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FINAL REPORT

Lubricant Life Tests on Ball Bearings for Space Applications

by
S. F. Murray
P. Lewis

October 28, 1965

Contract No. NAS5-9020

Prepared by
MECHANICAL TECHNOLOGY INCORPORATED
Latham, N.Y.

For
GODDARD SPACE FLIGHT CENTER
Greenbelt, Maryland

CR 36545

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ABSTRACT

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An experimental program has been conducted to compare the effective lives of ball bearings operating in vacuum with various types of MoS_2 solid films, and with a special high vacuum oil, as lubricants. The test bearings were size 205 bearings running at 30 rpm under a ten-pound radial load. Two particular combinations were also evaluated in oscillating motion tests. Torque was used as the criterion for failure.

The results of these tests showed that most of the solid film lubricated bearings were effective for the first several hundred hours, then gave high and erratic torque values as the result of debris being formed by wear of the lubricant film. A sodium silicate bonded solid lubricant film, which contained MoS_2 and graphite, was found to be particularly promising in both rotation and oscillation.

One particularly significant result of this work was the finding that the oil-lubricated bearings showed a sudden, large increase in torque after running effectively for about 1400 hours in vacuum. This behavior has often been predicted but has apparently never been observed experimentally, at least for rolling contact bearings.

TABLE OF CONTENTS

	Page
LIST OF ILLUSTRATIONS AND TABLES	iv
ACKNOWLEDGEMENTS	vi
INTRODUCTION	1
SUMMARY OF RESULTS AND RECOMMENDATIONS	2
BACKGROUND ON THE SELECTION OF LUBRICANTS AND TEST CONDITIONS	4
DESCRIPTION OF TEST EQUIPMENT	8
TEST PROCEDURE	12
RESULTS AND DISCUSSION	15
A. Rotating Test Results	15
B. Oscillating Test Results	25
REFERENCES	30
TABLE I	
FIGURES	
APPENDICES	

LIST OF ILLUSTRATIONS AND TABLES

Figure Number	
1	Effect of Speed on Torque of Gold-Plated Bearings. Bearing-Torque Tube Bearing with a Bore of 2.5625 inches. Thrust Load 0.6 pounds.
2	Bar Chart of Solid Lubricant Performance: R-4 size bearings, 3000 rpm, 0.8 pound thrust load, 10^{-6} mmHg.
3	Vacuum Systems for Bearing Evaluations.
4	Schematic of Vacuum System Layout.
5	Schematic of Ball-Bearing Test Rig Used for Vacuum Tests.
5a	Parts List for Ball-Bearing Test Rig.
6	Partially Assembled Bearing Test Rig.
7	Close-Up of Bearing Test Rig Mounted to Top Cover of Vacuum Chamber.
8	Clean Room Assembly Area.
9	Schematic of Static Load Test Rig.
10	Average and Peak Torque Values Measured on Three Test Bearings at 30 rpm. Bearings 52100 Steel, Size 205. Phenolic Retainers. Lubricant - none. Load - 10 pounds radial.
11	Typical Contact Areas on Dry, Clean Bearings.
12	Average and Peak Torque Values Measured on Three Test Bearings at 30 rpm. Bearings 52100 Steel, Size 205. Phenolic Retainers. Lubricant - silicate bonded MoS_2-graphite film. Load - 10 pounds radial.
13	Typical Contact Areas on Bearings Lubricated with Silicate Bonded MoS_2 - Graphite Film.
14	Disassembled Bearing with Tool Steel Retainer. Lubricant - Sodium Silicate-Bonded MoS_2 - Graphite Film.
15	Average and Peak Torque Values Measured on Three Test Bearings at 30 rpm. Bearings 52100 Steel, Size 205, Phenolic Retainers. Lubricant - Burnished MoS_2 Film. Load - 10 pounds radial.
16	Debris on Separator Plates of Bearing Lubricated with Burnished MoS_2 Film.
17	Typical Contact Areas on Bearings Lubricated with Burnished MoS_2 Film.
18	Average and Peak Torque Values Measured on Three Test Bearings at 30 rpm. Bearings, 52100 Steel, Size 205, Phenolic Retainers. Lubricant - Evaporated Metal Films with Outer Layer of MoS_2. Load - 10 pounds Radial.

LIST OF ILLUSTRATIONS AND TABLES (cont'd)

Figure
Number

19 Typical Contact Areas on Bearings Lubricated with Evaporated Metal Films and Outer Layer of MoS_2

20 Average and Peak Torque Measurements on Three Test Bearings at 30 rpm. Bearings, 52100 Steel, Size 205, Phenolic Retainers. Lubricant - High Vacuum Oil. Load - 10 Pounds Radial.

21 Continuation of Tests on Bearings Lubricated with High Vacuum Oil Showing Torque Failures and Subsequent Behavior.

22 Conclusion of Rotating Tests at 30 rpm on Bearings Lubricated with High Vacuum Oil.

23 Typical Contact Areas on Bearings Lubricated with High Vacuum Oil.

24 Average and Peak Torque Measurements on Three Test Bearings at 30 rpm. Bearings, 52100 Steel. Phenolic Retainers. Lubricant, MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film).

25 Typical Contact Areas on Bearings with MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film).

26 Oscillating Test. Torque Values on Three Test Bearings. Bearings, 52100 Steel. Phenolic Retainers. Lubricant - MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film).

27 Oscillating Test. Typical Areas on Bearings Lubricated with MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film).

28 Oscillating Test. Torque Values on Three Test Bearings. Bearings 52100 Steel. Phenolic Retainers. Lubricant-Silicate Bonded MoS_2 Graphite Film.

Frequency is 2 cycles per second and 14.3 cycles per second.

29 Oscillating Test. Typical Areas on Bearing with Silicate-Bonded MoS_2-Graphite Film.

Table Number

1 Appearance of Bearings After Test.

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INTRODUCTION

During the last eight years, increasing effort has been placed on the development of suitable lubricants and lubrication techniques for rolling contact bearings to operate in space. Low vapor pressure oils and greases have already been used successfully in many satellites. However, there are many reasonable doubts concerning the ability of these liquids to operate effectively under these conditions for long periods of time.

Solid lubricants appear to have many advantages over conventional oils for these applications. In general, these advantages include the following:

1. Extremely low vapor pressures.
2. High resistance to dissociation.
3. High radiation resistance.
4. The ability to function over a much wider temperature range.

The major drawback to the use of solid lubricants in rolling contact bearings is a lack of experience and design criteria which can be used to predict performance and life. Years of experience and innumerable test results on oil or grease-lubricated bearings have established a reasonable picture of the effects of operating variables on bearing life under normal conditions. In contrast, there is virtually no general information of this type available for solid lubricants in rolling contacts. Almost all of the emphasis has been placed on the feasibility of using these materials in particular operations.

The objective of this program was to develope systematic test data on the performance, in vacuum, of a limited number of ball bearing lubricants being considered for spacecraft applications and to evaluate the test data to determine those lubricant systems which are most suitable for long-life space applications.

A series of ball bearings tests were run in slow speed rotation under standardized test conditions with one variable, the lubricant itself. Two particular lubricants were also evaluated with oscillating motion. These tests were made in vacuum, using torque as the criterion for failure. This report describes and discusses the results obtained.

SUMMARY OF RESULTS AND RECOMMENDATIONS

1. Based on the results of these tests, the sodium silicate bonded MoS_2-graphite film appears to have the most promise for long life in slow-speed rotating or oscillating rolling-contact bearings. A period of run-in prior to using these bearings is essential for consistent operation. At the present time, a sixteen hour run-in at the desired load and speed appears to be adequate. It is necessary to measure the torque of these bearings before and after run-in to determine how this is affecting performance and to screen out faulty bearings.

2. The mode of failure in these solid-film lubricated bearings is wear of the lubricant film and subsequent jamming of the bearings by the loose debris. This would probably be even more serious if the bearings had been mounted on a horizontal instead of a vertical shaft. On the vertical shaft, much of the debris was able to fall out of the bearings.

3. The failures obtained with the oil-lubricated bearings indicate that there is a potential problem with conventional lubricants and that evaporation of the oil is not the only mode of failure.

These test results indicate that the sodium silicate bonded film is generally better than any of the other contenders. This film has certain inherent advantabes including the following:

1. A number of investigations have been published and more test information is available on the performance of this film than any other solid lubricant.

2. The method of application is straightforward and a complete description of the process is available.

3. The constituents of this film are completely inorganic and can be used in high radiation or high temperature applications.

However, there is considerably more information required under carefully controlled test conditions. In particular, the effect of bearing clearances, type and extent of loading and the effect of speed should be determined. Evaluations should also be made using other retainer materials besides the phenolics. By a suitable choice of retainer materials, it might be possible to use much thinner lubricant films, thus lessening the problem of debris generation. This type of information would be a significant contribution to our fund of knowledge on solid-lubricated bearings.

BACKGROUND ON THE SELECTION OF LUBRICANTS AND TEST CONDITIONS

Solid lubricants for rolling element bearings which must operate under vacuum conditions have been studied in several programs. A summary of the results of a literature search in this area is included as Appendix B of this report. In the following paragraphs, this background is discussed briefly as it applies to this particular work.

A. Comments on Lubricants

1. Solid Lubricant Films

For the most part, the work on solid film-lubricated bearings has been concentrated on bonded films of MoS_2 or Teflon, and on soft metal films such as gold, silver or bismuth. In our own previous work, a series of tests performed on silver, gold, and silicate-bonded MoS_2 films using large, low-speed ball bearings, had shown that torque levels on the order of .07 inch ounces at 300 rpm should be achieved when the bearings are properly lubricated. These were torque-tube bearings with a bore of 2.5625 inches. These tests also showed that the metallic films, particularly gold, were sensitive to operation at very low temperatures, on the order of -100 F. The silicate bonded MoS_2-graphite film, although its torque was somewhat higher than the metal platings, was more consistent over the entire temperature range from -100 F to +400 F. Except for extremely low temperature operation, gold was a good lubricant in large bearings ranging in bore from 20 mm to 35 mm, running at low speed under very light thrust loads. Some attempts were made to determine the effect of speed on bearing torque. As shown in Figure 1, under a light thrust load the torque increased linearly with speed in the range from 16 rpm to 400 rpm.

The performance of small, high speed rolling element bearings gave very different results than the low speed bearing tests. Figure 2 is a bar graph showing the life obtained with R-4 size bearings operating at 3000 rpm with a 0.8 pound thrust load. Gold and silver were the poorest candidates in these tests.

These results illustrate the danger of extrapolating data from one particular test to another, entirely different, set of conditions. Our own experience has shown that many factors must be considered including the tenacity of the film to the substrate, the mechanical properties of the film and the substrate, and the wear rate and nature of the debris which is formed. Generally, metal films such as gold or silver are very effective under light loads but these are very sensitive to both speed and load. Bonded solid lubricant films appear to have better load and speed capability, but these films are difficult to apply uniformly and will wear appreciably during run-in. This can result in jamming by loose debris or it can result in a significant change in the positioning function of the bearing.

2. Retainer Supply Techniques

When solid lubricant films are used, the effective life of the bearing is determined by the wear life of the film. Once the film wears through, metallic contacts will eventually cause bearing failure. Therefore, it is logical to consider some means for replenishing the film. One approach is to make the retainer as a source of solid lubricant with the expectation that the material transfer from the retainer to the balls and races will provide a constant supply of lubricant. In the following table, a few typical results of using this approach are given:

RETAINER - LUBRICATED ROLLING CONTACTS

Retainer	Load Lbs.	Speed RPM	Temp. °F	Life Hours	Ref.
Ag-PTFE-WSe_2	75	1800	80	> 100	1
Bronze PTFE-MoS_2	75	1800	80	> 100	1
PTFE-Glass Fiber-MoS_2	75	1800	80	> 100	1
MoS_2-Sodium Silicate in Reservoirs	5	10000	750	1148	2
PTFE-Ceramic Fiber-MoS_2	70	1750	80	>8000	3
PTFE-Ceramic Fiber-MoS_2	200	600	80	800	3
PTFE-Ceramic Fiber-MoS_2	10	5000	150	565	3
PTFE-Ceramic Fiber-MoS_2	10	6000	260	122	3

One limitation to this technique is the sacrifice in the structural strength of the retainer material. It is also doubtful if this approach could be used for lubricating bearings which oscillate rather than rotate. Unless the amplitude of oscillation is large enough to allow the balls to transfer material from the retainers to the races, the bearings will not be adequately lubricated.

B. Comments on Test Conditions

For the most part, bearing evaluations with solid lubricants have been for specific requirements and, as a consequence, general design data has not been obtained. One particular aspect which deserves more consideration is the level of torque produced by the bearings under a variety of operating conditions, and the consistency of this torque as a function of the acceptable life of the bearing. This has led to a situation where the term, life, is not completely meaningful unless a situation identical to the particular test arises.

It is also difficult to determine what test conditions should be selected for the bearing evaluations. For example, it is normally most convenient to test under conditions of loading where the weight of a floating housing or rotor provides a consistent load. Torque can easily be measured by restraining the housing from rotation. However, there are many applications where this method of loading does not simulate the actual conditions which will be encountered in the application.

Bearing geometry presents another problem. Under normal conditions in oil-lubricated bearings, the geometry can be clearly specified with regard to bearing size, permissible torque and radial clearances which are required to accomplish the positioning function. However, when the lubricant is a bonded solid film which may wear rapidly during the initial stages of operation, should the clearances be chosen to accommodate the film or to provide reasonably accurate positioning?

Material and Bearing Design Selections

The present test series undertaken for NASA sought to combine two objectives; (1) to obtain data that was pertinent to a series of applications for which NASA wished to determine the capability of various lubricants, and (2) to start on a logical and controlled test procedure which would yield design information under known operating conditions.

Most of the evaluation work on solid lubricated rolling element bearings has been done either under direct load conditions or in equipment which has considerable excess power and as a consequence would not be sensitive to changes in bearing torque. Although it was recognized that extremely wide internal clearances would be more benign to the operation of solid film lubricated rolling element bearings, the initial wear-in would result in a considerable change in radial play. Such changes are not compatible with precise locating requirements unless, of course, there is a mechanism, such as a load spring, which continuously removes play from the system. In the latter case there is always a question of the ability of the bearing to slip in its housing to accomplish this compensating function. As a consequence, it was felt that the specified radial clearance, .0008 to .0011 was a reasonable range which would provide acceptable radial location without the necessity for superimposed preloading. A review of potential applications indicated that the dominant loading to be expected was in the radial direction. Although it would have been much more convenient from the standpoint of load application and torque measurement to use a pure thrust load (the weight of the housing) it was felt that certain jamming characteristics inherent to the radial system would not be observed. Another critical aspect in generating useful design data was to specify the applied load and maintain it throughout the test conditions. Much of the available information specifies the load only as "light" and leaves much to the designers imagination.

For the present program it was decided that a ten-pound radial load constituted a realistic load level and the clearance range specified was compatible with location requirements of the various applications.

Solid lubricant films rather than composite retainer materials were selected for these evaluations since the bearings were to be run in both rotation and oscillation.

DESCRIPTION OF TEST EQUIPMENT

1. The Vacuum Systems

A photograph of two of the vacuum systems, and a schematic of the vacuum system design, are shown in Figures 3 and 4. Because of the nature of this work, no oil pumps were used, either for rough pumping or for the high vacuum pumps. This oil-free pumping system feature was considered essential to prevent accidental contamination of the test materials.

Each vacuum system consisted of a vertical stainless steel cylinder with an inside diameter of 14 inches and a working depth of 14 inches. A 100-liter-per-second triode ion pump was mounted in the bottom of the chamber. Rough pumping was done by a pair of sorption pumps. Additional pumping capacity was also available from a 300-liter-per-second sublimation pump mounted on one side of the vacuum chamber. However, during these tests, this pump was only used on rare occasions. Finally, the use of liquid nitrogen-cooled reservoirs and chill plates, which are described in more detail in the section on Evaporative Simulation, also provided very fast pumping for condensible vapors. The chamber vacuum was measured by means of a General Electric trigger gage mounted in the center of the chamber as shown in Figure 4.

These systems have no elastomers, copper O-rings were used throughout for sealing flanges. The systems were completely bakeable. Four feedthroughs were provided in the cover of the vacuum chamber for electrical connections. Each feedthrough could handle eight wire connections.

2. Temperature Controls

To permit tests to be run at low temperatures, a refrigeration coil was welded to the inside of the wall of the vacuum chamber. This coil was connected to an air-cooled cascade refrigeration unit which expanded Freon through the coil. The coil temperature could be controlled in the range from +50 F to -100 F. Additional cooling was also available from the liquid nitrogen-cooled condensing plates which viewed the bearings.

If required, heat could be supplied to run the bearings at higher temperatures by disconnecting the refrigeration system and using a bake-out mantle to heat the walls of the chamber. Nichrome wire-wound heaters in quartz tubing could also be mounted on the bearing housings.

3. Bearing Test Equipment

A schematic drawing of the bearing test equipment is shown in Figure 5. A list of the names of the essential parts are given on the page following this Figure (Figure 5a). With very few exceptions, all parts were made of polished austenitic stainless steel. Photographs of this equipment are shown in Figures 6 and 7.

This test rig consisted of a vertical shaft supported on each end by ball bearings mounted in end plates. These end plates were tied together by three mounting posts which also served to hold the torque measuring flexures and the radial load flexures. The whole bearing assembly was bolted to the top cover of the vacuum chamber.

Three test bearings, in stainless steel housings, were mounted on the center span of the vertical shaft with sleeve spacers between each bearing to provide mounting surfaces for the inner races. The top and bottom support bearings were lightly preloaded in the thrust direction with shims. Each of the three test bearings was subjected to a ten-pound radial load by means of a stainless steel wire pulling against the bearing housing. One end of this wire was fastened securely to the housing. The other end was attached to a cantilevered steel flexure by means of a bolt with a set of locking nuts for load adjustment. Strain gages were mounted on these steel flexures. The spring rate of the flexure was made small so that small changes due to expansion would have a negligible effect on the applied load. To apply the load, the bolt and nut assembly was adjusted until the strain gage signal indicated the desired load. Then the locking nuts were pulled up tightly in place. Initial tests showed that new stainless steel wire would stretch and relieve the load gradually under these conditions. However, it was found that if the wire was prestretched with a heavier load, this problem was eliminated.

These test bearings also had a 1.6 pound load in the thrust direction which was imposed by the weight of the bearing housing.

Torque was measured on each test bearing by restraining the bearing housing from rotation with a cantilevered flexure arm on which strain gages were mounted. The positions and contact points of these flexures were carefully adjusted prior to each run. Since the housing was also restrained from rotation by the radial load wire, a low spring rate of the load cantilever was required, and torque calibration was done with the load applied.

Temperatures were measured with four copper-constantan thermocouples. Three of these couples contacted the outer races of the three test bearings. The fourth couple was used to monitor the structure temperature. These temperatures were recorded on a Brown recorder.

The test rig was rotated by means of a magnetic drive acting through the top cover of the vacuum chamber. The driven magnet was mounted on the top end of the bearing test shaft inside the vacuum chamber. The drive magnet was mounted outside the cover and was turned by a variable speed DC motor. A recess was machined in the cover, so that these two magnets could be positioned as closely together as possible.

Two tests were also run with an oscillating motion instead of rotation. The scheme for achieving oscillation is described at the beginning of the section where these test results are reported.

Speed was measured with a magnetic speed pickup which sensed the teeth of a 60-tooth steel gear mounted on the lower end of the test shaft. The signal from this pickup was amplified by a Kintel amplifier and read off of a Hewlett Packard frequency meter.

4. Evaporative Simulation

When a molecule is desorbed or evaporated from a surface exposed to outer space, to all intents and purposes, it is lost. Only rarely will this molecule collide with another molecule and be returned to the surface. To simulate this condition in the laboratory, more than low pressure is necessary. Condensing surfaces must also be provided to trap the molecules which escape. To provide for this condition, a liquid nitrogen reservoir system was built into the top cover of the vacuum chamber. This reservoir system consisted of three two-inch diameter cylinders which extended down inside the chamber around the test rig. The cylinders were interconnected below the test rig by smaller diameter tubing which was coiled to compensate for thermal expansion and contraction. Copper condensing plates were bolted to the frame of these cold tubes and extended above and below each of the test bearings. The configuration of these copper plates provided an optically tight path from the evaporating surfaces of the bearings so that cross contamination was not possible. An automatic liquid nitrogen control system maintained the proper liquid level in the cryogenic reservoirs.

Working Area

The room in which these vacuum systems are located is a dust-free area. Atmospheric control is maintained by filtering the inlet air through Absolute filters. The room is maintained under a slight positive pressure in order to minimize dust contamination. An adjacent room has also been set up to do precision assembly work. These clean room facilities are considered essential to protect assemblies which have to be checked out in air before running vacuum tests. Figure 8 is a photograph of the assembly area.

Static Load Test Rig

This test rig was used to apply a 2100-pound radial load to each of the test bearings before the vacuum tests. The purpose of this test was to determine how well the solid film-lubricated bearings could withstand a typical launch load. The test rig is shown schematically in Figure 9. Four bearings, three of them test bearings and one a dummy bearing were mounted on a shaft and the whole assembly was set into the test fixture. A hydraulic jack was used to apply a 4200-pound load to the two center bearings (2100 pounds on each). The two end bearings equally shared the 4200-pound reaction load.

TEST PROCEDURE

1. General Notes

For each test, five bearings were required for the test rig. Three of these were test bearings and the other two were support bearings. In every case, the same bearing-lubricant combination was used for both the support bearings and the test bearings. This was done to avoid any possibility of contaminating the test bearings with alien materials.

The unpackaging of the test bearings and all test assembly work was done in the dust-free rooms, generally in Sterishield cabinets. No test bearings were handled with bare hands. Either nylon or polyethylene gloves were worn during assembly operations.

2. Test Sequence

a) The bearings were unpacked, examined, assembled if necessary, and mounted in the bearing housings on the bearing test rig shaft.

b) The bearing torques were measured in air at 30 rpm with a ten-pound radial load applied. If the results indicated that a run-in was necessary, it was done at this point and the torque was remeasured at the end of the run-in.

c) The bearings were removed from the test rig shaft and were mounted in the static load rig.

d) A 4200-pound static radial load was applied to each pair of bearings (2100 pounds per bearing) for fifteen minutes. The bearings were not rotated during this load cycle.

e) The bearings were taken out of the static load rig, mounted in the test rig, and torque was again measured in air at 30 rpm with the ten-pound radial load.

f) The test rig was then set in the vacuum chamber, evacuated, and the bearing temperature was brought down to +20 F.

g) The bearing torque was measured at 30 rpm and the vacuum life test was started.

3. Sequence and Duration of Life Tests

It was originally specified that each bearing-lubricant combination was to be run for 1000 hours (unless failure occurred) under the following test conditions:

Speed - 30 rpm in rotation
Radial Load - 10 pounds
Thrust Load - 1.6 pounds (housing weight)
Ambient Temperature - +20 F
Vacuum - 1×10^{-7} Torr or better

In certain tests, longer runs were made if the bearings were still in good condition at the end of 1000 hours.

Failure was defined as a two-fold increase in torque. Again, many tests were continued well beyond this point to be sure that the torque would remain high.

4. Description of Test Bearings and Lubricants

Test Bearings

All of the bearings used in these tests were ABEC-7,Type 205, angular contact bearings of 52100 steel. Nominal clearances (.0008 - .0011 inches) were specified. The retainer was a fabric-laminated phenolic.

Lubricants

a) A dry, unlubricated bearing with a phenolic retainer.
b) A sodium silicate bonded MoS_2-graphite film applied to the balls, races and retainers. Two sets of bearings were evaluated with this film. One set had phenolic retainers while the other had M-10 steel retainers.
c) A burnished MoS_2 film on the balls, races and retainers.
d) A proprietary film which consisted of depositing at least two metal layers

and coating this with sulfide films finished off with an MoS_2 film. This film was applied only to the balls.

e) A proprietary mineral oil-based low-vapor pressure oil. This oil was impregnated in the phenolic retainer and the rest of the bearing components were also generously lubricated with the same oil.

f) A proprietary film formed by running the bearing in a mixture of MoS_2 in a liquid, until a well-defined film was formed. The excess film was then wiped off, leaving a dry film on the surfaces.

RESULTS AND DISCUSSIONS

A. Rotating Test Results

1. Dry, Unlubricated Bearings

These bearings were received in the unassembled condition by MTI. The balls and races were cleaned with acetone and benzene, but nothing was done to the phenolic retainers.

During the initial torque tests in air, torque values ranging from .007 to .024 inch-pounds were measured with these dry bearings under a ten-pound radial load. After applying the static radial load of 2100 pounds to each bearing for 15 minutes, the torque values under the ten-pound radial load were lower and steadier, ranging from .004 to .008 inch-pounds.

A vacuum was drawn on the system and the bearing temperatures were allowed to stabilize for six days. The vacuum level at the start of this test was 3×10^{-8} Torr and remained roughly the same throughout the test.

The following table summarizes the data obtained during the start of this test:

Date	Pressure	Speed	Top Bearing		Center Bearing		Lower Bearing	
	Torr	RPM	Temp. ^{o}F	Torque in-lb	Temp.	Torque	Temp.	Torque
1/20/65 10 a.m.	$3x10^{-8}$	30	25	.003 avg .041 peak	24	.016	28	.018
1 p.m.	$4x10^{-8}$	30	20	.003 avg .033 peak	20	.028	25	.018
1/21/65 8.30 a.m.	$4x10^{-8}$	30	22	.003 avg .065 peak	22	.033	22	.039
9.20 p.m.		Bearings locked up. Restarted by reversing motor and raising temperature						
10 a.m.	$3x10^{-8}$	30	35	.003 avg .057 peak	35	.033	30	.053
1 p.m.	$3x10^{-8}$	30	25	.003 avg .016 peak	25	.033	25	.073

The test then had to be shut down for several days because of problems with the liquid nitrogen supply system.

Since the bearings were still in operating condition, in spite of the one failure, another attempt was made to run the 1000-hour test. The results from this point on are plotted as torque vs. time curves in Figure 10.

After 24 hours of reasonable operation on the three test bearings, the top bearing began to show very erratic torque values again. After about 100 hours, the middle and lower test bearings showed erratic torque while the top bearing improved. The test ran for 408 hours before the rig seized up. This time, the test could be restarted by reversing the motor, but the bearings seized again after a few minutes of running time.

These results bear out past experience on evaluating dry bearings in other vacuum systems. Generally, these bearings give extremely low and steady torque values at the start of the test, and then hang up in a short period of time. If the test can be restarted, by reversing rotation or jarring the fixture, the bearings will usually run for extended periods of time although the torque will be very erratic. The second time that the bearing seizes, it is usually not possible to restart the test and keep it running.

It is believed that the reason for this gradual deterioration is associated with the cleanliness of the bearing surfaces. Even the most scrupulous cleaning with solvents will not remove all of the contaminating films and, in this test, the use of a phenolic retainer certainly resulted in extensive outgassing in the bearing. In order to see a complete failure, it would be necessary to clean all of the bearing components by some rigorous procedure such as electrolytic degreasing in a trisodium phosphate solution and to use all metallic components. These contaminating films can help the bearing to survive for unusually long periods of time in spite of erratic operation. Eventually, the films will be depleted enough to permit extensive bare metal contact and this generally terminates operation very quickly.

