PERFORMANCE OF BALL BEARINGS IN AIR AND VACUUM WITH NO ADDED LUBRICATION

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FOREWORD

This report was prepared by Midwest Research Institute, 425 Volker Boulevard, Kansas City, Missouri 64110, under Air Force Contract No. F33615-69-C-1236, Phase II, which was initiated under Project No. 7340, "Nonmetallic Composite Materials," and Project No. 7343, "Aerospace Lubricants." The effective date of this 36-month contract was 1 January 1969.

The work reported herein was conducted from 1 January 1969 to 8 December 1971, and was monitored by the Air Force Materials Laboratory under the direction of Project Engineer, Mr. F. C. Brooks. The manuscript was submitted by the author in December 1971.

The author gratefully acknowledges the assistance of Messrs. C. D. Geisler and C. J. Windisch in the conduct of the experimental work.

This technical report has been reviewed and is approved.

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ABSTRACT

Forty-one experiments on six different designs of size 204 ball bearings were conducted. The results of these experiments, operating time and frictional torque values determined by coast-down times, are presented for various test atmospheres, loads, and speeds. The experimental approach is outlined and the different test apparatus are described. The distinctive design features of the production-type ball bearings are identified. Even though no lubrication was applied to any of these bearings, several of the specimens ran satisfactorily for over 10,000 hr at speeds of 1,790 and 3,600 rpm with light axial loads of 7 lb nominal. One specimen continues to operate in a vacuum environment after 45,600 hr at 1,790 rpm.

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I.

INTRODUCTION

Long component life is one of the goals of the lubrication engineer. Component life must be sufficient to satisfy system requirements, so that the lubricated element will perform its function for as long as the system is to operate. In striving to realize the goal of long life and to extend the system usefulness, the engineer must determine if individual components can demonstrate the expected life within the design conditions using either standard or specialty lubricants.

The operating conditions in which the lubricant is to function are usually well-defined. Much thought and analysis are devoted to the selection of the proper temperatures, pressures, loads, speeds, and atmospheres. Quite often, expensive test apparatus are developed to duplicate these conditions. However, the selection of which particular test element to use is not always based upon a thorough study of the available elements.

This report presents data that demonstrate the variation in performance of several different ball bearing test elements; bearing performance is measured by bearing operating life and frictional torque. All of the data reported are for bearings that have been chemically cleaned and have no applied lubrication in the form of dry powders, oils, or greases. The work is a continuation of previous effort (Ref. 1), with emphasis upon fulfilling Air Force requirements for long-term service of bearings in a vacuum environment.

This report presents the conclusion of the unlubricated bearing work. Included are sections on the experimental approach, descriptions of the two apparatus used for the work, and descriptions of the six bearing specimen designs. The results of the 41 experiments are also presented, along with various observations and conclusions regarding the results. A mathematical development of the coast-down time relationship to determine operating frictional torque is presented in Appendix A; the statistical handling of wear-life results is presented in Appendix B.

II.

EXPERIMENTAL APPROACH

The experimental approach used for this work was to rotate individual ball bearings until the frictional torque exceeded a predetermined value. Six different bearing designs were used, all bearings being size 204. (See Table II, p. 11 for size 204 bearing dimensions.) Forty-one experiments were conducted; 16 were in an air atmosphere; 21 were in chambers evacuated to pressures as low as 1.8×10^{-9} torr; and four were in air for a specified time, followed by operation in evacuated chambers.

Each of the bearings was chemically cleaned prior to installation in one of the test apparatus. The cleaning procedure was to ultrasonically clean the bearing in Stoddard solvent for 40 min and then to soak the bearing in clean, filtered acetone for an additional 60 min. Air drying after the acetone soak completed the bearing preparation. If the bearing was not placed immediately in one of the test apparatus, it was stored in a desiccator. No material was placed on any of the bearings to act as a lubricant.

After being weighed, the bearing test specimen was placed in one of the two types of test devices, described in Section III, Test Apparatus. The load, either a nominal 5- or 7-lb axial load, was applied and the test chamber was evacuated if a vacuum environment was to be used. Bearing operation was then initiated. Operation of each bearing experiment was interrupted periodically to check frictional torque and/or bearing weight loss. The frequency of interruption depended upon factors such as expected bearing operating life, previous period weight loss, and coast-down time (frictional torque). The individual experiments were considered completed when the bearing failed to restart after a frictional torque determination or when the frictional torque of the rotating bearing exceeded a specified maximum driving torque. Visual inspection of the bearing was made after removal from the test apparatus.

III.

TEST APPARATUS

For this work, two different types of test apparatus were used, a bearing rig and a Pope spindle.

A. Bearing Rigs

Eight bearing rigs were used. Each rig is capable of subjecting one size 204 bearing to a light load in either a vacuum or an air environment. Each of the eight rigs is a separate operating station, consisting of test chamber, bearing holder, drive motor, magnetic coupling, and vacuum pump (if a vacuum environment is desired). The loading of the bearing, a nominal 7-lb axial, is generated by the weight of the rotating portion and the attraction between two magnets. One magnet is inside the chamber attached to the rotating bearing inner race through a shaft and the other is outside attached to the driving motor. A schematic of one of the rigs is shown in Figure 1; the component parts are pictured in Figure 2.

The rotating portion of the load consists of the rotating shaft, spacer, washer, drive magnet, another washer, and the retaining screw, all of which are attached to the bearing inner race. The bearing outer race rests on a shoulder of the inner cylinder, which in turn is held in place by the long spacer, shield, and screw ring. The inner cylinder fits into the test chamber, is held in place by the snap ring, and is restrained from rotating by a pin inside the chamber that fits into a slot in the inner cylinder.

When a vacuum environment for the bearing is desired, the test chamber is attached to the ion pump, using the copper gasket to form the vacuum seal. When an air environment is desired, a piece of plexiglass is attached to the test chamber to keep foreign material from entering the test chamber and interfering with bearing rotation. The assembly, with either the vacuum pump or plexiglass cover, fits into a holder which positions the test chamber over the driving motor and magnet.

The driving motor and magnet can be raised and lowered, allowing the magnetic coupling between the driving and driven magnets to be established through the bottom of the test chamber. A rheostat is used in the driving motor circuit so that motor speed can be increased gradually. After full speed is obtained, the rheostat is removed from the circuit and full-line voltage powers the motor. The three different speeds used in this work on the bearing rigs were obtained by using different speed motors--1,800, 3,600, and 8,000 rpm.

