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DESIGN AND DEVELOPMENT OF A RADIAL LOAD BALL BEARING TEST SYSTEM AND SOME PRELIMINARY RESULTS

B. W. WARD, JR.
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MAY 1969

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SUMMARY

Man's venture into outer space has brought much attention upon the technology required to achieve reliable operation of machines in the space environment. Shaft support by ball bearings has been of particular concern because it is not always feasible to hermetically seal all mechanisms. Thus in the late 1950's and early 1960's a great amount of vacuum bearing testing was undertaken around the country. While many researchers performed a variety of tests, it was difficult to compare results. This fact was readily apparent in early 1964 when the development of the LOAD-LIFE (or SPEED-LIFE) test system was begun.

The difficulties in comparing results lay in the variety of test loads, speeds, and failure criteria indiscriminantly chosen for bearings by the various industry and Government laboratories. Thus the basic objective in developing the radial load tester described herein was to make it possible to gather statistical quantities of comparative data on the useful lifetimes of instrument size bearings under carefully controlled load, speed, and vacuum conditions. From this objective the name "LOAD-LIFE" tester has been coined. The information obtained is to be presented in the form of design curves. In order to accomplish a valid comparison, three eccentric weights were designed (2, 4-1/4, and 9 ounces) with a common eccentric moment of 1/4 oz-in. To obtain statistical results from one test, a fixture supporting five test weights mounted parallel to each other on a single shaft was designed. High speed shaft stability was obtained by supporting each end of the shaft by two bearings a finite distance apart. Procedures were developed to adjust out axial freeplay and yet keep axial loads at a very low level.

This report summarizes the above work and in addition reports the initial results from five complete tests. The essential features of the LOAD-LIFE tester may be seen in Figure 1.

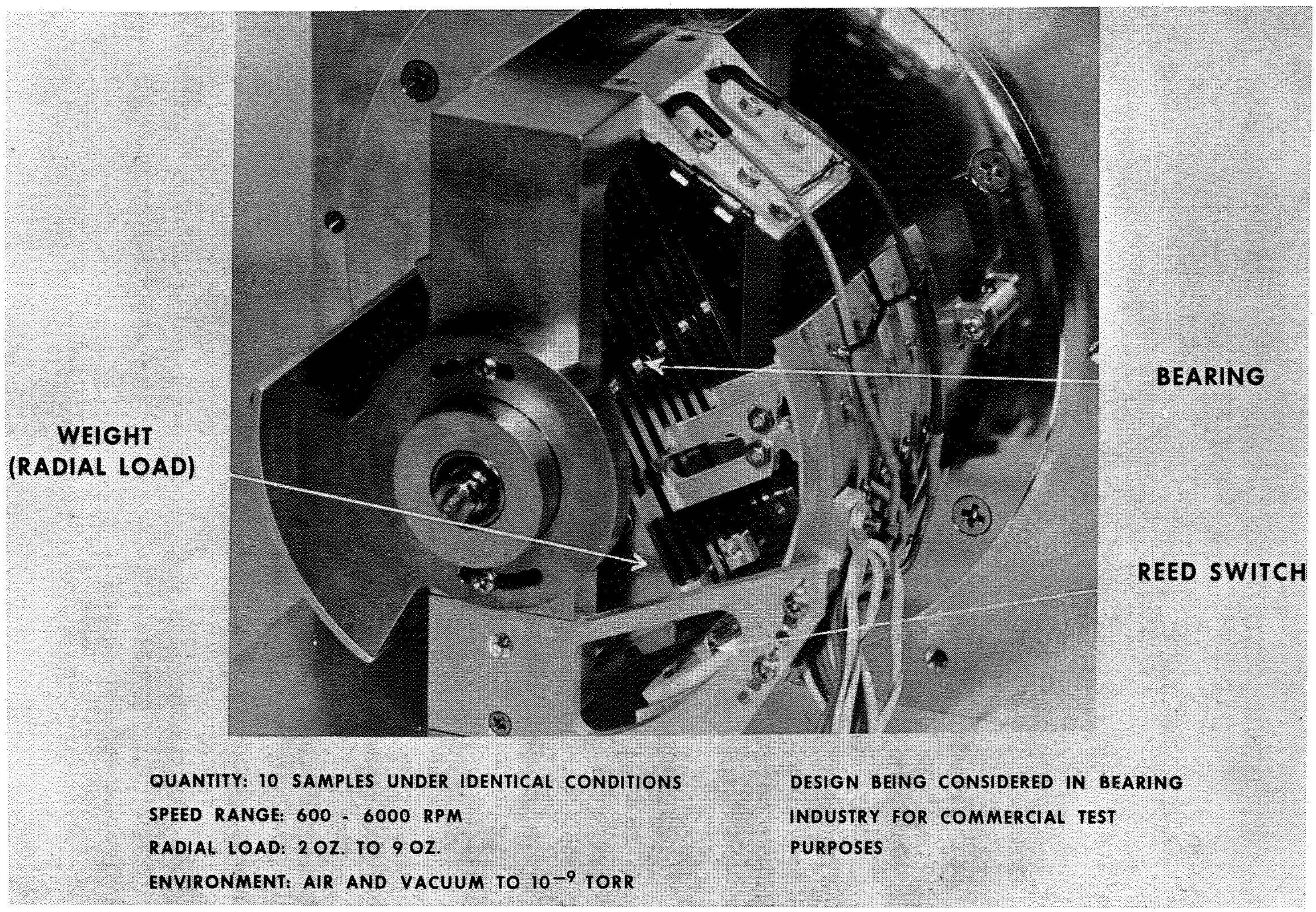

Figure 1–Radial load ball bearing test fixture

ACKNOWLEDGEMENTS

The authors should like to take this opportunity to acknowledge their indebtedness to fellow Mechanical Systems Branch personnel who aided them in the conduct of this project. We thank Mr. Charles E. Vest for his technical guidance, for handling the purchase and autopsy of the ball bearings, and for making arrangements to have the LOAD-LIFE test program continued on outside contract. Our appreciations go to Mr. Robert H. Peterson for his role in the design of the vacuum system and the test cabinet. Also, we express our gratitude to Mr. Richard F. Housel for his aid in designing the three test weights.

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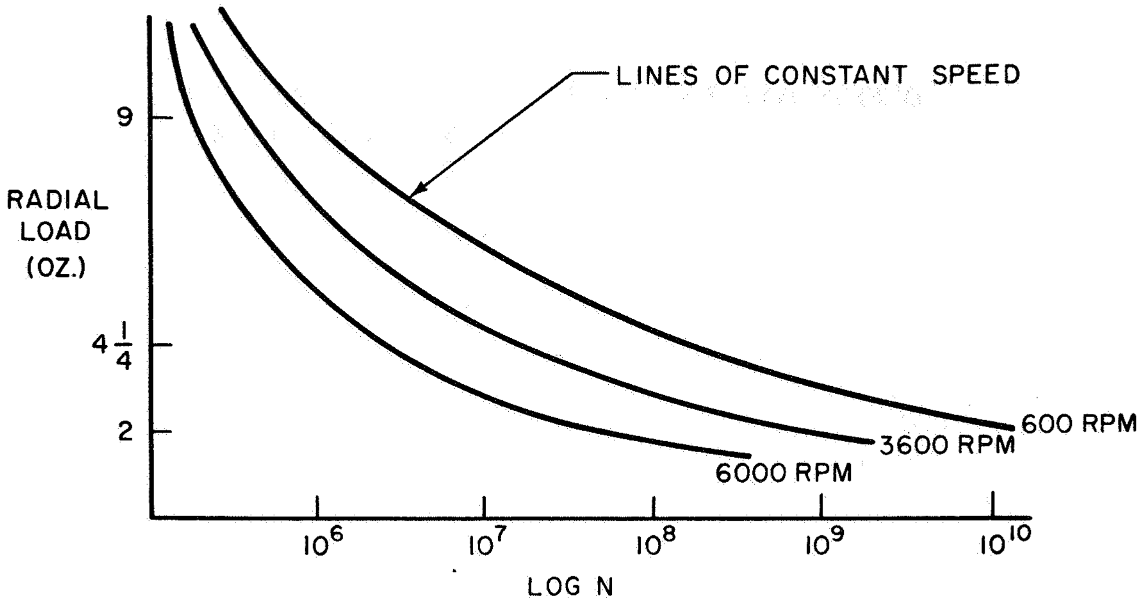
TEST OBJECTIVES AND DESIGN PHILOSOPHY

As stated in the Summary the ultimate objective of the LOAD-LIFE Bearing Test Program is to obtain design curves from tests of statistical quantities of identical bearings at three different conditions of radial load at each of three different angular velocities. This data can then be presented graphically in two families of curves which are seen in Figure 2. The first group of curves are plots of radial load versus the log of the average number of revolutions required to reach a predetermined running torque. Each individual curve represents a constant speed. The second group are plots of angular velocity versus the same abscissa. Individual curves represent constant load. (Note: Both families of curves are obtained by plotting the same nine data points.)

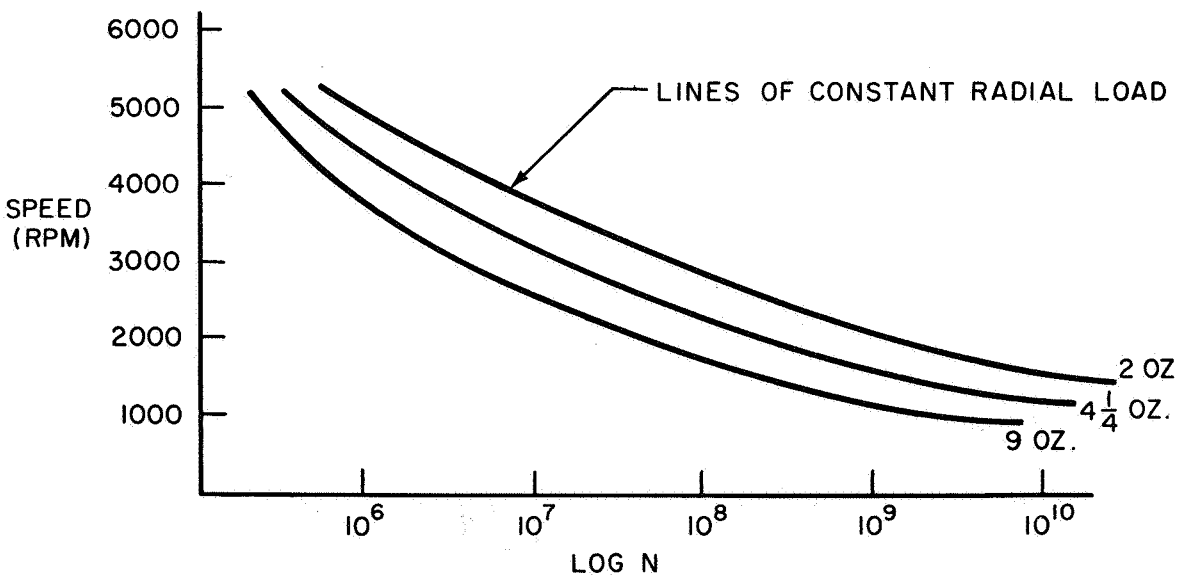
It will be noted that no specific loads or speeds have been mentioned at this point – but emphasis has once again been placed upon the use of statistical methods. The choices of specific test parameters were related to the requirement for statistical quantities of data because of test equipment limitations. For example, it would be possible to perform a very high speed test (e.g. 15,000 rpm) of a bearing pair in a single weight test fixture but difficult to accomplish tests at the same speed with a multiple weight fixture. A minimum sample of 10 events occurring under identical conditions is generally necessary to utilize statistical techniques.

Tests of bearings, like fatigue tests, can be machine sensitive–that is produce results which vary somewhat in response to the smoothness, vibration level, etc., of the individual test device. Thus in terms of the necessary statistical quantities, minimization of test fixture setups, and comparability of the data, it was decided to design a test fixture which would test five bearing pairs simultaneously on one shaft. While this design averages the condition of each bearing pair and yields only five data points instead of the desired 10, it was the optimum choice.

Another aspect of definitizing the tester design was the selection of a specific bearing size and type. Then test loads, test speeds, and a "failure" criterion which would be made common to all the individual tests were selected. Lastly, a decision was made to run duplicate tests in vacuum and normal atmospheric environments. Each of these areas is discussed as follows:



(a) RADIAL LOAD VS. LOG N



(b) SPEED (ANGULAR VELOCITY) VS. LOG N

Figure 2–Families of curves which can be constructed from the nine data points (3 different loads at 3 different speeds) obtained from a complete LOAD-LIFE test. N represents the total number of revolutions required to reach a predetermined running torque.