The test rig was disassembled and examined. To the eye, the bearings appeared to be in fairly good condition although a frosted track was visible on the raceway. However, microscopic examination disclosed many minute welds and material transfer. Figure 11 is a series of photographs of the test bearings. The photomicrographs were taken of the inner race ball track. Presumably, these tiny welds were the cause of the erratic torque which was measured on these bearings. The retainers appeared to be in good condition after these tests.

Failure in these tests was due to welding and material transfer on a microscopic scale.

2. Bearings lubricated with Sodium Silicate-Bonded MoS_2-Graphite Film

These bearings were received in the assembled condition. All bearing components had been coated with a heavy lubricant film. When the test was first run in air, considerable debris was noted on the separator plates. This was apparently from wear of the lubricant film. For this reason, the bearings were run-in for two hours, then the separator plates and the exposed areas of the bearings were cleaned out with a hand vacuum cleaner to remove excess debris. The bearings were then run for one more hour. Since the torque values were reasonably steady, and very little debris was still coming out of the bearings, the 2100-pound static radial load was applied to each bearing. The torque values after this step were considerably steadier. A vacuum was drawn on the system. When the bearings were pumped down initially, the vacuum levels were varying between the 10^{-9} and 10^{-10} scale. However, as soon as the test was started, the pressure increased and leveled out on the 10^{-7} scale. The pressure remained in the 10^{-7} to 10^{-8} Torr region for the rest of the test. This is believed to be the result of substantial outgassing from the silicate binder coupled with the fact that considerable loose debris was still being produced by wear of the lubricant film. Similar results were reported on this film in Reference 4. There was also a problem in maintaining temperature on these bearings. Most of the time, the bearings would only stabilize at temperatures just above 0 F. This seemed to be due to some peculiarity, such as the emissivity of the coating rather than the fault of the refrigeration system. The lower temperature levels did not appear to have any effect on bearing performance.

Figure 12 is a plot of the torque vs time trace on these bearings. After about 48 hours of running time in vacuum, the top bearing showed a sudden torque increase from about .02 to 0.1 inch pounds. The torque on this bearing remained high for the rest of the test. The other two test bearings gave low and steady torque values throughout the test.

The bearings were still running effectively after 1000 hours, so the test was continued at the request of the NASA project engineer. At the end of 1782 hours, there was still no indication of any significant torque changes. The speed was then increased from 30 rpm to 100 rpm. After about 72 hours at 100 rpm, the test rig became very noisy but neither the torque traces nor the speed pickup showed any significant changes. At the end of 188 hours of running at 100 rpm the rig appeared to be locking up periodically. The test was continued for a while longer but no improvement was observed.

The test was then shut down and the chamber was opened. It was found that the inside drive magnet had worked loose and was periodically slipping on the shaft. All of the bearings were still free, even under load, and rotated easily. These bearings apparently could have run effectively for a much longer period of time. Considerable debris from the lubricant film was found on the separator plates.

Figure 13 shows some typical photographs of the ball and race surfaces. The top bearing had a very thin coating but it was still intact. The other bearings still had considerable lubricant film on the contacting areas.

No failures were obtained in this test.

3. Bearings With Tool Steel Retainers Lubricated by Silicate-Bonded MoS_2-Graphite Film

An attempt was also made to run test bearings with M-10 tool steel retainers using the silicate-bonded MoS_2-graphite film. These bearings were in the assembled condition when they were received. The bearings were set up in the test fixture and run-in for an hour to remove any excess lubricant film. At the end of this run-in, the torque had increased substantially. The bearings were disassembled and examined. It was found that the balls were contacting the bottom of the retainer pocket rather than the side. This would indicate that the retainer was not properly dimensioned. Nothing further was done with these bearings. A photograph of one of the disassembled bearings is shown in Figure 14.

4. Burnished MoS_2 Films on Balls, Races and Retainer

These bearings were received by MTI in the assembled condition. Very high torque was noted when the bearings were being assembled in the test rig. For this reason, the bearings were "run-in" in air for one hour to be sure that they would be free enough to run in the vacuum tests. Even then, the peak torque values were high but there was a drop in the average values. Considerable debris from the lubricant film was observed on the separator plates after this run-in. After the static load test of 2100 pounds per bearing was applied, the peak torque levels were lower.

In the initial stages of the 1000-hour test, very high peaks in torque were measured on the numbers 2 and 3 bearings, but subsequent running resulted in considerable improvement. These data are shown in Figure 15. After about 200 hours, all three bearings were reasonably consistent, giving torque values in the range of .01 to .04 inch pounds. After about 620 hours, the numbers 1 and 2 bearings showed a definite increase in torque level. After 730 hours, the number 3 bearing showed a small, but definite, increase in torque. This test was stopped after 771 hours since there was no indication that the torque would decrease again. The vacuum levels during this test ranged from $7x10^{-8}$ to $1x10^{-7}$ Torr.

When the test rig was removed from the chamber, considerable loose, black debris was observed on the separator plates below each of the bearings. A photograph of two of the separator plates is shown in Figure 16. This debris appeared to be predominantly MoS_2.

The bearings were very stiff but, after the radial load was released and some of the loose debris was shaken out, the bearings became free again. Examination of the bearings showed that the two support bearings (these are lightly thrust-loaded) were polished and smooth. The balls in the best bearings were generally dull except for a few which still had a polished appearance. Apparently these were either not coated as heavily or were slightly undersized.

Photomicrographs of typical contact surfaces are shown in Figure 17. Failure in these tests was apparently due to an excessive amount of loose debris which tended to jam the bearings.

5. Evaporated Metal Films With Outer Layer of MoS_2

These bearings were received in a disassembled condition. The balls were coated with the solid lubricant film but no coating was apparent on the races or retainer.

Low and steady torque values were measured in air before and after applying the 2100-pound static radial load. The first attempt to run these bearings in the vacuum chamber was a failure because the inside magnet was rubbing on the top plate. This was remedied and the test was restarted.

Initially, all the torque values were very low and steady, about .01 inch pounds. These data are plotted in Figure 18. After 24 hours, the top bearing gave high torque readings but these decreased after about 90 hours. After 400 hours, the torque rose gradually, reached .2 inch pounds after 550 hours, and remained high for the rest of the test.

The number 2 bearing gave very low and steady torque values for 720 hours, then increased from .01 to .09 inch pounds and remained at this level for the rest of the test.

The number 3 bearing started very low and steady, became slightly erratic and gave higher torque after 48 hours of running, but dropped back to low torque after 100 hours and remained low and steady for 720 hours. After 720 hours, isolated peaks were observed on the torque trace at random intervals. These peaks seemed to indicate an occasional "grabbing" action of the bearing. This was not consistent and was no worse after 930 hours. The extent of the torque increase, which was measured when this unstable behavior was noted, is shown in Figure 18 as a dotted line with an arrow pointing to the peak value.

The test was concluded at this point since there was no indication that the performance of the top and middle bearings would improve with further running. The vacuum levels during this test were generally in the 10^{-9} to 10^{-10} Torr range.

When the test rig was disassembled, some debris was observed on the separator plates. Under the microscope, these appeared to be soft, dull, black particles. The shaft turned roughly under load but was free as soon as the load was removed.

Figure 19 shows photographs of typical contact surfaces on the balls and races. The balls had a silvery appearance indicating that the evaporated metal films had not worn down to the base steel surfaces. Some of the film from the balls had transferred to the steel races.

The torque failures on these bearings were due to the formation of loose debris which tended to jam the races.

6. Special High Vacuum Oil

These bearings were received in the assembled condition. The phenolic retainers had been impregnated with oil and a plentiful supply of oil was also visible on the balls and races. No attempt was made to wipe off the excess. Low, steady torque values were measured in air before and after the static 2100-pound radial load had been applied to each bearing.

During the initial pumpdown on these bearings, the vacuum held on the 10^{-6} Torr range for about three days, then suddenly dropped into the 10^{-8} to middle 10^{-9} Torr for the rest of the test. This three-day plateau was apparently due to outgassing of the oil.

Although the initial torque values in vacuum were low, these values were less steady than the values in air, probably because of an increase in the viscosity of the oil at lower temperatures. The lower bearing showed moderately high torque values throughout the 1000-hour test while the top and middle test bearings gave comparatively low and steady torque values. These data are shown in Figure 20.

At the end of 1000 hours, all of the test bearings appeared to be in good operating condition, so this test was continued. At the end of 1440 hours, bearings 1 and 2 showed a sudden torque increase. Forty-eight hours later, bearing 3 also showed a sharp increase in torque. This is shown on the torque-time trace in Figure 20.

This was an unexpected development. In the case of the solid-lubricated bearings, torque failures were all apparently due to the generation of loose debris which tended to jam the bearing. In no other case did all three of the test bearings fail within such a short period of time. There appeared to be two reasonable explanations for these torque failures:

1. Some constituent (or most of the oil) had evaporated away with time. This seemed most plausible.

2. Debris was generated in the top bearing and this dropped down into the other bearings thus causing an increase in torque. This seemed very unlikely since the bearings were protected from each other by an optically tight series of cryogenically-cooled baffles.

To evaluate the cause of failure, the bearings were first allowed to warm up to room temperature. This was done by cutting off the liquid nitrogen supply and the Freon refrigeration system. Unfortunately, this also released all of the organic material which had been condensed on the LN_2-cooled baffle plates so that considerable vacuum was lost and some bearing contamination probably took place at the same time. The torque did decrease on all three test bearings, although it was still high, particularly for bearings 1 and 2. Then the system was brought up to ambient pressure by bleeding nitrogen into the chamber. The torque levels dropped rapidly while this was being done. Finally, the system was opened to air.

In Figure 21, the torque traces for the latter portion of this test are plotted on an expanded scale to show the bearing performance during this time.

These results indicated that failure was not the result of debris in the bearings since the torque dropped when air was admitted.

Without removing the test rig from the chamber, a vacuum was drawn on the system again to see if this high torque behavior was reproducible. The test was run for 1072 more hours at 30 rpm. No significant changes in torque were observed up to this point. Then the speed was increased to 60 rpm. After 168 more hours of running, all three test bearings showed a sudden and substantial increase in torque. The torque vs time plot for this second test is shown in Figure 22.

When the bearings were removed from the test fixture, a considerable amount of oil was still visible on the bearing components. This showed that failure was definitely not due to a loss of oil.

Photographs of some of the contact surfaces are shown in Figure 23. Definite damage was visible on the raceways in the form of narrow but torn tracks. The support bearings had a much wider track with a satin finish. Very little damage was observed on the balls and the retainers appeared to be in good condition both on the guiding lands and in the ball pockets.

At the present time, only hypotheses can be advanced for these torque failures. Many investigators have predicted that conventional lubricants might not function satisfactorily for long periods of time in vacuum. A variety of reasons have been advanced to support this prediction. However, to the best of our knowledge , this is the first time that a definite torque failure has been observed on rolling contact bearings under this condition.

In Reference 4 , dioctyl phthallate was used as the lubricant in a ball bearing test which was run in an ion-pumped system. The authors reported extensive lubricant degradation which left patches of polymerized oil on the balls and races. They hypothesized that the ion pump might have caused cracking of the oil. This did not appear to be the case in these tests since there was plenty of lubricant in the bearings after the run and no evidence of decomposition was observed.

The most logical explanation appears to be a lack of water vapor or oxygen. Knowing the affinity of oils, and of materials such as the laminated phenolics, for water vapor and oxygen, it is not too surprising that more than 1000 hours of running time in vacuum could be accumulated before all traces of these impurities were removed. The importance of water vapor and oxygen in the lubrication process is a well-established fact. For example, Tingle (Ref. 5) has shown that freshly cut metal surfaces cannot be lubricated by conventional additives if air or water vapor is excluded. Godfrey (Ref. 6) pointed out the importance of oxide films in the lubrication process. In another paper (Ref. 7) it was noted that the load carrying capacity of oils was decreased more than 50 percent in vacuum.

7. Bearings Lubricated By Run-In In a Suspension of MoS_2 in liquid (Wiped Dry)

These bearings were received in the assembled condition. During the preliminary torque check in air, the center test bearing gave high peak torque values but, after running-in the bearings for ten minutes, all three bearings showed satisfactory readings. After the static load test, the lower bearing was found to be jammed. The bearing was worked loose by hand. It then appeared to be free and in good condition. No explanation has been found for this behavior.

The vacuum level at the beginning of the 1000-hour life test was on the low 10^{-7} range. For most of the test, the pressure was about 5×10^{-8} Torr.

Figure 24 shows the torque vs time curves for these bearings. The center bearing gave somewhat erratic torque values for about 344 hours, then showed a sharp increase in torque from .03 to .32 inch pounds and remained high for the rest of the time. The lower bearing showed a sudden torque increase after about 108 hours, but dropped back to a low steady value after running overnight. After 936 hours had been accumulated, the torque on this lower bearing suddenly increased from .03 to .12 inch pounds and was still increasing when the test was concluded at the end of 948 hours. The top bearing gave excellent results for the first 700 hours, then became more erratic although the torque was still low when the test was concluded.

Some dark debris, apparently from the lubricant film, was observed on the separator plates after the test. Photographs of the contacting surfaces on these bearings are shown as Figure 25.

The cause of the torque failures on two of these bearings was loose debris which tended to jam the bearings.

B. Oscillation Test Results

To obtain oscillating motion in the bearing test rig, a cantilevered, flexible, stainless steel beam was attached to the mounting structure and to the lower end of the test shaft. When the drive and driven magnets were aligned in phase, the shaft tried to rotate in one direction against the restraining beam. When the force on the beam reached a certain value, the magnets slipped out of phase and the beam recovered, thus rotating the shaft in the opposite direction. This motion was reproduced four times per revolution because of the four like and four unlike poles on each magnet. The actual arc of oscillation was measured with a dial indicator before the test was started.

In both of these oscillation tests, the conditions were as follows:

a) Load - 10 pounds radial on each bearing

b) Frequency - As specified (two frequencies, 2 cps and 14.3 cps were used)

c) Amplitude - One degree, single amplitude

d) Test Temperature - -25 F

Two solid lubricant coatings were evaluated in these tests. One was the bearings lubricated by run-in in a suspension of MoS_2 in liquid (wiped dry) and the other was the sodium silicate bonded MoS_2-graphite film. The results are described in the following paragraphs.

1. Bearings Lubricated by Run-In in a Suspension of MoS_2 in Liquid (Wiped Dry)

Previous tests had indicated that most of these solid film lubricated bearings required some run-in time to get rid of excess film and debris. Therefore, these bearings were run for three hours in rotation at 30 rpm under a ten-pound radial load. The results were as follows:

TORQUE-INCH POUNDS

Time	Number 1	Number 2	Number 3
Start	.065	.016	.01
1 Hour	.08	.02	.01
2 Hours	.095	.028	.01
3 Hours	.125	.028	.01

Since the torque on the number 1 test bearing was getting worse with time, one of the support bearings was exchanged for the number 1 test bearing.

After reassembling the test rig, the three-hour rotational run-in in air was repeated with the following results:

TORQUE-INCH POUNDS

Time	Number 1	Number 2	Number 3
Start	.032	.01	.01
3 Hours	.03	.01	.01

Then the test was converted to an oscillating motion of two cycles per second. The initial readings in air under a ten-pound radial load were:

Number 1	Number 2	Number 3
.032	.024	.012

After applying the 2100-pound static load to each bearing, the oscillating torque values in air under a ten-pound radial load were:

Number 1	Number 2	Number 3
.016	.032	.012

After drawing a vacuum on the system, and cooling the bearings to -25 F, the torque values were:

Time	Number 1	Number 2	Number 3
Start	.01	.012	.024
8 Hours	.01	.019	.012
24 Hours	.01	.012	.012

The vacuum level during this test varied in the range 10^{-9} to 10^{-10} Torr. After 204 hours, the number 1 bearing torque jumped from .01 to .16 inch pounds but then dropped back to .01 again. After 432 hours, the torque again increased to .031 inch pounds on this same bearing, then gradually dropped back to .01.

These occasional high torque values were probably the result of minute amounts of loose debris from the lubricant film. No significant torque changes were observed with the other two bearings during the first 1032 hours of the test. These torque data are shown in Figure 26.

At the end of 1032 hours, about 7.43 x 10^6 cycles had been accumulated on these bearings. The frequency of oscillation was then increased to 14.3 cycles per second and the test was continued at this higher frequency for 972 more hours.

As shown in Figure 26, the top bearing continued to give low and steady torque values but the other two test bearings, especially the lower bearing, showed marginal performance. The erratic torque values are believed to be the result of loose debris in the contact areas. The test was suspended at this point. A total of 57.47 x 10^6 cycles had been accumulated on these bearings at 2 cps and at 14.3 cps.

When the test rig was disassembled, a small amount of loose debris from the bearings was observed on the separator plates. This was apparently caused by wear of the lubricant film. Photographs of the areas where the ball was oscillating in contact with the race are shown in Figure 27.

2. Sodium Silicate Bonded MoS_2-Graphite Film

An attempt was also made to run-in these bearings for three hours in rotation in air but, at the end of this period, the torque values were still very high. These were:

	TORQUE-INCH POUNDS		
Time	Number 1	Number 2	Number 3
Start	.023	.05	.032
3 Hours	.032	.12	.16

Since a much longer run-in period was required, the run-in, in rotation, was continued overnight. The readings at the end of this time were:

Number 1	Number 2	Number 3
.032	.028	.032

The test was then converted to oscillating motion. In oscillation, the readings were:

Number 1	Number 2	Number 3
.016	.03	.08

After the 2100-pound static load on each bearing, the readings in oscillation in air were:

Number 1	Number 2	Number 3
.01	.012	.015

Under vacuum at -23 F, the readings in oscillation were:

Number 1	Number 2	Number 3
.01	.018	.032

These bearings were run in oscillation for 336 hours at a frequency of two cycles per second. The test had to be shut down at that point because of accidental damage to the vacuum system. Up to that time, reasonably low and steady torque values had been observed. Approximately 2.42 x 10^6 cycles had been accumulated in this run. These torque data are plotted in Figure 28.

The test rig was then moved to another vacuum system, without disturbing the bearings, and the test was restarted. Bearings 1 and 2 were unchanged, but bearing 3 showed slightly lower torque after this change. The data are shown in Figure 28 as a plot of torque vs time.

In general, the vacuum levels held on the 10^{-8} to 10^{-9} Torr ranges during this oscillating test. Apparently, the outgassing of the silicate film was not as much of a problem when the bearing was oscillating around one point as it was when the motion was complete rotation.

The test was run at a frequency of two cycles per second for 516 hours which represents 3.72 x 10^6 cycles of oscillation. Then the frequency was increased to 14.3 cycles per second and the test was continued for 504 more hours at this higher frequency. This amounted to 25.95 x 10^6 cycles. A failure in the liquid nitrogen supply took place at this point and the test was concluded.

These bearings had run for a grand total of 32.09 x 10^6 cycles (including the first part of the test) with very low and steady torque values.

Some debris was observed on the separator plates when these bearings were removed from the test rig. Figure 29 shows the contact areas on the balls and raceways.

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TABLE I. APPEARANCE OF BEARINGS AFTER TEST

1. Dry Clean Bearings, Rotating Test.

General Remarks: These bearings were cleaned in alcohol and benzene and were assembled by MTI. No lubricant was used.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Balls have a shiny but mottled appearance. Inner race has abrasion scratches in track which go in direction of rotation. Outer race has definite "pullouts" in three places on the track.	Balls appear to be fairly well polished but have a mottled appearance. Inner Race-under the microscope the track is rough and shows metal transfer. Outer Race-Similar appearance.	Balls appear to be fairly well polished. Races have fine, frosted track. Under the microscope, numerous tiny welds and metal transfer was observed.	Balls are dull and appear to be pitted under the microscope. Inner and outer races have fairly smooth tracks to the eye.

2. Bearings Lubricated with Silicate-Bonded MoS_2-Graphite Film, Rotating Test.

General Remarks: These bearings had a coating on the balls, races and retainers. They were received in the assembled condition.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Coating intact and in good condition.	Coating intact but noticeably thinner than other bearings.	Coating intact and in good condition on all components.	Same as center bearing.

3. Bearings with M-10 Steel Retainers Lubricated by Silicate-Bonded MoS_2-Graphite Films, Rotating Test

General Remarks: These bearings could not be run-in. There appears to be a poor contact between the balls and the retainers.

4. Bearings Lubricated with Burnished MoS_2 Film, Rotating Test.

General Remarks: Coating was originally applied to ball, race and retainer surfaces. A heavy coating was observed on all surfaces. These bearings were received in the assembled condition.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Balls were generally smooth and polished with a few dull spots. Inner Race-Ball track dull and unusually wide. Outer Race-Similar.	Balls were dull and mottled. Inner Race-Ball track indicates that ball motion was not steady but occasionally jumped off track. Outer Race-Track dull and wide.	Balls were dull and mottled. Inner Race-Track dull and wide. Outer Race - Similar.	Balls were dull and mottled except for three which were still polished. These balls were probably slightly undersize. The races were similar to Center Bearing. This bearing was still effective but the torque was rising.

5. Evaporated Metal Films With Top Layer of MoS_2, Rotating Test.

General Remarks: Only the balls were coated in this bearing. The bearings were assembled at MTI.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Balls showed light swirl patterns but some of the coating was intact. Inner Race-Smooth track, appears to have transferred lubricant.	Balls have very small islands of coating left. Inner Race-Smooth and polished. Ball track widens at one point. Outer Race-Similar in appearance.	Balls have a definite silvery color. A few have an MoS_2 coating on one area of the ball surface. Inner Race-Smooth and polished track. Outer Race-Similar but loose, dark debris on edges of track.	Ball surfaces have shiny, silvery color with speckled coating. Inner Race-Track polished but loose, dark debris on edges of track. Outer Race-Track smooth and polished. This bearing was still operating effectively except for small torque peaks.

6. Bearings Lubricated With High Vacuum Oil, Rotating Test.

General Remarks: Retainers were vacuum impregnated with oil. Balls and races also had oil on surfaces. These bearings were received in the assembled condition.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Balls show no evidence of damage. Outer race had wider track than test bearings. Retainers in good condition. Oil film visible.	Balls appear to be in good condition. Outer race has one thin roughened track. Inner race shows slightly less damage. Retainer in good shape. Plenty of oil left.	Same as top bearing.	Same as top bearing.

7. Bearings Lubricated By Running-In With Liquid Suspension of MoS_2, then Cleaning Off Liquid to Leave a Dry Film, Rotating Test.

General Remarks: These bearings were received in the assembled condition.

Support Bearings	Top Bearing	Center Bearing	Lower Bearing
Balls are polished with swirl pattern which is probably lubricant film. Races have smooth tracks with small amount of debris deposits.	Balls polished, no visible film. Races have smooth tracks with heavy deposits of brownish colored debris caked on track.	Balls shiny but with very light pits. Races similar to top bearing but track is slightly displaced at one point.	Balls are not as shiny and have more pits. Races have rougher tracks with considerable debris.

8. Oscillating Motion, Same Lubricant as Number 7.

General Remarks: Same as Number 7.

Support Bearing	Top Bearing	Center Bearing	Lower Bearing
Balls have swirl pattern in contact area. Races have transferred material built up in contact area.	One ball has heavy transfer of what seems to be lubricant film. Races show heavy lubricant debris transfer but no metal damage.	Balls are partially dull. Outer and inner races appear to have heavy pull-out or transfer in one area.	One ball shows heavy transfer. Races are similar to center bearing.

9. Oscillating Motion, Bearing Lubricated With Silicate-Bonded MoS_2-Graphite Film.

General Remarks: These bearings were received in the assembled condition and required a 16-hour run-in to get steady torque readings.

Support Bearing	Top Bearing	Center Bearing	Lower Bearing
Balls have lightly mottled appearance. Races have dull, black, heavy lubricant coating.	Balls have mottled appearance from lubricant film smearing on surface. Races have dull tracks with heavy lubricant films.	Same as Top Bearing.	Same as top bearing.

FIGURES

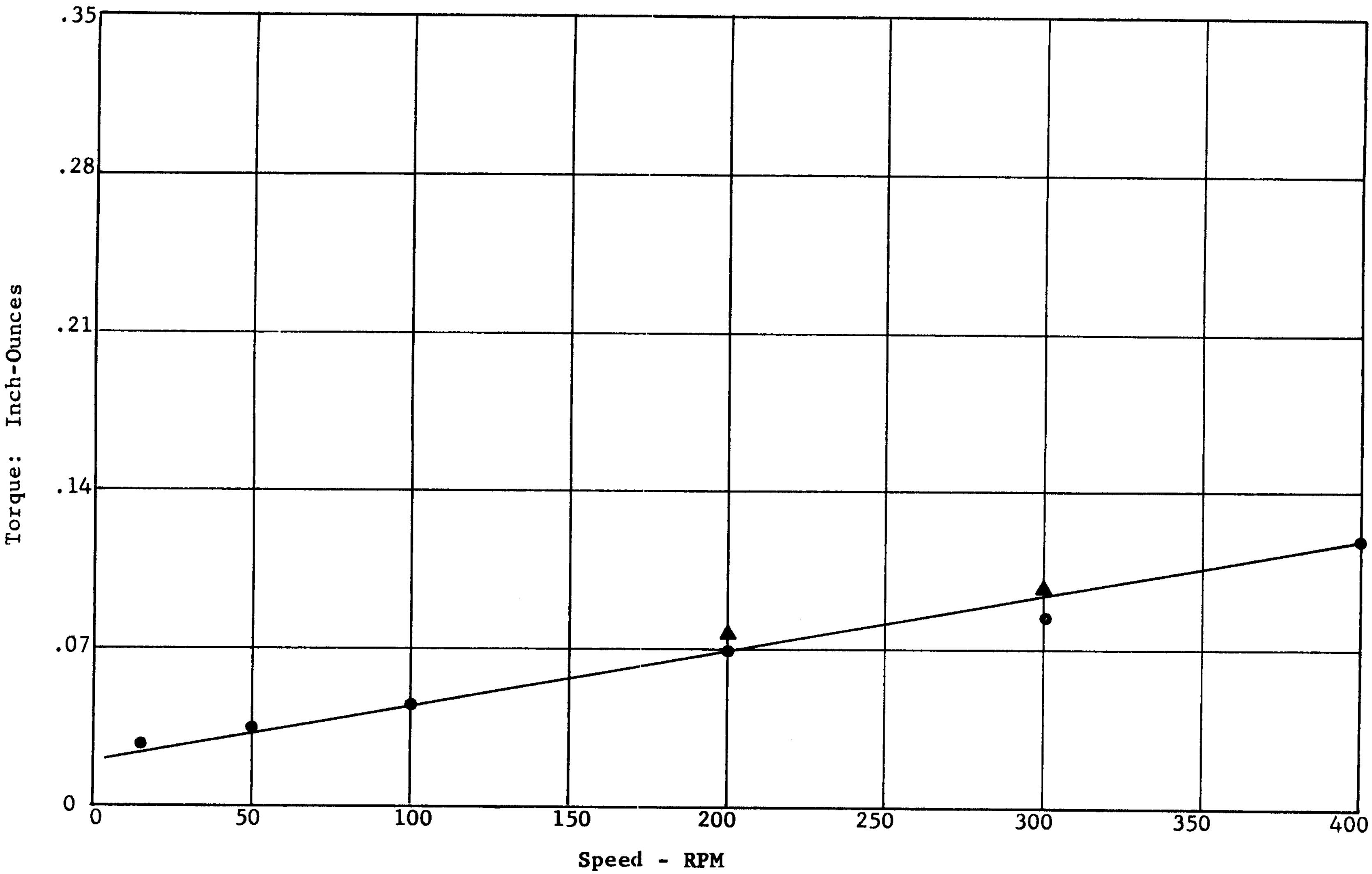


Fig. 1 Effect of Speed on Torque of Gold-Plated Bearings Bearing - Torque tube bearing with a bore of 2.5625 inches. Thrust load 0.6 pounds.

