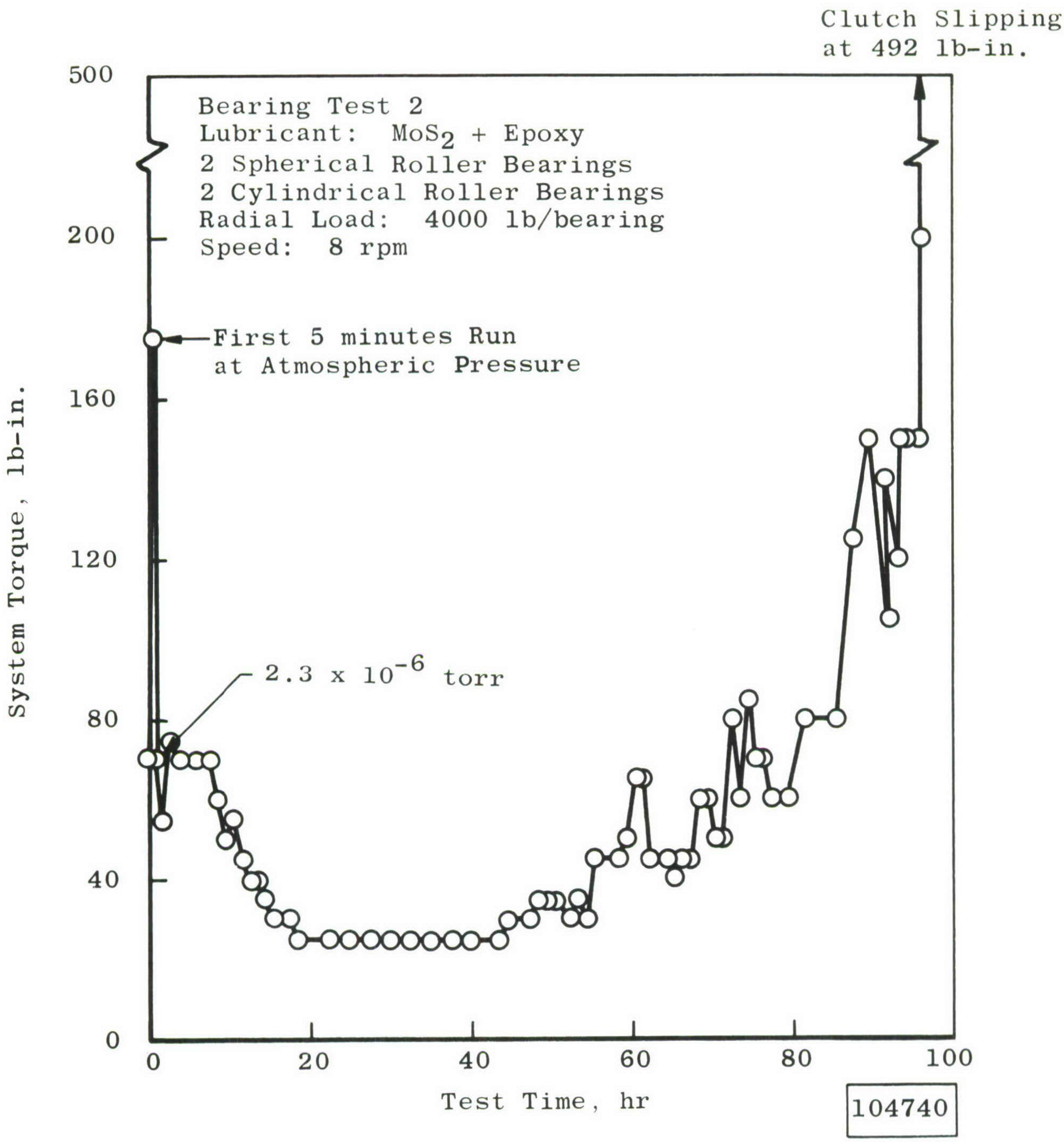


Fig. 18 Torque Versus Time for Bearing Test 2

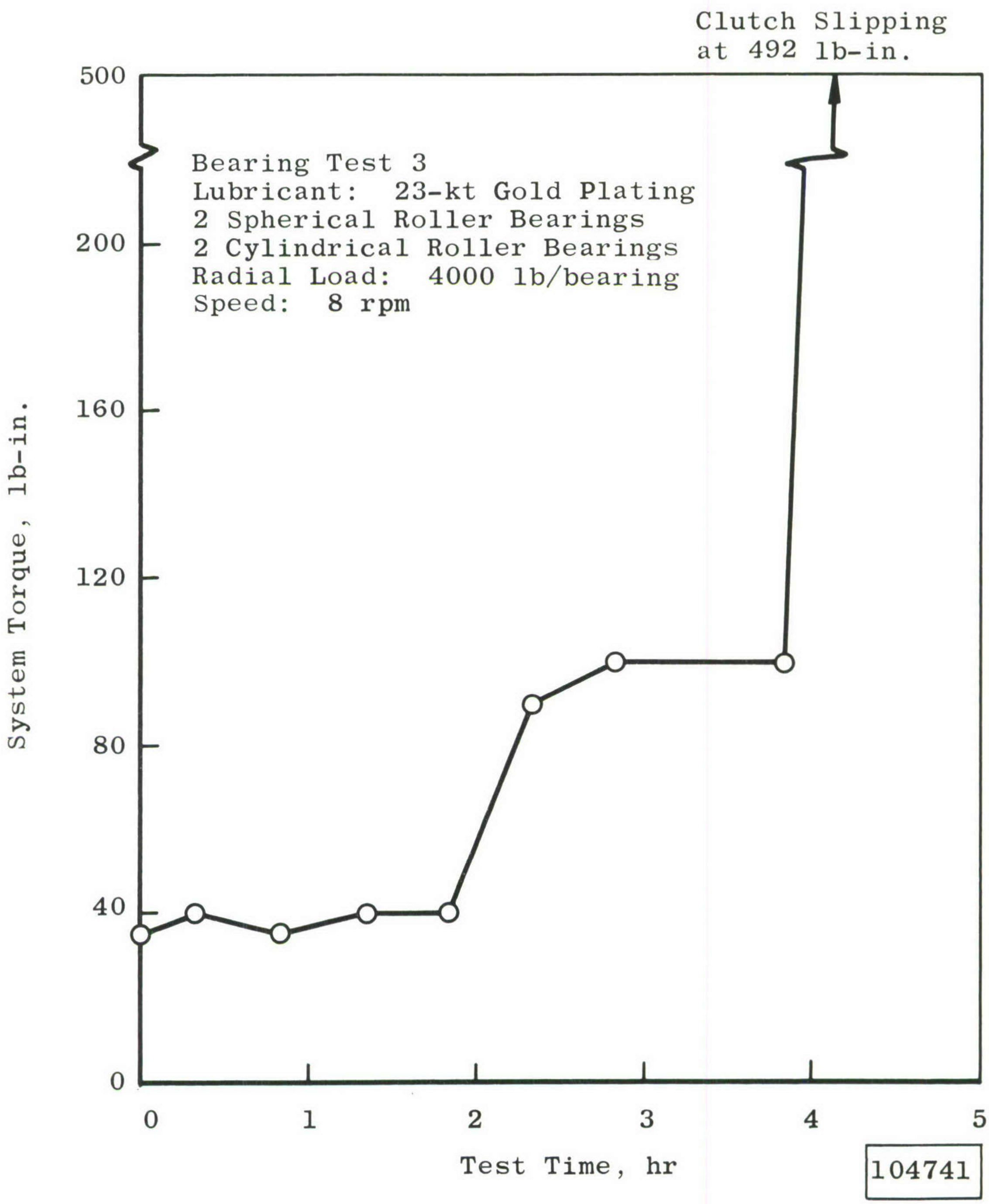


Fig. 19 Torque Versus Time for Bearing Test 3

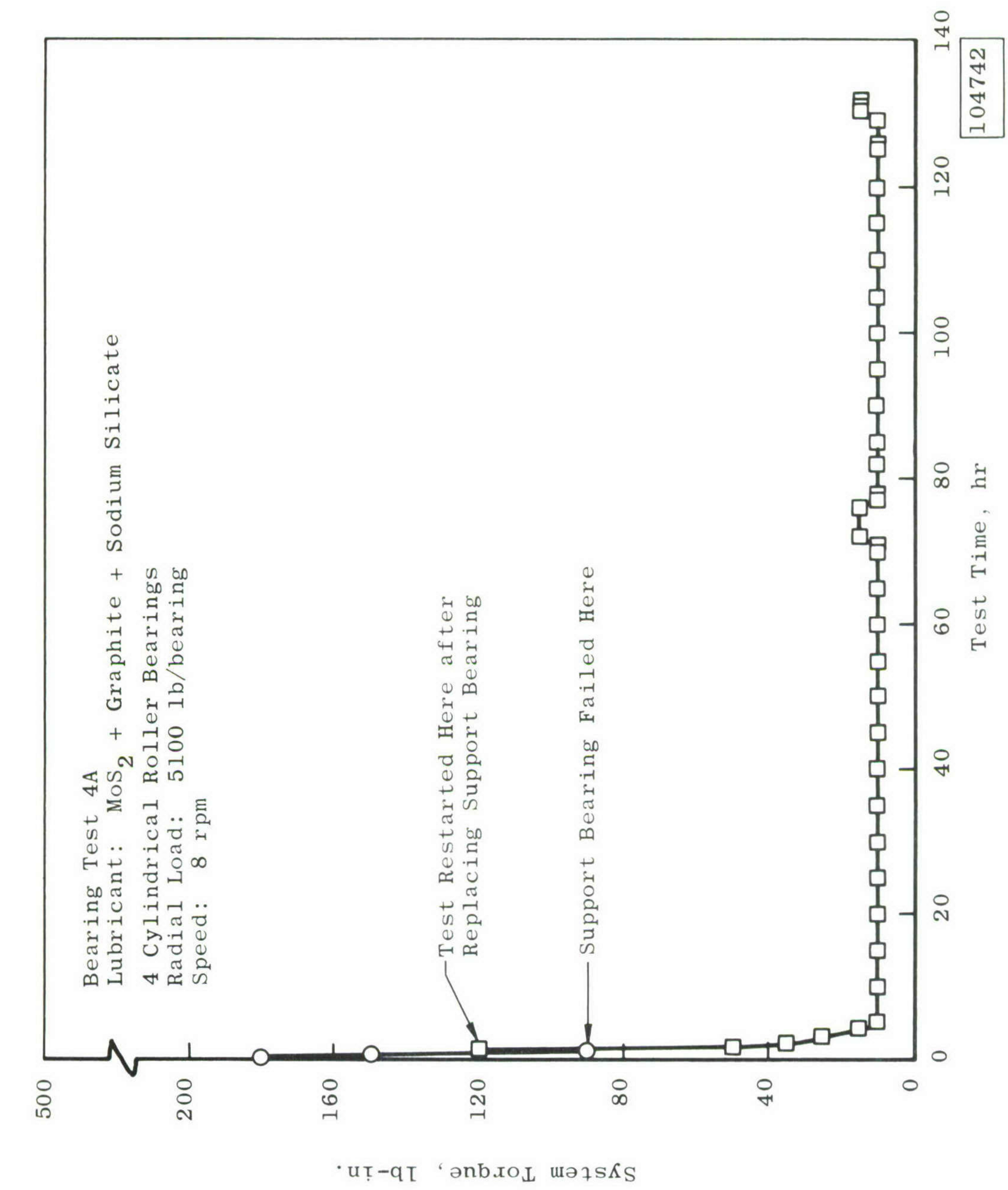


Fig. 20 Torque Versus Time for Bearing Test 4A

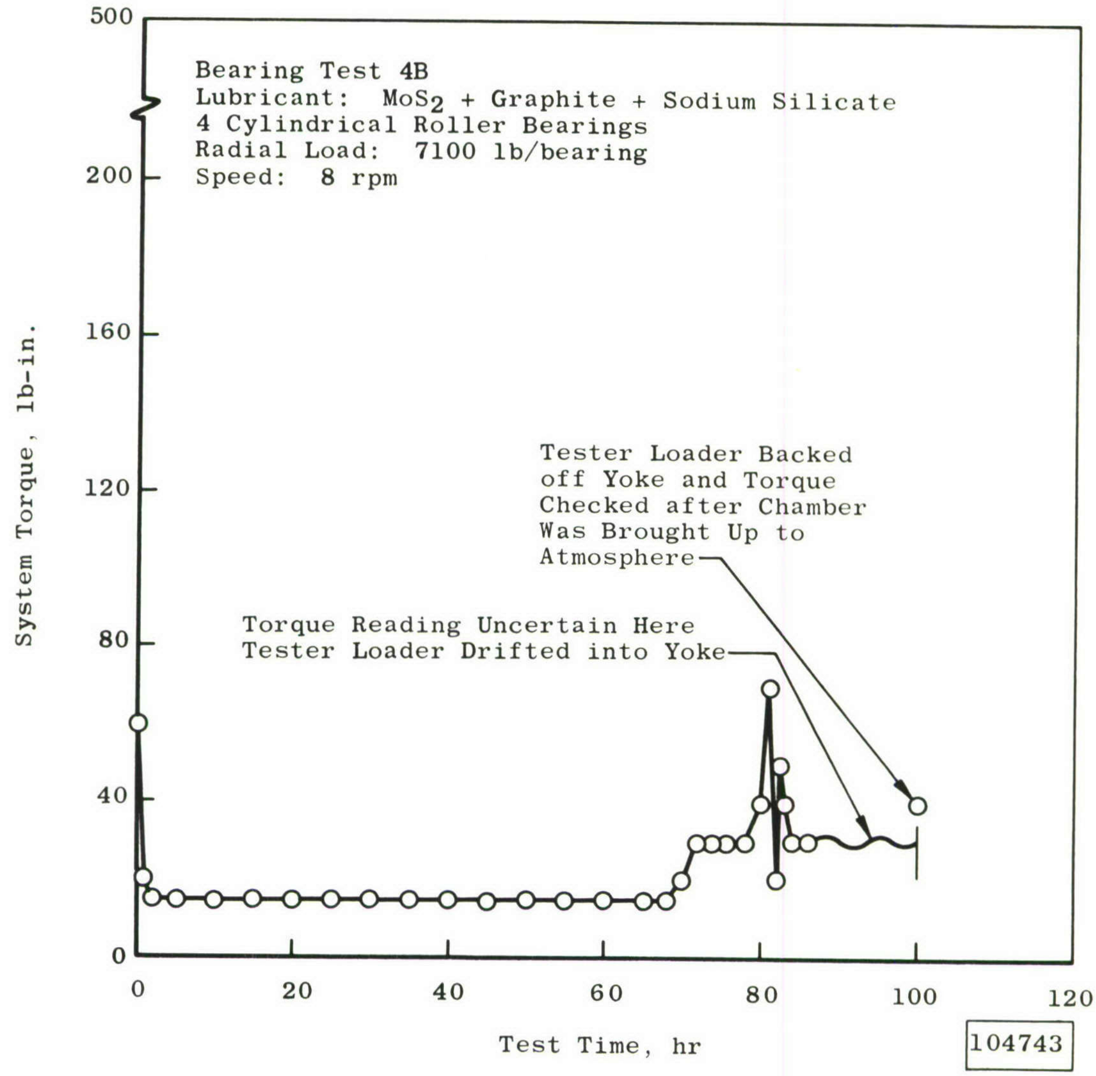