1. SELECTION OF BEARING TYPE

The fundamental criterion used in selecting a given bearing type for this extensive series of controlled tests was applicability to current space mechanism designs. It was found through a design survey of electric motors, small gearheads, optical scanners, and other space instruments that the SR2-6 size bearing is frequently used. This has a 0.1250" bore and 0.3750" outside diameter and is .1094" wide. Bearings of this width are available with one-piece fully machined retainers and with double shielding if required.

For space application the stainless steel type 440C vacuum melt has been found to give the best results and is most commonly used. This is designated by the prefix "S" in the number. Absence of this prefix indicates that chrome steel type SAE 52100 or its equivalent is standard. Standard tolerances for this type of instrument bearing are Class ABEC-7 or better.

It was not the objective of this program to design a ball bearing tester which would be limited to gathering design data for a particular lubricant scheme for use on SR2-6 size bearings. However, the most reasonable approach was to size the test loads and failure criteria upon a lubricant scheme showing good promise for space application. All initial tests would then be run to evaluate this lubricant scheme and later series of tests would be run to compare the results of alternative lubrication methods with that originally tested. Thus to a certain degree the load, speed, and failure parameters were sized for SR2-6 bearings utilizing the lubrication scheme discussed next.

Previous testing reported by Evans (1) et al, had shown that the one-piece fully machined retainer concept can be used to advantage in the space environment. By machining this retainer out of a self-lubricating material, it can serve not only to separate the balls but also supply a replenishing lubricant film. Good results had been obtained with a composite of polytetrafluoroethylene (PTFE) and MoS_2 powders, reinforced by glass fibers. There are several trademarked names given this composite by bearing manufacturers but they all have a similar composition. The particular material chosen was a composite of approximately 59 v/o (Volume percent) PTFE, 39 v/o fiberglass, and 2 v/o MoS_2 (hereafter called PTFE-MoS_2 retainer). In addition to using this lubricant material in the initial series of test bearings it was also decided to use it in the larger sized shaft support bearings.

Let us now turn our attention to the difficult problem of selecting a reasonable range of test loads and speeds.

2. SELECTION OF TEST LOADS AND SPEEDS

The objective of this program was to test statistical samples of identical bearings at three different radial loads at each of three different angular velocities. To a large degree the success of the entire program depended upon the selection of a realistic range of loads and speeds. In a life test of this type test conditions must be chosen so that extremes in cycles to failure are avoided. Harse test conditions lead to running times too short to be significant whereas light or easy conditions may lead to runs of several years' duration. Neither of these extremes is suitable for the conduct of a test intended to gather design data.

At this point it is necessary to stress the difference between life testing bearings used in conventional earth applications and those utilized in the space environment. In the former the lubrication provided is very adequate so that high running torques and/or excessive vibration are not encountered until one or more localized fatigue failures occur on the raceway or rolling element surfaces. In other words the life obtained is dependent upon the number of load carrying cycles the contact surfaces can withstand without spalling (cracking or chipping).

In contrast the life testing of bearings for space flight use is in essence a test of the wear life of the lubricant film itself. The static loads in orbit are very light due to the essentially zero gravity condition. Since the stresses are well below the endurance limit of the contact materials, no fatigue failures occur on the contact surfaces. Thus running torque and vibration increase as the contact surfaces become roughened by wear or by the buildup of lubricant wear debris. Direct stick-slip action and the accompanying deterioration of the surfaces does not occur of course until the lubricant film is worn away or no longer replenished at a sufficient rate. This fact accounts for the rather sudden "failures" of bearings lubricated with dry film lubricants.

With the above background information, it can be seen that the selection of test radial loads had to be based largely upon previous test experience. In addition, several practical design limitations governed the final selections.

"Applied loads for bearings with PTFE-MoS_2 retainers should be kept low in order to minimize wear rates in the ball pockets and at the piloting surface. At the same time, the wear rate must be sufficient to provide minimum lubricant or film transfer from the self-lubricating retainer materials to the bearing raceways and balls. In miniature bearings the maximum applied loads should not exceed eight ounces in a radial direction and a range of one to three pounds thrust. In the case of the SR2-6 test bearings, maximum recommended loading is eight ounce radial and/or one pound axial." As a comparison, a static load rating for a SR2-6 high

speed, class ABEC-7 bearing with a fully machined retainer utilizing oil lubrication is 26 pounds.

Prior to the choice of the LOAD-LIFE test parameters some advance testing was done with existing 4-1/4 ounce eccentric test weights. This weight's design provided for support by two SR2-6 bearings. (It may be viewed in Figure 8 in the next part of this paper.) For test purposes it could be adjusted to several different eccentric moments. Thus, tests were run at several different eccentric moments across a speed range from 1800 to 8000 rpm. This was an effort to determine if the 4-1/4 ounce load in conjunction with a realistic "failure" torque criterion might be a suitable test condition. These advance tests and their results are discussed in more detail in Section 3 entitled SELECTION OF "FAILURE" CRITERION.

The essential result was that meaningful torque buildups occurred within approximately 2000 hours at 1800 rpm. This appeared to be a reasonable test interval so efforts were then directed to geometrical and design limitations. These constraints included available chamber size, permissible shaft deflection, and the necessity to maintain identical eccentric moments for the three different test weights. A careful appraisal of these factors led to the final choice of 2, 4-1/4, and 9 ounce test weights.

The final choices for test speeds were 600, 3600, and 6000 rpm. The speed was limited to a maximum of 6000 rpm because of the difficulties encountered in maintaining stability at 8000 rpm in the advance tests. It was observed that vibration levels were notably reduced at a little above 6000 rpm and thus this value was picked as a limit. With a "failure" torque setting of about 1/8 ounce-inch and a test speed of 3600 rpm, a median time to failure of about 1000 hours was expected. Without the benefit of specific advance tests, guesstimates of 300 to 500 hours and 5000 to 7000 hours were forecast for the 6000 rpm and 600 rpm test speeds, respectively.

In concluding this section, it can be said that specific choices for test radial loads and for test speeds were made with the objective of gathering meaningful data. Note that the 9 ounce maximum radial load (4-1/2 ounces per bearing) only applied 56% of the recommended radial load for composite retainer bearings of this type and thus was a conservative test value. At the time these choices were made, it was felt that the 2 ounce - 600 rpm test condition might create tests of a longer duration than desired, but a data point was needed at this condition. With load and speed values chosen, the final adjustment for producing tests of a reasonable duration lay in the choice of the "failure" torque criterion. This is discussed in detail in the next section.

3. SELECTION OF "FAILURE" CRITERION

The principal basis for selecting the specific value of eccentric moment to design into the test weights was the maximum allowable losses that a typical small electric motor utilizing SR2-6 size bearings can sustain. A survey revealed that 3 watt motors are used quite extensively to drive spacecraft mechanisms through gear reducers. Most motor designers provide for up to 10% power losses in the bearings.

A few calculations suffice to show the range of maximum permissible bearing tunning torques applicable to such motors. Typical motor speeds of interest are in the 3000 - 3600 rpm range. The calculations are as follows:

$$\text{Permissible power loss} = 0.10 \times 3 \text{ watts}$$

$$\doteq 0.3 \text{ watt}$$

$$P = 0.000742\ TN$$

where

P = Power in watts

T = Torque in oz-in.

N = Speed in rpm

At 3000 rpm:

$$0.3 = 0.000742\ T\ (3000)$$

or

$$T = 0.1123 \text{ oz-in.}$$

$$= 80{,}862 \text{ mg-mm}$$

At 3600 rpm:

$$0.3 = 0.000742\ T\ (3600)$$

or

$$T = 0.1348 \text{ oz-in.}$$

$$= 97{,}035 \text{ mg-mm}$$

Thus it was seen that a "failure" torque level of 80,000 to 100,000 mg-mm brackets the range of interest for small motor applications. This value is for a bearing pair such as that by which each test weight is supported. These calculations served as the primary basis for making the "failure" torque selection. However, as discussed in Section 2 above, some advance testing was done to arrive at the final design value.

A few words regarding the means of experimentally reading out the data from an eccentric weight test is now in order. A test weight is designed so that its center of mass is offset from the position of the test bearings by which it is supported. As with any simple pendulum, the rest position of the weight occurs when the center of mass hangs directly below the pivot point. At this position of static equilibrium, there is no gravitational torque acting upon the weight. This condition may be viewed in Figure 3a. However, as the weight is rotated as shown in Figure 3b, an eccentric moment or static torque occurs as the center of mass moves off the vertical centerline. This torque is expressed mathematically as follows:

$$T_s = W(d)\ (\sin\theta)$$

$$T_s = \text{static torque in oz-in.}$$

where

$$W = \text{total weight in oz.}$$

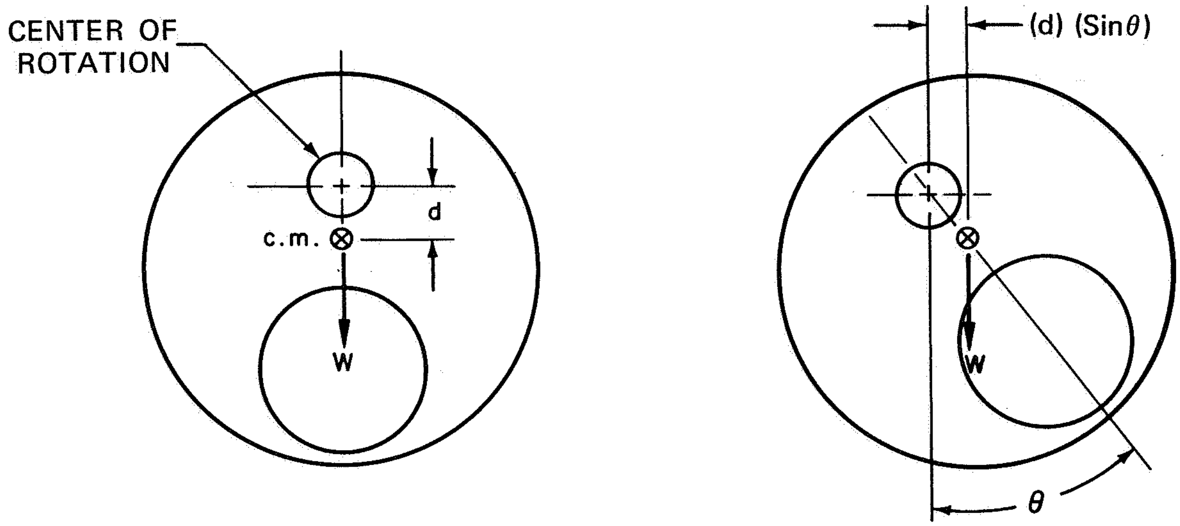
$$d = \text{distance to c.m. in inches}$$

$$\theta = \text{rotation angle}$$

It is trivial to point out that the maximum moment occurs when the weight is suspended out horizontally (or $\theta = 90°$).

Ideally an analog readout system capable of giving the instantaneous position of the test weight from 0 degrees to 90 degrees would be desirable. Such a system would not only have to be vaccum sealed but would have to operate without applying any forces on a test weight itself. In addition any system would have to be compact enough to be accessible to all five disks. Because a system to meet these requirements was not available on an off-the-shelf basis, it was decided to utilize a digital system which would give a signal at two discrete positions.

The two most logical positions for placement of sensors were determined to be 90 and 30 degrees. Here the attainment of the maximum eccentric moment and one-half its value occur, respectively. Continuous recording of sensor condition would yield a time history of the attainment of these two torque values.



(a) Rest position: Static torque equals zero

(b) CCW rotation: Static torque equals W (d) (Sinθ)

Figure 3–Principle of eccentric test weight design. Note that the maximum static torque opposing the dynamic torque of the bearing pair occurs at θ = 90 degrees.

With this two-point readout system it was clear that the design point for the eccentric weights (2, 4-1/4, and 9 ounces) would have to be the 30 degree rather than the 90 degree position. This choice was based on the following analysis:

(a) Any test weight attaining the 90 degree position because of an increase in running torque of the bearings is in a dynamically unstable condition. This is because the driving torque is generally unsteady and there is a decreasing opposing torque beyond the 90 degree point. Also, since the opposing torque is a function of sin θ, it is very nearly maximum at the 80 degree position. Thus, the possibility of an extremely rapid excursion past the 90 degree sensor and flip-over is imminent once a test weight attains a steady position much above 70 degrees. This is undesirable both because the sensor may fail to record this single rapid event and also because the remaining four test weights may be expected to be subjected to high vibration levels once this flipped-over weight is being driven against a stop.