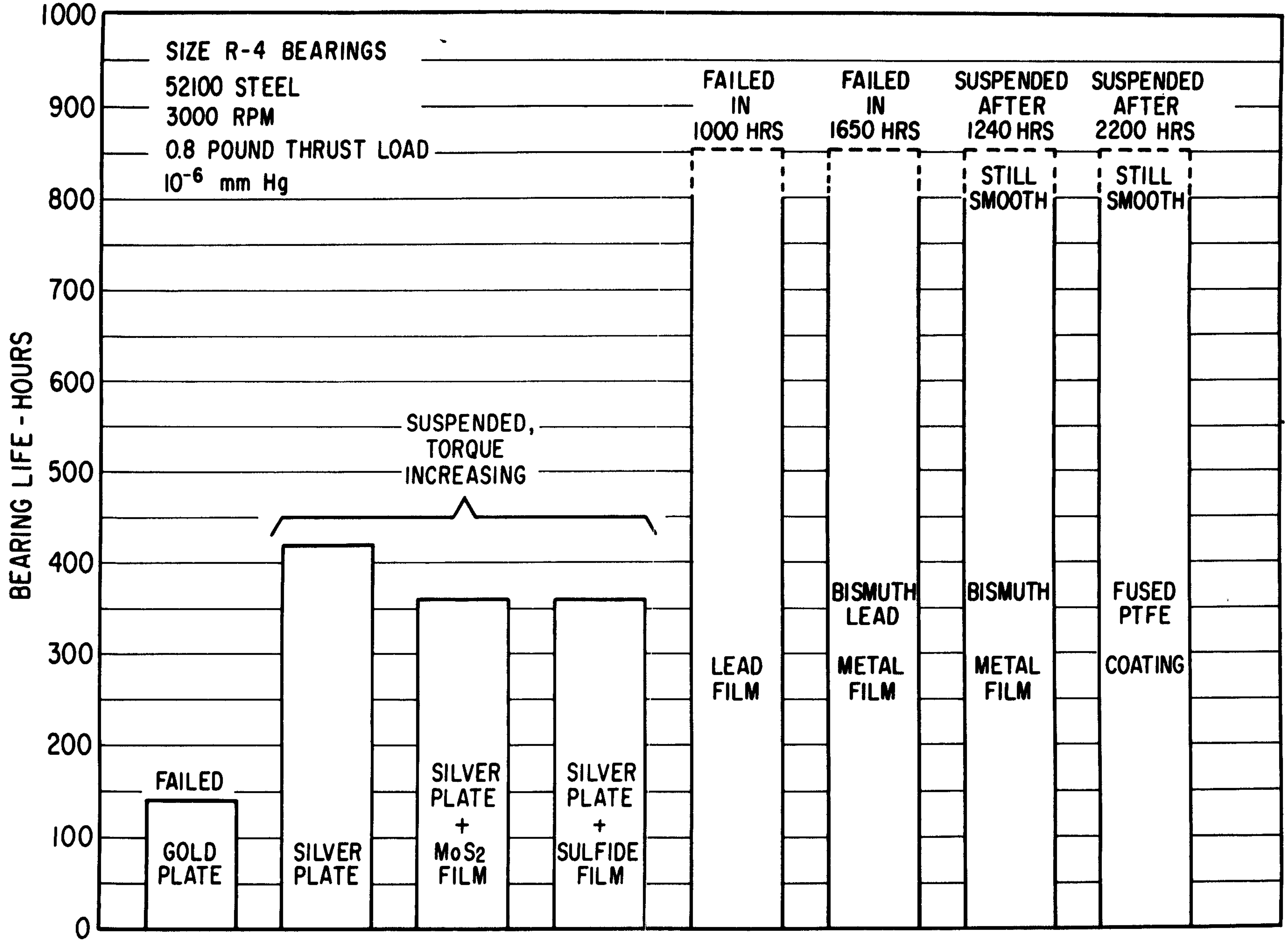


Fig. 2 Bar Chart of Solid Lubricant Performance: R-4 Size Bearings, 3000 RPM, 0.8 Pound Thrust Load, 10^{-6} mmHg

Fig. 3 Vacuum Systems for Bearing Evaluations

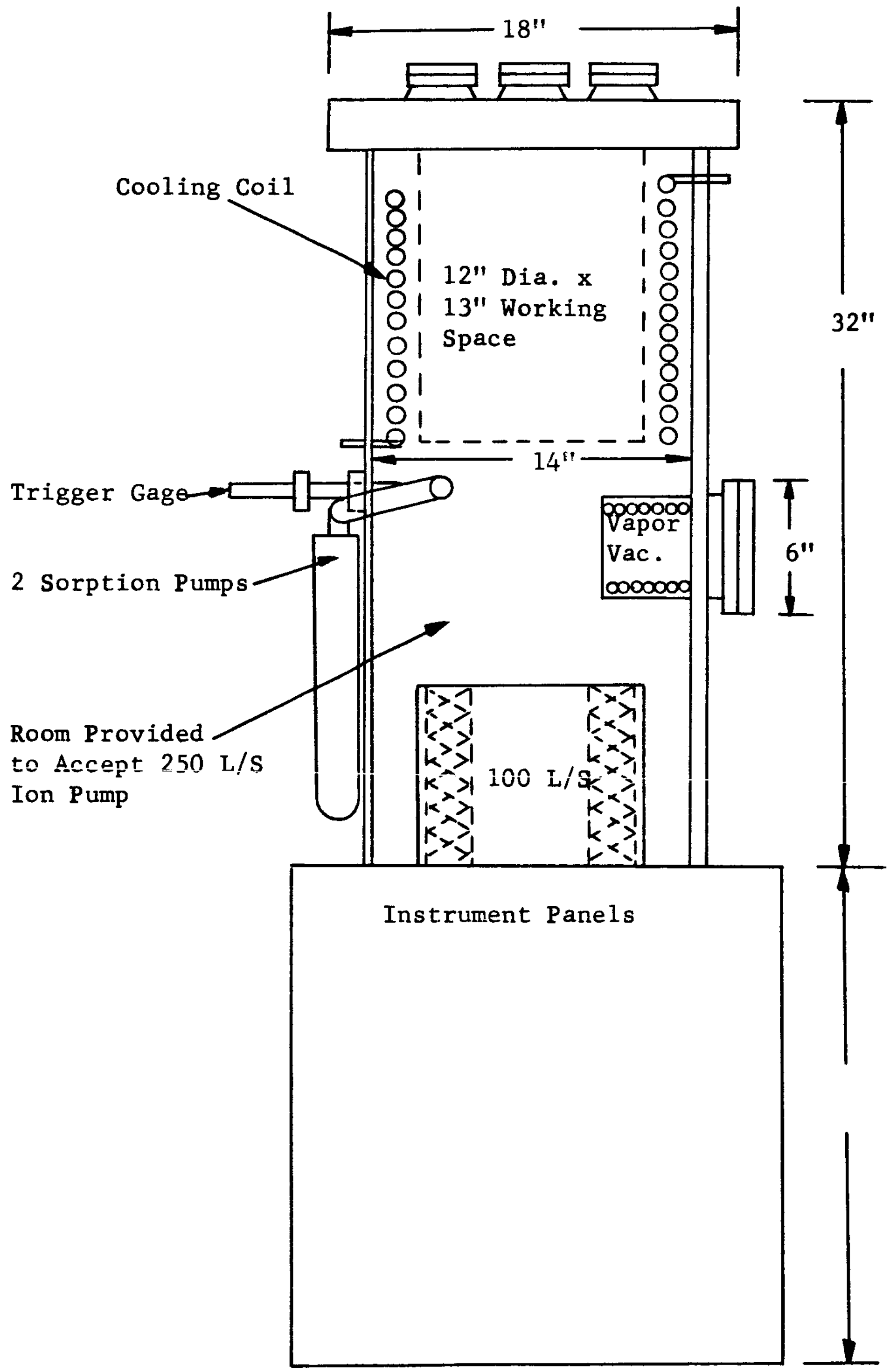


Fig. 4 Schematic of Vacuum System Layout

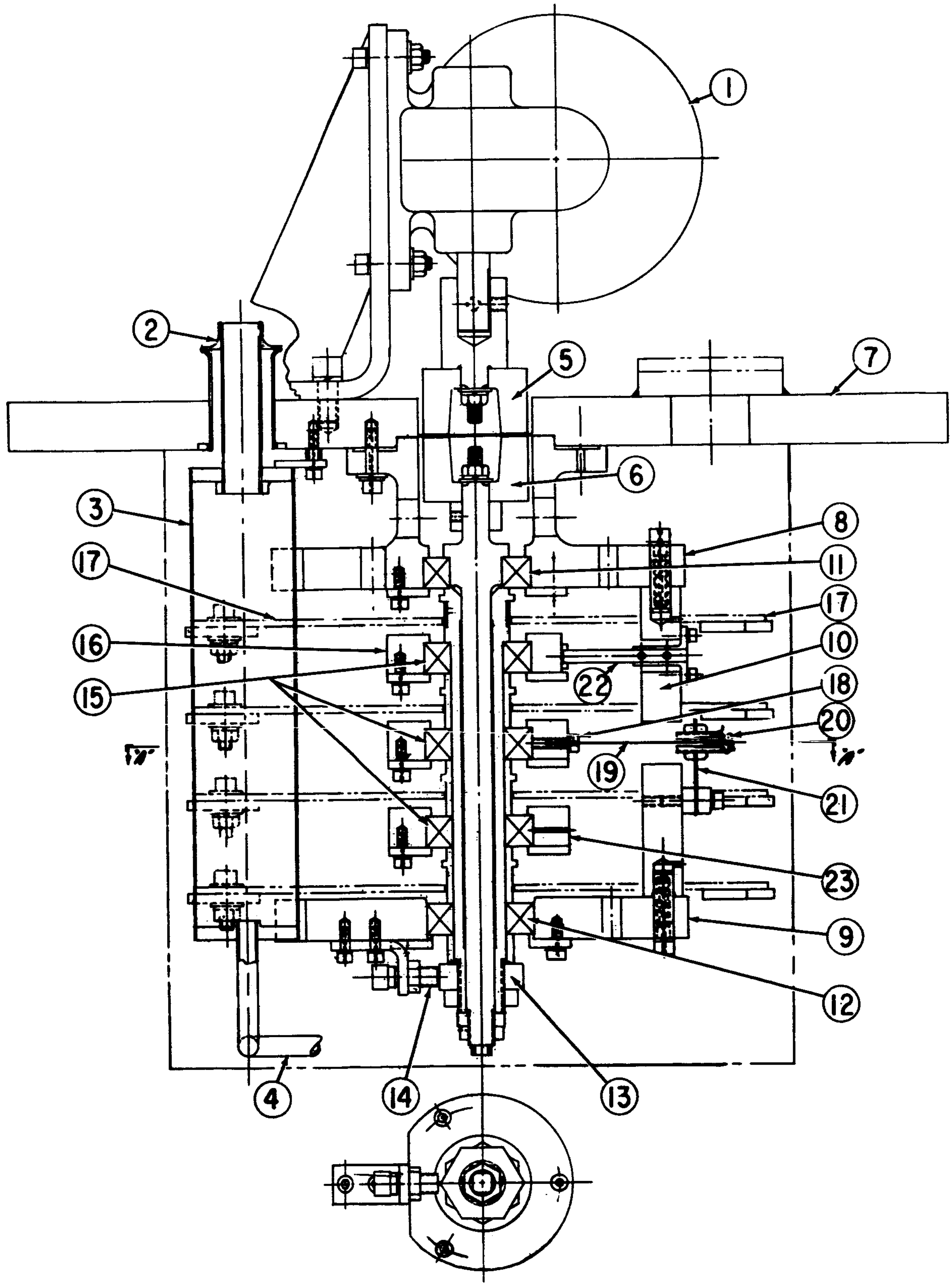


Fig. 5 Schematic of Ball-Bearing Test Rig Used for Vacuum Tests

Fig. 5a. Parts List for Ball Bearing Test Rig

1. Drive Motor
2. Liquid Nitrogen Filling Tubes (Three Spaced 120° apart)
3. Liquid Nitrogen Reservoirs (Three Spaced 120° apart)
4. Tubing Interconnecting Liquid Nitrogen Reservoirs
5. Driving Magnet
6. Driven Magnet
7. Top Cover of Vacuum Chamber
8. Top Plate of Test Rig
9. Bottom Plate of Test Rig
10. Connecting Posts (Three Spaced 120° apart)
11. Upper Support Bearing
12. Lower Support Bearing
13. Gear Used for Speed Signal
14. Magnetic Pickup for Speed Measurement
15. Test Bearing
16. Test Bearing Housing
17. Copper Separator Plates for Condensing Vapors (Used Above and Below Each Bearing)
18. Retaining Bolt for Radial Load Wire (One for Each Bearing)
19. Radial Load Wire (One for Each Bearing)
20. Adjustments for Radial Load (One for Each Bearing)
21. Flexure with Strain Gages for Measuring Load (One for Each Bearing)
22. Torque Arm Flexure with Strain Gages (One for Each Bearing)
23. Thermocouple Well (One in Each Test Bearing Housing)