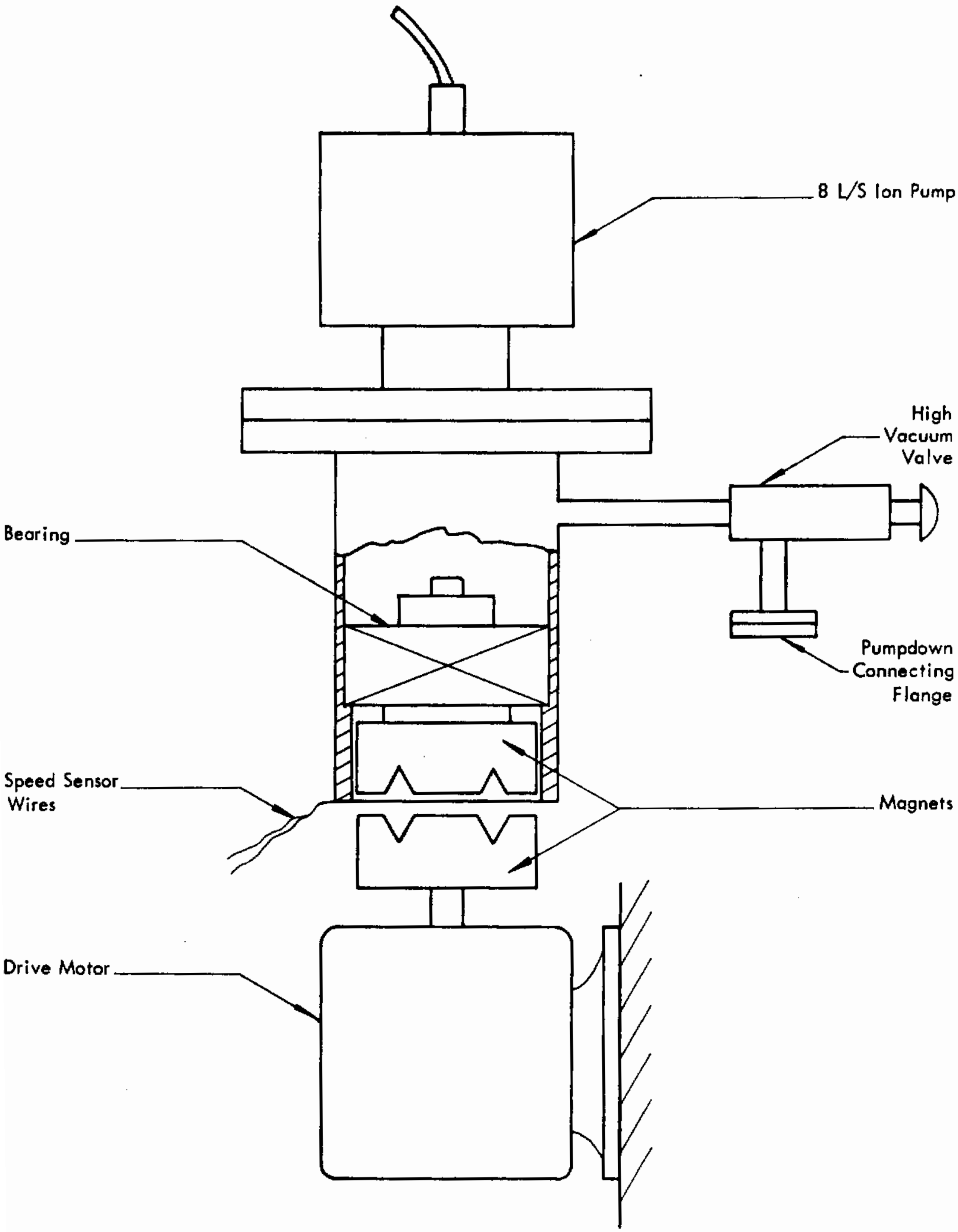

Figure 1 - Bearing Rig Schematic

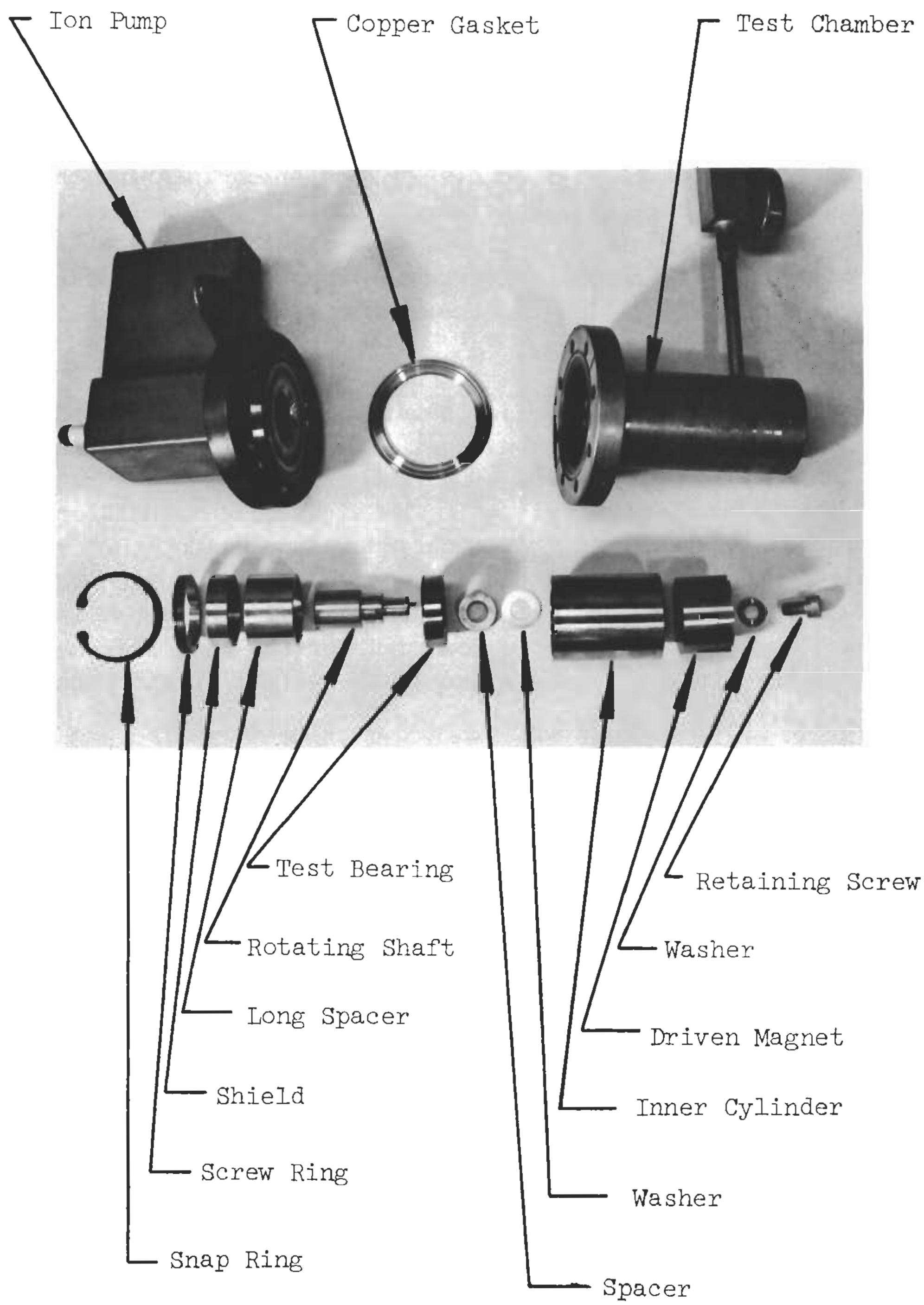

Figure 2 - Bearing Rig Component Parts

Bearing frictional torque is determined from the coast-down time of the bearing in the chamber. Coast-down time is measured by loosening a holding screw and quickly lowering the motor and driving magnet approximately 2 in. A voltage, induced in a coil wrapped around the lower end of the test chamber by the rotation of the driven magnet, decays to zero as the rotating portion comes to rest. The time between motor lowering and voltage decay to zero is the coast-down time. Frictional torque has been calculated and plotted as a function of coast-down time. (See Appendix A for the mathematical development and curves of torque versus coast-down time.) If the coast-down time from 1,800 rpm is less than 1 sec--the lowest readable value--the frictional torque is recorded as greater than 3 oz-in. If the bearing frictional torque exceeds 44 oz-in, the magnetic coupling torque is exceeded and the bearing cannot be driven.

The magnetic attraction between the driven and driving magnets contributes to the load on the test bearing. This magnetic force is measured before bearing rotation begins and after the experiment (or a major phase of the experiment) has been completed. The eight-pole magnets are indexed at the four possible locations. At each indexed position, the driving magnet is loaded with dead weight until the attractive force is overcome. Then the break away force, the dead weight, is measured. The force is determined twice for each location. The force recorded for the test bearing load is the average of these eight values added to the weight of the rotating portion (which is 1.15 lb).

B. Pope Spindle

All of the 10,000 rpm testing was done on a Pope spindle. This machine has two bearings mounted on a rotating shaft; one bearing is the test bearing while the second is a support bearing. The horizontal shaft is belt-driven at 10,000 rpm by a step-up pulley from a 3,500 rpm drive motor. Axial loading of the test bearing is accomplished by using wavy washers (axial force springs); radial load is controlled by using part of the weight of the motor acting on the spindle through the driving belt. The loads used for this work were 5-lb axial and 5-lb radial. A schematic of the Pope spindle is shown in Figure 3.

On the left end of the rotating spindle are mounted the left end flinger and the test bearing, which seats against a shoulder on the spindle. On the right end of the spindle are mounted the support bearing, seated against the shoulder opposite the test bearing, and the driven pulley. End caps, attached to the spindle housing, position the bearing outer races, the outer seals, and the wavy washer.