Fig. 21 Torque Versus Time for Bearing Test 4B

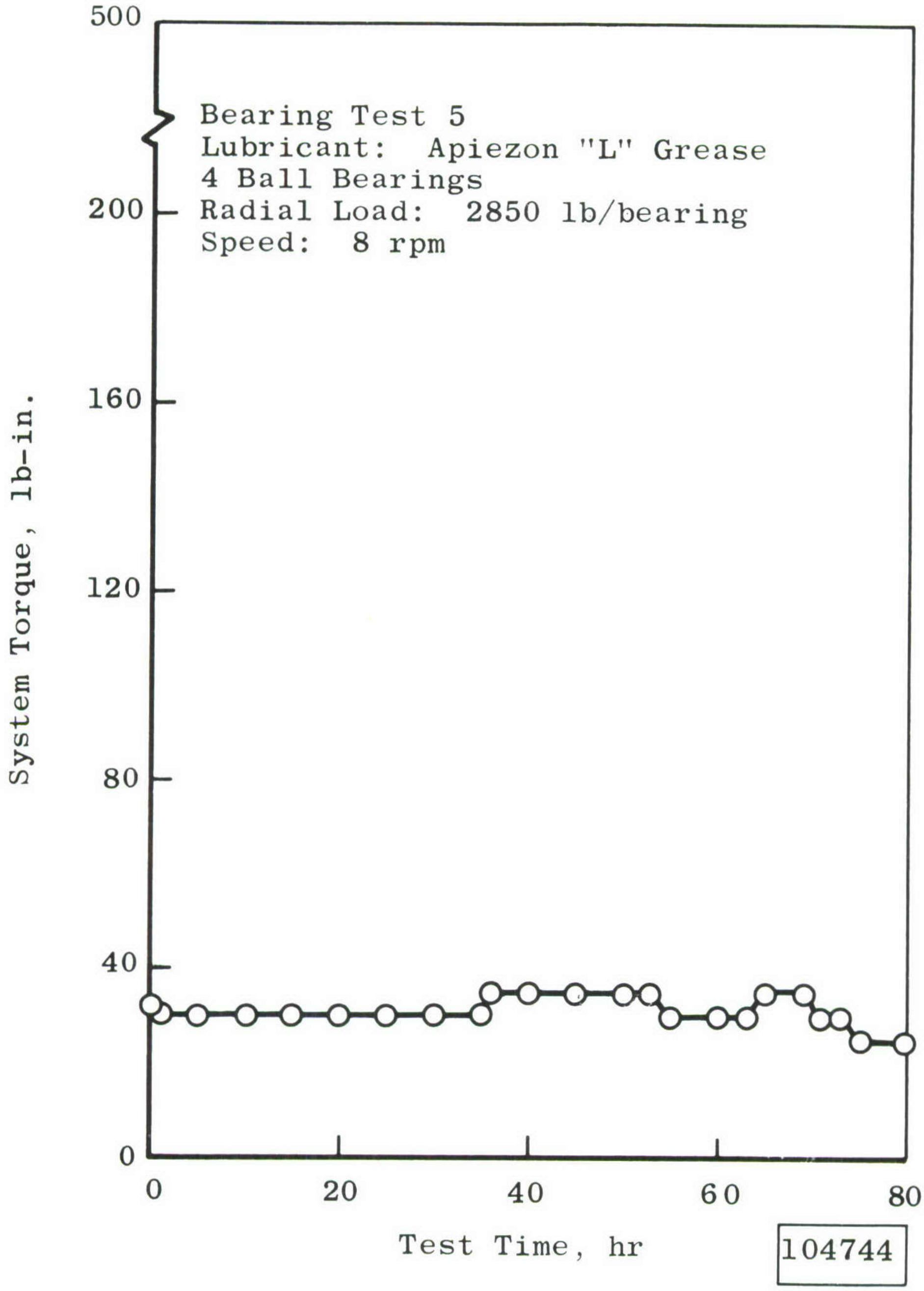


Fig. 22 Torque Versus Time for Bearing Test 5

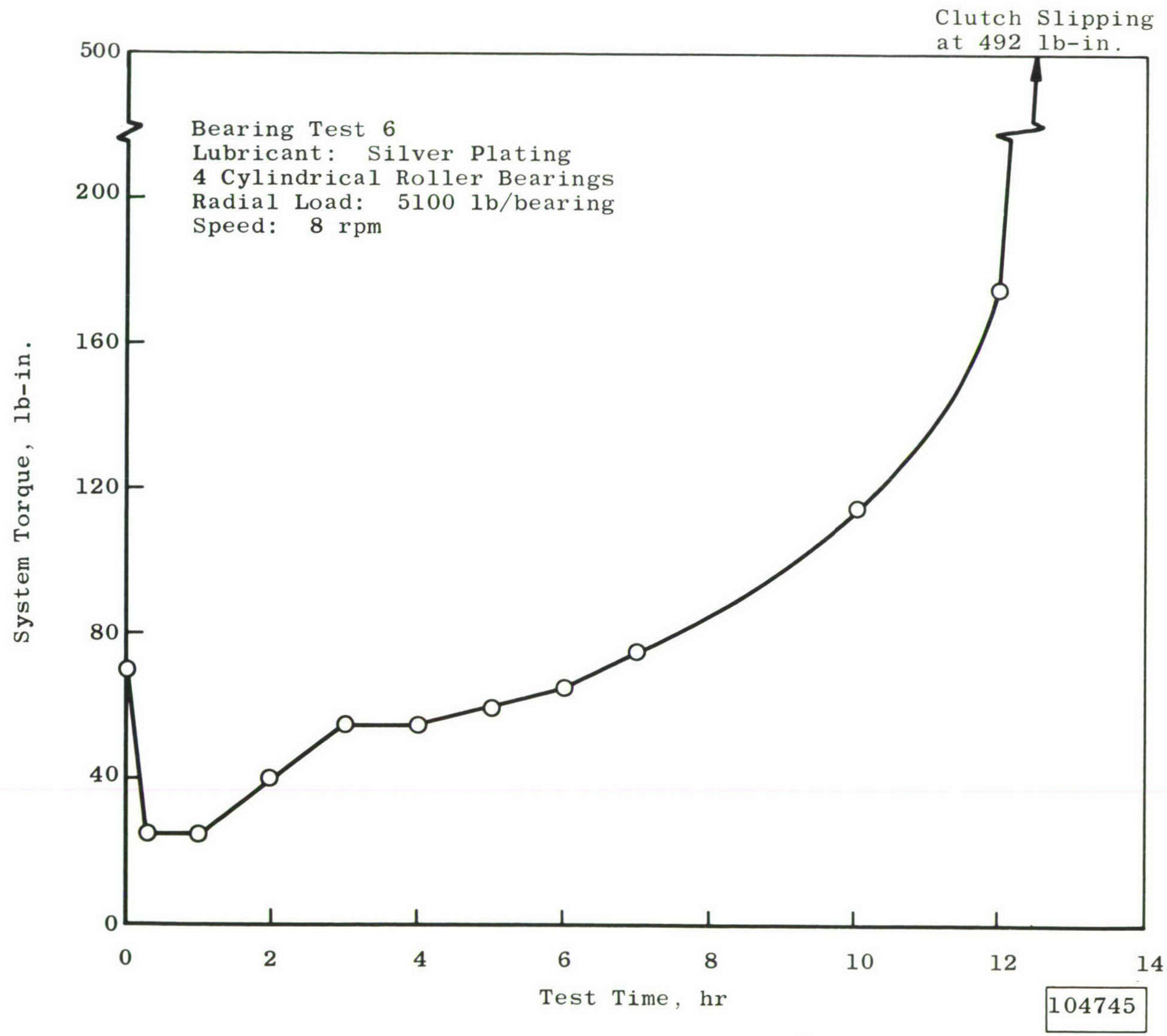


Fig. 23 Torque Versus Time for Bearing Test 6

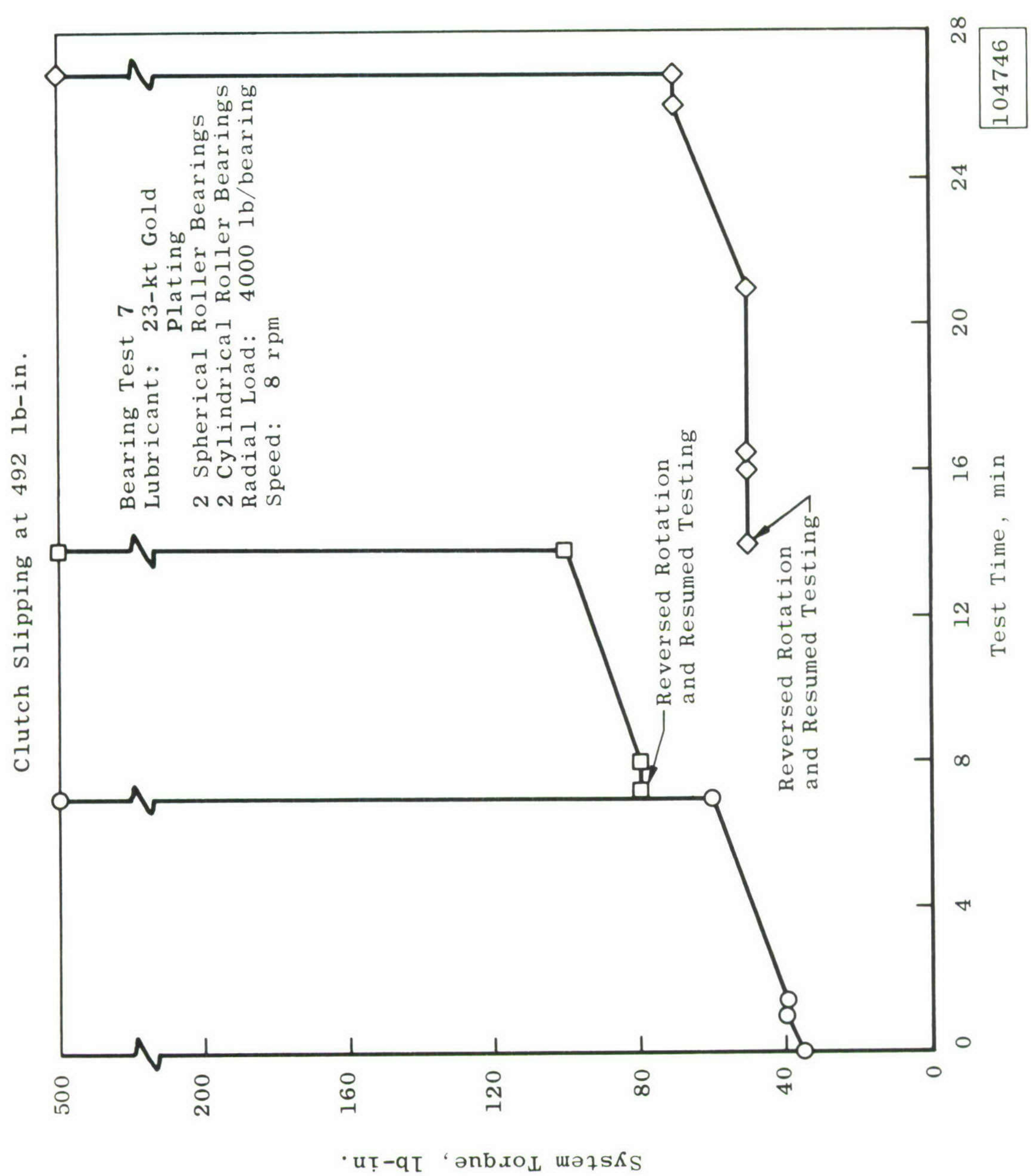


Fig. 24 Torque Versus Time for Bearing Test 7