2

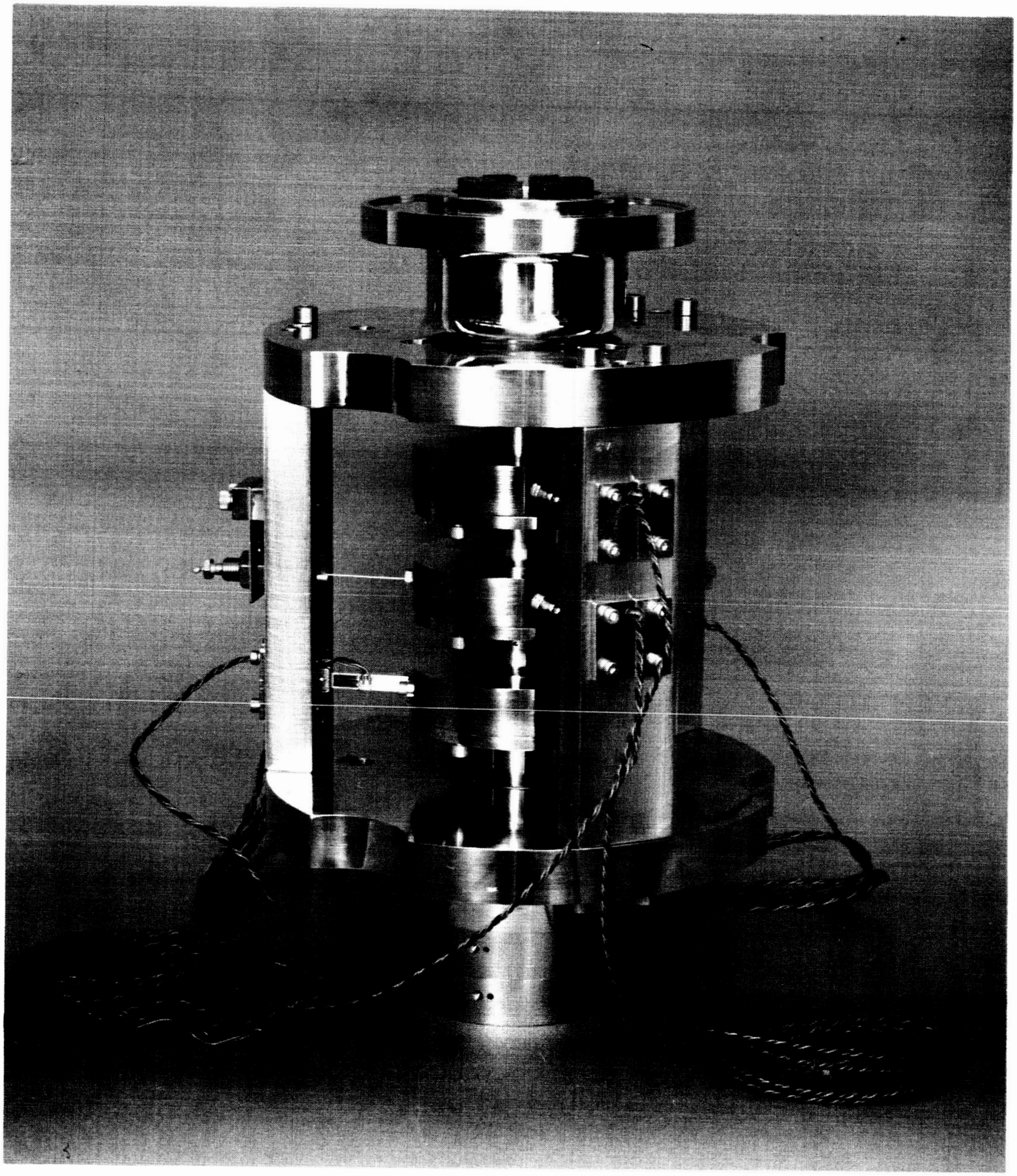


Fig. 6 Partially Assembled Bearing Test Rig

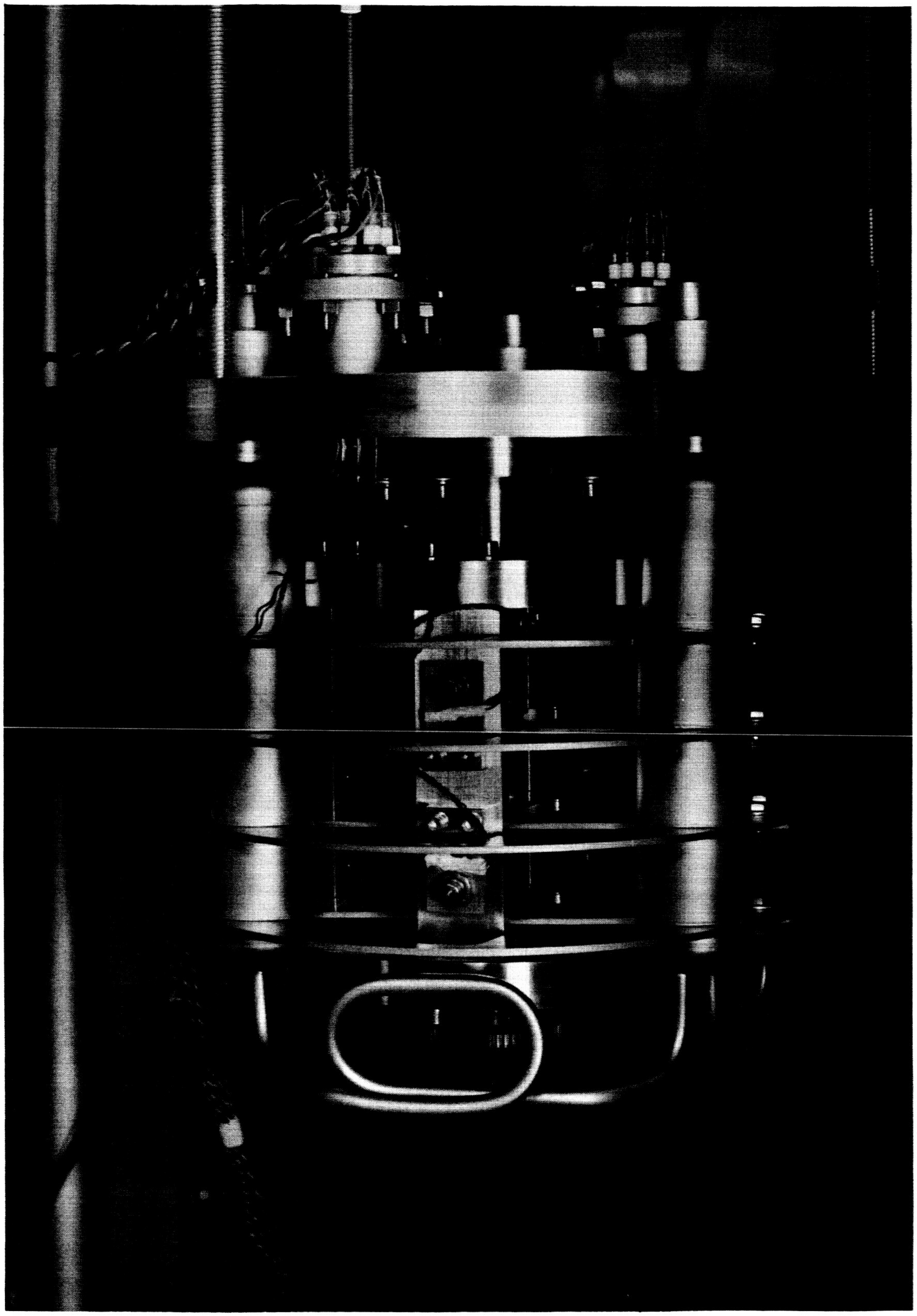


Fig. 7 Close-Up of Bearing Test Rig Mounted to Top Cover of Vacuum Chamber

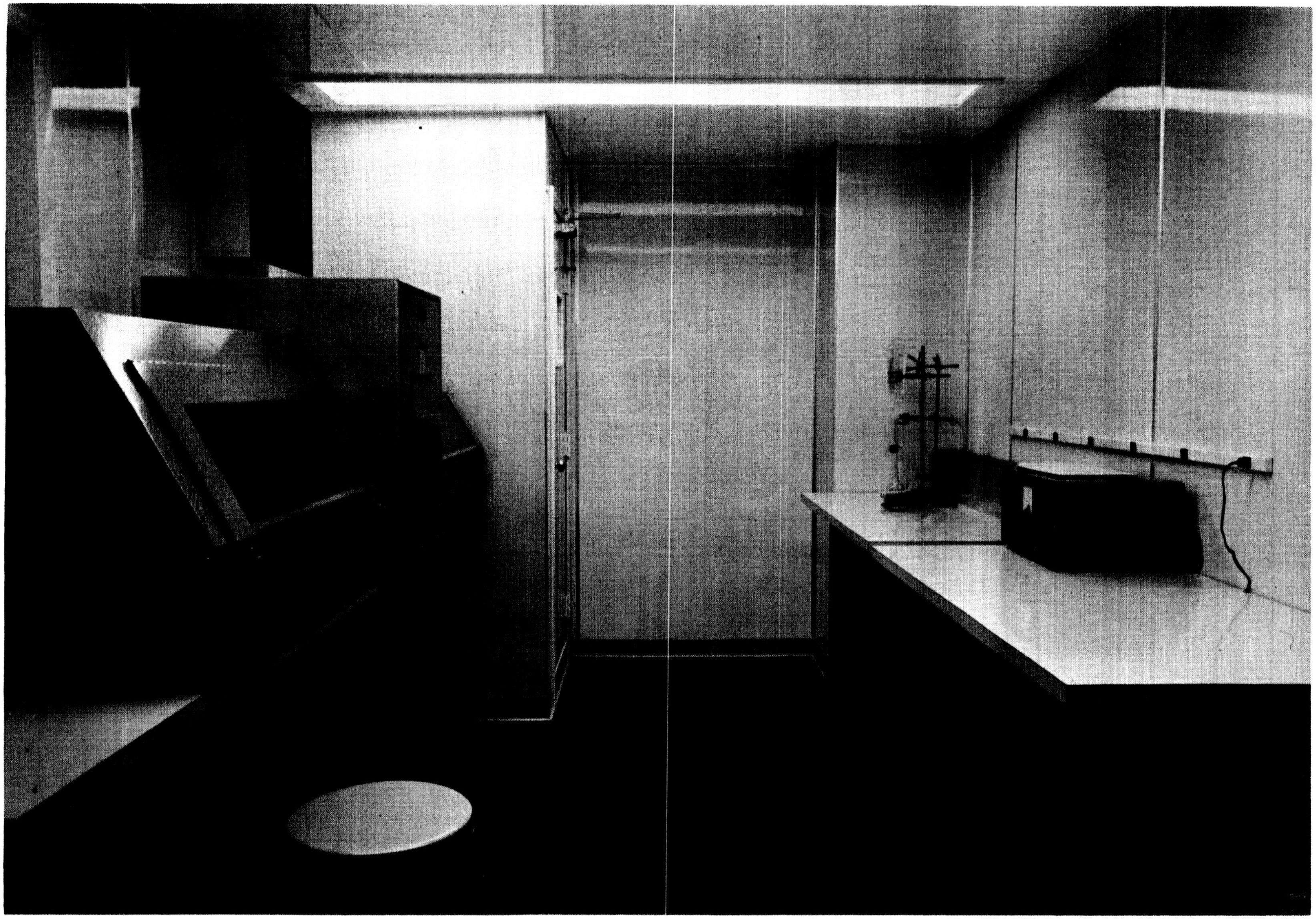


Fig. 8 Clean Room Assembly Area

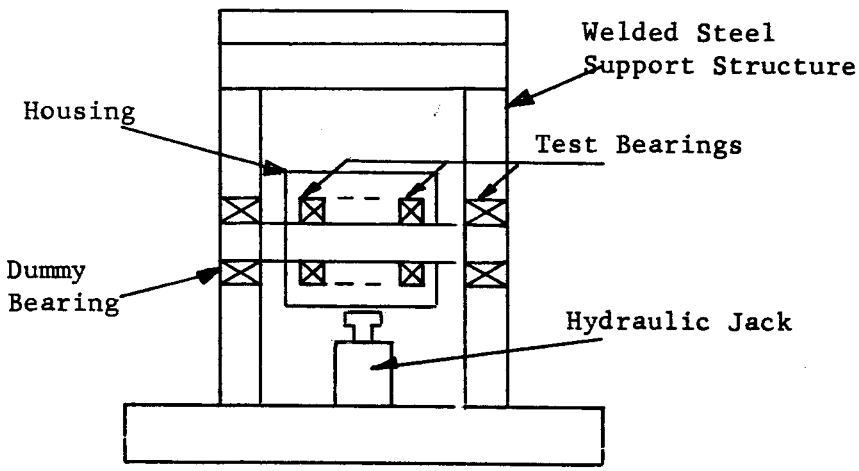


Fig. 9 Schematic of Static Load Test Rig

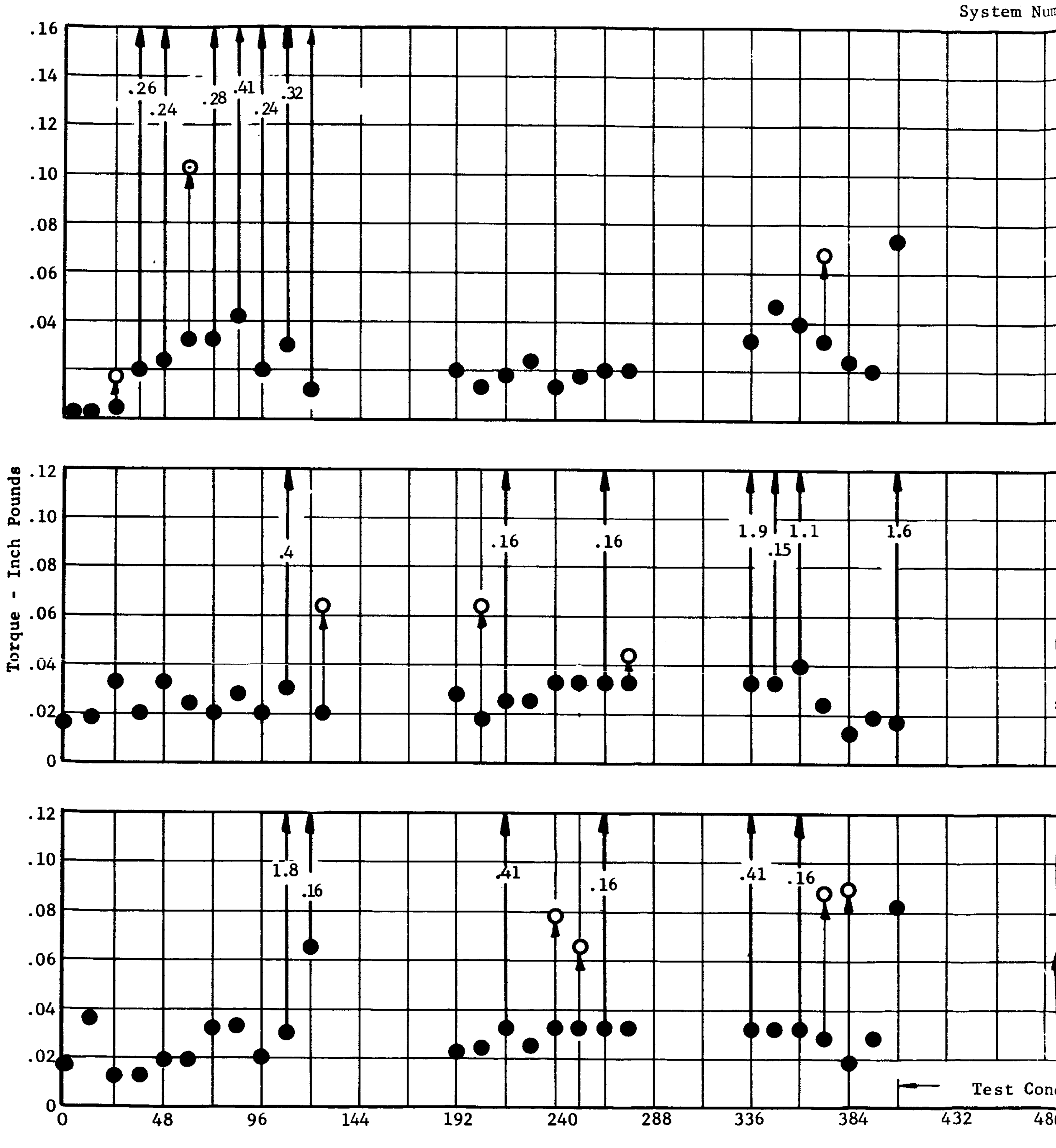
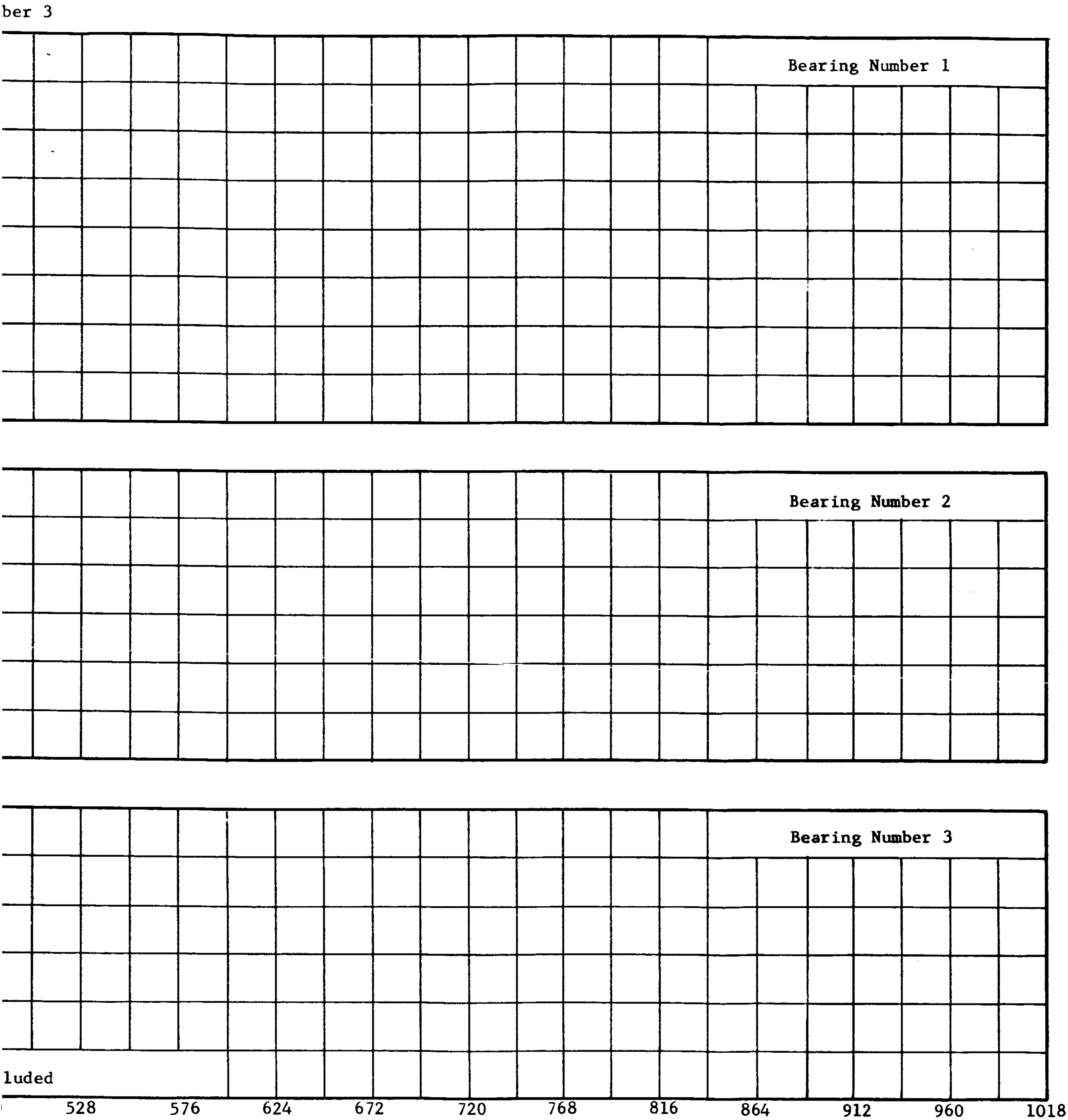
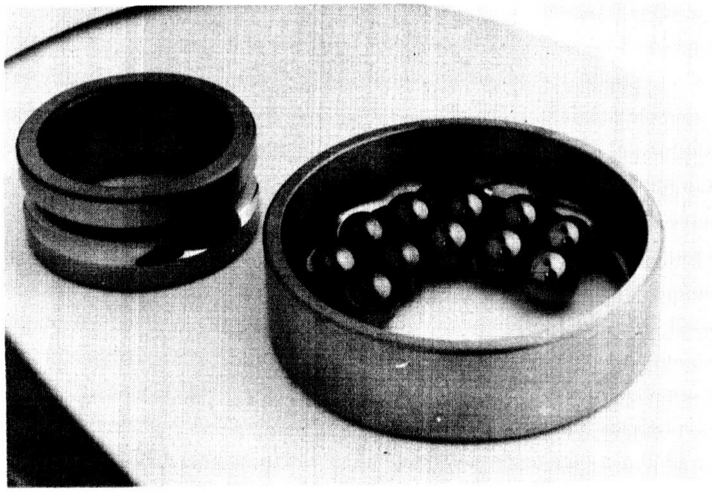


Fig. 10 Average and Peak Torque Va
at 30 RPM. Bearings 52100
Lubricant - None. Load -

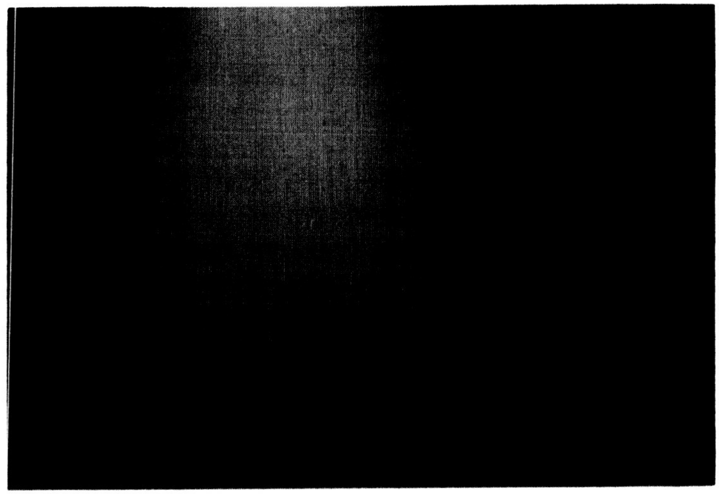


.ues Measured on Three Test Bearings
Steel, Size 205. Phenolic Retainers.
.0 pounds Radial.

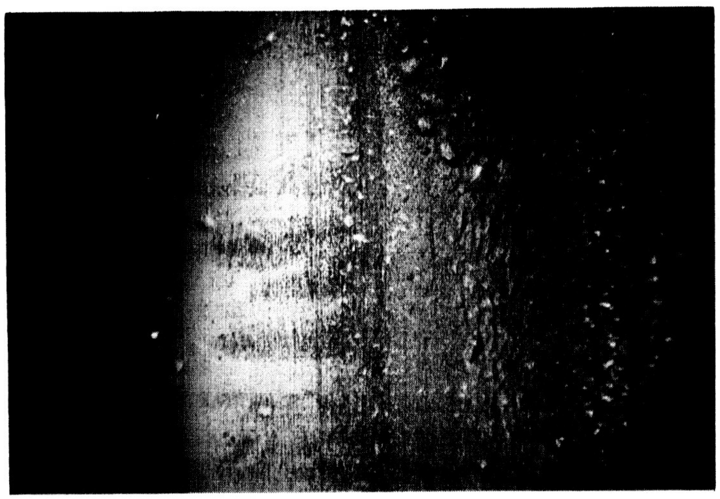
2

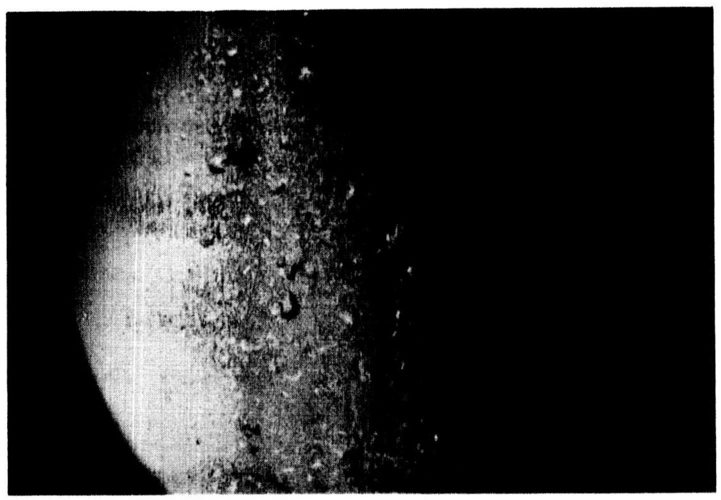


Overall Appearance of Bearing — After Test



Typical Surface Finish of Race — Before Test — X100





Surface Damage on Race After Test — X100

Fig. 11 Typical Contact Areas on Dry, Clean Bearings

Figure 12

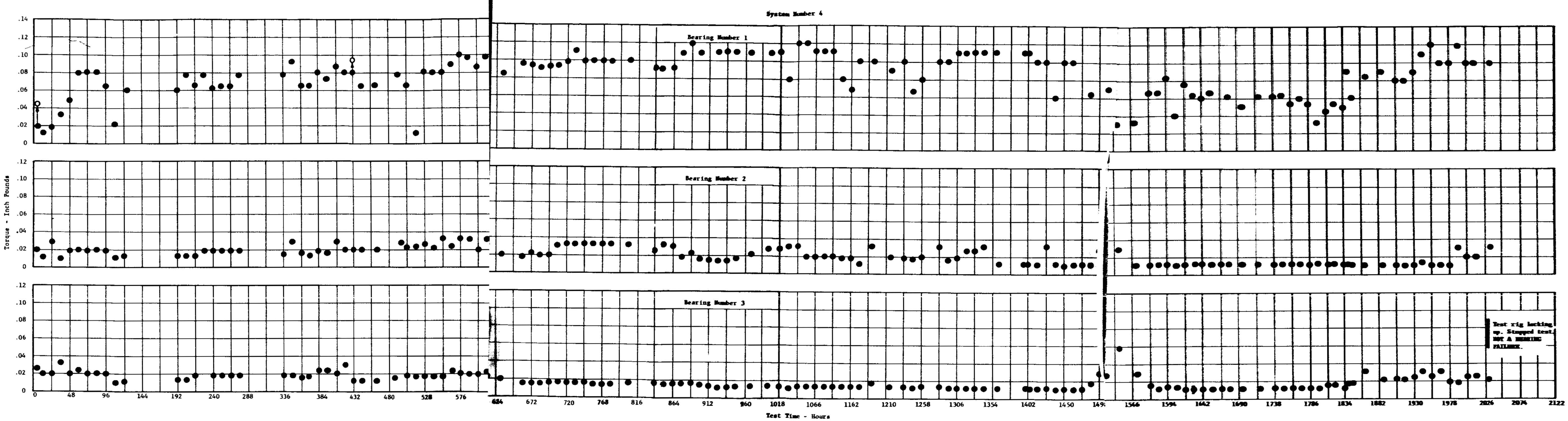
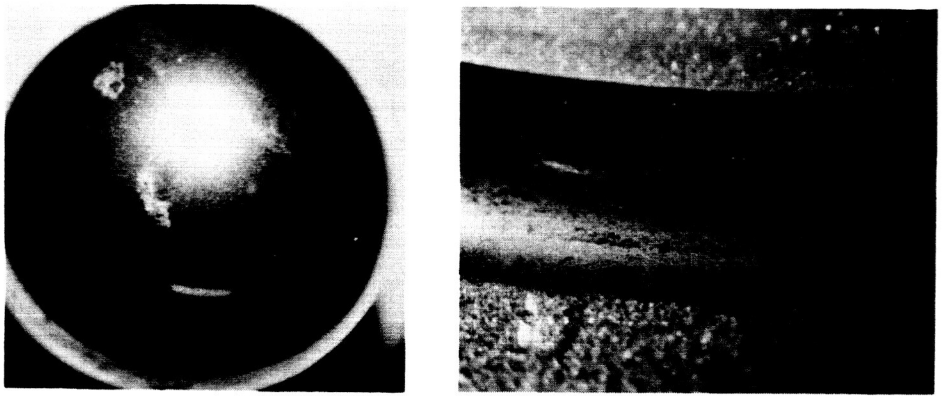


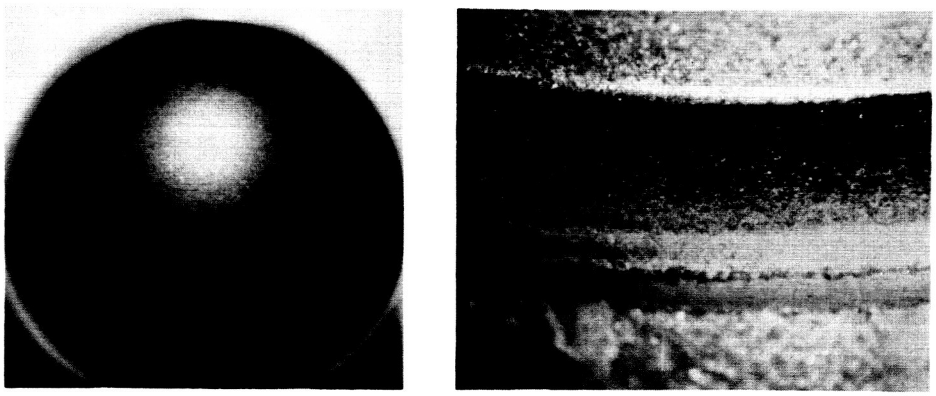
Fig. 12 Average and Peak Torque Values Measured on Three Test Bearings at 30 RPM. Bearings 52100 Steel, Size 205. Phenolic Retainers. Lubricant - Silicate Bonded MoS_2-Graphite Film. Load-10 Pounds Radial

1 2 3

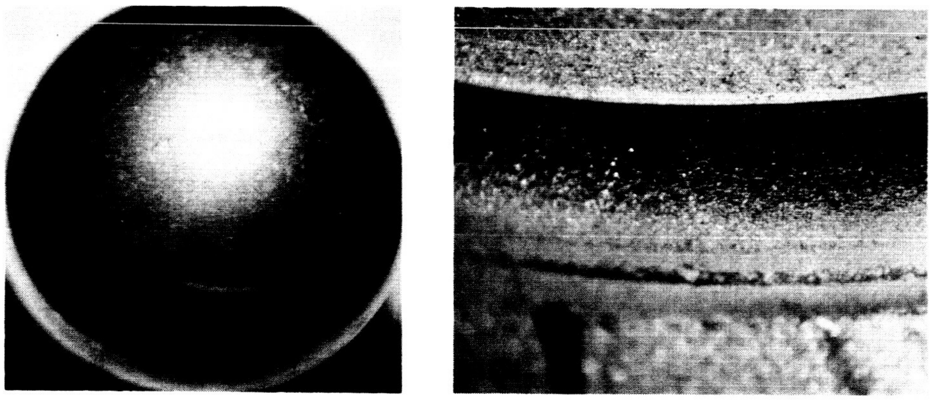
Top Test Bearing



Center Test Bearing



Lower Test Bearing



Support Bearing

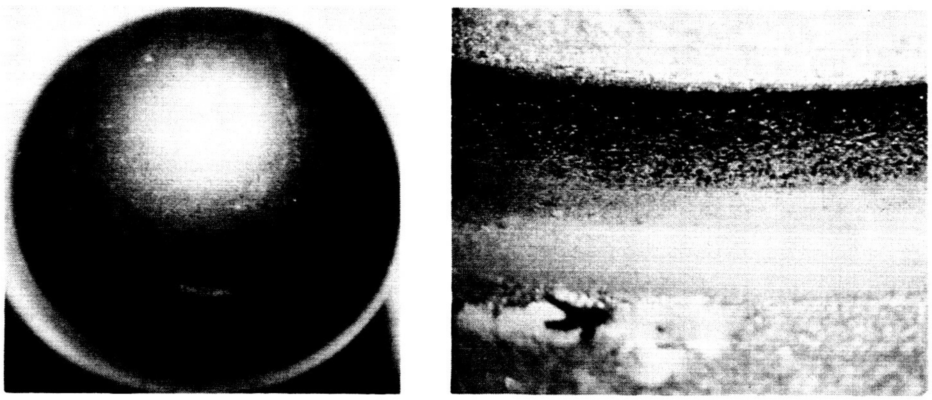


Fig. 13 Typical Contact Areas on Bearings Lubricated with Silicate Bonded MoS_2 - Graphite Film.

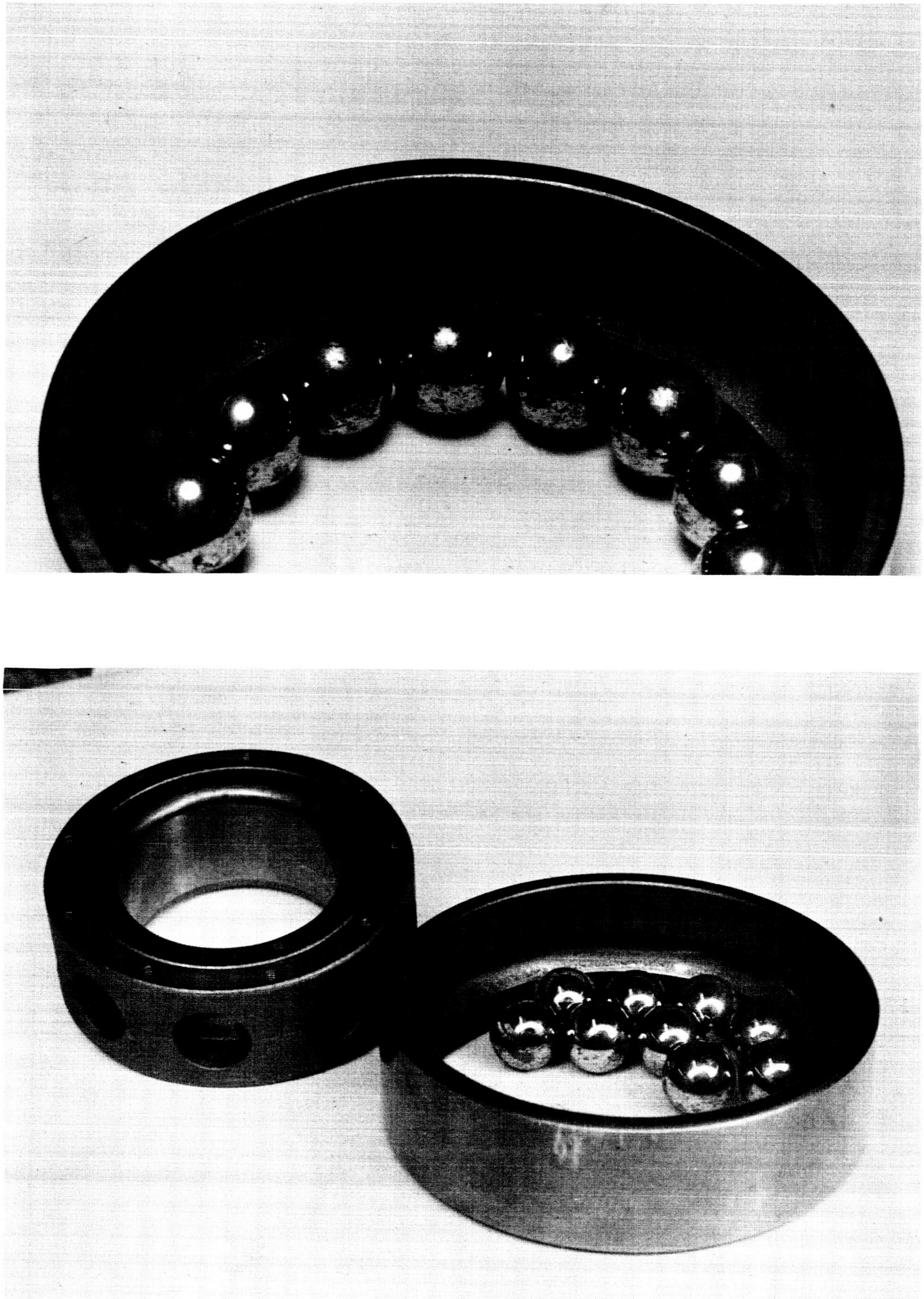


Fig. 14 Disassembled Bearing with Tool Steel Retainer.
Lubricant - Sodium Silicate-Bonded MoS_2-Graphite Film

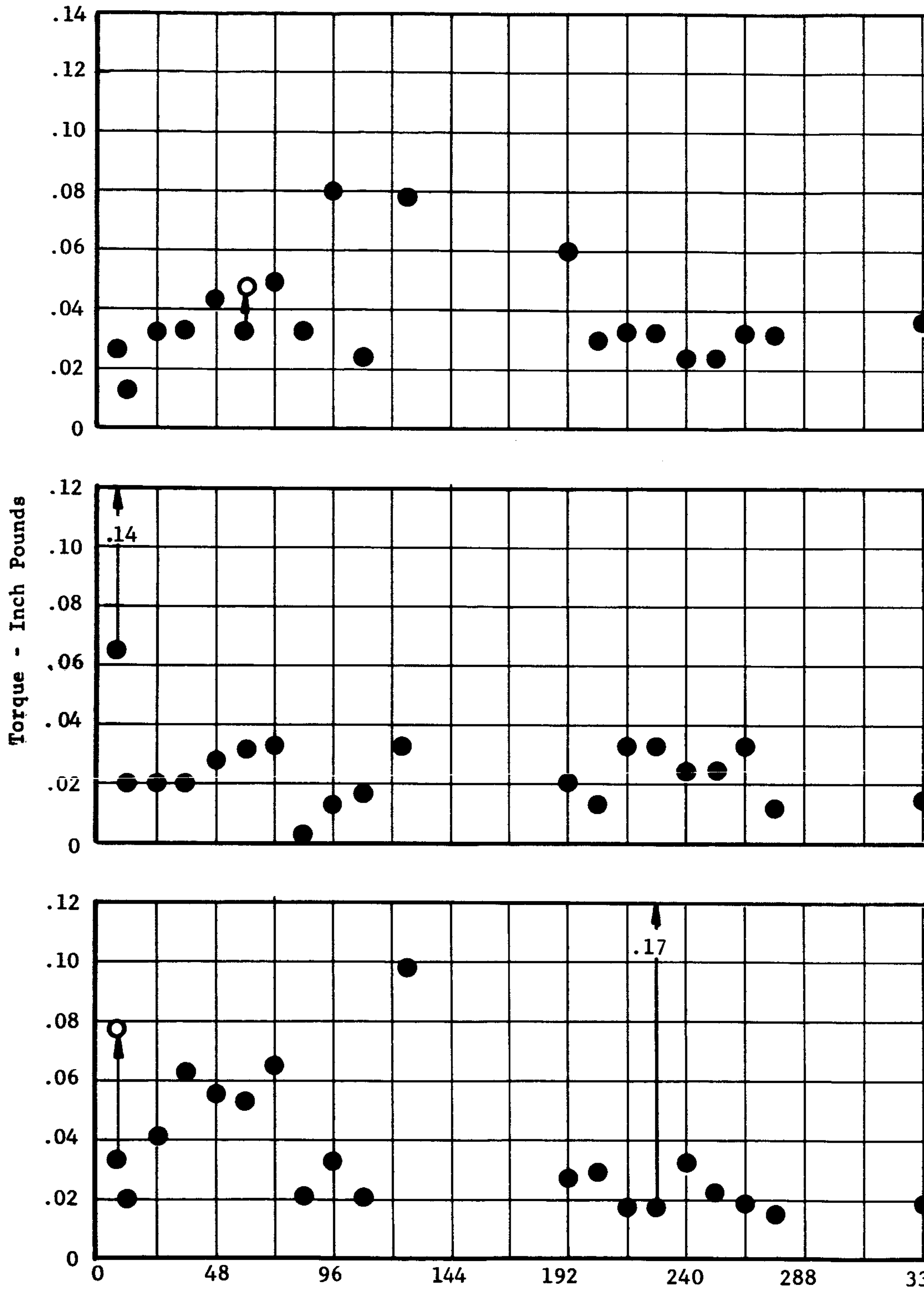
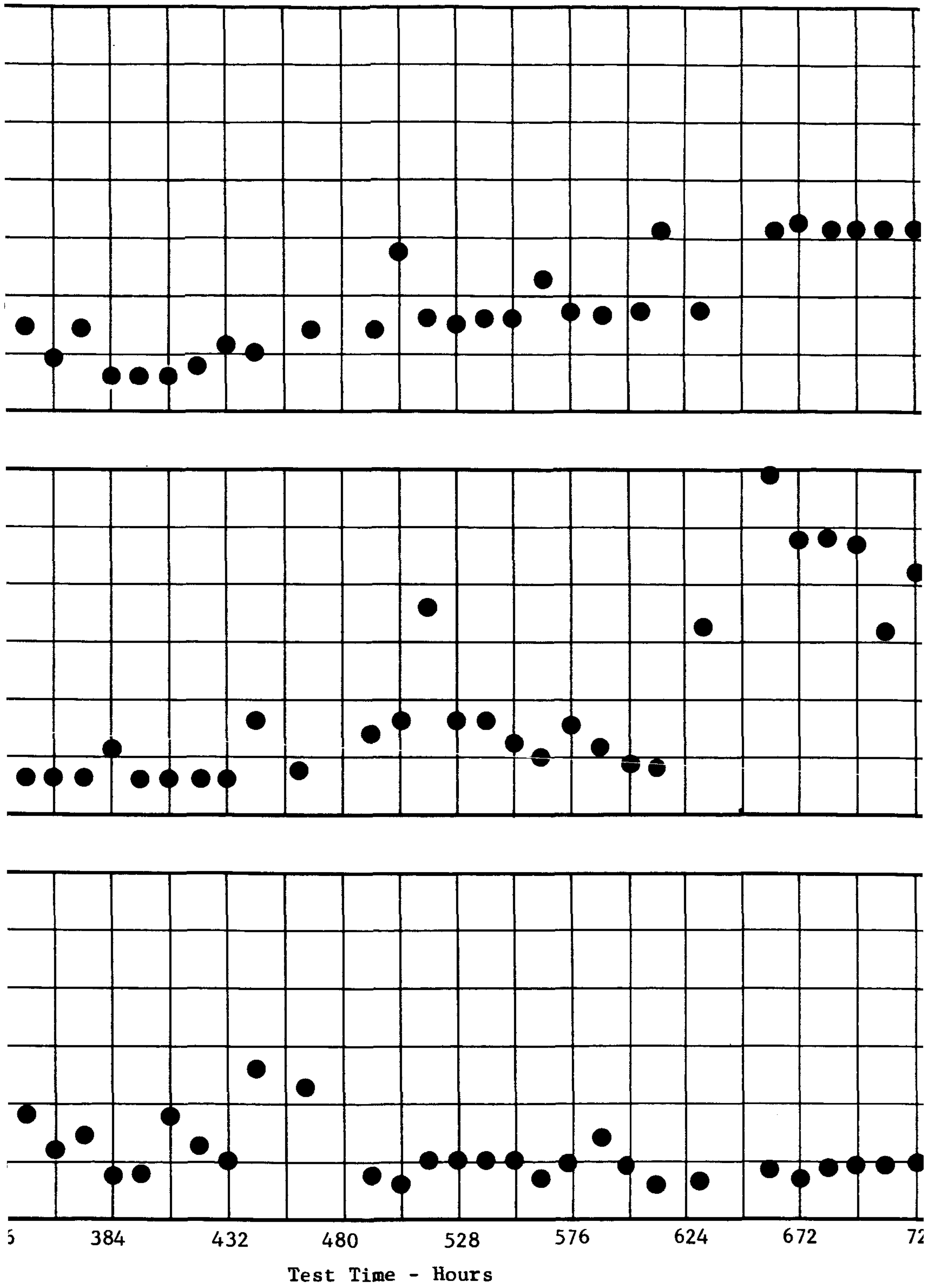