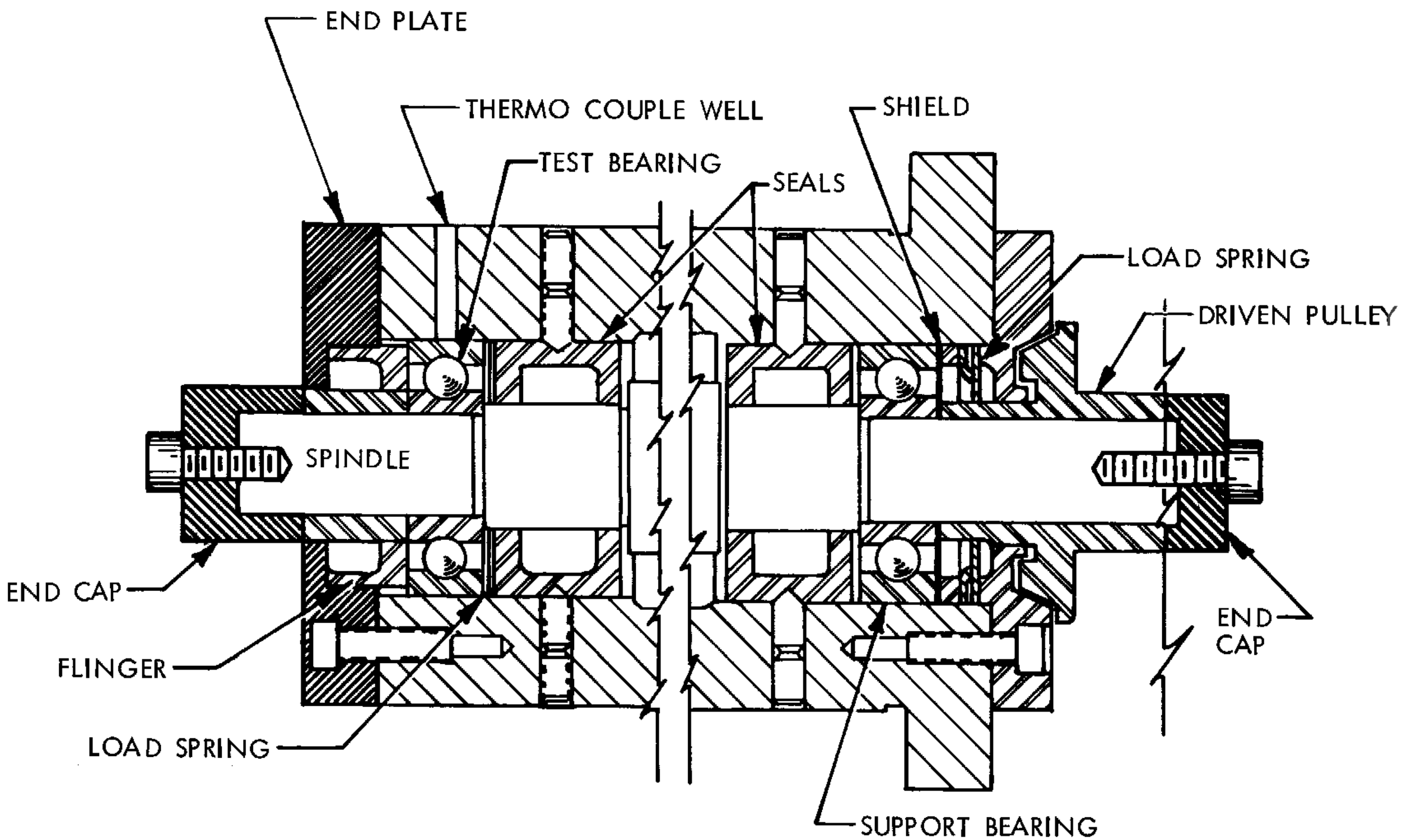

Figure 3 - Pope Spindle Schematic

The shoulder area of the spindle, separating the test and support bearings, also serves as the mating surface for the seals separating the test area from the support area. The support bearing is lubricated with conventional greases; adequate seals and reservoir volumes, coupled with the high rotational speeds, keep the support bearing lubricant from contacting the test bearing.

The spindle assembly fits into a mounting bracket on the machine base. The base holds the pivot-mounted, spring-supported motor plate and motor, with provisions for adjusting the amount of spring force necessary to provide the required belt tension for the radial loading of the spindle assembly. Motor speed controls (high, low, off), heater controls (if applied heat is required), and temperature monitoring provisions are also included on the machine base. No applied heat was used in any of this work. Also, there are no provisions for any atmospheres for the test bearings other than normal laboratory air.

IV.

TEST SPECIMENS

In this work, six different bearing designs were used, all of which were size 204 bearings which have a 20-mm bore, 47-mm outside diameter, and 14-mm width. The bearing quality ranged from high precision bearings (ABEC-7) (Annular Bearing Engineers Committee standards) to low precision bearings (ABEC-1). The performance of these bearings is not to be considered a reflection upon the manufacturer of the bearings; the bearings have been subjected to environments and conditions other than those for which they were designed. The bearing designs and outstanding construction details are presented in Table I for comparison.

Bearing A is an ABEC-5 precision bearing, made by Marlin Rockwell Corporation. (See Table II, adapted from Ref. 2, for the ABEC tolerances for the size 204 bearing dimensions.) The bearing is made from 52100 steel and utilizes an inner-land-riding, high nickel cast iron separator. The separator has the shape of a thick cylinder, occupying 90% of the space between the inner and outer races.

Bearing B is an ABEC-7 precision bearing, employing 11 balls 0.3125 in. in diameter. This bearing, made by the Barden Corporation, is constructed from 440-C stainless steel. The outer-land-riding separator is made from a leaded tin bronze, identified as SAE 792 bronze. The separator is in the shape of a thin cylinder, being 0.040 in. thick, 0.521 in. wide, and weighing 9.170 g.

Bearing C is another Marlin Rockwell Corporation unit of ABEC-3 precision. The eight balls and both races are made from M-10 tool steel. The silver-plated iron-silicon bronze separator is a ball-riding separator, stamped into two ribbons that are riveted together between each of the bearing balls.

Bearing D is made by New Departure as an ABEC-5 precision bearings. This unit uses eight balls, separated by a U-channel-shaped separator. The balls, races, and separator are made from M-2 tool steel. The U-channel separator operates as an outer-land-riding type, with the "prongs" of the U in contact with the lands of the outer race.

TABLE I

BEARING DESIGNATION AND DESCRIPTION

Bearing Identifying Code	Manufacturer's Name and Number	ABEC[a/] Precision Class	Number of Balls	Race and Ball Material	Type of Separator	Separator Material	Separator Design
A	Marlin Rockwell Corporation MRC 204R	5	9	52100	Inner-land riding	High nickel cast iron	Thick cylinder
B	Barden Corporation S204 HX100K6	7	11	440-C	Outer-land riding	SAE 792 bronze (leaded tin bronze)	Thin cylinder
C	Marlin Rockwell Corporation MRC 204S17	3	8	M-10	Ball-riding	Silver-plated iron-silicon bronze	Stamped, ribbon type, riveted
D	New Departure X-14047-D	5	8	M-2	Outer-land riding	M-2 tool steel	Machined, U-channel
E	Norma Fag Bearings Bearings Corporation 204[a/]	1	7	52100	Ball-riding	AISI 1008 steel	Stamped, ribbon type, bent tabs
F	Jack & Heintz JH 20BC02	1	8	52100	Ball-riding	Steel	Stamped, ribbon type, bent tabs

a/ ABEC - Annular Bearing Engineers Committee.