(b) Basing the design failure torque upon the 30 degree position, however, eliminates these objections. An increasing opposing torque tends to keep the driving and opposing torques balanced—resulting in a stable running condition of some finite duration. Also, because this torque level is only one-half the maximum or flip-over torque level, it might be hoped that all five test weights would obtain the 30 degree level before

any one weight flipped over. (Note in PRELIMINARY TEST RESULTS that this expectation did not materialize in some of the tests run.)

For the reasons stated above, the eccentric moment of the test weights at 30 degrees was made the design point. With this decision firm, additional advance testing offered a means for "fine tuning" the previously discussed range of realistic "failure" torques.

Advance Testing: As mentioned in Section 2 some advance testing was conducted with an existing 4-1/4 ounce test weight at speeds ranging from 1800 to 8000 rpm. The main objective of these tests was to finalize the "failure" torque criterion for the design of the LOAD-LIFE bearing test system. However, this problem was not simple and clear-cut but actually involved an adjustment of axial spacer width as well.

As a start two five-weight vacuum tests were run at 1800 rpm with their 30 degree eccentric moments set at 100,000 mg-mm and 200,000 mg-mm, respectively. To make certain that axial loading was not a factor, no axial spacer was used at first. By the end of 2084 hours, two test weights had occurrences of 100,000 mg-mm indications, (first occurrence at approximately 1725 hours) but none had 200,000 mg-mm indications. The 200,000 mg-mm level was beyond the "realistic" range desired but was included in case earlier than anticipated indications occurred at the 100,000 mg-mm level. Because such indications did not occur, the 200,000 mg-mm level was changed to 75,000 mg-mm without changing the test bearings. Then both the 100,000 and 75,000 mg-mm fixtures were restarted. Within a few hours of startup, all five 75,000 mg-mm weights and three out of five 100,000 mg-mm weights had given "failure" indications. A continuation of these two tests for 200 hours did not alter this data.

Before commenting on the meaning of the above results, it would be useful to shift our attention to the other side of the "failure" torque selection problem. That was the choice of axial spacer thickness for axial freeplay control. The objective was to keep axial loading at or near zero, but it was also recognized that freeplay would have to be eliminated to attain a stable running condition. This included the possibility of needing a very light axial load (< 1 oz).

It was expected that proper freeplay adjustment would present the greatest difficulty in getting the highest speed test to run stably with a low value of "failure" torque. Thus a five-weight air test was conducted at 8000 rpm with the 30 degree failure level set at 50,000 mg-mm. The one-way axial freeplay or "stickout" was measured for each of the bearings and inner race spacers honed to be equal to the sum of the stickouts of the two bearings paired together in a test weight. Assembly of these test weights resulted in a too tight condition and an immediate flipover of the test weights when the test was started.

Individual 8000 rpm trial runs were then made with spacers .0007", .0055", .0045", and .0003" less than the sum of the stick-outs of each particular bearing pair, respectively. The most favorable results were obtained when the sum of stick-outs was reduced by .0003 to .0004". Here random 60 degree indications (86,700 mg-mm) were obtained upon startup. However, even with this adjustment, stable conditions did not occur until the speed was reduced to the 6000 rpm level. A literature search revealed that additional high speed stability can be obtained with the spacer located between the outer races rather than the inner races.

Therefore, outer race spacing which is referred to as "back to back" mounting was chosen for the LOAD-LIFE test weights. The shim thickness formula previously found most acceptable was retained for this configuration which is illustrated in Figure 4. A final advance test was conducted at 3600 rpm utilizing this spacing technique but the results were inconclusive because of fixture problems.

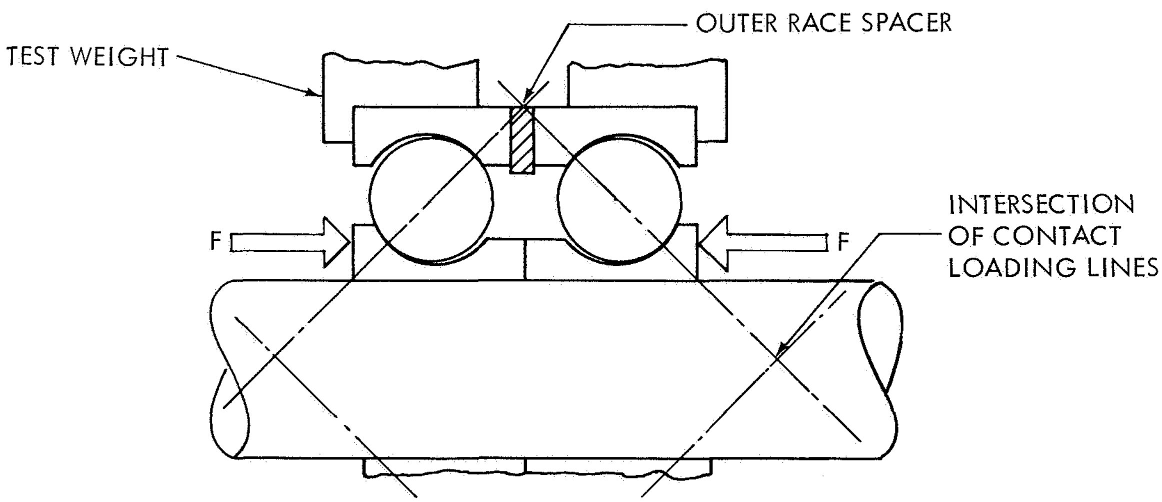


Figure 4—Outer race spacing or "back to back" mounting of test bearing pair chosen for LOAD-LIFE tester. The force F which is applied by the stack-up of inner race spacers brings the inner races into contact. Outer race spacing provides a comparatively wide spacing between the intersections of the force line of contact which gives greater stability at high speeds.

In making the final choice of "failure" torque for the LOAD-LIFE tests, the following points were considered:

(i) a range of 80,000 to 100,000 mg-mm bracketed the permissible torque losses of a typical electric motor used in space applications.

(ii) Only two out of five 100,000 mg-mm and none of five 200,000 mg-mm indications had been obtained by 2084 hours in vacuum at 1800 rpm.

(iii) Upon change of the 200,000 mg-mm setting to 75,000 mg-mm and restart of the test, all five weights gave "failure" indications within a few hours.

(iv) A 6000 rpm test required a 30 degree "failure" level setting of slightly over 86,700 mg-mm to eliminate indications at startup.

A careful analysis of the above four points plus several others too detailed to be discussed here led to the choice of 90,000 mg-mm as the 30 degree "failure" torque criterion for the 2, 4-1/4, and 9 ounce test weights. It was felt that this level would provide significant results at the three test speeds of 600, 3600, and 6000 rpm and yet not prove to be high enough to cause excessively long runs at the lowest speed.

One additional useful benefit derived from conducting the advance tests was the clear demonstration of the need for a more rigid mounting of the test shaft. The test shaft in these tests was simply supported by a single ball bearing on each end. This means of shaft support permits rotation of the ends of the shaft through the bearings which leads to significantly greater center deflections upon loading. The means of solving this problem is discussed in the next part of this paper.

4. DUPLICATE TESTS IN VACUUM AND AIR

In formulating the plans for the LOAD-LIFE radial load test system, the final major decision concerned the gathering of data in air as well as vacuum to obtain comparative results. Several researchers on the problem of obtaining long life of mechanical elements in a high vacuum environment had reported superior results in a vacuum with some dry lubrication schemes. For example, Federline (2) reported improved lifetimes in a vacuum environment for gears lubricated by a PTFE-MoS_2 idler gear. Also, as reported in Reference (1), long lifetimes without a significant torque buildup had been obtained on PTFE-MoS_2 lubricated bearings. These and other results established the need to run duplicate tests in vacuum and in air.

The questions to be answered by running identical tests under the two different environmental conditions include the following:

(i) are the lifetimes obtained sufficiently close that the added difficulty and expense of conducting vacuum tests are unnecessary?

(ii) are the lifetimes obtained related to the specific type of dry lubricant tested so that duplicate tests are necessary to obtain the complete picture for a given lubricant type?

(iii) Should some good vacuum lubricating schemes for bearings carry a prohibition against preliminary running in a normal atmosphere?

(iv) Is the moisture level of a normal atmosphere a major contributing factor in the rapid deterioration of some lubricant types run in air.

As demonstrated by the above questions the issue of vacuum versus air testing of dry lubricant films can become very complex.

The decision was thus made to conduct duplicate LOAD-LIFE tests in vacuum and in air. For the initial series of tests (bearings lubricated with the PTFE-MoS_2 retainer technique) it was hoped that an answer to at least the first question (i) above might be obtained. Subsequent series of tests with other lubricant schemes could serve to provide additional answers.

TEST SYSTEM DEVELOPMENT

In the preceeding part of this paper, the basic test objectives, the choice of test parameters, and the design philosophy behind the ball bearing tester have been discussed. In addition to the necessity to obtain five data points from a given test setup, the requirement to make nine different test setups for each lubricant type and surrounding environmental condition determined that a complete test system be developed. This system includes six radial load ball bearing testers contained within a single cabinet which houses all supporting vacuum equipment and data recording instrumentation. Three of the testers are run in a hard vacuum and the remaining three in a normal atmospheric environment. In the following four sections each of the major parts of the test system are discussed – beginning with the eccentric weight tester.

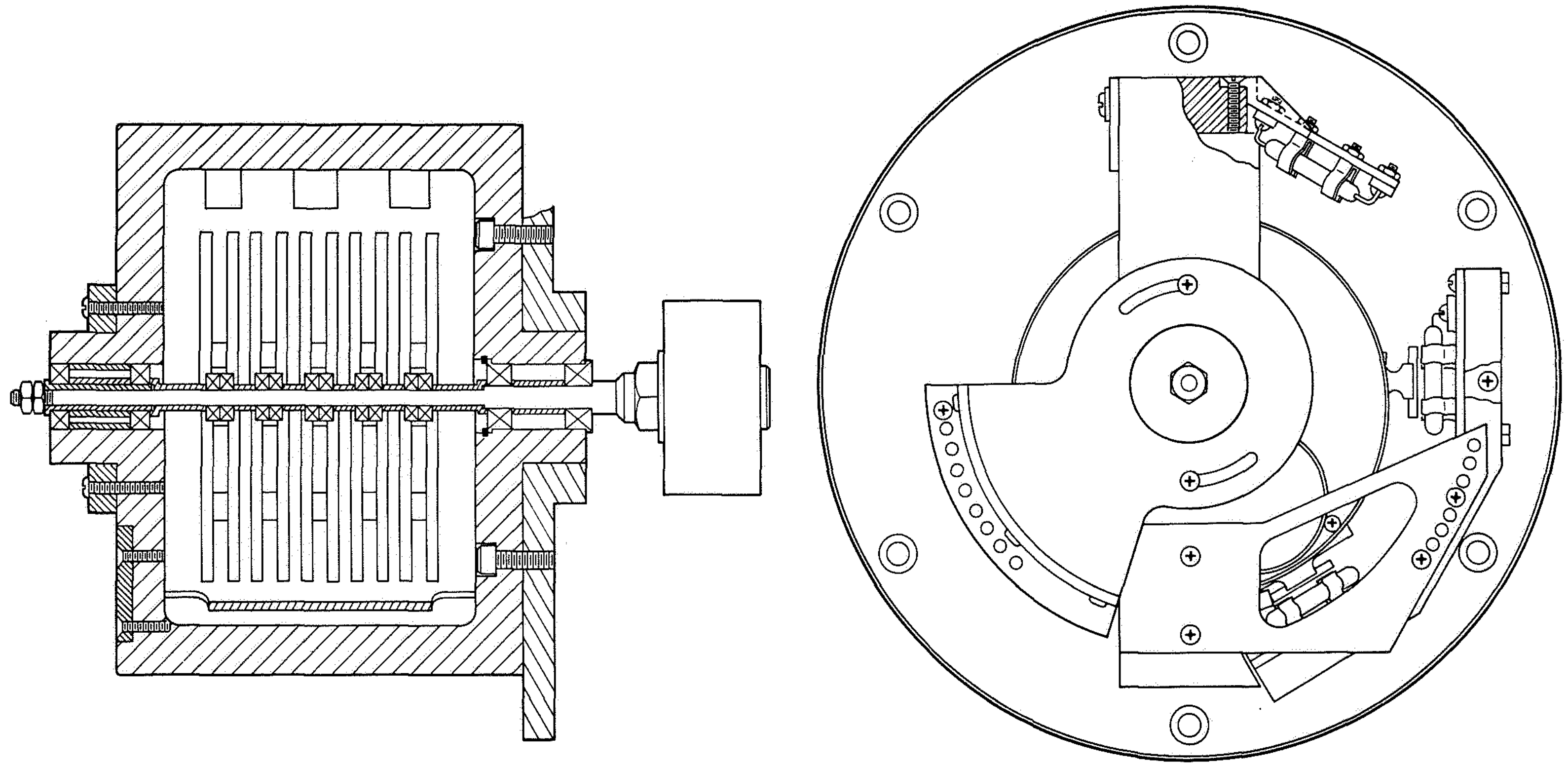
1. RADIAL LOAD BALL BEARING TESTER

The tester shown in Figure 1 was designed and developed to accomplish the above discussed objectives. In Figures 5a and 5b cross-section and end views of the tester are shown, respectively. The most important design requirements were as follows:

(i) Shaft stability at high speeds (minimization of shaft deflection)

(ii) Maintaining identical eccentric moments with three different test weights

(iii) Reliable indication of the rising of the test weights to preset angular positions



(a) Cross-section view. Note the stack-up design of the inner race spacers to remove all freeplay from the drive magnet to the opposite end upon the tightening of the end nut.