Torque - Inch Pounds
.14
.12
.10
.08
.06
.04
.02
0
.14
.17
48
96
144
192
240
288

Fig. 15

1

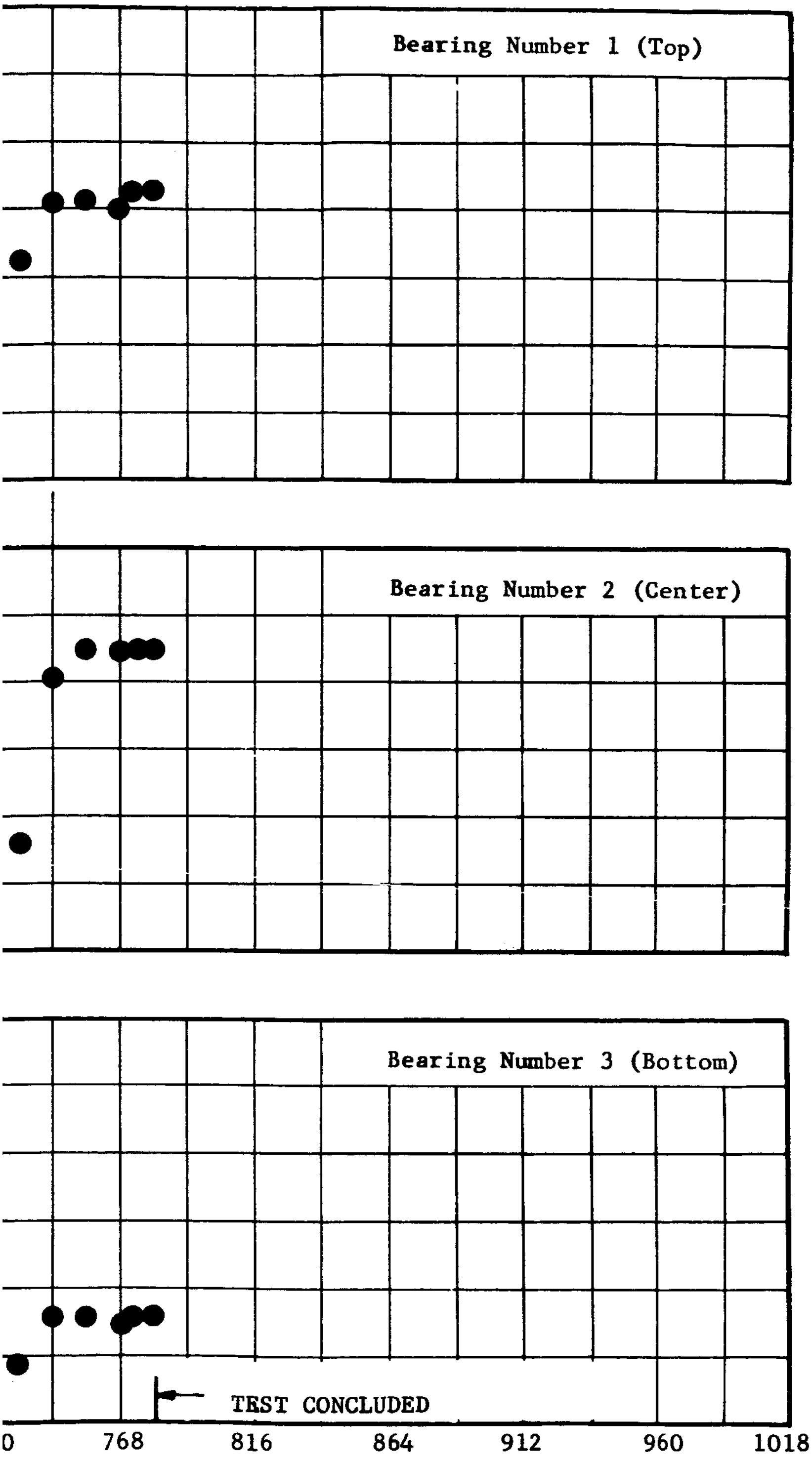
System Number 1



Average and Peak Torque Values Measured on Three Test Bearings at 30 RPM. Bearings 52100 Steel, Size 205, Phenolic Retainers. Lubricant - Burnished MoS_2 Film. Load - 10 pounds Radial

2

Figure 15

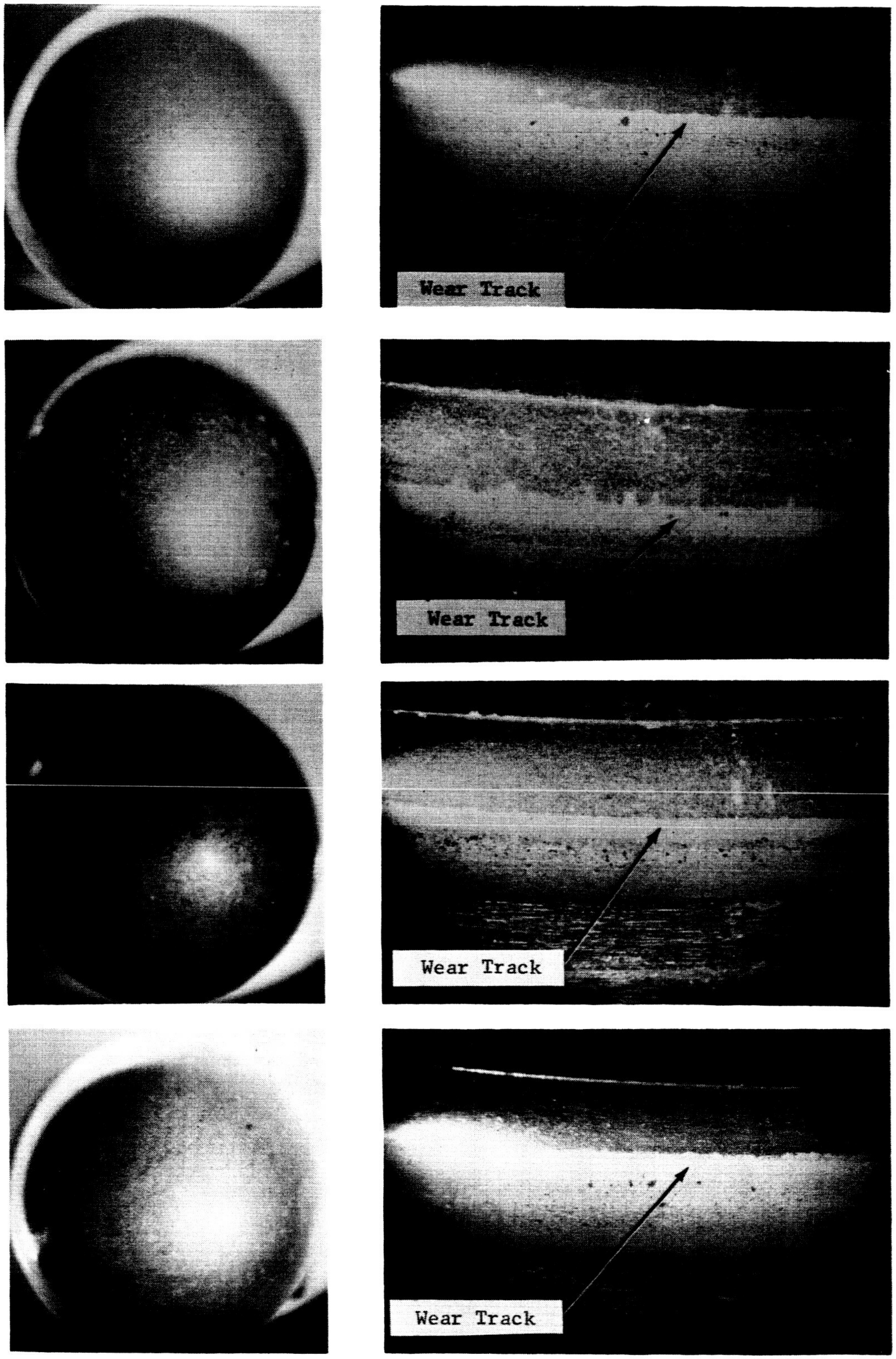

Bearing Number 1 (Top)
Bearing Number 2 (Center)
Bearing Number 3 (Bottom)
TEST CONCLUDED
0
768
816
864
912
960
1018

Support Bearing

Top Bearing

Center Bearing

Lower Bearing



Typical Ball

Portion of Wear Track on Outer Race

Fig. 17 Typical Contact Areas on Bearings Lubricated with Burnished MoS_2 Film

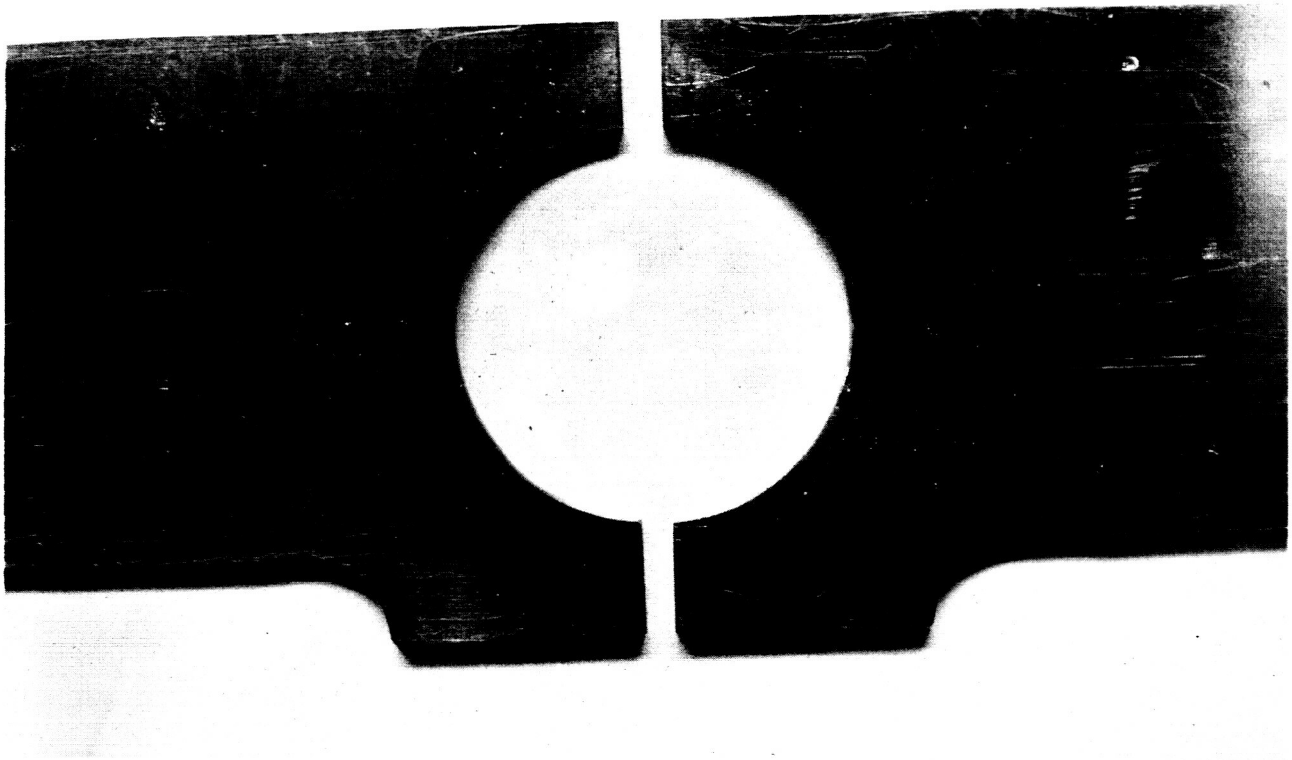


Fig. 16 Debris on Separator Plates of Bearing Lubricated with Burnished MoS_2 Film

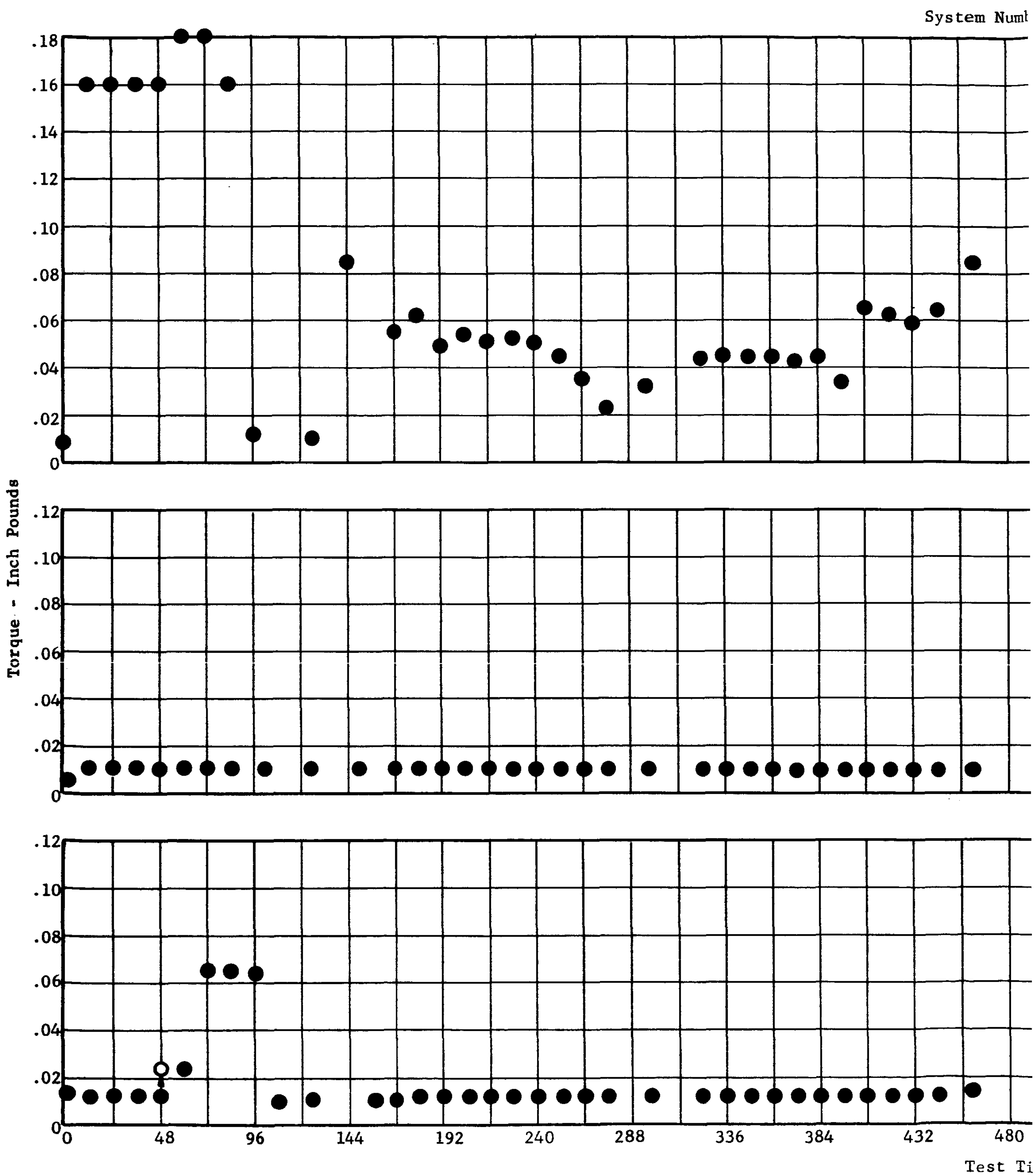
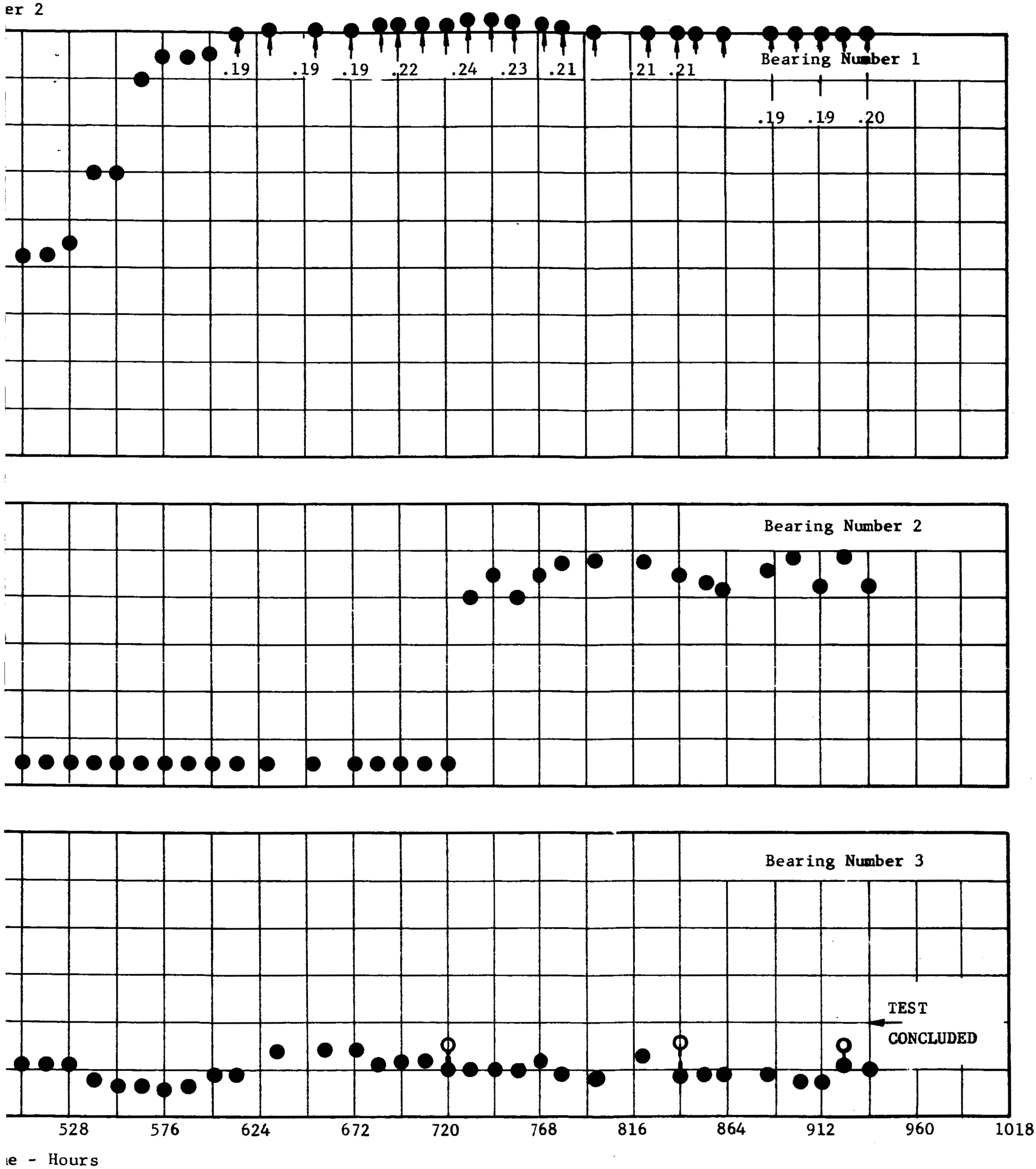


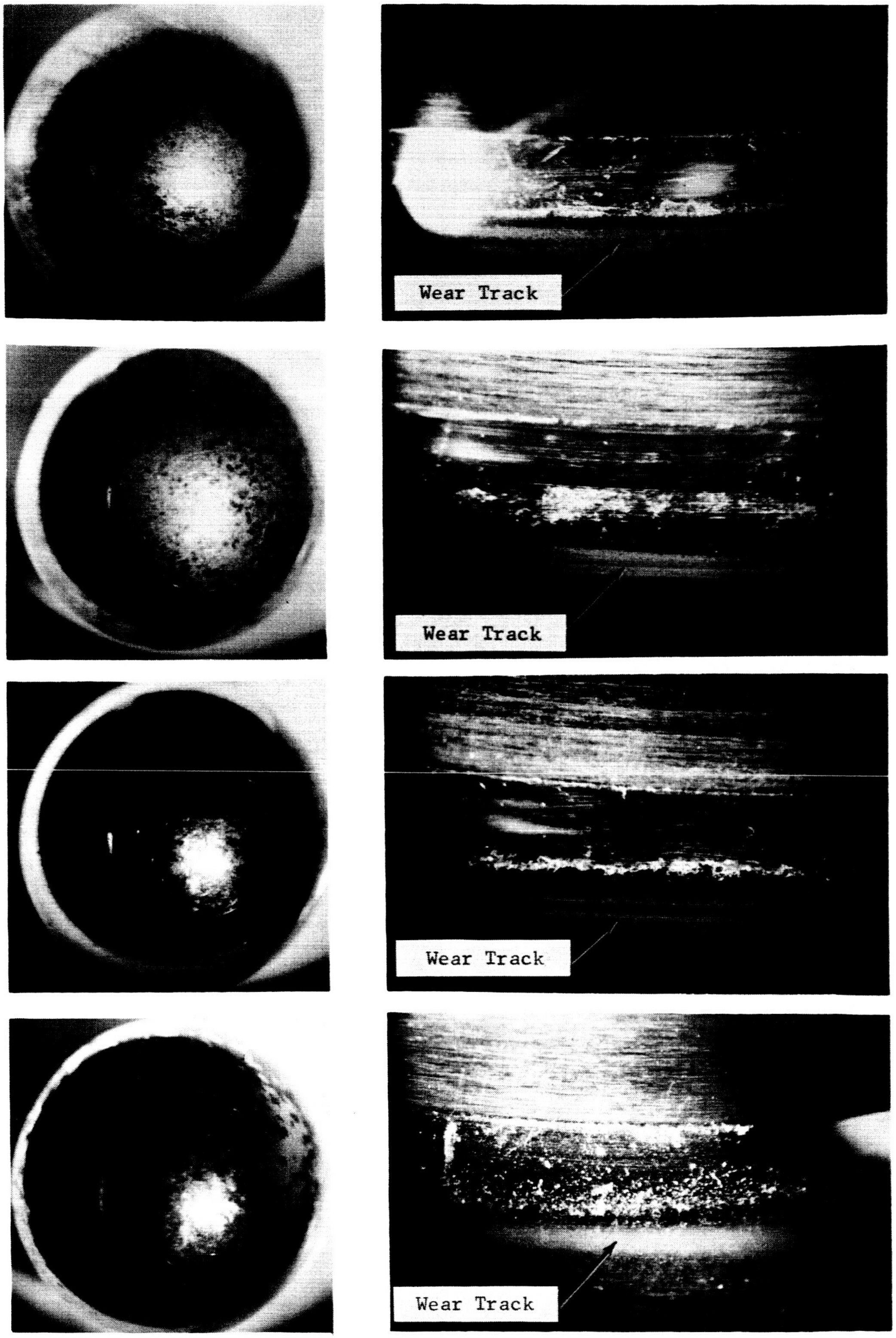
Fig. 18 Average and Peak Torque V
at 30 RPM. Bearings, 521
Lubricant - Evaporated Me
Load - 10 pounds Radial.

Figure 18



lues Measured on Three Test Bearings
0 Steel, Size 205, Phenolic Retainers.
al Films with Outer Layer of MoS_2.