TABLE II

BEARING DIMENSIONS AND ABEC TOLERANCES
FOR SIZE 204 BEARINGS

	Basic Size (in.)	ABEC Tolerances (in.) -1	-3	-5	-7
Bore diameter (20 mm)	0.7874	+0.0000 -0.0004	+0.0000 -0.0002	+0.0000 -0.0002	+0.0000 -0.00015
Eccentricity of inner race		0.0005	0.0003	0.0002	0.00015
Outer race outside diameter	1.8504	+0.0000 -0.0005	+0.0000 -0.0003	+0.0000 -0.0002	+0.0000 -0.0002
Eccentricity of outer race		0.0008	0.0004	0.0002	0.0002
Width	0.5512	+0.000 -0.005	+0.000 -0.005	+0.000 -0.005	+0.000 -0.005
Balls	No standard size or number given for all 204 bearings.				

Bearings E and F, made by Norma Fag Bearings and the Jack & Heintz Company, respectively, are similar in construction. These bearings are both ABEC-1 precision, with the balls and races constructed from 52100 bearing steel. Both bearings use the stamped, ribbon-type separator with bent-tab construction. The separator material for both bearings is a plain carbon steel. The major difference between the two bearings is only in the number of balls used--the Norma bearing uses seven and the Jack & Heintz uses eight.

Bearing selection for this work was made to enable the evaluation of several distinctly different design features. Again, bearing performance is not to be considered a reflection upon the manufacturers; the bearings were subjected to environments and conditions other than those for which they were designed. The specific features could have been selected from other sources just as easily as from the source selected.

V.

EXPERIMENTAL RESULTS

The results of the 41 experiments are presented in Table III. Bearing performance was considered to be the operating life and the frictional torque, with greater emphasis being placed upon operating life. A partial summary of the average operating times is presented in Table IV, showing only the results for the 1,790 rpm operation using one environment per bearing. The range of the bearing operating times is quite large, ranging from 1 min to over 5.2 years, although all of the bearings were cleaned and handled in the same manner and tested in identical rigs.

A. A Bearings

The A bearings had wear-lives of 1.3 hr and 1.0 hr. The second experiment did not attain full speed at any time during the 1 hr of attempted operation. The "average" time shown in Table IV is based upon the total number of revolutions of both bearings divided by the nominal speed of the experiments (1,790 rpm) averaged between the two bearings. The separator, made of Niresist (a high nickel cast iron), is in the shape of a thick cylinder, bored to receive the nine balls. The relatively large amount of rubbing surface presented to the balls apparently allowed the generation of a sufficient amount of wear debris to jam the bearings and stop rotation.

B. B Bearings

The performance of the B bearings has been much better than expected. The first bearing of the entire test sequence is still operating. This bearing was installed in the test rig on 27 June 1966. Operation of the bearing was restricted to 8 hr/day until bearing performance required that provisions be incorporated for unattended operation and automatic cutoff at failure. The range of determined values of frictional torque has been 0.03 to 0.19 oz-in, with most of the values being in the 0.06 to 0.08 oz-in range.

Another of the B bearing specimens was operated for 1,604 hr in the same test equipment, using the same cleaning techniques and test procedure. Operation was in a vacuum environment of 5×10^{-9} to 5×10^{-8} torr.

TABLE III

TEST RESULTS

Bearing	Vacuum (torr)	Speed (rpm)	Axial Load (lb)	Frictional Torque (oz-in)	Operating Time (hr)
A	4×10^{-8} to 1.1×10^{-7}	1,790	6.50	> 3	1.3
A	3×10^{-7} to 3×10^{-6}	a/	6.46	> 3	1.0
B	1.8×10^{-9} to 2×10^{-7}	1,790	7.20	0.03 to 0.19	> 45,672b/
B	5×10^{-9} to 5×10^{-8}	1,790	7.88	0.08 to 0.13	1,604c/
B	3.5×10^{-7} to 3.6×10^{-6}	1,790	6.26	0.02 to 0.24	4,261d/
B	9×10^{-9} to 3.4×10^{-7}	1,790	7.02	0.03 to 0.17	27,600e/
B	760	1,790	7.73	0.12 to 0.30	73.58
B	760	1,790	7.62	0.12 to 0.24	175.47
B	760	1,790	8.36	0.12 to 1.45	89.67
B	760	10,000	5.0		66.8
B	760	10,000	5.0		19.9
B	760	10,000	5.0		8.33
B	2×10^{-8} to 2×10^{-7}	3,600	9.14	0.062 to 3.2	16,000e/
B	1.3×10^{-8} to 1.8×10^{-7}	8,000	7.73	0.2 to 0.3	213f/
B	760	1,790	7.76	0.065 to 0.60	50
	2×10^{-8} to 3.5×10^{-7}	1,790	7.76	0.029 to 0.15	13,300
B	760	1,790	7.40	0.01 to 0.12	50
	7×10^{-9} to 2.5×10^{-7}	1,790	7.40	0.016 to 0.10	12,500e/
C	6×10^{-9} to 5×10^{-7}	1,790	5.85	0.1 to 1.5	529.5
C	1.4×10^{-8} to 2×10^{-6}	1,790	5.08	0.24 to > 3	327.5
C	2.5×10^{-8} to 2.5×10^{-7}	1,790	5.43	0.3 to > 3	277.8
C	760	1,790	6.91	0.37 to > 3	27.25
C	760	1,790	7.16	0.24 to 1.5	30.75
C	760	10,000	5.0		0.02
C	760	1,790	7.02	0.016 to 0.029	20
	2×10^{-7} to 5×10^{-6}	1,790	7.02	0.325 to > 3	8.13
C	760	1,790	6.97	0.43 to 1.45	20
	3×10^{-6}	1,790	6.97	0.9	0.5
D	9×10^{-8} to 4×10^{-7}	1,790	6.21	0.3 to > 3	24.5
D	8×10^{-8} to 5×10^{-7}	1,790	6.47	0.3 to 1.5	110.8
D	1.2×10^{-8} to 4×10^{-6}	1,790	5.52	0.5 to > 3	8.25
D	1.7×10^{-7} to 1.5×10^{-6}	1,790	5.59	1.5 to > 3	45.8
D	760	1,790	7.61	0.10 to 0.48	16.83g/
D	760	1,790	7.57	0.06 to 0.48	22.0
D	760	1,790	6.41	> 3	0.67
D	760	1,790	5.95	> 3	0.13
D	760	10,000	5.0		6.4
D	760	10,000	5.0		10.3
D	760	10,000	5.0		73.7
E	1.9×10^{-7} to 1.8×10^{-6}	1,790	6.08	> 3	1.5
E	3×10^{-8} to 3×10^{-6}	1,790	6.46	> 3	0.58
E	8×10^{-6}	1,790	5.43	> 3	0.2
F	1.7×10^{-8} to 5×10^{-6}	1,790	5.40	> 3	6.7
F	2.4×10^{-8} to 1.8×10^{-6}	1,790	7.25	> 3	4.0
F	3×10^{-8} to 2×10^{-6}	1,790	6.00	> 3	0.05

a/ This bearing would not operate faster than 450 rpm.
b/ This bearing is still operating.
c/ This bearing did not fail. The test was terminated for bearing analysis.
d/ The vacuum pump failed.
e/ These tests were stopped without bearing failure.
f/ This bearing did not fail. The drive motor operation was erratic.
g/ This thrust bearing was installed upside down.

TABLE IV

PARTIAL SUMMARY OF TEST RESULTS

Bearing	Average Wear-Life in Air	Average Wear-Life in Vacuum
A	-	0.78[a]
B	112.91	> 45,672 hr[b]
C	29.00	378.27
D	9.91	47.34
E	-	0.76
F	-	3.58

Operating speed: 1,790 rpm

a/ Weighted average since the second bearing failed to reach full operating speed.

b/ No failures have occurred--one bearing still operating; others were stopped.

The experiment was stopped (no bearing failure) for examination of the bearing. There were no signs of wear on either the inner or outer race; not even a small wear track could be found. The outer-land-riding separator has been worn slightly where the separator contacted the outer race. Some wear in the separator ball pockets was also noted. The wear debris was in the form of fine bronze powder, with some of the small flakes having been burnished or rolled onto the bearing surface until they became so thin that they appeared to be transparent under microscopic examination. These thin, rolled-on powder flakes looked like small drops of oil that had crept onto the surface. When these small "drops" were scraped off the balls and races, they were found to be small, thin bronze particles.