(b) End view. The *30 and 90 degree reed switches, including the* 60 degree displaced switches for the #2, #4 position test weights may be seen. The stop bar is adjustable from -10 to +30 degrees in 5 degree increments. The reed switch basket is adjustable in the CCW direction *in three 3-1/2 degree increments. Preliminary* tests were run with the stop bar at 30 degrees and the reed switch basket at 33-1/2 degrees.

Figure 5–Design features of the radial load ball bearing tester

(iv) Initial setting of test weight position.

(v) Provisions for mounting in vacuum and in air.

The accomplishment of each of these requirements is discussed briefly under the subsequent headings.

A. Test Shaft Design: Reference to Figure 5a reveals that a test shaft six inches long and one-eighth inch in diameter is supported by spaced tandem pairs of bearings on either end. About 5/8" center to center spacing between each pair of support bearings provides a long moment arm to counter a tendency of the shaft ends to rotate upon the application of loads. With this mounting scheme a calculation of static shaft deflection can be made with the assumption that both ends are fixed. Such a calculation shows that a downward deflection of approximately 0.0006" will occur at the center of the approximately 3 inch unsupported span when five 10 ounce concentrated loads are applied at the positions of the test weights as shown. Since this figure is of the order of magnitude of the optimum straightness of this shaft that can be maintained during manufacture, it was deemed satisfactory. It should be pointed out that a similar shaft supported by single bearings on its ends would experience about 0.003" deflection at the center with the same loading configuration. Thus the design incorporating the spaced tandem pairs of SR3 and SFR3 support bearings was instrumental in obtaining shaft stability.

Adjunct to proper mounting of the test shaft was the incorporation of a means to apply a compressive axial force to the inner races of all the bearings on the shaft, both test and support. This was accomplished through the design of a set of interconnecting spacers which run from the shaft end nut to the support bearings on the drive magnet side. By torquing the inner end nut to 28 oz-in. (a value recommended by the bearing vendor), all the inner races and inner race spacers are brought into metal to metal contact. As discussed under TEST OBJECTIVES AND DESIGN PHILOSOPHY axial freeplay in the five test bearing pairs is removed when the inner races are brought into contact by an appropriate axial spacer placed between the outer races. The freeplay is similarly removed from the two support bearing pairs through the use of differential length inner and outer race spacers – the latter being slightly wider. A locking nut is provided to secure the initial torque setting. Each of the major elements of the test shaft assembly may be viewed in an exposed condition in Figure 6.

A check of the shaft resonant frequencies revealed no difficulties for the desired speed range.

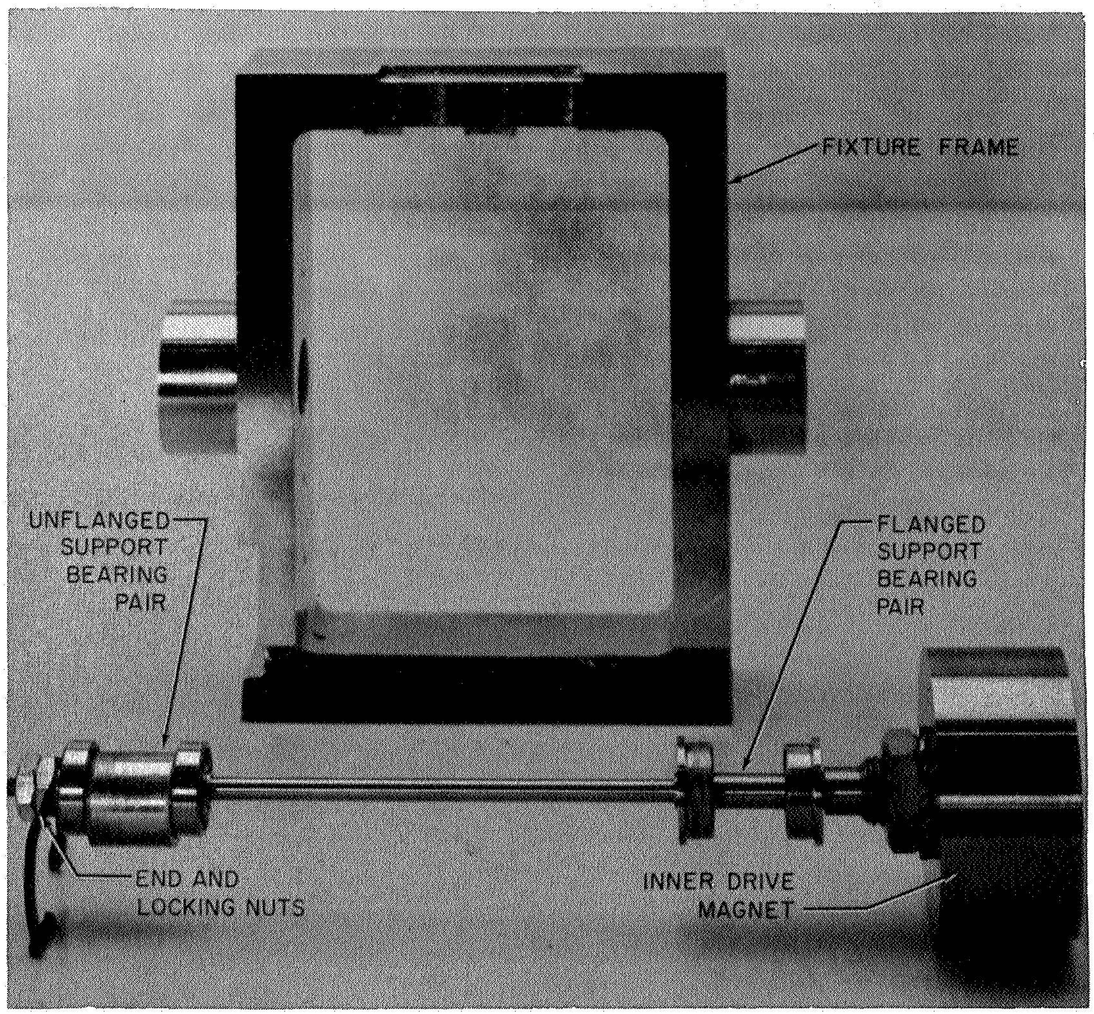


Figure 6-Test shaft support bearing pairs

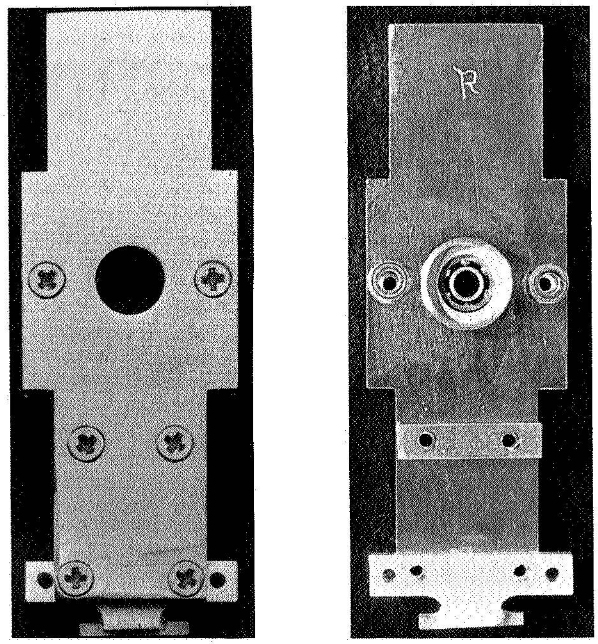
B. Test Weight Design: The original task was to design 2, 4-1/4, and 9 ounce test weights with a common eccentric moment of 180,000 mg-mm at 90 degrees to the vertical. These weights were to be symmetrical about their vertical centerlines. The geometrical constraints included limitations on the width and the radial distance to the magnetic reed switch sensors. To maintain good vacuum capabilities the material was generally limited to stainless steel. The grade used had a density of 4.57 oz/in^3. Some small

hardware was made from anodized aluminum, and Heavy-Metal (density = 9.786 oz/in^3) was used for the balance weight of the 9 ounce weight.

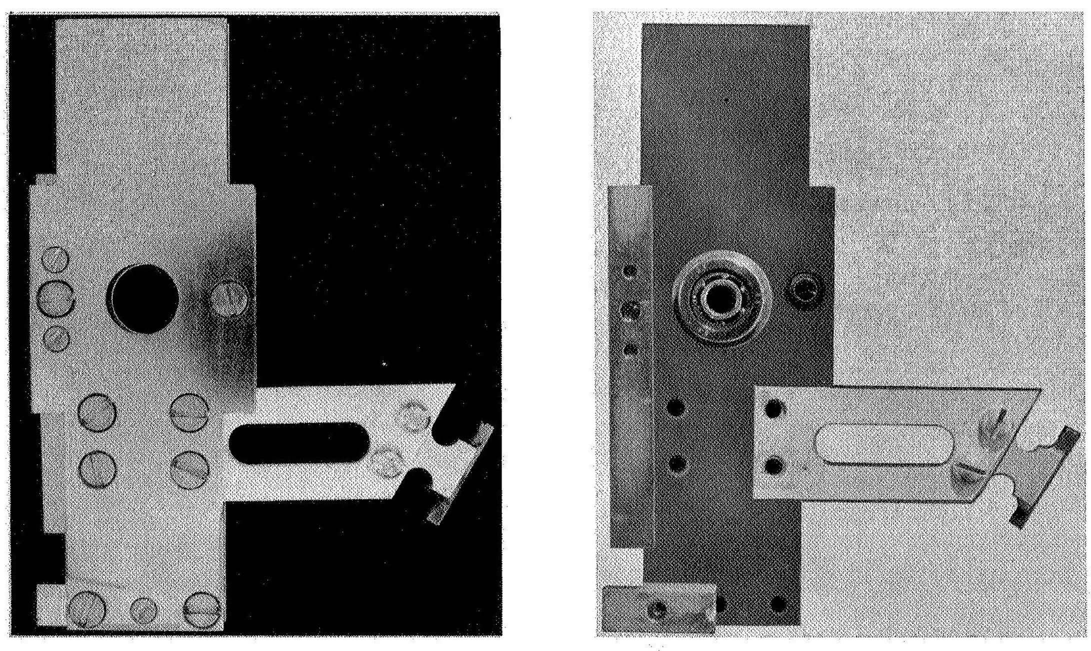
Upon the completion of an initial test fixture, however, it was discovered that adjacent sensor magnets attached to the bottom of each test weight developed repulsive-attractive forces which were equivalent to about 30,000 mg-mm of applied torque. Thus a design change was made to place the sensor magnets on the #2 and #4 test weights 60 degrees above the bottom center position. The corresponding 30 degree sensors were placed in the normal 90 degree position and the 90 degree sensors were placed at 150 degrees in a special holder. This change required the design of precision balance weights for the #2, #4 position versions of each of the 2, 4-1/4, and 9 ounce test weights in addition to the balance weights for the standard test weights.