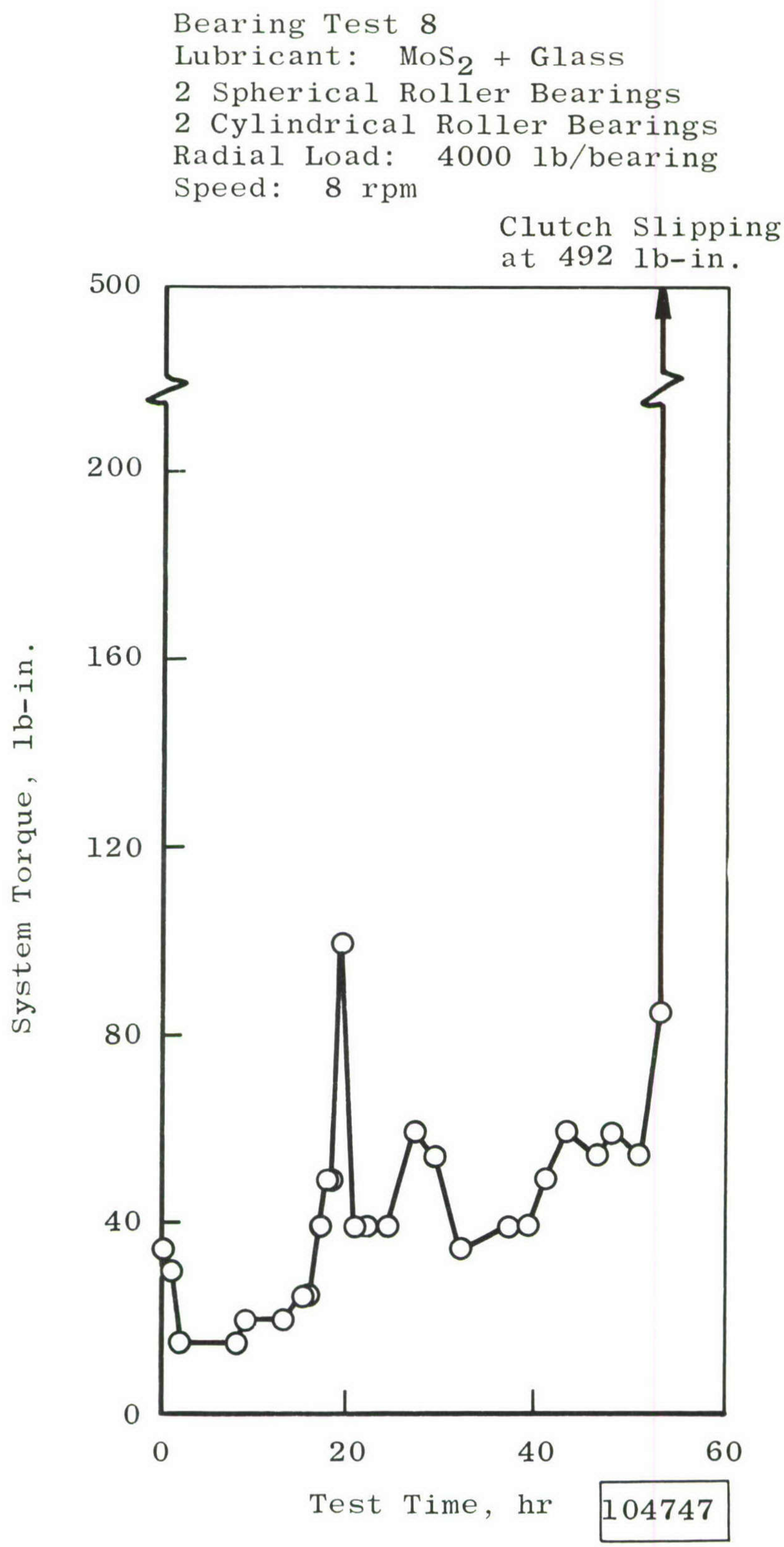


Fig. 25 Torque Versus Time for Bearing Test 8

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4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*

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N/A	Arnold Engineering Development Center Air Force Systems Command Arnold Air Force Station, Tennessee

13. ABSTRACT

This report contains the results of a test program set up to determine the operational characteristics of dry, thin film lubricated bearings and gears, uniquely adapted for operation in a simulated space environment. Results indicate that dry, thin film lubricants and soft metal plating lubricants can be applied satisfactorily to certain types of bearings and gears and are capable of sustaining heavy loads at slow speeds in space environments.

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
bearings						
dry lubrication						
ball bearings						
roller bearings						
gears						
helical gears						
vacuum lubricants						
thin film lubricants						

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It is highly desirable that the abstract of classified reports be unclassified. Each paragraph of the abstract shall end with an indication of the military security classification of the information in the paragraph, represented as *(TS), (S), (C), or (U)*.

There is no limitation on the length of the abstract. However, the suggested length is from 150 to 225 words.

14. KEY WORDS: Key words are technically meaningful terms or short phrases that characterize a report and may be used as index entries for cataloging the report. Key words must be selected so that no security classification is required. Identifiers, such as equipment model designation, trade name, military project code name, geographic location, may be used as key words but will be followed by an indication of technical context. The assignment of links, rules, and weights is optional.