2



Support Bearing

Top Bearing

Center Bearing

Lower Bearing

Typical Ball

Portion of Wear Track on Outer Race

Fig. 19 Typical Contact Areas on Bearings Lubricated with Evaporated Metal Films and Outer Layer of MoS_2

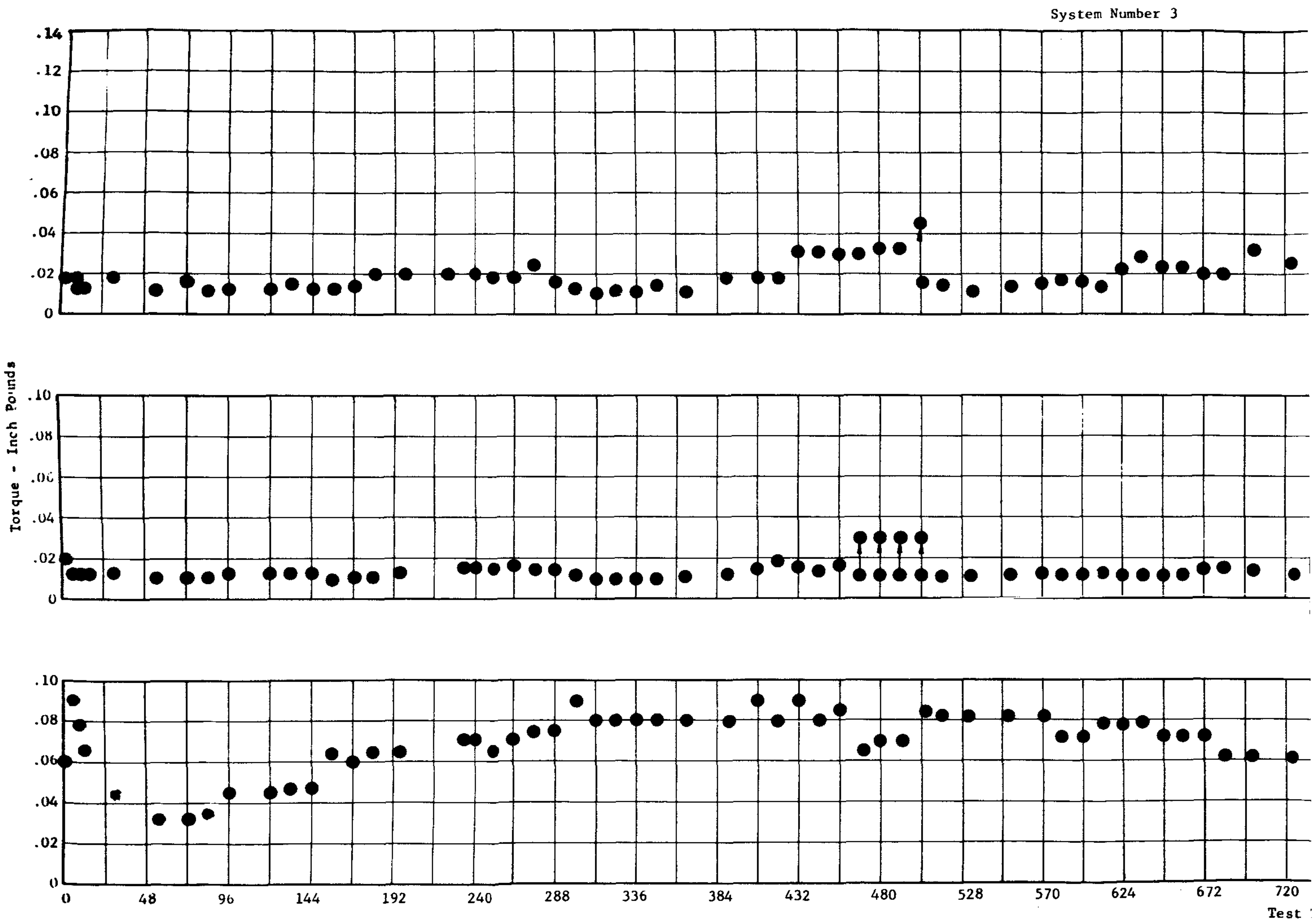
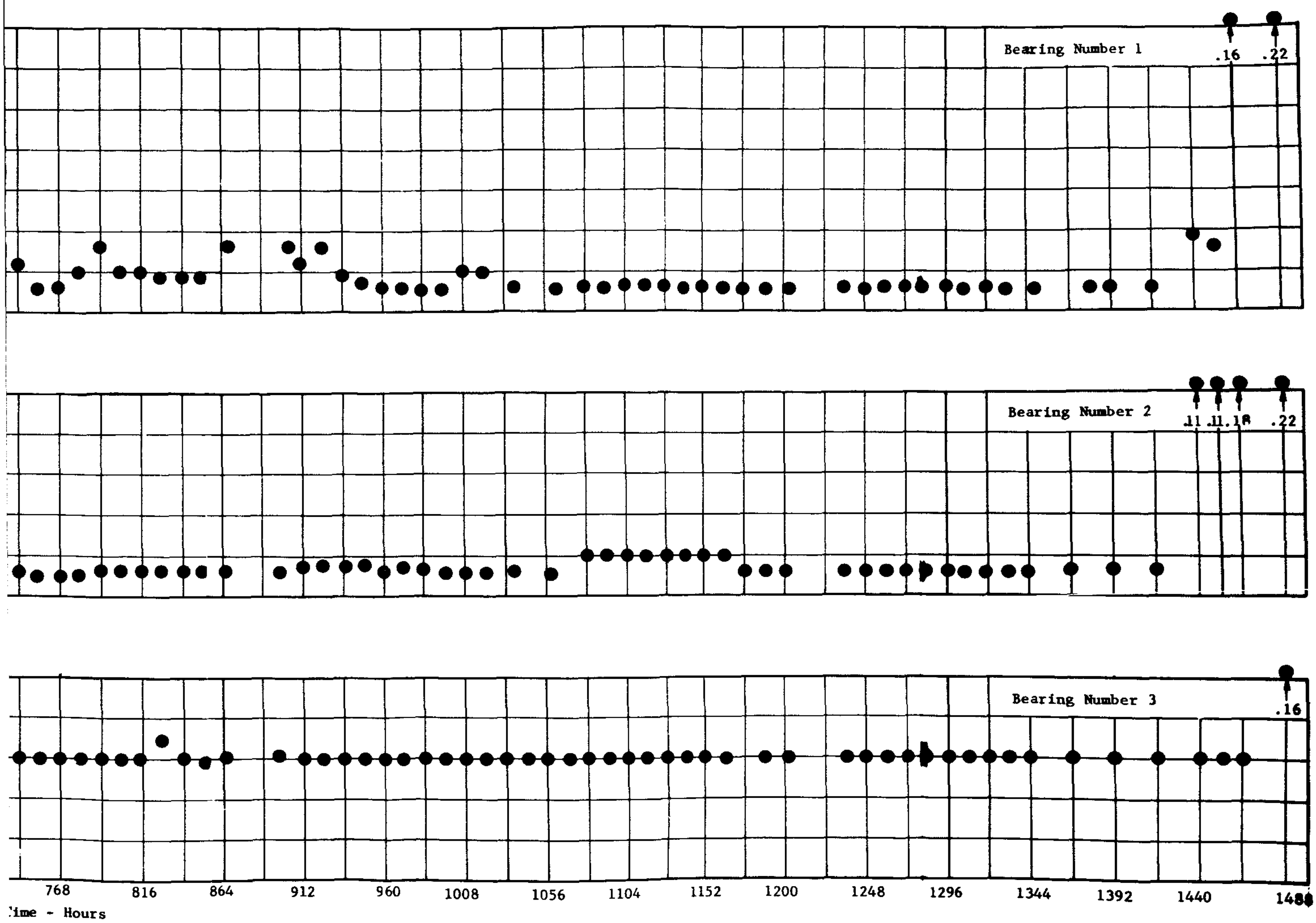


Fig. 20 Averagd and Peak Torque Meas
30 RPM. Bearings, 52100 Ste
Lubricant - High Vacuum Oil.



urements on Three Test Bearings at
el, Size 205, Phenolic Retainers.
Load - 10 Pounds Radial

2

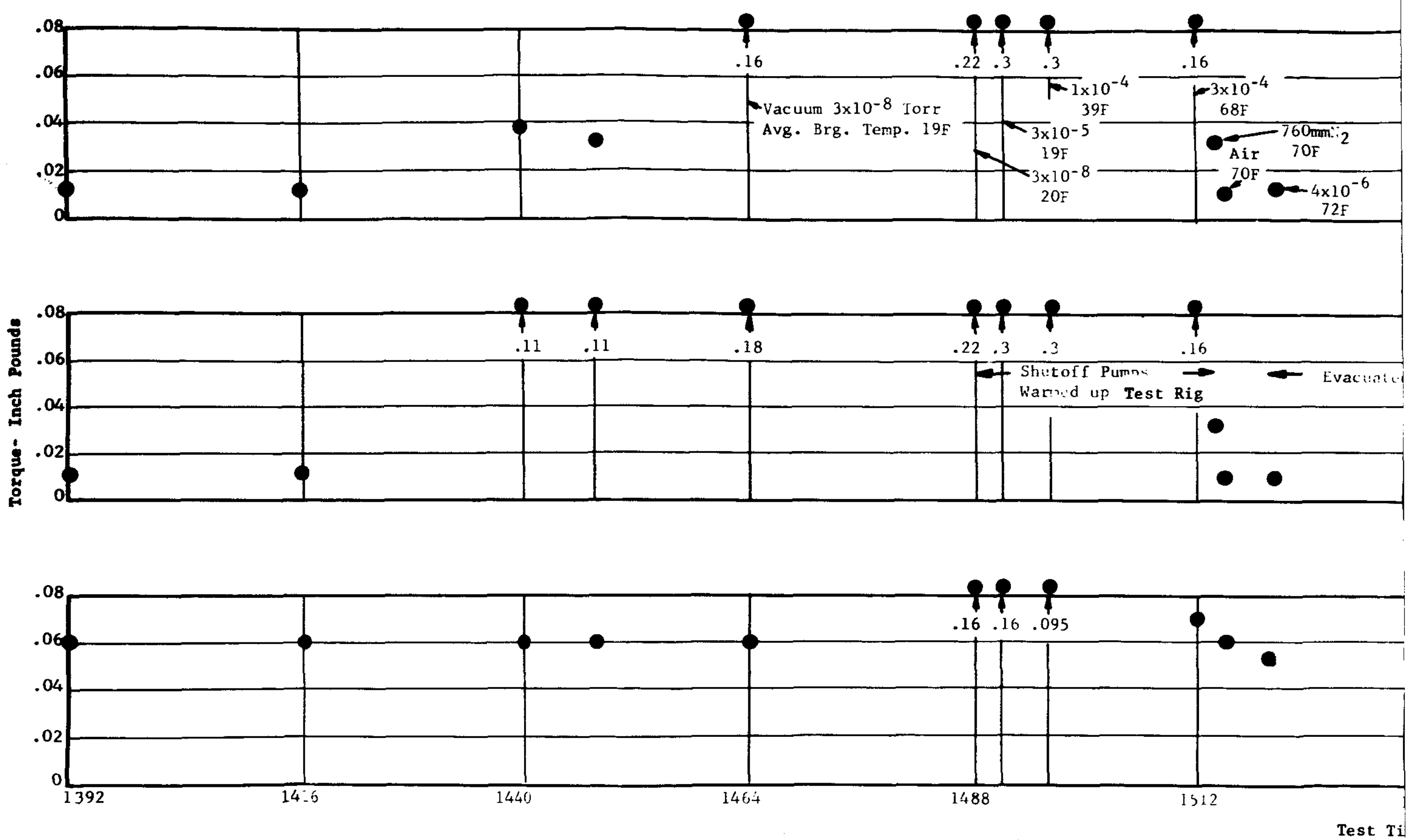
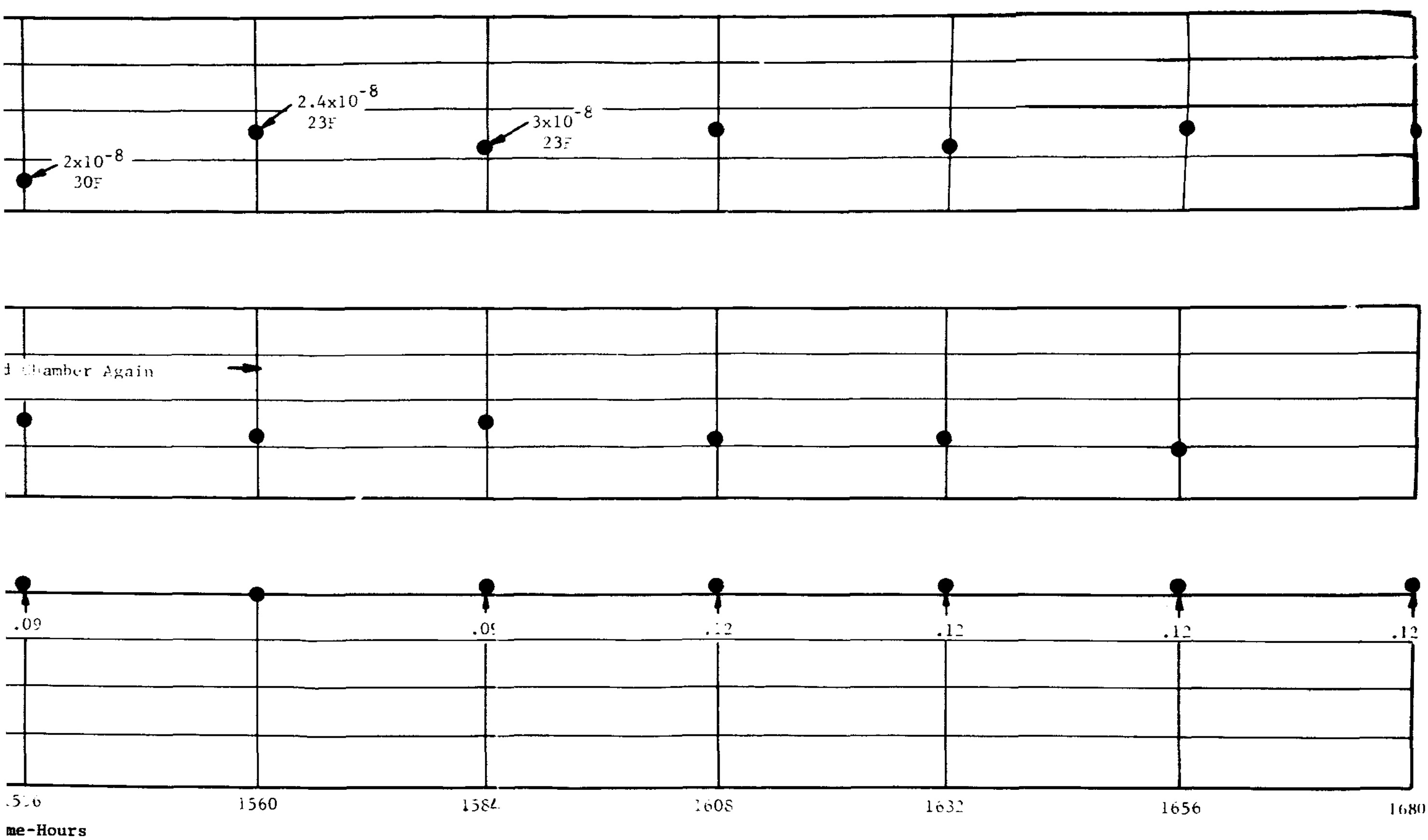


Fig. 21 Continuation of Tests on
Oil Showing Torque Failu



Bearings Lubricated with High Vacuum
res and Subsequent Behavior

2

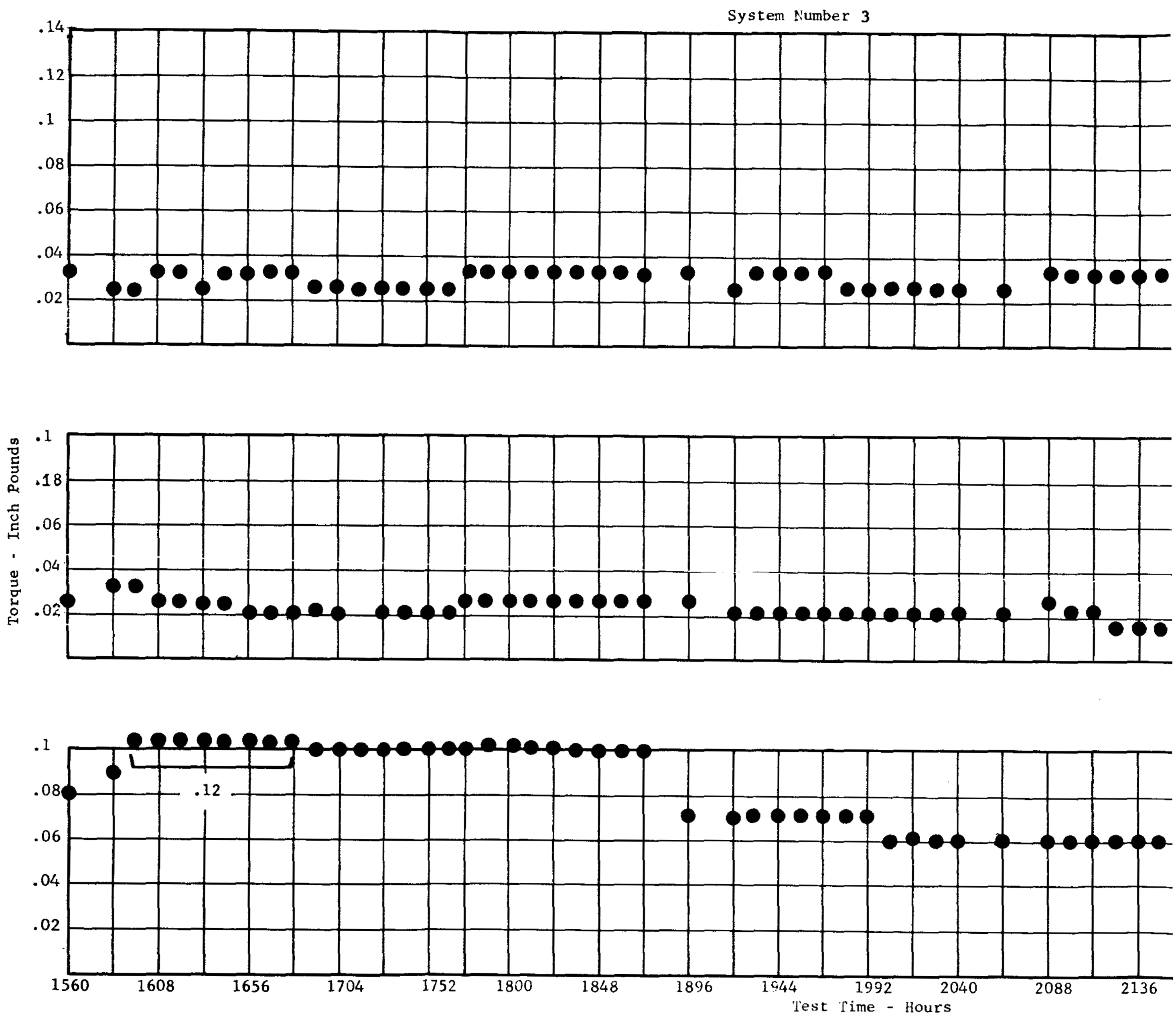
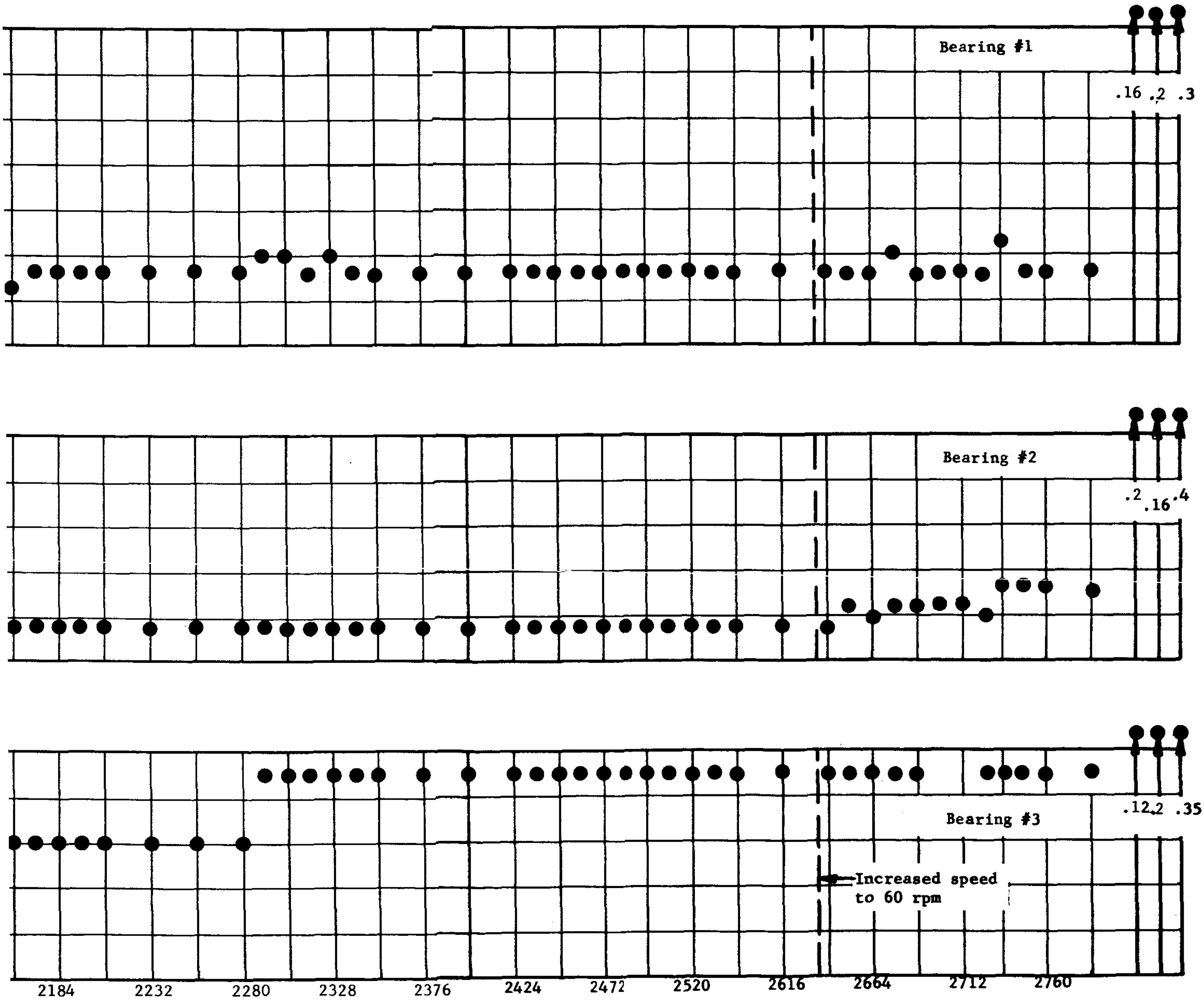