This particular bearing was further examined to determine if any preservative oil was present on the bearing surface. The entire bearing was boiled in chloroform (after more than 1,600 hr of exposure to a vacuum of at least 5 x 10^{-8} torr). After boiling for 2 hr, the bearing was removed and the chloroform evaporated. A small amount (less than 0.01 ml) of oily residue remained. This residue was transferred to a cessium iodide plate, and an infrared spectrum was recorded. The absorption bands in the spectrum were weak, but the spectrum was similar to an infrared spectrum of the oil used in packaging the bearing, MIL-L-6085A, a diester-based fluid.

The next B bearing was operated for 4,261 hr before termination when the vacuum pump failed. The bearing showed minor signs of wear but had not failed.

The 4,261 hr bearing and the 27,600 hr bearing were both tested to verify the results that were being produced by the first bearing studied. The 27,600 hr bearing was stopped simply because there was other work to be done using these bearing rigs. Performance of the first bearing had been shown to be reproducible, at least to 1,604 hr, by three other specimens.

The performance life of the B bearings operating at 1,790 rpm in air averaged 112.91 hr for the three specimens. The failure in each of the experiments was a jamming of the bearing caused by ball separator wear. The separator ball pockets had worn into an elliptical shape, with the major axis increasing in length until the separator was worn into two pieces. The lower portion would attempt to fall clear of the bearing but, due to space limitations within the chamber, would generally jam either the bearing balls or the driven magnet. In one instance, the separator only partially separated, cocked, and stopped bearing rotation. Although the average life in air is relatively short, the value of at least 70 hr of bearing performance in air, without external lubrication, does give some degree of useful life. For a potential space application of this type of bearing, a duplex test sequence was initiated. The objective of this test sequence was to show that limited time air atmosphere operation of a bearing could be successfully followed by prolonged vacuum operation, as might be encountered with an air-environment check out of equipment to be used later in space vehicles. Two of the B bearings were each operated in air for 50 hr. At the end of the 50 hr, each of the test chambers was evacuated to a pressure of less than 4×10^{-7} torr. Bearing rotation was resumed until, after periods of 12,500 and 13,300 hr, the tests were stopped to allow other work to progress. These two bearings had not failed. Greater wear was observed in the separator ball pockets than had been observed in the 27,600 hr vacuum bearing, but the wear was not to the same stage as the three air-operation bearings in which the separator had worn into two pieces.

Operation of the B bearings at speeds greater than 1,800 rpm was limited by the number of continuous tests that could be conducted. The 3,600 rpm bearing test was stopped for other work after 16,000 hr of operation in vacuum. Minor wear on the separator was noticed. The 8,000 rpm bearing experiment in vacuum was terminated due to drive motor problems. The 10,000 rpm experiments on the Pope spindles resulted in lives of 8.33, 19.9, and 66.8 hr for the unlubricated bearings operating in air. These results show excessive scatter and many more tests under identical conditions would have to be performed to determine the order of magnitude of the operating life. The results obtained do not warrant an extensive program in this area.

C. C Bearings

The operation of three C bearings at 1,790 rpm in a vacuum environment resulted in lives of 277.8, 327.5, and 529.5 hr, with the average being 378.27 hr. Failure in each case was due to excessive wear debris buildup in the separator, causing the bearing to jam. The riveted, ribbon type separator has more ball contact area and does not allow the debris to fall away from the bearing as easily as with the thin cylinder of the B bearing. The silver plating on the separator appears to have helped the performance (see Table III, bearings E and F), but the confining ball-pocket design does not appear to help.

The performance of the C bearings in air at 1,790 rpm produced results of 27.25 and 30.75 hr for the two specimens used, for an average of 29.00 hr. This performance life is approximately one order of magnitude less than that in vacuum.

Based upon the usable life in air, the duplex test sequence was used for two of the C bearings. The initial operation in air was 20 hr, followed by chamber evacuation and continuation of rotation. Apparently the air operation was more severe than anticipated, because the vacuum operation was only 0.5 and 8.13 hr for the two specimens.

The single attempt at 10,000 rpm operation in air produced a failure just as the spindle shaft reached operating speed. The Pope spindle automatic overload protection shutdown the operation. Inspection of the bearing revealed excessive wear and sloughing of the silver plating, with the debris jamming the bearing. Removal of the debris by an air stream revealed that some of the debris had been rolled onto the surface, making bearing operation rough. No further attempts at higher speed operation were made.

D. D Bearings

Statistical evaluation of the wear-life results (see Appendix B for the statistical analysis techniques) for the D bearings would reveal that any operating time less than 164 hr would be just as acceptable* as the 47.34 hr average used here for comparison. There is no explanation for the

* The word "acceptable," as used here, means that, statistically, the range specified has an 88.9% probability of containing the actual average wear-life for the type of bearing being discussed. The specified range of $\pm$ 3 σ (three sigma), where sigma is the standard deviation, would include any type of unlimited distribution of values (Tchebycheff type) and not be restricted to normal, bell-shaped distribution. (See Ref. 7, pp. 65-66.)

order of magnitude spread in the wear-lives for the four specimens. The values (8.25, 24.5, 45.8, and 110.8 hr) show no tendency to be in any particular range of wear-life.

The same extreme spread of operating time was evidenced in the 1,790 rpm operation in air. The values of 0.13, 0.67, 16.83, and 22.0 hr have an arithmetic average of 9.91 hr and, statistically, any value under 39 hr would be acceptable (see footnote, p. 16).

The high speed operation (10,000 rpm) produced better results than the 1,790 rpm operation in air. The somewhat lower load (5 lb as compared to 7 lb) would not have made the difference. (The 3,800 hr life load rating for the size 204 bearing is 550 lb.) Apparently the greater inertia of the faster-rotating spindle shaft and drive motor of the Pope spindle enabled the bearing to "run over" the wear debris that was not thrown away from the contract zone by the greater centrifugal forces. Any wear debris that was left in the ball races or ball-separator contact zone apparently took longer to accumulate and jam the bearing. The operating times were 6.4, 10.3, and 73.7 hr, for an average of 30.13 hr, with any value less than 122.7 hr being statistically acceptable (see Appendix B for the method of calculation) (see footnote, p. 16).

The D bearing ball separator is machined from M-2 tool steel. The U-channel shape has the "fingers" of the U extending outward so that a cavity is formed between the outer race and the inner portion of the separator. This cavity would provide a convenient reservoir for lubricant; but for this unlubricated work, the cavity seems to provide a place for the accumulation of wear debris.

E. E and F Bearings

The E and F bearings are very similar in both construction and performance. Both are ABEC-1 precision bearings with stamped, ribbon-type ball separators. The vacuum operation at 1,790 rpm produced results of 0.2, 0.58, and 1.5 hr for the E bearings and 0.05, 4.0, and 6.7 hr for the F bearings. Statistical interpretation of these times (see Appendix B for method of calculation) would allow any value of performance life below 2.4 hr for the E bearings and 11.8 hr for the F bearings to be acceptable (see footnote, p. 16).

Based upon the results of the vacuum operation of these two types of bearings, no air operation or higher speed operation was attempted.

VI.

OBSERVATIONS AND DISCUSSION

Several interesting observations can be made of the work being reported. These observations are not in themselves "conclusions," but they are worthy of further discussion.

All of this work has been performed on bearings that have no added lubrication in the form of dry powders, oils, or greases. None of the bearings were intended to be operated without adequate, added lubrication, but yet the performances, both life and friction, have been quite varied. The separators used were not intended to provide sacrificial material for establishing a transfer film, yet two types of bearings (B and C) have demonstrated performance that must be due to sacrificial wear.

The self-lubricating properties of the leaded tin bronze as a bearing separator were not well-publicized when this work was originally undertaken. With the normal 20-20 hindsight, this selection was proved to be sagacious. Performance of this separator in a bearing has been remarkable, with both air and vacuum environments. The demonstrated long life in vacuum (over 12,500 hr without failure) after a 50-hr air operation would allow adequate air-operation checkout of space equipment and still provide the capacity for over 1-1/2 years of space utilization. The light loads used in this work would also be appropriate for space applications.