Simple static balancing calculations involving first moments of masses were used to arrive at the finished designs — with certain key pieces slightly overweight for final trimming during laboratory calibration. The balancing of the symmetrical (#1, #3, #5 position) weights was far simpler in that only a one-dimensional balance was required. This placed the c.g. of the given weight the desired distance from the bearing center on the vertical centerline so that a static torque of 180,000 mg-mm occurs when the weight is rotated upward 90 degrees. The balancing of the unsymmetrical (#2, #4 position) weights was tedious because the off-center placement of the sensor magnet and holding pieces required the placement of a counter balancing mass on the opposite side to return the c.g. to the vertical centerline. Also, this and other balancing pieces had to be placed in the proper vertical position to arrive at the desired c.g. offset from the bearing center to obtain the desired 180,000 mg-mm eccentric moment. A final requirement was that the total weight of each modified test weight remain essentially the same as that of its symmetrical counterpart. These objectives were achieved with the three test weight designs shown in Figures 7, 8, and 9. Each weight's characteristics is summarized in the respective subtitle.

C. Reed-Switch Basket Design: As discussed in the preceeding part of this paper, it was determined that a digital data readout system giving signals at 30 and 90 degrees would be utilized. Based upon the factors of reliability and compactness, a miniature magnetic reed switch was chosen as the basic sensor element. This choice required the placement of an actuating magnet on each test weight. Also, a means of accurately mounting an array of 10 reed switches for a given test fixture had to be devised. Some flexibility for adjusting the exact switch position was also desirable. This task consisted of choosing the optimum reed switch, determining its operating characteristics, and designing a basket-like frame to mount the switches in.

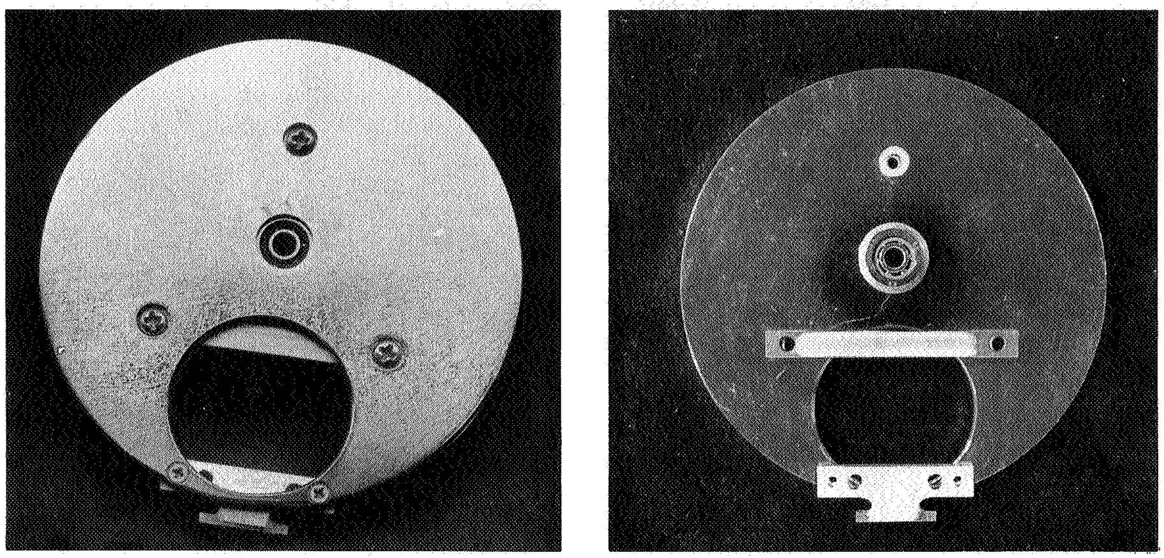


(a) Standard weight

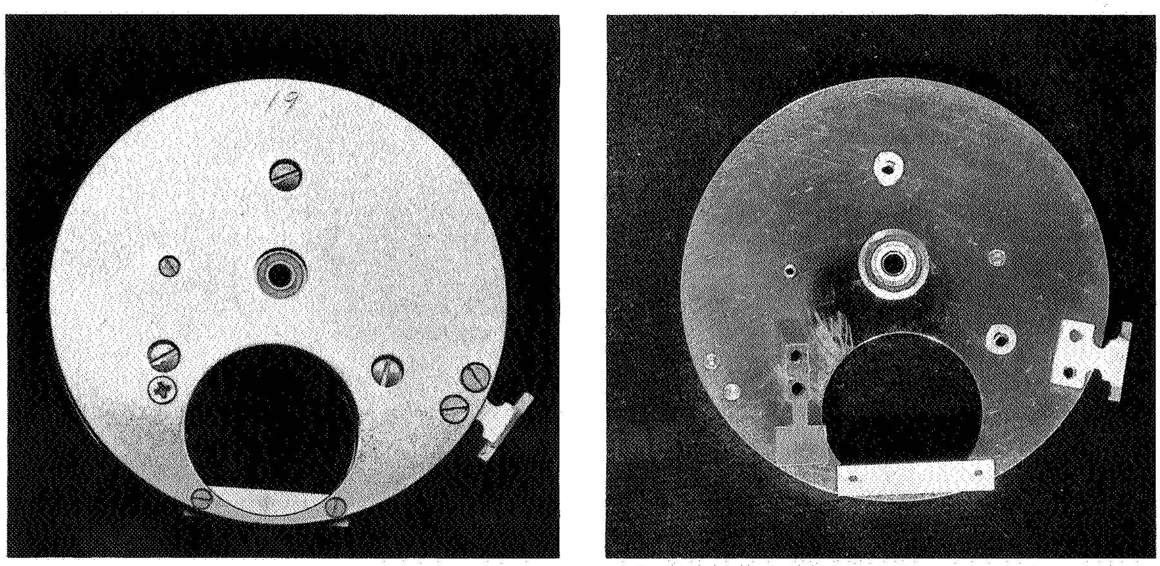


(b) #2 and #4 position weight

Figure 7—Two ounce test weight. The location of the balancing weights necessary to maintain a 180,000 mg-mm eccentric moment at 90 degrees may be seen in the pictures of the open weights at the right.

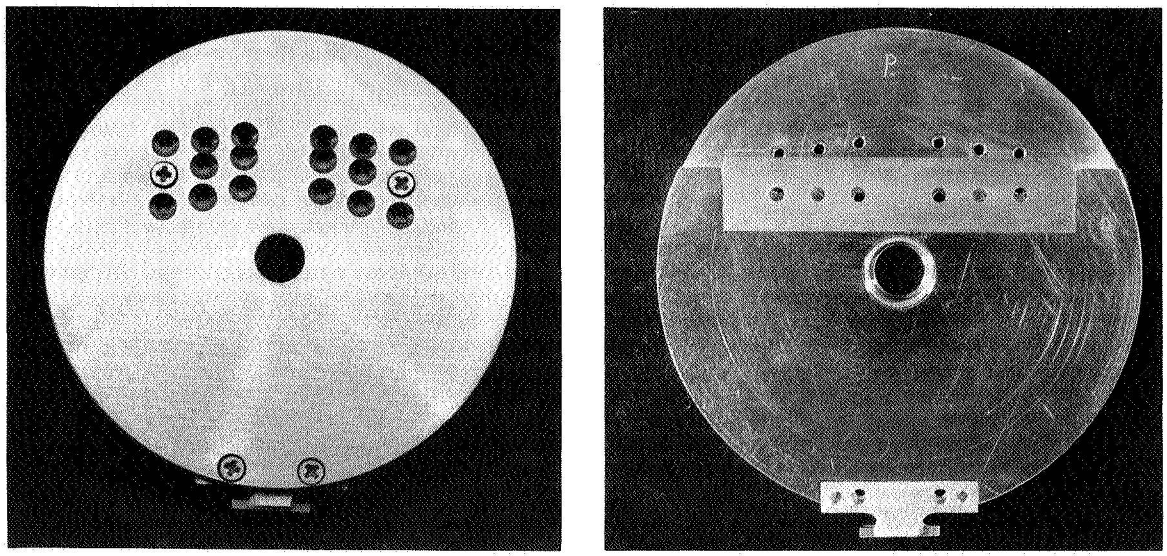


(a) Standard weight

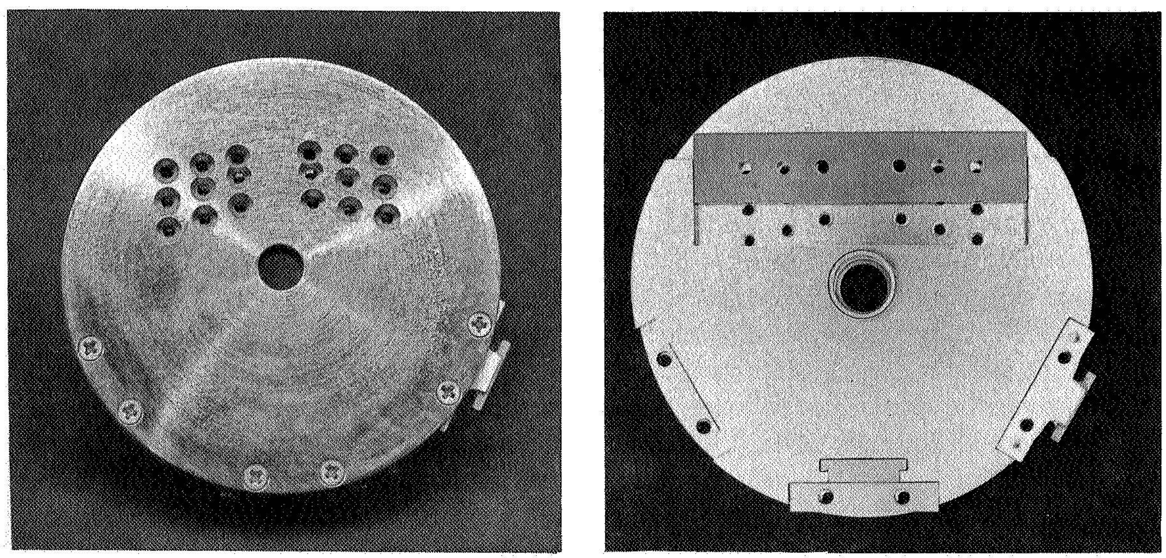


(b) #2 and #4 position weight

Figure 8—Four and one-quarter ounce test weight. As in Figure 7 the internal features may be seen in the photographs on the right.



(a) Standard weight



(b) #2 and #4 position weight

Figure 9—Nine ounce test weight. This weight is calibrated for tests at 150,000; 180,000; 200,000; 250,000; 300,000; and 350,000 mg-mm with no change in total weight. The desired setting is accomplished by moving the Heavy Met balance weight progressively inward to the predetermined hole positions. Tests may be run at a 1 oz-in. setting (720,000 mg-mm) by removing the 1.01 ounce balance weight. Three additional calibrated positions can be obtained by making new Heavy Met balance weights with off-center holes.

The particular reed switch chosen is approximately 5/8" long and 1/8" in diameter. The contact points are rhodium tipped for maximum resistance to oxidation and the expected lifetime is 2×10^6 cycles at the specified low current rating. The companion actuating magnet is 1/16" × 1/16" × 1/2" with two simple poles.

Simple experiments showed that the switch would close reliably with a distance between the magnet and the switch of about .110". The particular switch purchased specified a close tolerance on the magnetic field strength required to close the contact points. This gave an assurance that the switch closure will occur no more than one or two degrees before the center of the magnet reaches the center of the switch.

The design achieved for the reed switch mounting frame or basket as it is commonly referred to is shown in Figure 10. The 10 individual switches are mounted on clips with the leads protruding through PTFE strips for wiring attachment on the back side. The basket is located on the fixture frame so that the actuating magnets clear the reed switches by about 1/16". This gives a good margin of safety for reliable actuation of the switches. The standard mounting holes for the basket place the two rows of switches at 30 and 90 degrees. Three additional hole positions allow upward rotational adjustments of 3-1/2, 7, and 10-1/2 degrees, respectively.

D. Initial Setting Capability: The last important feature required for the test fixture was a means of restraining angular oscillations of the test weights. These random motions are caused by transient running torques which are smaller in magnitude than the 30 degree torque level. It was determined that more meaningful test results could be obtained by setting the test weights at a predetermined angle upward from bottom center. Then the preset torque would have to be exceeded before a test weight could begin to move. Thus early oscillations would be eliminated. The physical means of accomplishing the initial setting would be a stop bar. It was determined that it would be desirable to have the capability to vary the initial setting from -10 to +30 degrees of test weight rotation in 5 degree increments. The design achieved for the stop bar is illustrated in Figure 11. In addition the back edge serves as a stop for test weights which have exceeded 90 degrees and flipped over.