Fig. 22 Conclusion of Rotating Tests
High Vacuum Oil



at 30 RPM on Bearings Lubricated with

Top Bearing

Center Bearing

Lower Bearing

Support Bearing

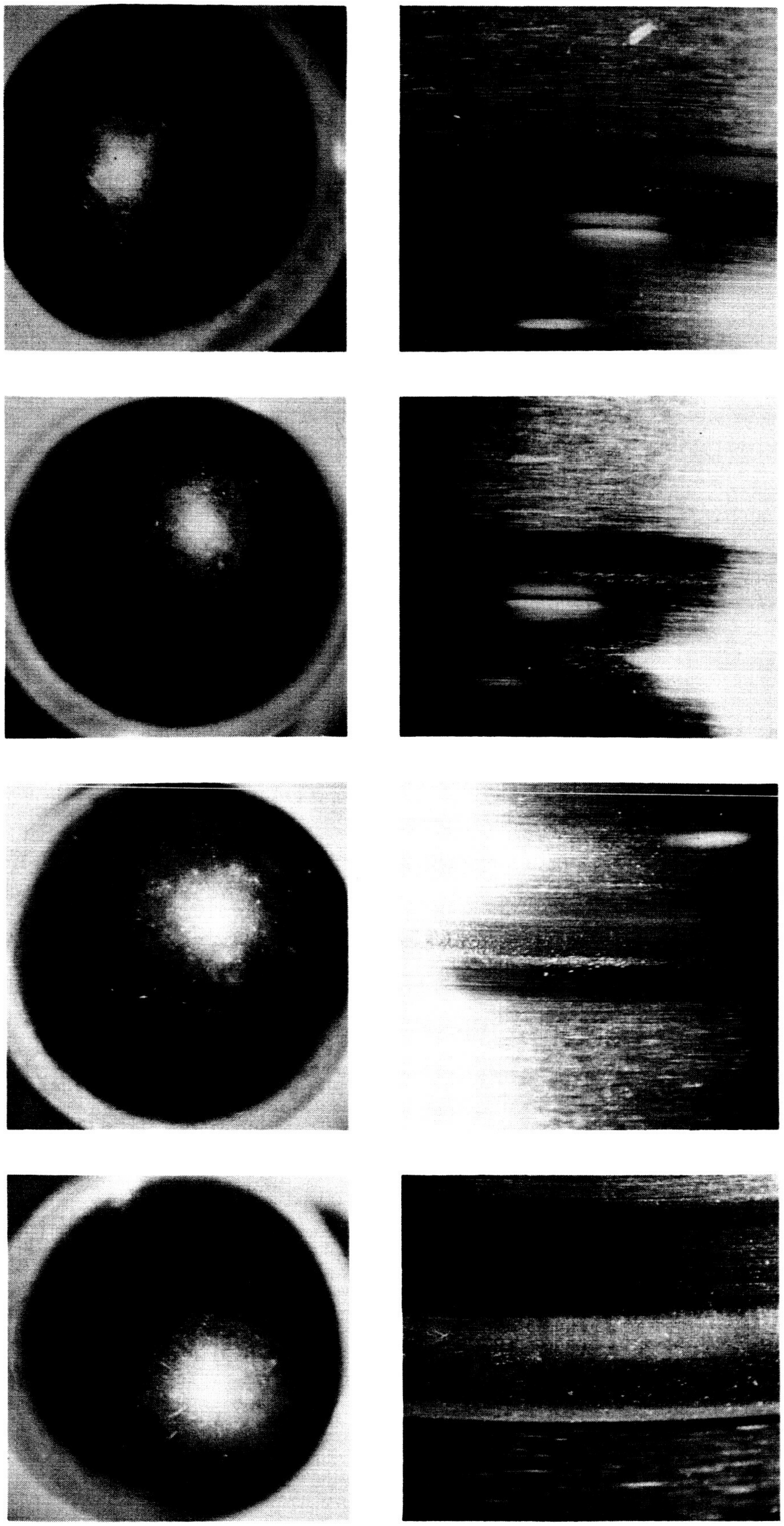
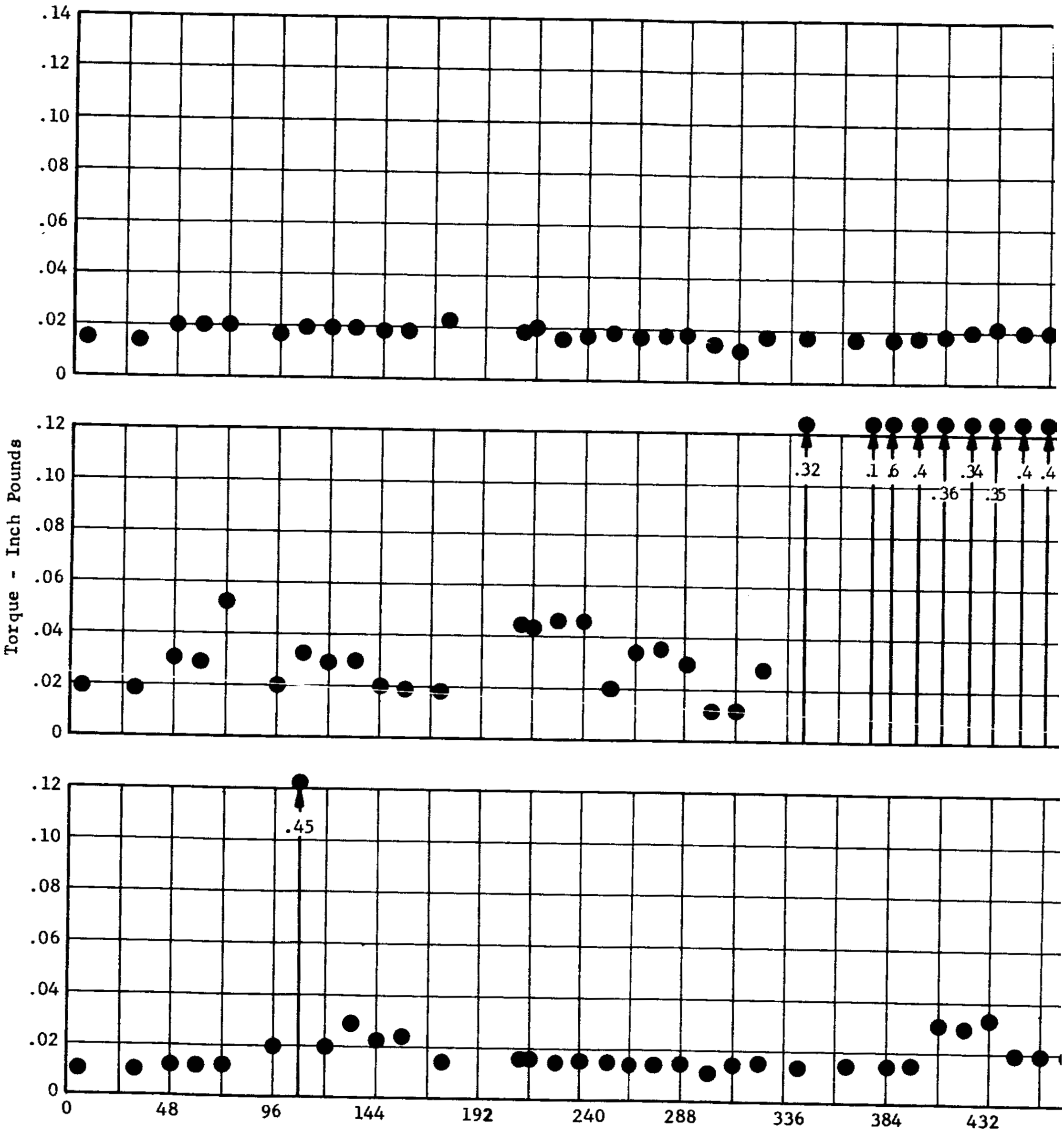
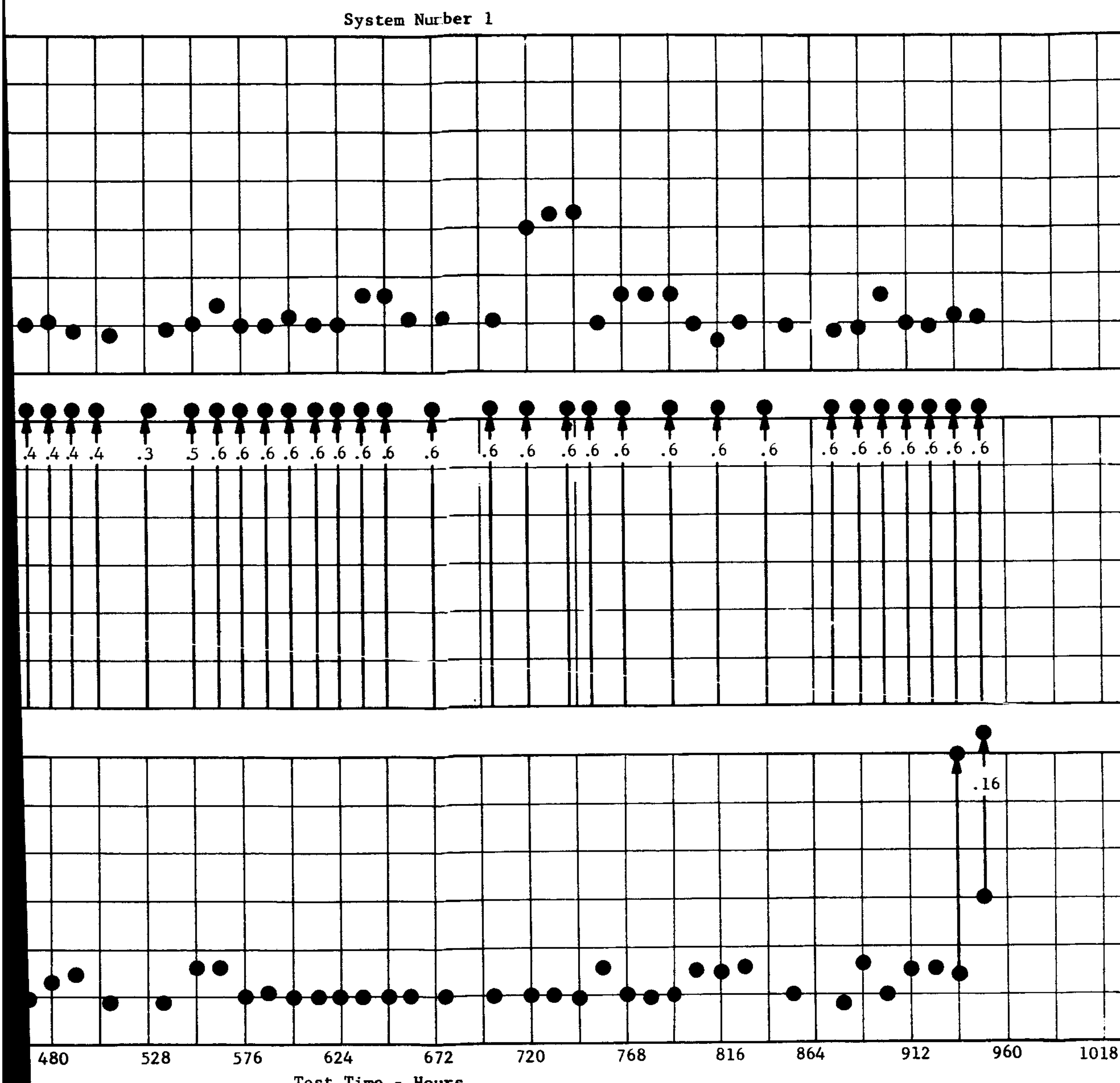


Fig. 23 Typical Contact Areas on Bearings Lubricated with High Vacuum Oil



Torque - Inch Pounds
.14
.12
.10
.08
.06
.04
.02
0
.32
.1
.6
.4
.36
.34
.35
.4
.4
.45
0
48
96
144
192
240
288
336
384
432

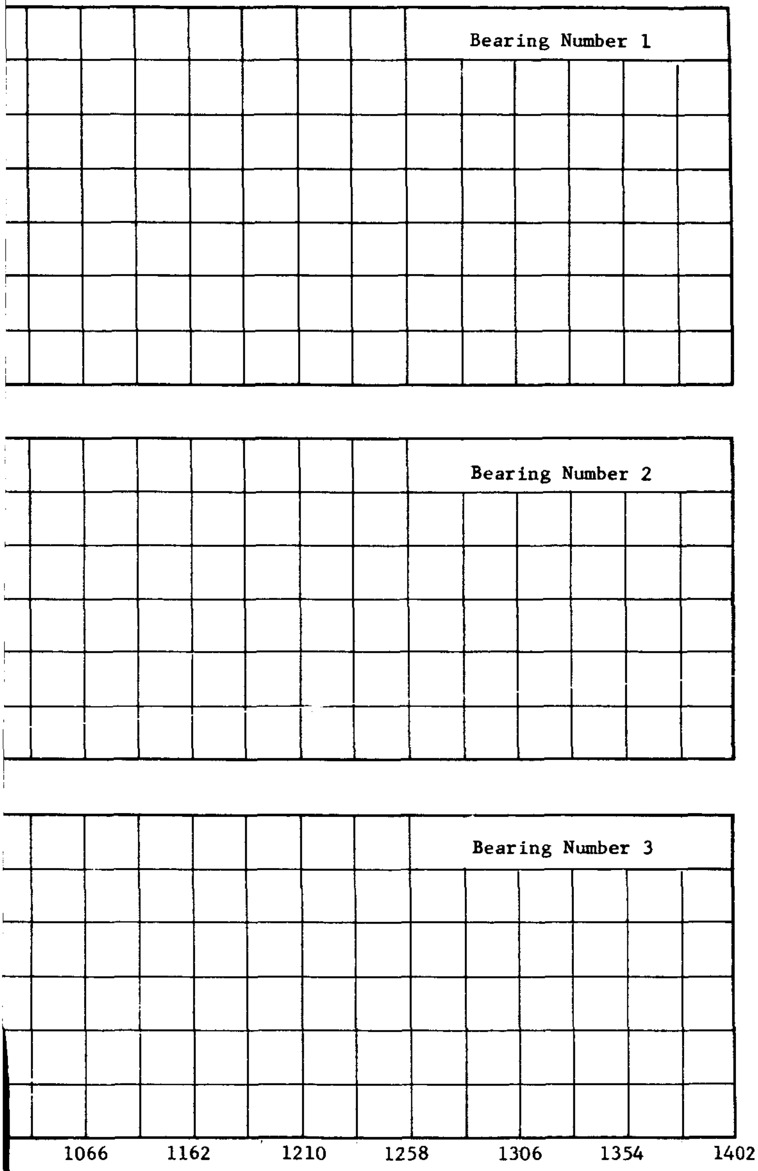
Fig. 24



Average and Peak Torque Measurements on Three Test Bearings at 30 RPM. Bearings, 52100 Steel. Phenolic Retainers. Lubricant, MoS_2 film generated by Run-In From Liquid Suspension (Dry Film)

2

Figure 24


Bearing Number 1
Bearing Number 2
Bearing Number 3
1066
1162
1210
1258
1306
1354
1402

3

Top Bearing

Center Bearing

Lower Bearing

Support Bearing

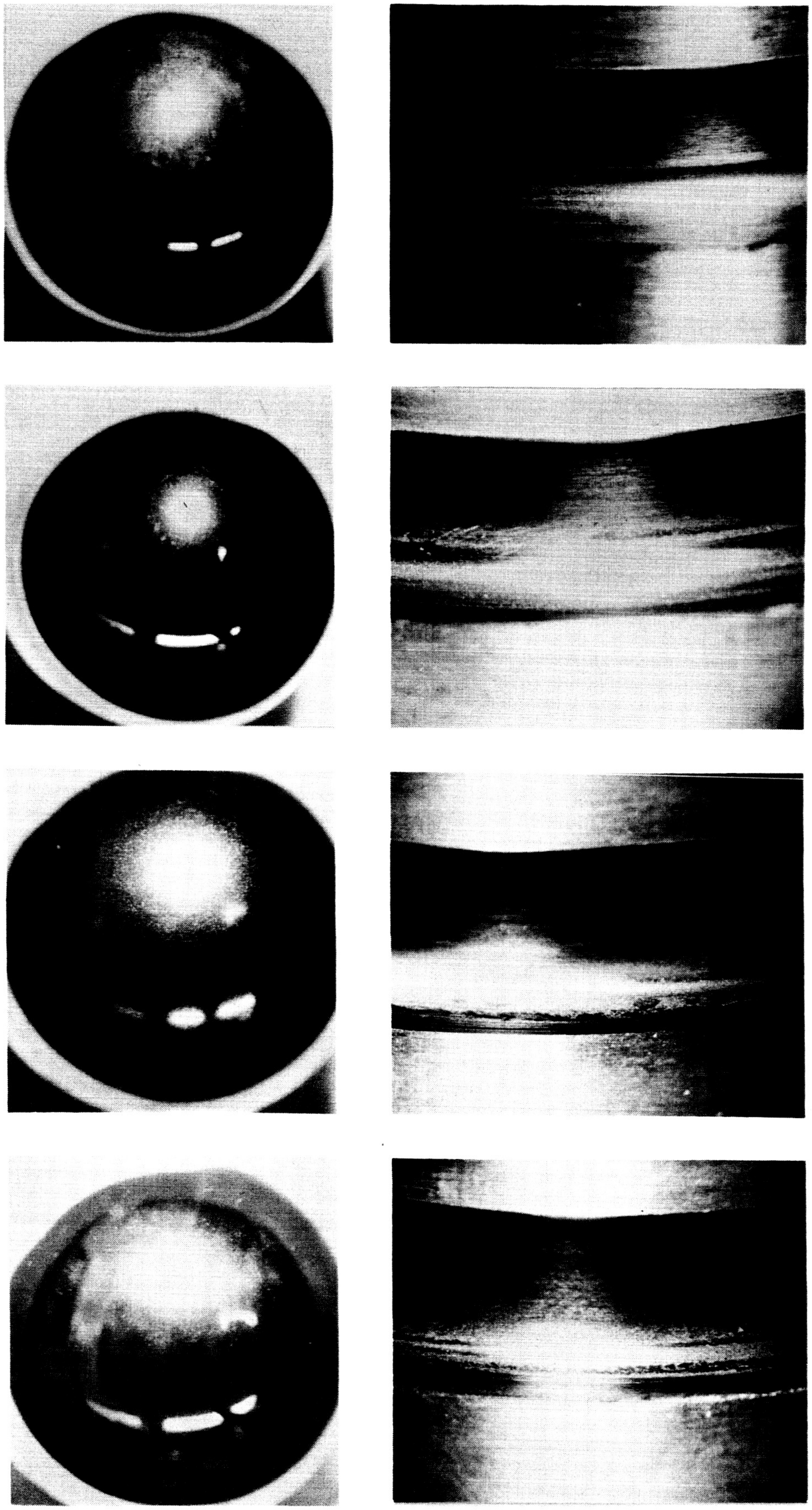


Fig. 25 Typical Contact Areas on Bearings with MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film)

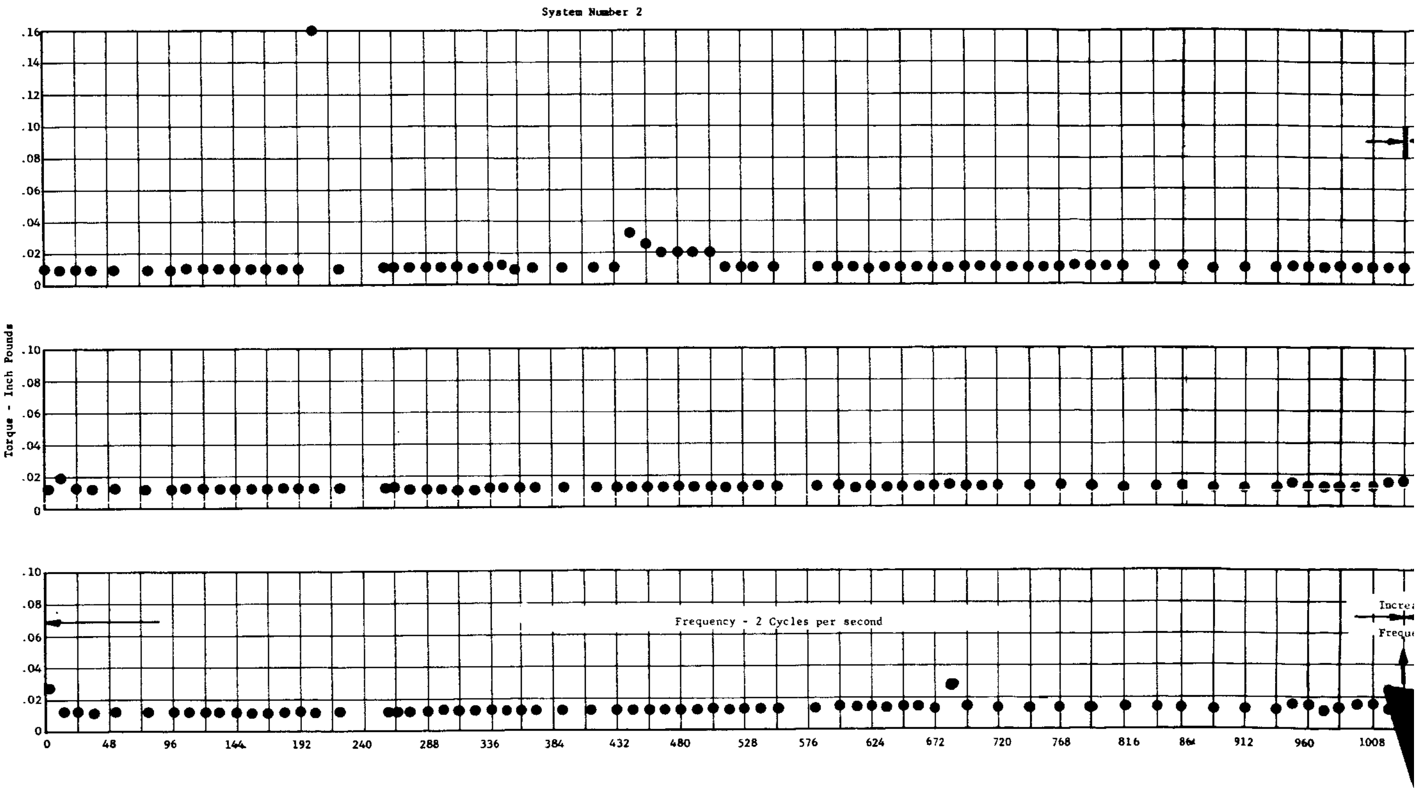
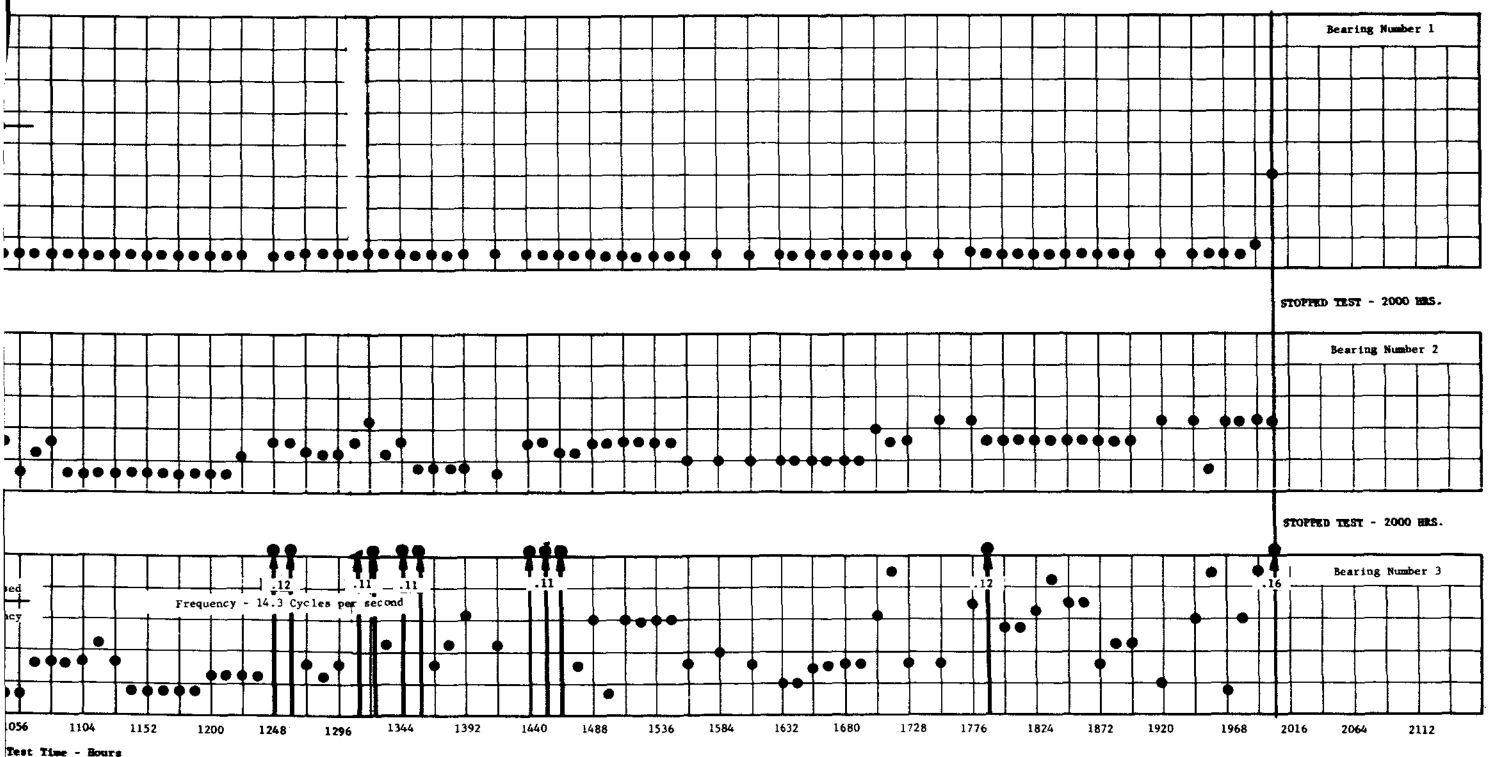


Fig. 26 Oscillating Test. Torque Va
52100 Steel. Phenolic Retai
by Run-In From Liquid Suspen



lues on Three Test Bearings. Bearings,
ers. Lubricant - MoS_2 Film Generated
sion (Dry Film)

Top Bearing

Center Bearing

Lower Bearing

Support Bearing

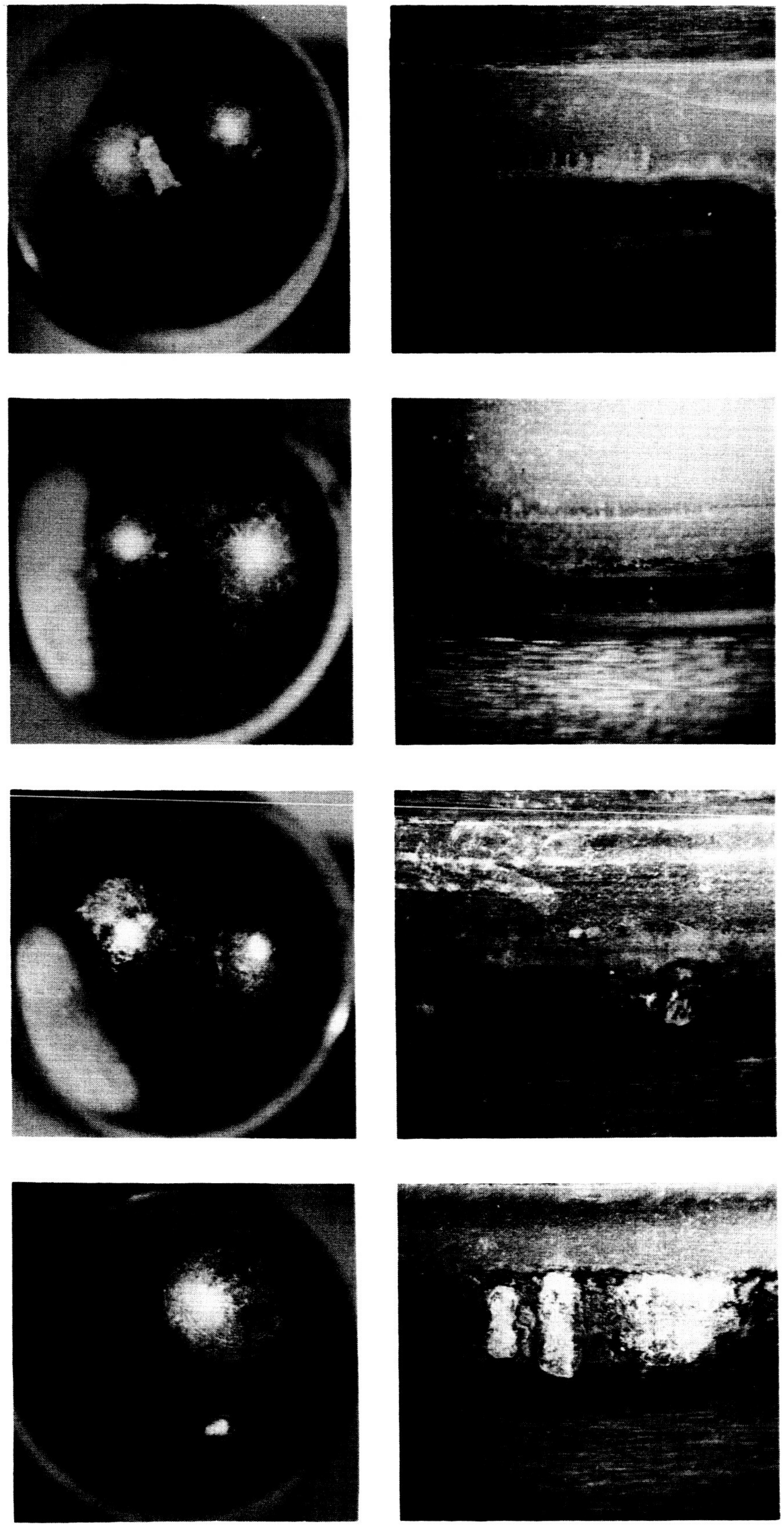
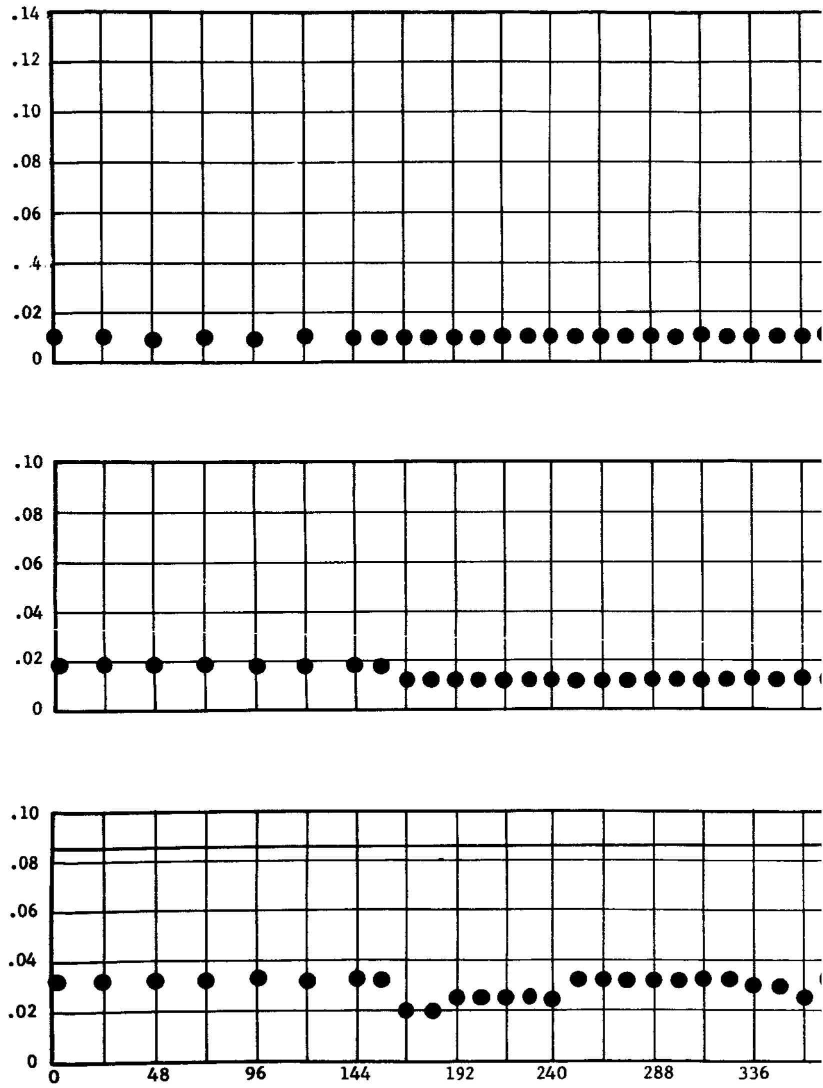
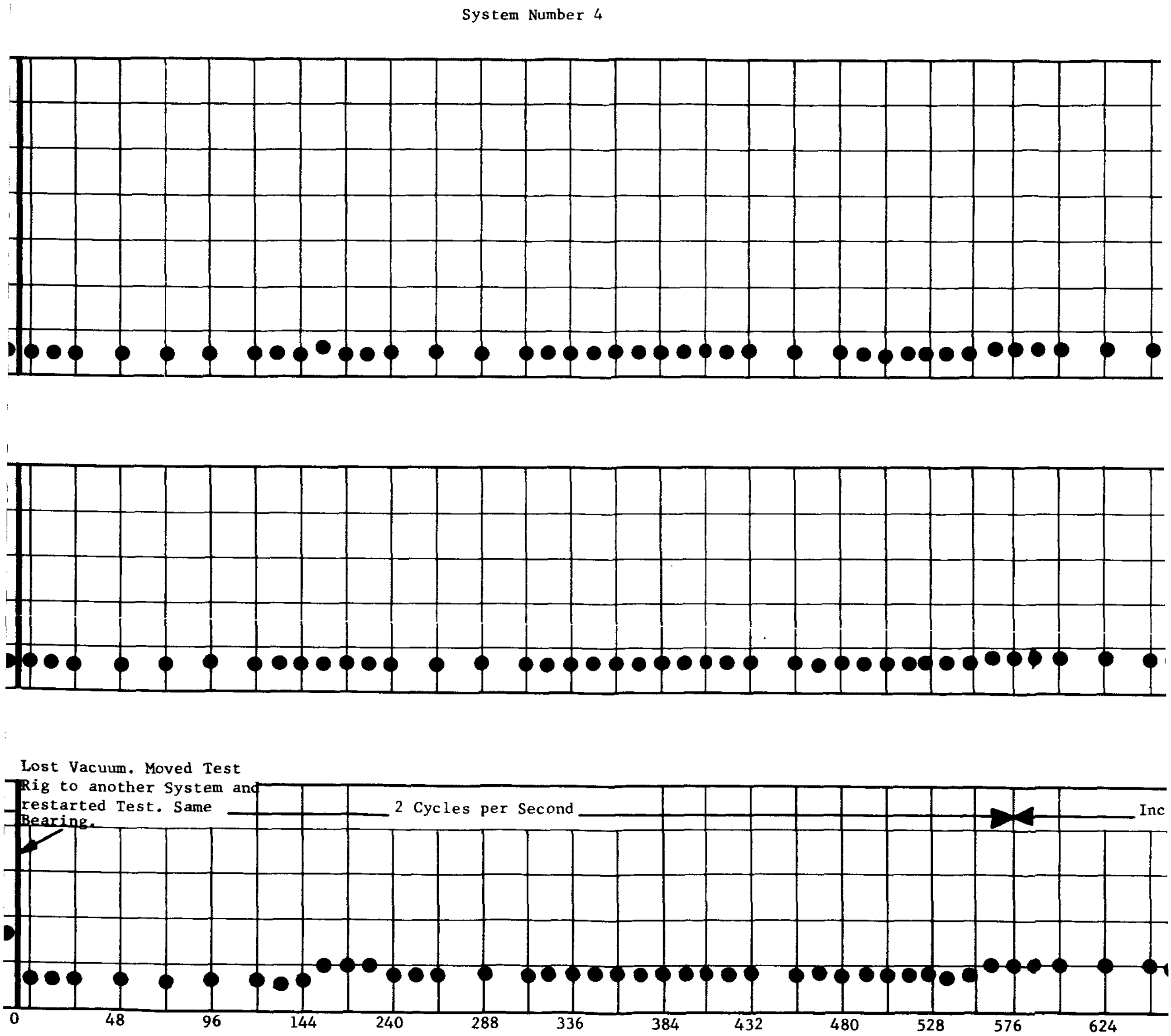