As might be expected, the value of silver-plating was demonstrated as one method of prolonging bearing operating life when operating in a vacuum environment. The silver-plated, ball-riding separators of the C bearings provided an average wear-life of 378.27 hr as compared to the plain, ball-riding separators of the E and F bearings, with their composite average wear-life of 2.17 hr. As might also be expected, the frictional torque was less with the silver plated-separator bearings (0.1 to > 3 oz-in) as compared to the plain separator bearings (> 3 oz-in). The weighted average of frictional torque per unit of operating time would produce a torque comparison of 0.38 as compared to > 3 oz-in.

Bearing wear-lives appear to be a complicated function of bearing precision, separator design, and separator material. There is not enough comparable information to determine just what the relationship is, but there does seem to be some evidence that the ball/separator contact area influences bearing performance. As the contact area increases, the bearing operating times decrease. The types of bearings that have the ball-riding separators, with greater contact area, also have higher frictional torque than do the bearings with the edge contact between balls and separators.

The variation in bearing vacuum performance as compared to air atmosphere operation in some respects is unusual but in other respects is as might be expected. The metal-to-metal contact protection supposedly supplied by the metal oxide should provide for less likelihood of cold-welding failure in the air atmosphere than in vacuum. With the vacuum levels used in this work (less than 10^{-6} torr), it takes over 1 sec to provide a new monolayer of oxide coating, during which time the bearings can rotate 30 revolutions. The continued existence of an oxide coating in the vacuum environment is thus unlikely. But yet, cold-welding in vacuum was not found.

VII.

CONCLUSIONS

As a result of the work presented in prior sections,the following conclusions are appropriate.

1. The service life demonstrated by the B bearings has been remarkable. This is the first known work that has produced over 45,600 hr of rotation at 1,790 rpm in a vacuum environment with no lubrication applied to the bearing.

2. The B bearings (and the C bearings, to a limited extent) will tolerate a brief checkout operation in air that can be followed by further, prolonged operation in a vacuum environment.

3. Short-time (up to 175 hr) air operation without lubrication is possible with some of the bearings studied.

APPENDIX A

MATHEMATICAL DEVELOPMENT OF FRICTIONAL TORQUE AS A FUNCTION OF COAST-DOWN TIME

Just as in linear motion,

$$F = Ma$$

where F = force

M = mass

a = linear acceleration

we have, for radial motion,

$$T = I\alpha$$

where T = torque

I = moment of inertia

α = radial acceleration

The expression for torque can be written in terms of the time rate of change of radial velocity, as:

$$T = I\alpha = I \frac{d\omega}{dt}$$

where ω = radial velocity

t = time

The rotating torque for the rotating bearing and magnet inside the metal test chamber consists of two components--frictional torque and eddy current generation. The frictional torque comes from the composite of the metal-on-metal frictional junctions inherent in the ball bearing. The eddy

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See Ref. 3 for other basic relationships and Ref. 4 for another approach to the same problem.

current generation (electrical power) comes from the rotation of the magnet inside the metal container. While friction is basically not a function of rotational speed (only rubbing forces), eddy current is proportional to the radial velocity. That is:

$$T = K + p\omega$$

where K = constant related to frictional torque

p = eddy current power

What do we know about K ? Does K have a value that can be determined? If the coast-down determination is made in a nonmetallic container, such as glass, then there is no eddy current generation and we find, for a retarding frictional torque:

$$T = -K = I\frac{d\omega}{dt}$$

$$dt = -\frac{I}{K}\,d\omega$$

$$\int_{o}^{t_o} dt = -\frac{I}{K}\int_{\omega_1}^{o} d\omega$$

$$t_o = \frac{I}{K}\,\omega 1$$

$$K = \frac{I\omega_1}{t_o}$$

That is, K has the value of the product of the original radial velocity of the shaft times the system moment of inertia divided by the time required for the rotation to stop when the magnet is in a nonmetallic container.

The actual application involves a metallic chamber with eddy currents. Since the frictional torque of a specific bearing should be the same whether the container is glass or metal, the value of p (eddy current power constant) can be determined for the system. Once this value has been determined, any value of frictional torque can be related to the coast-down time.

To determine the value of p , we therefore need to determine the coast-down times of one specific bearing in both a glass container and the actual test chamber.

The value of p would come from the following:

$$T = - K - p\omega = I \frac{d\omega}{dt}$$

$$dt = \frac{-I}{K + p\omega} d\omega$$

$$\int_0^{t_1} dt = \frac{-I}{p} \int_{\omega_1}^{o} \frac{p d\omega}{K + p\omega}$$

$$t \Big|_0^{t_1} = \frac{-I}{p} \ln (K + p\omega) \Big|_{\omega_1}^{o}$$

$$t_1 = \frac{-I}{p} \ln \left(\frac{K}{K + p\omega_1} \right) = \frac{I}{p} \ln \left(\frac{K + p\omega_1}{K} \right)$$

where t_1 = coast-down time in metal

ω_1 = radial velocity from which coast-down starts

All that remains is to evaluate the constants of the problem.

The "system moment of inertia" requires a numerical value. However, some preliminary analysis is required since all of the components do not rotate at the same radial velocity. The magnet, rotating shaft, bearing inner race, and fastening devices all rotate at the stated radial velocity. The balls and separator move at a reduced radial velocity, which can be determined from the geometry of the bearing component parts. The specific bearing being used has an inner race radius of 0.508 in., with a ball diameter of 0.3125 in. Thus, the inner race instantaneous velocity at the point of contact of the ball is:

$$V_R = r_R \, \omega_R$$

$$= 0.508 \, \omega_R$$

The instantaneous velocity of the center of the ball would be one-half of the instantaneous velocity of the extreme moving point, since the outer extreme point is stationary. That is:

$$V_{B_c} = \frac{V_R}{2} = \frac{0.508}{2}\,\omega_R$$

The radial velocity of the ball would be

$$V_{B_c} = r_{B_c}\,\omega_{B_c}$$

where r_{B_c} = radius of the ball center

$$= \left(0.508 + \frac{0.3125}{2}\right)\text{in.}$$

$$= 0.664 \text{ in.}$$

Hence we find that the radial velocity of the balls and separator is, in terms of the shaft radial velocity:

$$V_{B_c} = 0.664\,\omega_{B_c} = \frac{0.508}{2}\,\omega_R$$

$$\omega_{B_c} = 0.3825\,\omega_R$$

The system moment of inertia has two major components. The moment of inertia of the balls rotating about their own axes was neglected because it is small. The two major components are the moment of inertia of the shaft and all attached parts and the moment of inertia of the balls and separator, as:

$$T = I_S \frac{\omega_S}{t} + I_B \frac{\omega_S}{t} = I \frac{\omega}{t}$$

To find an equivalent I for the system

$$I\omega = I_S\omega_S + I_B\,\omega_B = I_S\,\omega_S + I_B\,(0.3825\,\omega_S)$$

The determination of the system moment of inertia resulted from adding the moments from all the components. The result was:

Component	Moment of Inertia (g cm^2)
Inner race	47.19
Magnet	1,128.36
Rotor	112.39
Cap screw	3.34
Large washer	19.34
Small washer	2.38
Separator	32.10 x 0.3824
Balls	64.28 x 0.3824
	1,349.86

which, for computational purposes, was taken as 1,350 g cm^2.

The other physical constants of the problem that remained to be determined were the coast-down times from the various initial speeds to be used for a given bearing operating in a glass environment and in the metal environment of the test chamber. A glass environment was selected instead of no cover at all so that the windage effects would be as nearly similar as possible.

Determination of the coast-down times produced the results:

Initial Speed (rpm)	Coast-Down Times (sec)	
	Glass Enclosure (t_o)	Metal Enclosure (t_1)
1,800	94.8	15.6
3,600	130.2	24.0
8,000	132.6	30.6

The equation

$$t_1 = \frac{I}{p} \ln\left(\frac{k + p\ \omega_1}{k}\right)$$

does not lend itself to an easy analytical solution for p . A different approach was used, involving selecting a value for p , substituting into the equation with the known t_o $\left(K = \frac{I\ \omega_1}{t_o}\right)$, and determining a value for t_1 . By graphing the selected values of p versus the derived values of t_1 , the point of intersection of the graphed curve and the actual t_1 would evaluate p . These results were:

Initial Speed (rpm)	Values of $p\left(\frac{g\ cm^2}{sec}\right)$
1,800	254.5
3,600	156.2
8,000	108.2

Using these values for p, the equation was evaluated for frictional torque for any coast-down time while in the metal test chamber. These frictional torque values (converted to ounce-inch units) were then plotted as functions of the corresponding values of coast-down time to give the three curves shown in Figures 4, 5, and 6. These graphs were used for the frictional torque values given in Table III.