E. Provisions for Mounting: The basic test fixture had to be capable of being mounted on either a vacuum flange or air test stand. The designs achieved are shown in Figure 12a and 12b, respectively. Note that the air test stand incorporates a rather heavy design; also, the magnetic coupling was retained – both features for the purpose of maintaining a close similarity between the vibration and damping characteristics of the vacuum and air test configurations.

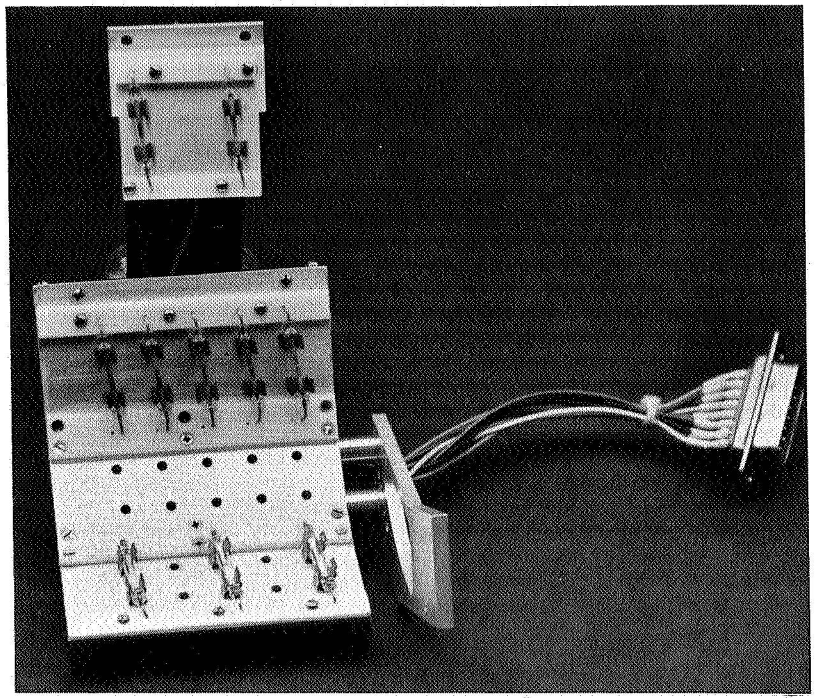


Figure 10–Reed switch basket

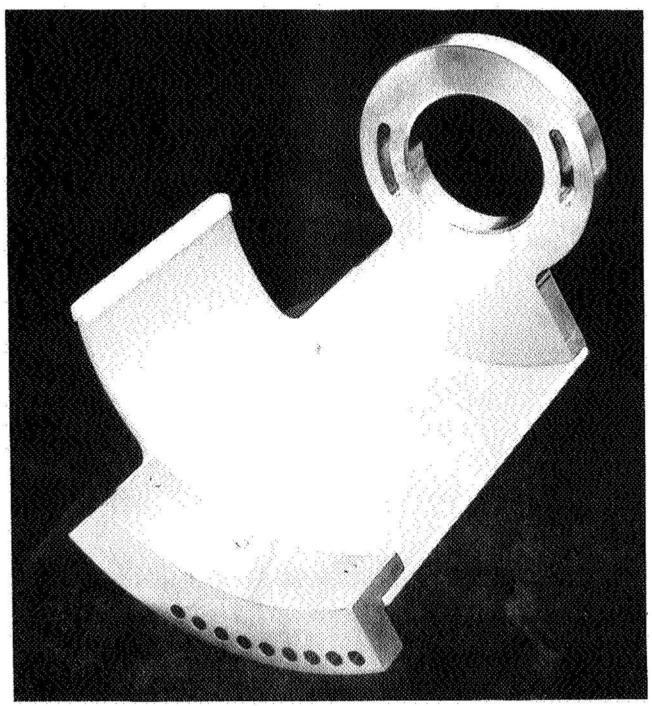
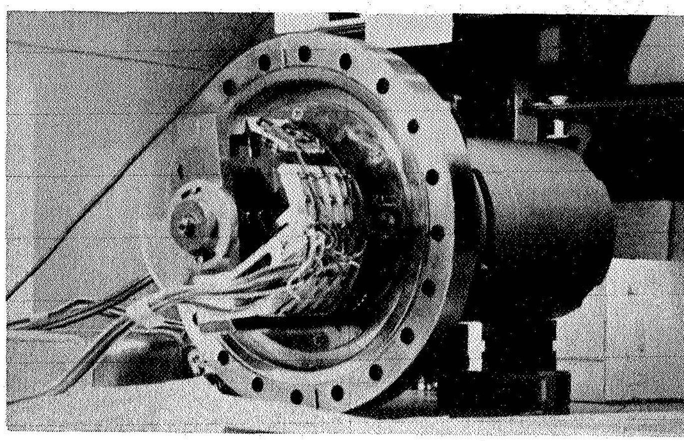
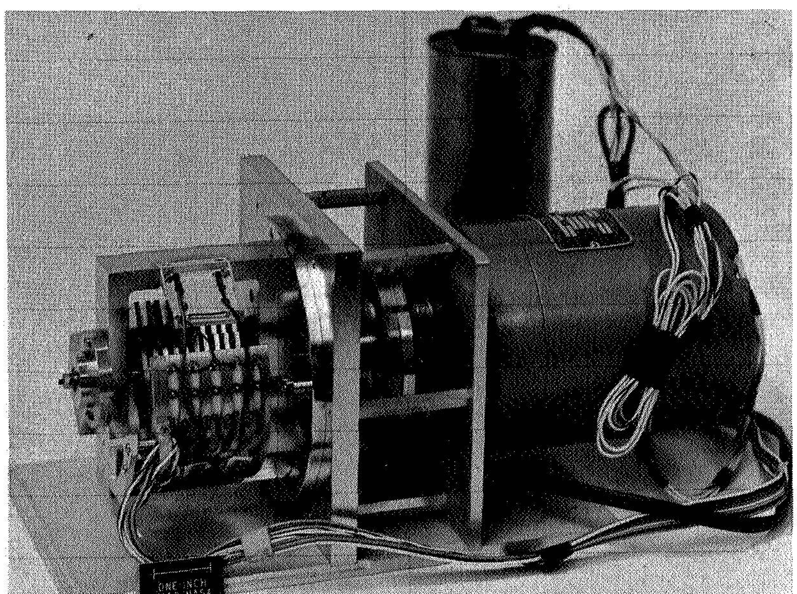


Figure 11–Stop bar assembly. The holes in the lower frame are spaced 5 degrees apart for initial settings of the test weights from −10 to +30 degrees from bottom center.



(a) Fixture mounted on vacuum flange



(b) Fixture mounted on air test stand

Figure 12—Vacuum and air test assemblies

2. DATA RECORDING

The test system requires the real-time recording of 60 channels of information regarding position sensor condition – 10 per individual test fixture. Not only does each closure of a reed switch have to be recorded reliably, but in addition easy check-out capability of the instrumentation system is necessary – both prior to and during the conduct of a given test. Also, the system must be operable for long durations with a minimum of servicing.

These objectives were achieved through the choice of a 20 channel ink-writing chart recorder capable of a 3/4" per hour chart speed. Three of these recorders provided the required 60 channels. The ink writing feature was deemed superior for this application because recorder pen failures are instantly observable. Also, at low chart speeds, the hot tip method of writing used on many recorders was not deemed as reliable. The three recorders were mounted in the base of the test cabinet as discussed subsequently in Section 4 (See Figure 13).

3. VACUUM EQUIPMENT

For the vacuum tests it was decided to utilize ion-getter pumps plus titanium sublimation elements. This type of pumping insures a "clean" vacuum free of any oil molecules which might serve to lubricate the bearings being tested and thu negate the results of the test. For roughing down to the 5-10 micron range, a cyrogenic system with molecular sieve was selected. Cycling bakeouts were employed to improve the vacuum obtained by ion-getter pumps prior to utilization of the titanium sublimation elements.

Separate parallel vacuum systems were provided for each of the three vacuun test chambers with the exception that a common header interconnecting the three chambers was provided for cyro pumping. This enabled the two molecular sieves to be connected to one chamber. Each chamber was provided with a 75 liters/second ion-getter pump and a 350 liters/second titanium sublimation element. The power supply unit for each pump is provided with a gauge for vacuum level at the pump. This gauge is relatively insensitive to fluctuations in chamber pressure because of the distance between the chamber and pump. Thus a single precision Hastings vacuum gage was provided with a direct access plug to each chamber.

The design of the chamber itself warrants some discussion. It is a "Tee" shaped design with provision for large flanges on either end of the main chamber and a neck and flange for connection to the pump in a transverse direction. Opposite the pump neck is a small tube with a flange for a 20 pin feedthrough. The main feature of the chamber itself is that the test fixture mounts on one flang and a large viewport is provided on the opposite flange so that a test in progress

can be viewed directly. The overall arrangement of the equipment is discussed in the next section.

4. TEST CABINET

The final requirement for the test system was to design a cabinet to contain all six test fixtures and their related support equipment. This was accomplished through the design of the facility shown in Figure 13. The significant features are discussed briefly.

For convenience in visually observing the tests in progress, the test fixtures are located at or below eye level. The three vacuum chambers are at the cabinet's center and the three air test fixtures are located on a shelf immediately above. The chart recorders are located at the bottom and the power supplies for the ion-getter pumps are located at the top of the cabinet. In the photograph a complete vacuum assembly cannot be viewed because each ion-getter pump is located behind a chart recorder. Each pump is directly under the vacuum chamber to which it is connected. Between each ion-getter pump and chamber a titanium sublimation pump is perpendicularly mounted to the front. These three pumps may be seen in the picture – immediately above the chart recorders. Also, between the titanium pumps the two cyrogenic sieves may be seen at about the same height.

Other chamber features include an internally contained color-coded wiring harness with permanently mounted connector plugs adjacent to each test bay. The interconnecting vacuum manifold has valving so that each pump can operate independently. The use of this for cyro pumping has already been discussed in the preceeding section. Also, a shelf was provided immediately below the vacuum chambers for convenience in making final checks upon a vacuum test fixture just prior to installing it in the chamber. Full height doors close the back side of the cabinet for dust protection and yet permit easy access for viewing vacuum tests in progress or for servicing the equipment.

CALIBRATION, ASSEMBLY, AND TEST PROCEDURES

The validity of test results depends in a large measure upon the precision and accuracy with which test equipment is fabricated and assembled. For the LOAD-LIFE testers the highest standards were employed in the manufacture of all parts and in the initial calibration of the individual test weights. Also, for each test assembly precision methods were developed to insure the proper adjustment of axial preload – both in the test bearings and in the support bearings. Lastly, a step by step procedure was developed to insure a correct final assembly for each test. Each of these areas is discussed briefly below and expanded into more detail in related appendices.