Fig. 27 Oscillating Test. Typical Areas on Bearings Lubricated with MoS_2 Film Generated by Run-In From Liquid Suspension (Dry Film)

Torque - Inch Pounds



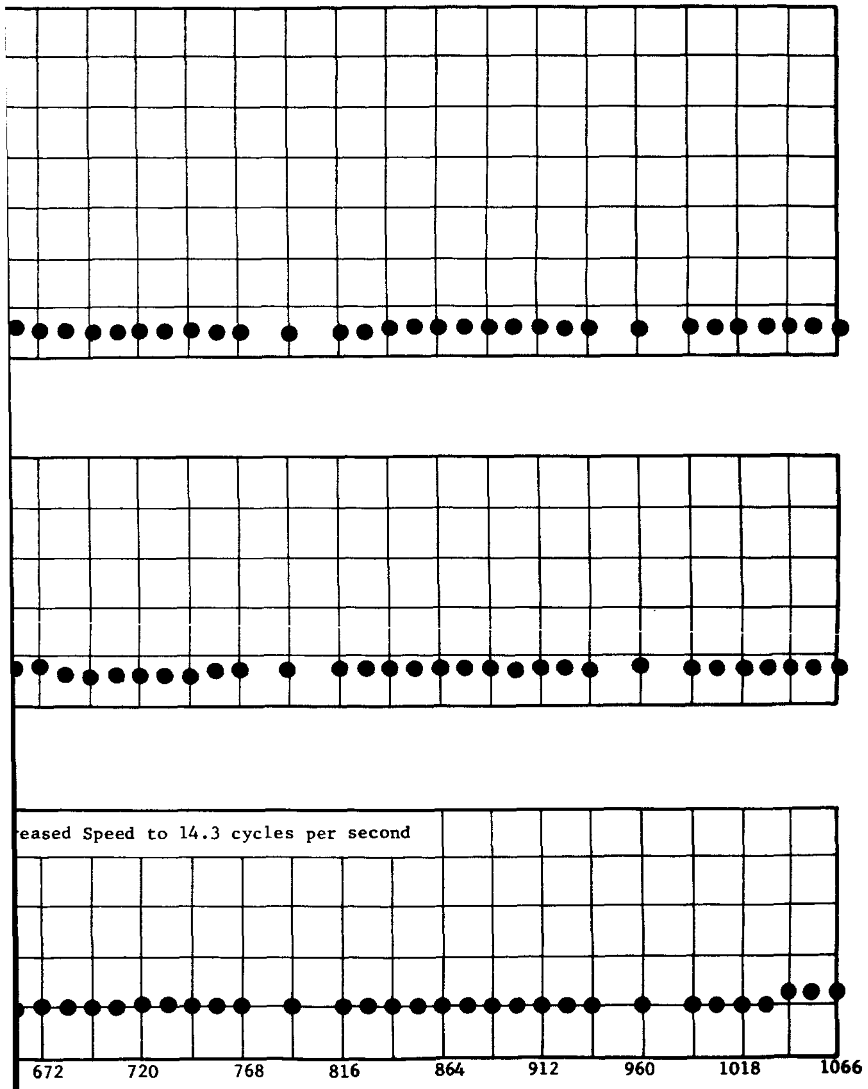


Test Time - Hours

Fig. 28 Oscillating Test. Torque Values on Three Test Bearings. Bearings 52100 Steel. Phenolic Retainers. Lubricant Silicate-Bonded MoS_2-Graphite Film.

Frequency is 2 Cycles per second and 14.3 cycles per second.

Figure 28



eased Speed to 14.3 cycles per second
672
720
768
816
864
912
960
1018
1066

3

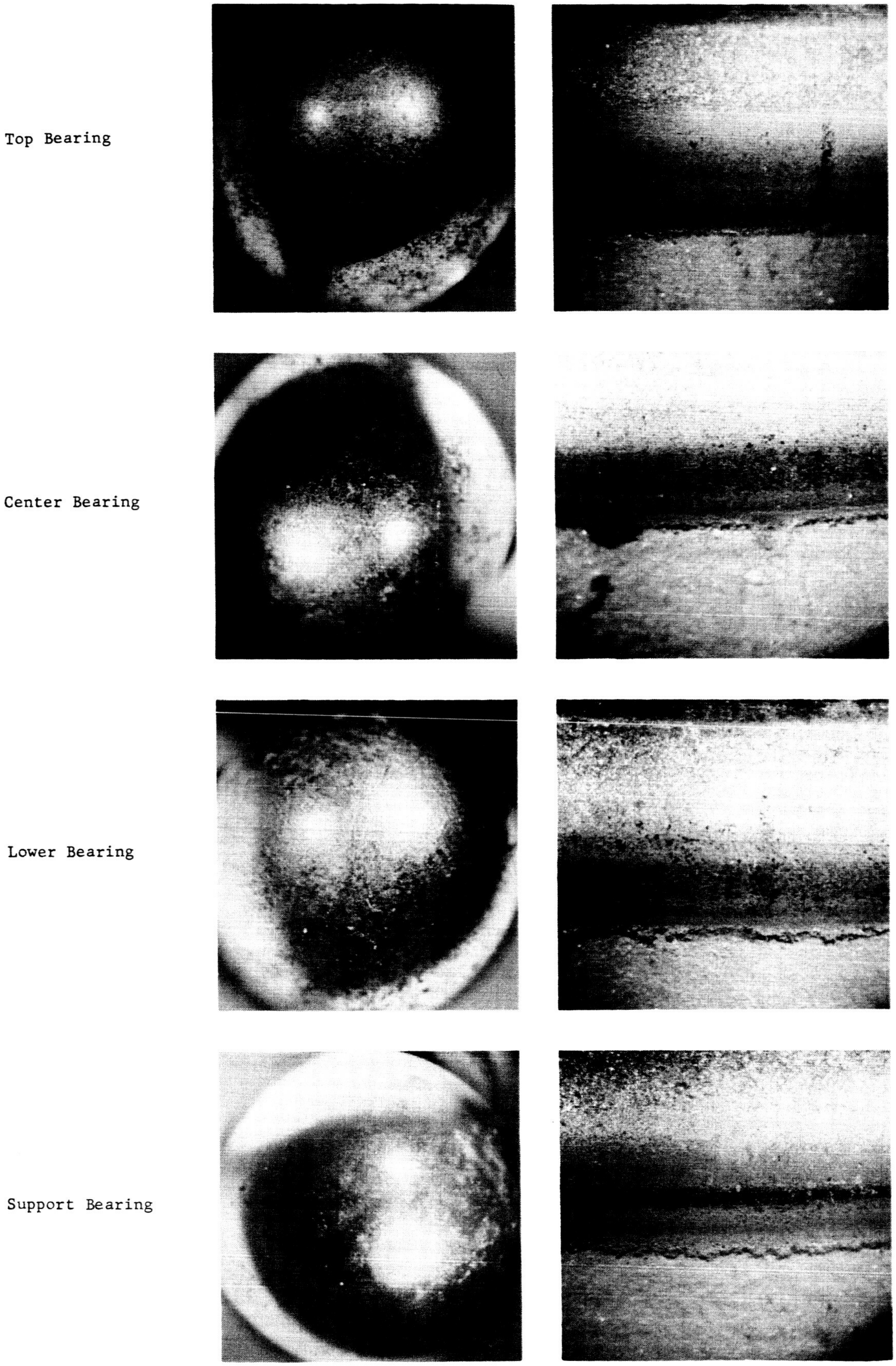


Fig. 29 Oscillating Test. Typical Areas on Bearing with Silicate-Bonded MoS_2-Graphite Film.

APPENDIX A: TECHNIQUE FOR APPLYING SODIUM SILICATE-BONDED MoS_2-GRAPHITE FILM TO BEARINGS

The processing of these bearings was done by the Aeronautical Materials Laboratory. The essential steps were:

1. Precleaning races and balls by vapor degreasing.
2. Applying vapor blasting pretreatment to phenolic retainers.
3. Applying MoS_2 powder (MIL-M-7866) to phenolic retainers by burnishing.
4. Spray deposition of NAEC-AML-23A inorganic solid film lubricant (MoS_2 - 71 wt %, graphite - 7 wt %, sodium silicate - 22 wt %) to races and retainers.
5. Curing solid film by exposing to 77 F - 0.5 hours; 180 F - 2 hours; 250 F - 2 hours.

Each solid-lubricated bearing was then assembled and packaged.

FORMULATION, PREPARATION AND APPLICATION OF MoS_2-GRAPHITE SODIUM SILICATE SOLID-FILM LUBRICANT (SPRAY CONSISTENCY)

1. Formula	Grams
MoS_2 powder (Military Specification MIL-L-7866 (ASG)	70
Graphite powder (military Specification MIL-G-6711)[a]	7
Sodium-silicate solution (pct ration $Na_2O:SiO_2$ = 1:2.9, 43 pct solids, viscosity = 960 centipoises)	50
Water	60

[a] Sieve - Utilize only that which passes through 325 - mesh screen.

2. Preparation

(a) MoS_2 powder (70 g) and graphite powder (7 g) are mixed in a beaker. Water is added while stirring the mixture thoroughly. Continue addition of water, stiffing to insure complete wetting of the powder mixture, until a pourable slurry is obtained. The total quantity of water is approximately 60 g. (NOTE: An excess amount of water will result in separation. If separation is observed, the mixture should be discarded and a second preparation using fresh constituents initiated.)

(b) The pourable slurry is added to sodium-silicate solution (50 g) with stirring to obtain a uniform sprayable mixture.

NOTE: Sodium-silicate solution should be used within date specified on the container.

3. Application

(a) The mixture is placed in a spray bottle (8 oz. capacity) and the spray bottle attached to a spray gun.

(b) Mixture should be agitated immediately before spraying.

(c) Spray pressure is approximately 40 psi.

(d) Apply lubricant to surfaces of retainer, races and lands.

(e) Allow film to air dry prior to applying successive coats.

4. Curing

The coated specimens are then subjected to the following cure cycle:

(a) Air dry - 1/2 hour.

(b) 180 F - 2 hours.

(c) 300 F - 2 hours.

5. Assembly and Run-In

When angular contact bearings are being used, the bearing components should be coated and the film cured. Then the bearings should be assembled by heating the outer race to 300 F and cooling the inner race to approximately -50 F. Heavy snap fits should be avoided because of possible damage to the lubricant film. The bearings should then be worked in carefully by hand under very light pressure to assure free rotation. Loose debris should then be removed by blowing the bearings out with about 20 psig dry nitrogen or dry air, or the bearings should be cleaned with a hand vacuum cleaner.

The bearings should then be set up in a suitable fixture and run-in under the required load and speed conditions for about sixteen hours. The purpose of this run-in is to determine the consistency of the bearing torque and to remove excess lubricant film. Where the application loads and speeds are high, it would be preferable to increase the load or speed by increments rather than subject the bearings to the full load or speed immediately.

A final cleaning with dry nitrogen or air, or with a vacuum cleaner, should be made to remove loose debris.

Torque should be measured and recorded before and after run-in. While it is not possible to specify a particular level of torque without knowing the actual test conditions, excessively high torque or erratic torque behavior should be cause for rejection.

APPENDIX B: SURVEY OF THE LITERATURE ON THE USE OF SOLID LUBRICANTS FOR BEARINGS OPERATING IN VACUUM.

1. Buckley, D.H., Swikert, M.A., and Johnson, R.L., Friction, Wear and Evaporation Rates of Various Materials in Vacuum to 10^{-7} mmHg, Trans. ASLE, Vol. 5, No. 1, April 1962, p. 8.

   This was a study of the behavior of various solid and liquid lubricants in vacuum. Evaporation studies and basic sliding data were obtained on various MoS_2-bonded films. The results showed that epoxy-phenolic resins and silicone resins were more effective binders than sodium silicate for MoS_2 films in sliding tests at low temperatures.

   Teflon and all of the bonded MoS_2 films were satisfactory for use up to at least 300 F as far as evaporation rates were concerned.

   Oil pumps were used for the evaporation tests.
   Ion pumps backed by a mechanical pump which was cryogenically baffled were used for the sliding tests.

2. Evans, H.E. and Flatley, T.W., Bearings for Vacuum Operation - Retainer Material and Design, Presented at the Aviation Conference, Washington, D.C., June 26-28, 1962. ASME paper No. 62-Av-11.

   Motor tests were run at 10,000 rpm in a vacuum of 10^{-7} Torr, using size R2-5 bearings with various metal films as lubricants. These were very lightly-loaded bearings. Over 1600 hours of running time were obtained on a gold-plated bearing with a machined, silver-plated "Circle-C" retainer. A machined S-Inconel retainer also looked promising with a gold-plated bearing.

   It was found that 23+ carat gold, with plating additives to improve its adherence and hardness, was superior to pure 24 carat gold plate.

   Oil free, ion-pumped systems were used.

3. Bowen, P.H., Dry Lubricated Bearings for Operation in a Vacuum, Trans. ASLE. Vol. 5, No. 2, November 1962. p 315.

   Vacuum tests at 10^{-7} Torr were made with size 204 bearings which had various self-lubricating retainers. The bearings were run at 1800 rpm with a 75-pound radial load and a five-pound axial load. These were 100-hour tests although support bearings were operated for as long as 405 hours.

   Best operation was obtained with retainers made of Teflon-glass fiber-MoS_2, bronze-Teflon-MoS_2, and Silver-Teflon-WSe_2.

   An oil-pumped system was used.

4. Buckley, D.H. and Johnson R.L., Gallium-Rich Films as Boundary Lubricants in Air and in Vacuum to 10^{-9} mmHg. Trans. ASLE. Vol. 6, No. 1, January 1963.

   Gallium films were evaluated in sliding tests and were found to be promising lubricants for certain material combinations. Corrosion resistance is an important consideration.

   An ion-pumped system with a mechanical forepump trapped by a cryogenically-cooled baffle was used.

5. Lewis, P., Murray, S.F., Peterson, J.B., and Esten, H., Lubricant Evaluation for Bearing Systems Operating in Spatial Environments. Trans. ASLE, Vol. 6, No. 1, 1963. p. 67.

   This paper summarizes the results of a number of bearing evaluation programs which were performed for space applications. The evaporation characteristics of a number of promising oils and greases were determined at various temperatures in vacuums ranging from 10^{-5} to 10^{-6} Torr.

   Several R-4 size instrument bearings lubricated with solid lubricants and soft metal films were evaluated at 3000 rpm with a 0.8-pound thrust load. A fused Teflon coating and a bismuth film showed the best life at 10^{-6} Torr.

   This work was done in oil-pumped systems.

6. Brown, R.D., Burton, R.A., and Ku, P.M., Evaluation of Oscillating Bearings for High Temperature and High Vacuum Operation. Trans. ASLE, Vol. 6, No. 1, January 1963. p 12.

   Oscillating plain bearings were evaluated over a temperature range from -90 F to +1750 F at vacuums from 10^{-3} to 10^{-6} Torr. The tests were run at 31 cycles per minute with a travel of 25° single amplitude and stresses from 4400 to 12,000 psi.

   Best results were obtained with MoS_2 powder in the bearing cavity. A resin-bonded MoS_2 film was effective until the film was worn away.

   Higher friction was measured at -90 F.

   This was an oil-pumped system.

7. Devine, M.J., Lamson, E.R., and Bowen, J.H., Jr. The Effect of the Chemical Composition of Metals in Solid Lubrication. Presented before Div. of Petroleum Chemistry, ACS, Los Angeles Meeting, March 31 - April 5, 1963.

   This is a general paper reviewing the results of extensive work on the effect of the base metals on the performance of sodium silicate-bonded MoS_2-graphite films.

Tests were run in air at 10,000 rpm, 750 F, using size 204 bearings with a 5-pound thrust and a 3-pound radial load. Molybdenum retainers gave lives on the order of 1000 hours. At 5 x 10^{-6} Torr, a bearing lubricated with this film ran effectively for 100 hours at 1000 F.

This was probably an oil-pumped system.

8. Young, W.C., Clauss, F.J. and Drake, S.P., Lubrication of Ball Bearings for Space Applications. Trans. ASLE. Vol. 6, No. 3, July 1963. p. 178.

Instrument-size ball bearings were evaluated as motor support bearings in vacuum at pressures of 10^{-7} to 10^{-9} Torr using several oils and dry film lubricants. Radial loads of 155 to 454 grams were used. The bearing speed was 8000 rpm.

Best performance with solid-lubricant films was obtained on a sodium silicate-bonded MoS_2 film which ran 2213 hours before failure. However, other tests with the same film gave lives of 1862, 1796, 28 and 25 hours.

Two self-lubricating retainer materials showed promise. Both of these were reinforced Teflon. A 60 percent Teflon, 40 percent glass fibre composite with MoS_2 was still running after 4353 hours under an axial load of 0.25 pounds.

Bearings lubricated with oils and greases have been operated for more than a year at 10^{-8} Torr.

Ion-pumped systems were used with all dry lubricant films and composites.

9. Boes, D.J. and Bowen, P.H., Friction-Wear Characteristics of Self-Lubricating Composites Developed for Vacuum Service. Trans. ASLE. Vol. 6, No. 3, July 1963. p. 192.

Composites made of solid lubricants + Teflon + metals were investigated in basic sliding tests. The optimum compositions (Teflon-WSe_2-silver and Teflon-WSe_2-copper) were then evaluated in size 204 and 305 bearings. One hundred-hour tests were run in vacuums from 10^{-7} and 10^{-8} Torr. Radial loads from 75 to 250 pounds were used at speeds of 1800 rpm and 35 rpm. Test temperatures were varied from -180 F to +300 F. Promising results were obtained.

These were run in oil-pumped systems.

10. Lavik, M.T. and Clow, W. L., Friction and Wear Characteristics of a Ceramic-Bonded Solid Lubricant Film. 1963 USAF Aerospace Fluids and Lubricants Conference. San Antonio, Texas.

A Pbs: MoS_2:B_2O_3 lubricant film gave good wear life and frictional behavior in both air and vacuum of about 10^{-6} Torr. The wear life increased by a factor of three in vacuum.

This was an oil-pumped system.

11. Proceedings of Space Lubrication Conference. Fuels and Lubricants Division, Office of the Director of Defense Research and Engineering. Report FL210/53 September 1963.

This is a compilation of papers which summarize the efforts of various government agencies and private companies. Much of the work is covered in these abstracts.

12. Reichenbach, G.S., Shaw, R. Jr., and Foster, R.G. Lubricant Behavior in High Vacuum. Trans. ASLE, Vol. 7, No. 1, January 1964. p. 82.

Pin-on-disk sliding friction tests and crossed-cylinder load-carrying tests were run in air and in vacuum with various oils.

Friction was not changed significantly by pressure but the load-carrying ability of the oils was decreased by more than 50 percent in vacuum.

These tests were run in oil-pumped systems.

13. Bowen, P.H., Solid Lubrication of Gears and Bearings in a Space Environment. Trans. ASLE, Vol. 7, No. 3, July 1964. p.227.

A study was made of heavily loaded bearings and gears running at speeds from 30 to 50 rpm at vacuums of 10^{-8} to 10^{-9} Torr. This was an oil-pumped space chamber.

Bearings with retainers of Ag-Teflon-WSe_2 and Cu-Teflon-WSe_2 were used. Idler gears of the same compositions were used to supply lubricants to the gear teeth.

Promising results were obtained.

14. Brown, R.D., Burton, R.A., and Ku, P.M., Long Duration Lubrication Studies in Simulated Space Vacuum. Trans. ASLE, Vol. 7, No. 3, July 1964. p. 236.

Both sliding and rolling contact bearing tests were run in ion-pumped systems.

Sliding tests on resin-bonded MoS_2 and silicate-bonded MoS_2 showed comparable friction results at room temperature but the silicate film showed higher friction at 300 F.

Ball bearings, size 204,were run at 1800 rpm with a 0.75 pound axial load. Dry 52100 steel bearings with pressed steel retainers failed in 15-30 minutes. With a bronze retainer, slightly better life was obtained.

Best ball bearing life was obtained with an epoxy-MoS_2 composite retainer. A Teflon-glass-MoS_2 retainer was much less effective.

A dioctyl phthallate-impregnated porous bronze retainer ran for 150 days but extensive oil decomposition was observed.

A silicate-bonded MoS_2-graphite film gave two widely different results. One ball bearing failed in thirteen days, the other ran very effectively for 206 days.

15. Craig, W. D., Jr., Friction Variation of PTFE and MoS_2. Lubrication Engrg. Vol. 20, No. 7, July 1964. p.273.

Starting friction characteristics of fused Teflon and sodium silicate-bonded MoS_2-graphite films were studied in small journal bearings at 10 psi stress. Temperatures from -130 F to +200 F were used at vacuum levels of 10^{-7} to 10^{-8} Torr. Exposure to vacuum for as long as five months did not affect the coatings. The friction of the fused Teflon was affected by temperature in the range from -100 F to +150 F. The silicate-bonded MoS_2-graphite film varied only slightly in friction with temperature.

16. Campbell, M.E. and Van Wyk, J. J., Development and Evaluation of Lubricant Composite Materials. Lubrication Engrg. Vol. 20, No. 12, December 1964. p. 463.

A large number of composite materials containing MoS_2 were fabricated and screened in sliding tests.

The most promising composite (90 percent MoS_2, 8 percent Fe, 2 percent Pt) was made into a retainer and was run in a K162B titanium carbide ball bearing at speeds ranging from 5000 to 15,000 rpm, radial loads from 37.5 to 75 pounds,thrust loads of 12.5 to 25 pounds, temperatures from 300 to 680 F and a vacuum of 1 x 10^{-5} Torr. The test was terminated after 140 minutes because of a malfunction in the test equipment.

This appeared to be a promising approach.

Oil-pumped equipment was used for vacuum.

17. Devine, M.J., Lamson, E.R., Cerini, J.P., and McCartney, R.J., Solids and Solid Lubrication. Lubrication Engrg., Vol. 21, No. 1, January 1965. p. 16.

This paper is a general review of solid lubricants and a large number of typical examples and the test parameters are given.

18. Hopkins, V. and Gaddis, D. Friction of Solid Film Lubricants Being Developed for Use in Space Environments. Lubrication Engrg. Vol. 21, No. 2., February 1965, p. 52.

Basic sliding evaluations were run at light loads and at temperatures from 80 F to 400 F in air and in a vacuum of 10^{-6} Torr. Sodium silicate-bonded MoS_2 and graphite, and modifications of this film by the addition of gold, bismuth or molybdenum, showed promise. Additions of gold or bismuth appeared to improve friction in vacuum, particularly at 80 F. These additives may also increase the wear life of this film.

These tests were made in an oil-pumped system.

19. Flom, D. G., Haltner, A. J., and Gaulin, C.A., Friction and Cleavage of Lamellar Solids in Ultra-High Vacuum. Trans. ASLE. Vol. 8, No. 2, April 1965. p. 133.

Evidence is presented that MoS_2 does not require adsorbed gases or vapors to be an effective lubricant in vacuum.

Work was done in 10^{-9} Torr range using ion-pumps.

20. Other References Which Were Not Abstracted:

21. Westmoreland, R., and Reed, J.D. Vacuum Testing Bearings. Space Aeronautics. Vol. 37, 1962. pp. 175-177.

22. Boes, D. J., Long Term Operation and Practical Limitations of Dry, Self-Lubricated Bearings. Lubrication Engrg., Vol. 19, No. 4, April 1963, p. 137.

23. Proceedings of a Symposium on Lubrication in Space. Arthur D. Little, 1963.