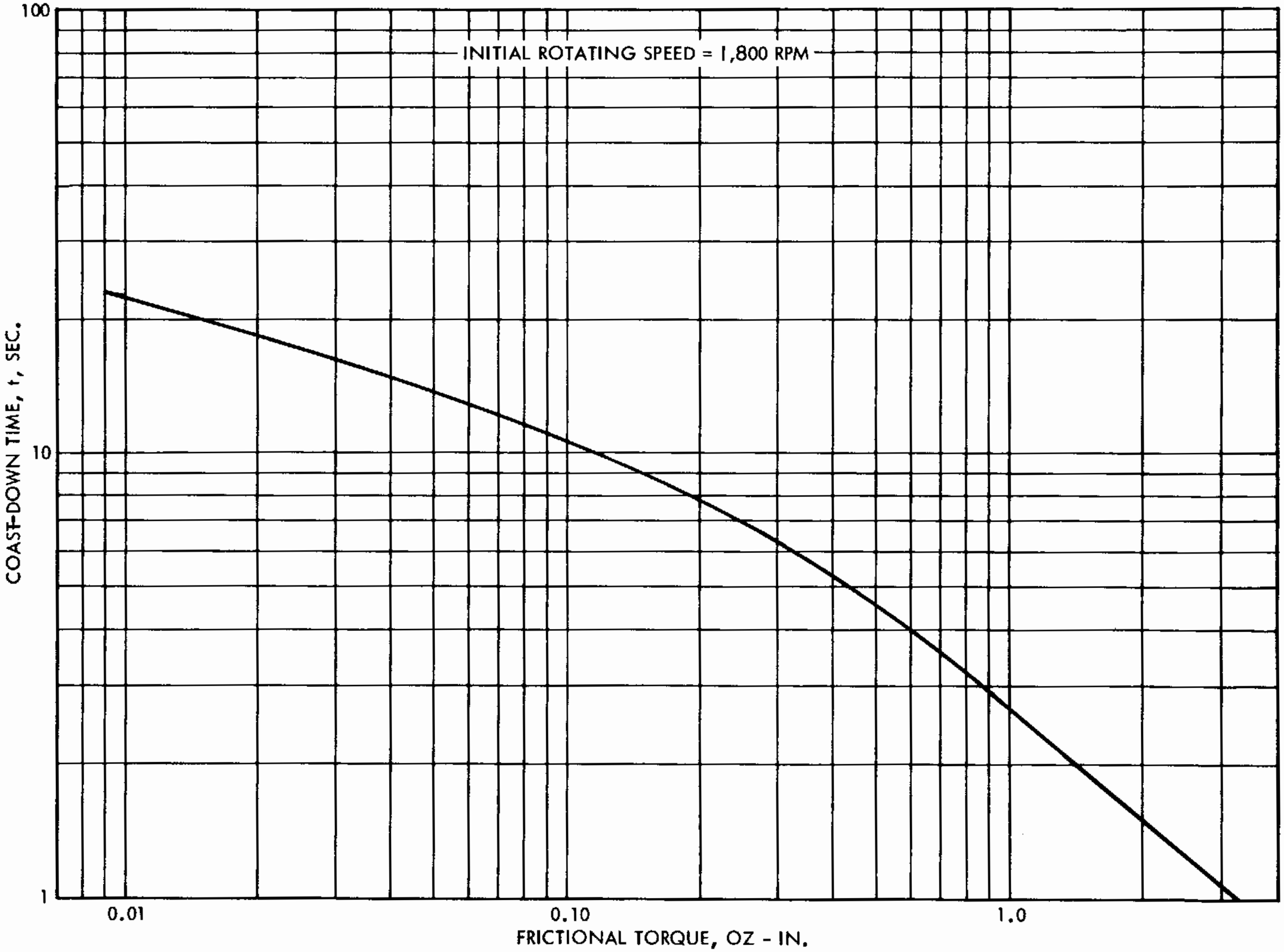

Figure 4 - Coast-Down Time Versus Frictional Torque, 1,800 rpm

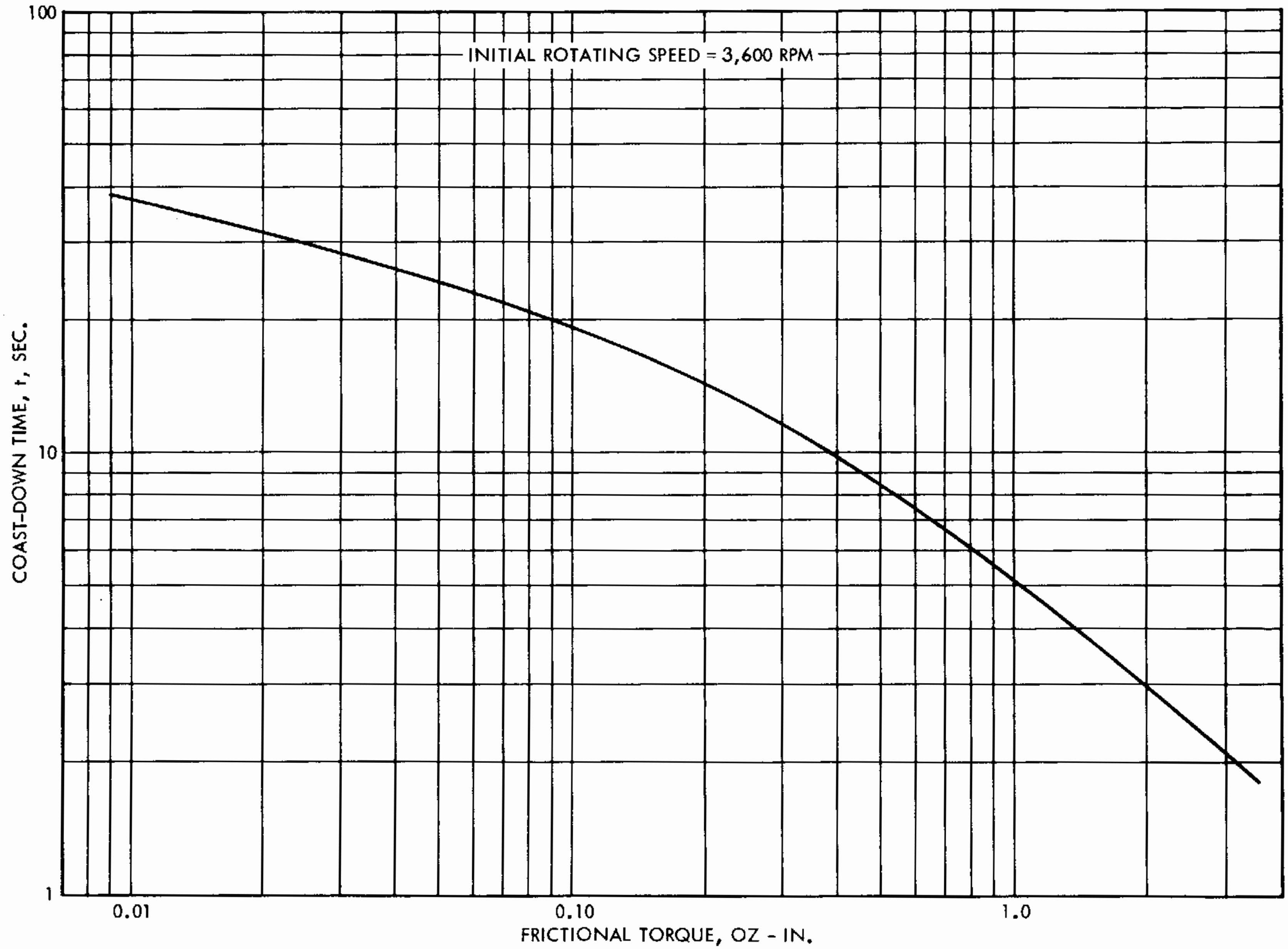

Figure 5 - Coast-Down Time Versus Frictional Torque, 3,600 rpm

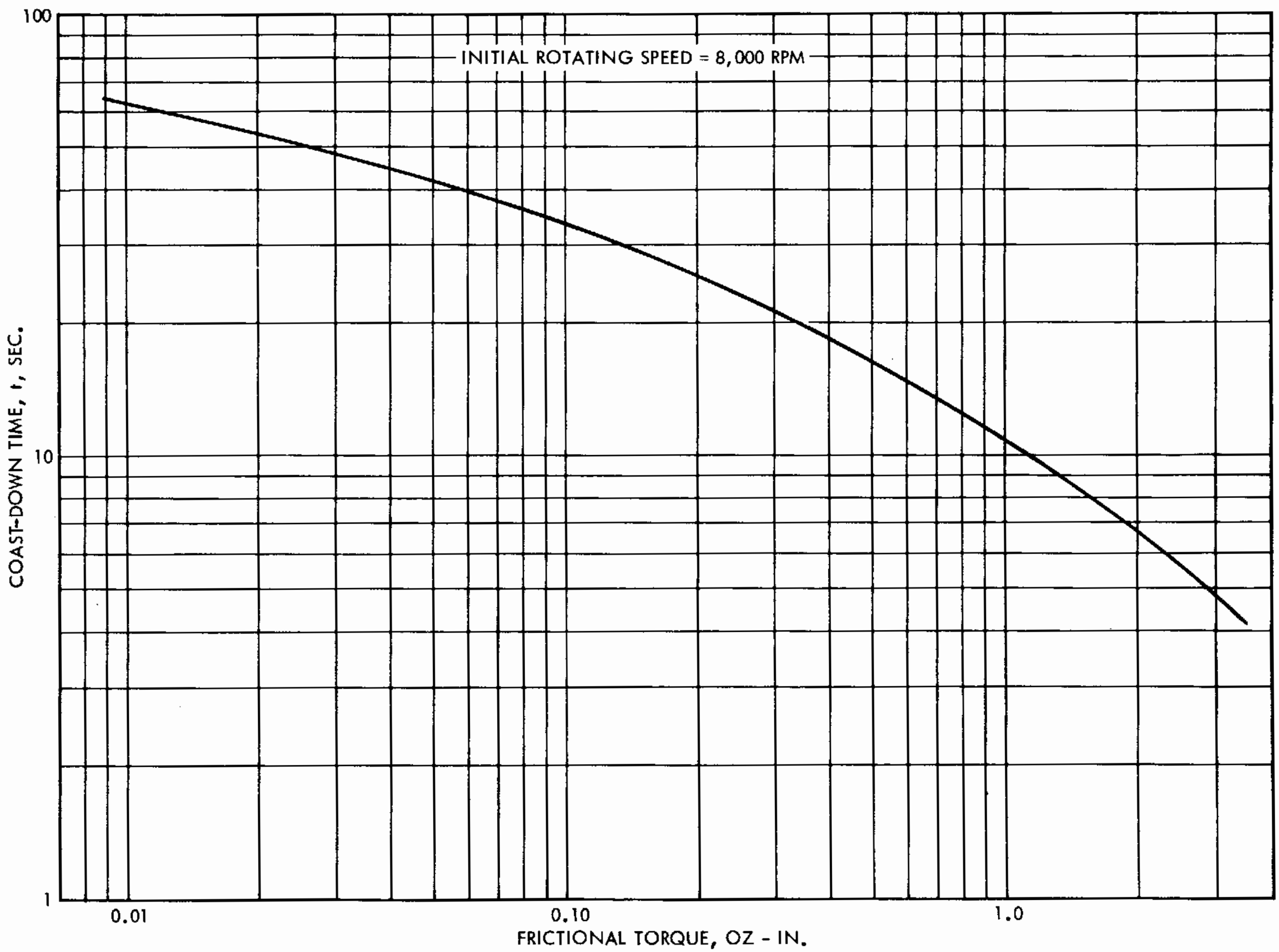

Figure 6 - Coast-Down Time Versus Frictional Torque, 8,000 rpm

APPENDIX B

STATISTICAL ANALYSIS TECHNIQUES FOR BEARING WEAR-LIFE RESULTS

To illustrate how statistical techniques are used to determine the range of wear-lives as presented in the text (see Section IV-D), the following example for the D bearing calculation is presented.

The values determined by experiment were 8.25, 24.5, 45.8, and 110.8 hr. The arithmetic average was first calculated and found to be (8.25 + 24.5 + 45.8 + 110.8) hr/4 = 47.34 hr. Next the individual value differences were calculated, these differences squared, and the squared differences averaged, as shown below:

Value	Value - Average	$(\text{Value - Average})^2$
8.25	-39.09	1,528.03
24.5	-22.84	521.67
45.8	- 1.54	2.37
110.8	63.46	4,027.17
	Sum =	6,079.24

$$\text{Average} = \frac{6{,}079.24}{4} = 1{,}519.81$$

This average is the sample variance; the square root of the sample variance is the standard deviation, generally referred to as sigma.

$$\text{Sigma} = \sqrt{1{,}519.81} = 38.98$$

For normal (bell-shaped) distribution, a range of three sigma, or three times the standard deviation, should include 99.73% of all the possible values of the statistically acceptable values of the average wear-life (see Ref. 7). However, the wear-life results determined for this work are not known to be of normal distribution. The Tchebycheff or general type of distribution must therefore be assumed, in which a range of $\pm 3\sigma$ (plus or minus three sigma) would have an 88.9% probability of containing the actual average wear-life. Thus, the range used in the text became:

See Refs. 5, 6, and 7 for further explanation of the statistical techniques.

$$\text{Range} = 47.34 \pm 3(38.98) = 164.28,\ -69.60$$

The negative value has no physical meaning and any value of operating time (or wear-life) less than 164 hr would be just as acceptable as the 47.34 hr arithmetic average.