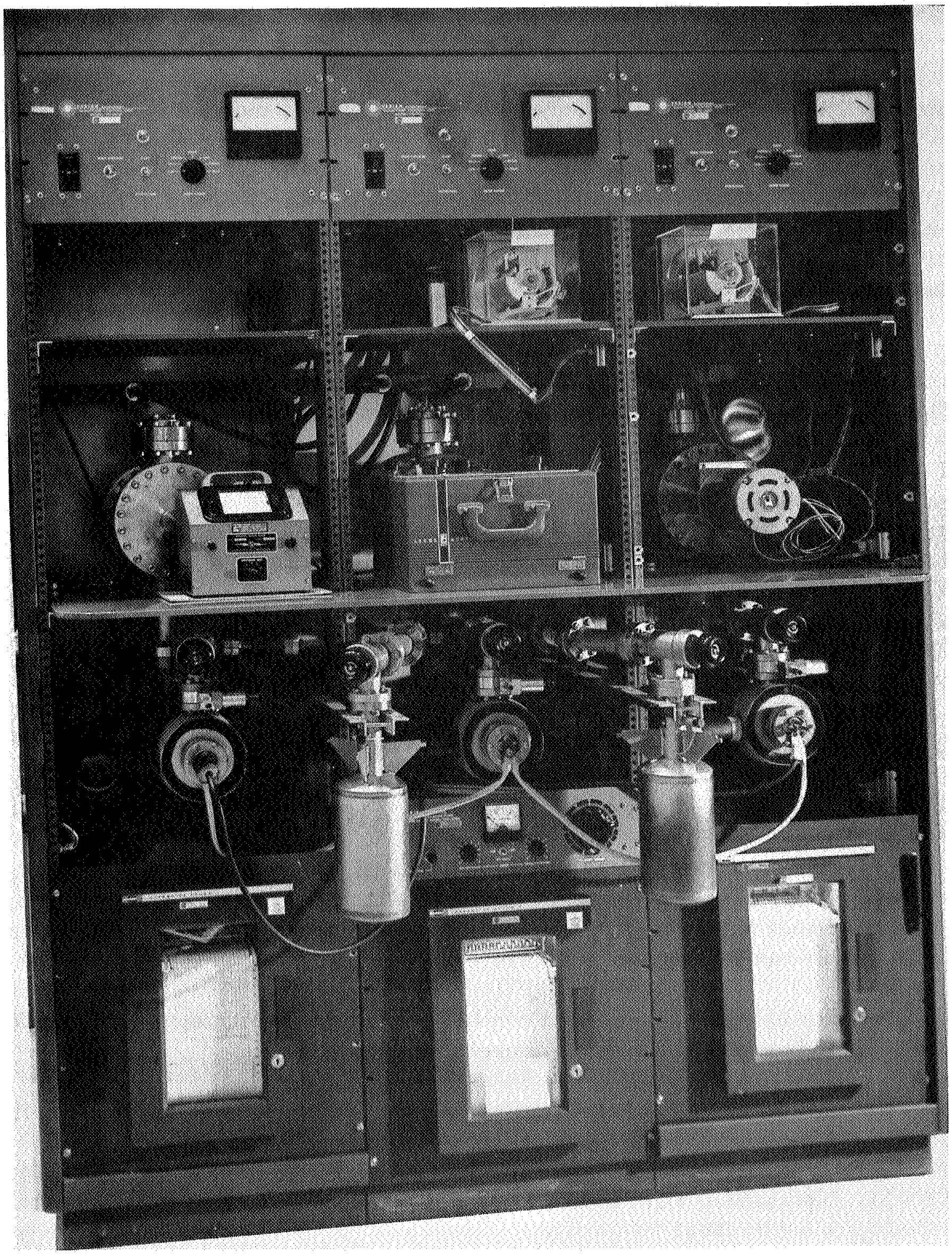


Figure 13-Test cabinet. This enclosure houses the entire test system. There are three separate bays in which from top to bottom the power supplies, air test fixtures, vacuum test chambers, titanium sublimation pumps, ion-getter pumps, and chart recorders are housed.

1. CALIBRATION OF TEST WEIGHTS

To insure the uniformity of the "failure" criterion for all tests to be performed, all test weights were calibrated to an eccentric moment value of 180,000 ± 2000 mg-mm. The basic theory of eccentric moment design has been previously discussed in the first part of this paper under SELECTION OF "FAILURE" CRITERION and illustrated in Figure 3. Each test weight design included at least one balancing piece fabricated slightly oversize so that it could be trimmed to bring the test weight into the desired calibration (see Figures 7, 8, and 9).

A special fixture having parallel knife edges and a balanced, two-inch diameter pulley wheel were designed for making static torque measurements. The inner and outer races of two bearings were fixed relative to each other and the pair were used as a hub on a 1/8" diameter shaft. The test weight being calibrated was then assembled on this hub with the balanced wheel attached to the outside. By means of a nylon cord attached to the rim, the weight of a small bucket of lead shot was applied tangent to the wheel at a distance of 1.000" from the center. This setup is shown in Figure 16 on page 40. By adding lead shot to the bucket until a test weight was balanced at the 90 degree position, a calibration was obtained. For readers interested in a more detailed discussion of this method including tables of final calibrated values for all 30 test weights, Appendix I is included.

2. PRELOAD CONTROL

The LOAD-LIFE tests were conceived to gather information regarding the effect of radial loads upon the lifetimes of instrument sized bearings run at several different speeds, in vacuum and in air. As discussed previously it was decided to mount test and support bearings "back to back" with shims between the outer races to take up axial freeplay. The mounting arrangement for a test bearing pair has been shown in Figure 4 on page 10.

In order to mount the test and support bearings with consistency and accuracy, the following procedures had to be developed.

(i) Determination of bearing stickout under a very light axial load to within a range of .0001"

(ii) Determining axial spacer thickness and honing to a uniform dimension to within .0001"

(iii) Checking for uniformity of preload condition in all test setups.

After experimentation, appropriate laboratory procedures were developed for each of the above areas. They are discussed briefly in the succeeding three paragraphs. For additional information, refer to Appendix II.

A. Bearing Stickout Determination: The test bearings are ordered from the vendor with the requirement that the one-way free-play or stick out measurements be supplied. As a precaution prior to assembly of test and support bearings, all stick out readings are rechecked. Through use of an electronic displacement gauge, a dial indicator force gauge, and a special holding fixture, an in-house recheck is made. Readings can generally be duplicated to within 0.0001 to 0.0002".

B. Axial Spacer Thickness: In making an assembly of a bearing pair, the following formula is used for determining the initial thickness of the axial spacer (differential thickness of inner and outer race spacers for support bearing pairs):

$$\text{Thickness} = \text{Stickout}_1 + \text{Stickout}_2 - 0.0003^*$$

where all dimensions are to the nearest 0.0001 in inches. For the test bearing pairs, circular brass shims of .003, .004, and .005" are stocked. Through careful hand honing techniques and frequent micrometer checks of the thickness at several points around the circumference, the appropriate standard thickness shim is reduced to the width determined in the above formula. The thickness around the circumference must vary by no more than 0.0001".

C. Uniformity of Preload Condition: It is very important to have the five bearing pairs in the test weights adjusted to the same preload condition (approximately zero). This is accomplished by making adjustments in the individual shim thicknesses until each of the test weights oscillates 25 ± 2 times upon being released from the 90 degree angular position. This procedure which is called the Swing Test is discussed completely in Section C of Appendix II.

Because the number of oscillations that occur before the weight comes to a complete rest depends upon several geometric variables other than the spacer thickness alone, there was some doubt that the Swing Test effectively insures uniformity of preload condition. To verify the uniformity of their preload condition, five test weight assemblies which met the Swing Test criterion were checked at the vendor's plant on a Dynamic Deflection/Preload

*0.0004 for support bearing pairs

Gauge. This checkout showed that a generally uniform preload condition of several ounces existed. This is considered sufficient for test purposes. Also, all five assemblies loosen up slightly after undergoing a few hours of testing so the small initial preloading is probably desirable.

3. FIXTURE ASSEMBLY

A check list was developed for the assembly of test fixtures to insure that a thorough and uniform assembly would be accomplished each time. With prior assemblies of – the yoke to the adapter plate, reed-switch basket, stop bar, and test shaft with inner magnet, the complete fixture assembly consists of the following ten steps:

(i) Measurement of stickouts of test and support bearings

(ii) Installation of support bearings with differential width spacers

(iii) Assembly of five test weights with appropriate shims

(iv) Torquing of end-nut to 28 oz-in. and performance of Swing Test

(v) Mounting on vacuum flange or air test stand

(vi) Setting reed-switch basket to the desired angular position and checking for clearance of magnets over reed switches

(vii) Setting stop-bar to the desired angular position

(viii) Checking electrical continuity from reed switches to recorder pens

(ix) Trial run of fixture to check the shaft rotational speed and for smooth operation

(x) Installation in vacuum chamber or placement of dust cover on air test assembly.

4. TEST PROCEDURES

The objective in running each test setup is to obtain 30 degree indications on all five test weights. Operational procedures have been developed to assure adequate control of test conditions from start up to conclusion. In this section pumpdown procedures for vacuum tests, special observations of the start up period, and daily monitoring routines are discussed briefly.

A. Vacuum Pumpdown: Either one or both molecular sieves cooled by liquid nitrogen are utilized to pump down to the 5 to 10×10^{-3} Torr range. When this pressure range is reached, the ion-getter pump is used to get down to the 10^{-6} Torr range. Cycling bakeouts aid in getting down to the mid 10^{-8} Torr range at which time the titanium sublimation elements are employed to get down to the mid 10^{-9} Torr range.

B. Initial Start up: The shaft speed is checked by stroboscope and each test weight is visually observed for any unusual vibrations. Early 30 degree "failure" torque indications of a sporadic nature are disregarded provided they cease within several hours but recurrent indications require stopping of the test for further adjustments.

C. Daily Observations: Following the initial period several conditions are visually noted twice daily and recorded on an appropriate test log. The full-time recorder tape is checked for any indications of reed-switch closure and the individual test weights observed as a cross-check. The exact times of closures are recorded and the visually observed condition noted under Remarks. In addition the chamber pressure and temperature are recorded.

Toward the end of a test these observations should be made more frequently with particular emphasis upon the general condition of test weight motion – whether steady or intermittent. Jerky, intermittent motion of a test weight is usually indicative of an imminent 90 degree "failure" and flip-over which ends the test.

PRELIMINARY TEST RESULTS

Actual experience with the LOAD-LIFE tester was necessary in order to evaluate the concepts incorporated in the design. Thus preliminary tests were run – four at the 2 oz - 3600 rpm condition and one at the 4-1/4 oz - 600 rpm condition. For all these tests the stop bar has been set so that at start up the test weights rest in a position rotated 30 degrees counterclockwise from bottom center. The reed switch basket is set with the bottom row of switches at 33-1/2 degrees from bottom center. With a switch pull-in tolerance zone of from minus 1/2 to 1 degree, the actual recorder indication occurs when a test weight has risen 2-1/2 to 3 degrees off the stop bar.

The advantages of the above described setup are several. The static setting of the test weights eliminates random oscillations across bottom center. By placing the switch closure position 2-1/2 to 3 degrees above the static position, it is assured that a dynamically applied torque in excess of the preset static

torque has occurred for a sufficient duration to overcome the rotational inertia of the test weight and trip the switch. Because the inertias of the 2, 4-1/4, and 9 ounce test weights could not be designed to be equal to each other, identical torque pulses do not produce the same responses. However, the same general principle applies – namely that the preset torque level has been exceeded whenever switch closures occur. Proper interpretation of the data tape is the means by which variations in test weight response are accounted for.

Through study of the data tapes from the four 2 oz - 3600 rpm tests a tentative standard for picking times to the 90,000 mg-mm "failure" level has been established. It was decided that sporadic, random switch closures should not be counted in gathering designer data. The times to "failure" recorded herein are those at which switch closures generally occurred twice each hour for a period of at least 2 or 3 hours. This formula proved to be satisfactory for the limited number of tests described herein. However, for the proposed future tests engineering judgement must be employed in determining the exact "failure" times to record. This is particularly important when a wide scatter in the occurrence of switch closures has occurred within running times of only a few hundred hours.

1. INDIVIDUAL TESTS

The preliminary test results obtained at GSFC divide basically into two categories. In 1965 early in the LOAD-LIFE test program a 4-1/4 oz — 600 rpm air test was started. This test was run continuously for about three years. The 2 ounce fixtures were completed next and a series of four tests were run at the 2 oz - 3600 rpm condition. These four tests provided some unexpected results. The times to "failure" for all five tests are summarized in Table I. Complete information including stickouts, spacer thicknesses, and "before and after" Swing Tests are contained in Appendix III.

A. 4-1/4 oz - 600 rpm Test: This single air test was run for 22,960 hours with no indication of the attainment of the 90,000 mg-mm (30 degree) torque level. This fixture had not had its #2 and #4 position magnets moved up 60 degrees (as was done prior to all subsequent tests) but the small magnetic interaction was not a factor in the obtaining of such long life. In total cycles this was the longest duration without obtaining an indication of the 90,000 mg-mm torque level of any of the five original tests – and it is not known how much longer the test might have run before the first 30 degree indication would have been obtained.

B. 2 oz - 3600 rpm Tests: Four tests were conducted in the following sequence:

(i) Air Test I: This test was run to a conclusion with all five test weights achieving 90,000 mg-mm torque indications. Fixture No. 1 was utilized. The first indication occurred at 163 hours and the fifth at 236 hours. A mean life of 202 hours was obtained. A 90 degree indication occurred at 231 hours for the test weight which first achieved a 30 degree switch closure at 163 hours. There was no difficulty in obtaining all five 30 degree indications. The test was stopped because of the excessive vibration which occurred after the center test weight flipped over against the back stop. (See Figure 14a.) A large amount of wear debris was in evidence around the bottom of the test fixture. (See Figure 14b.)