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UNCLASSIFIED

Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY *(Corporate author)* Midwest Research Institute 425 Volker Boulevard Kansas City, Missouri 64110	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP N/A

3. REPORT TITLE

PERFORMANCE OF BALL BEARINGS IN AIR AND VACUUM WITH NO ADDED LUBRICATION

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

Technical Report (1 January 1969 to 8 December 1971)

5. AUTHOR(S) *(First name, middle initial, last name)*

Karl R. Mecklenburg

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
March 1972	37	7

8a. CONTRACT OR GRANT NO. F33615-69-C-1236	9a. ORIGINATOR'S REPORT NUMBER(S)
b. PROJECT NO. 7340 and 7343	
c.	9b. OTHER REPORT NO(S) *(Any other numbers that may be assigned this report)* AFML-TR-72-73
d.	

10. DISTRIBUTION STATEMENT

Approved for public release; distribution unlimited

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	Air Force Materials Laboratory Air Force Systems Command Wright-Patterson AFB, Ohio

13. ABSTRACT

Forty-one experiments on six different designs of size 204 ball bearings were conducted. The results of these experiments, operating time and frictional torque values determined by coast-down times, are presented for various test atmospheres, loads, and speeds. The experimental approach is outlined and the different test apparatus are described. The distinctive design features of the production-type ball bearings are identified. Even though no lubrication was applied to any of these bearings, several of the specimens ran satisfactorily for over 10,000 hr at speeds of 1,790 and 3,600 rpm with light axial loads of 7 lb nominal. One specimen continues to operate in a vacuum environment after 45,600 hr at 1,790 rpm.

DD FORM 1 NOV 65 1473

UNCLASSIFIED

Security Classification

UNCLASSIFIED

Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Tribology						
Ball Bearings						
Dry Friction and Wear						
Long-Term Tests						
Self-Lubricating Retainers						
Performance in Vacuum						

UNCLASSIFIED

Security Classification