(ii) Vacuum Test I: This test was run to a conclusion with all five test weights obtaining 90,000 mg-mm indications and one a 180,000 mg-mm indication. Fixture No. 2 was utilized. The earliest 30 degree indication occurred at 1990 hours and the fifth one at 3129 hours. The mean life was 2738 hours. The fourth test weight reached the 30 degree level at 2889 hours and attained a 90 degree indication at 3127 hours. This test weight then flipped over against the back stop where it developed excessive radial and axial play. It then cocked and jammed into the magnet boom of the adjacent test weight. (See Figure 15.) A considerable amount of wear debris was in evidence around the lower parts of the fixture as reported for Air Test I.

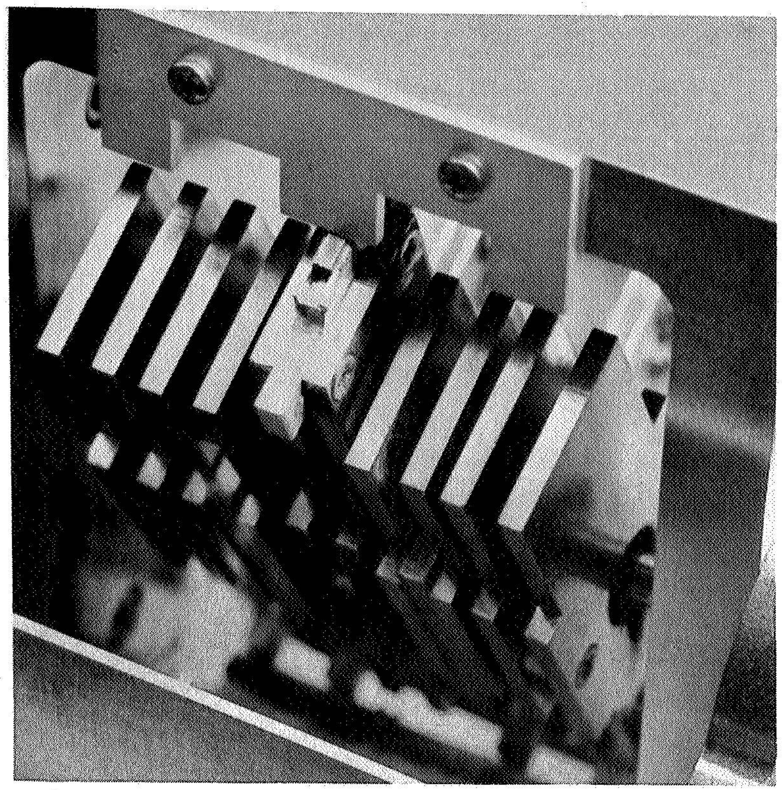
(iii) Air Test II: The order of magnitude increase in mean running time for Vacuum Test I over Air Test I led to some speculation that the heavier mounting base of the vacuum test fixture might have been a factor. Thus the second 2 oz - 3600 rpm air test (Fixture No. 1 again) was run in a vacuum chamber vented to a normal atmosphere. The first 90,000 mg-mm indication occurred at 237 hours, the second at 280 hours, and the third at 355 hours. The fourth and fifth test weights never attained this level because the weight with the initial indication at 30 degrees flipped over and jammed on the stop bar following a 90 degree indication at 388 hours. These two test weights were still running smoothly when the test was stopped at 429-1/2 hours. The mean life for the three 30 degree indications was 290-2/3 hours – not significantly above that obtained in Air Test I.

(iv) Dry Nitrogen Test: A final 2 oz - 3600 rpm test was run under identical conditions (in Fixture No. 1) as that described for Air Test II except that the chamber was filled with dry nitrogen under a positive pressure instead of a normal atmosphere. Haltner (3) and Winer (4) discuss the detrimental effects of the presence of water vapor and/or oxygen upon the lubricating qualities of MoS_2. A nitrogen atmosphere eliminates the presence of oxygen and water vapor and thus is sometimes

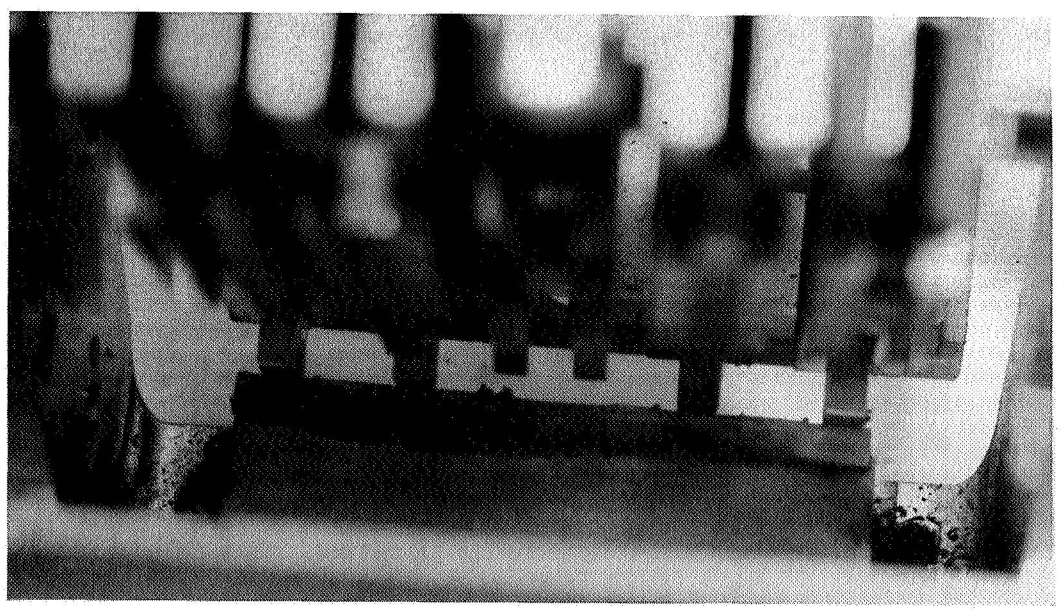
Table I
Torque Buildup Histories of Preliminary Tests

Test Weight		Hours to Indicate Torque Level: 90,000 mg-mm	Hours to Indicate Torque Level: 180,000 mg-mm
A	4-1/4 oz - 600 rpm (None of the five test weights had any indications in 22,960 hours)		
B	2 oz - 3600 rpm		
	(i) Air Test I		
	A	204	
	B	236	
	C	163	231
	D	189	
	E	219	
	Mean	202	
	(ii) Vacuum Test I		
	R	1990	
	S	2818	
	T	2865	
	U	3129	
	V	2889	3127
	Mean	2738	
	(iii) Air Test II		
	A	237	388
	B	280	
	C		
	D	355	
	E		
	Mean	291*	
	(iv) Dry Nitrogen Test		
	A	694	1159
	B	988	
	C		
	D		
	E		
	Mean	841*	

*Mean from incomplete data set



(a) Center test weight jammed against upper stop



(b) Wear debris around lower parts of fixture

Figure 14–Conclusion of Air Test I. This was a 2 oz - 3600 rpm test in which all five 90,000 mg-mm indications were obtained with a mean life of 202 hours.

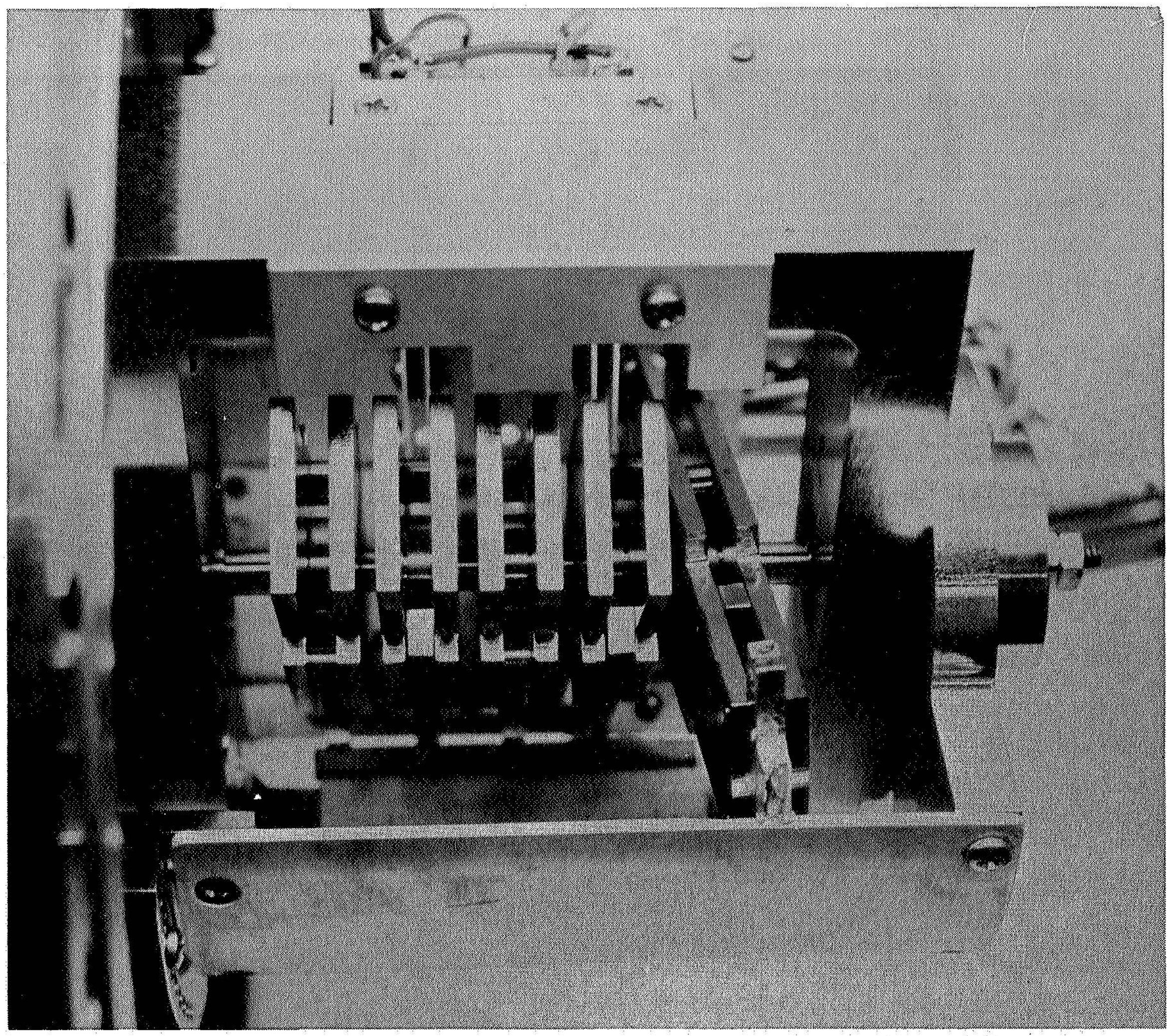


Figure 15–Conclusion of Vacuum Test I. This 2 oz - 3600 rpm test experienced all five 90,000 mg-mm indications with a mean life of 2738 hours prior to the flip-over and jamming of the end test weight.

used to simulate a high vacuum environment. The first 90,000 mg-mm indication occurred at 940 hours and the second at 988 hours. The test had to be stopped at 1159 hours because the test weight which first experienced a 30 degree indication gave a 90 degree indication and flipped over. The stopping of a test with 3 test weights failing to achieve a 30 degree indication led to consideration of ways to continue a test after a 90 degree failure has occurred. (See CONCLUSIONS.)

2. AUTOPSY REPORT ON TEST BEARINGS

The test and support bearings from three out of the four 2 oz - 3600 rpm tests were sent to the bearing vendor for autopsy. A complete autopsy report was

prepared and forwarded to GSFC. While its contents are too detailed to be included here, it was felt that the introductory and summary information might be of interest to some readers. This information was abstracted and placed in Appendix IV.

3. DISCUSSION OF RESULTS

The results from the four 2 oz - 3600 rpm tests provide some interesting comparisons. Vacuum Test I produced the most predictable result with a mean life of over 2700 hours. Class ABEC-7 instrument sized bearings lubricated with PTFE-MoS_2 retainers might be expected to run approximately this long before reaching 90,000 mg-mm of running torque. The mean lifetimes of 200 to 300 hours obtained in Air Tests I and II do not appear to be realistic. Both tests were run in Fixture No. 1 so there was some concern that the vibration characteristics peculiar to that fixture might have been related to these results. The results from the Dry Nitrogen Test which was also run in Fixture No. 1 dispelled this concern. The fact that only 2 out of 5 test weights had attained the 90,000 mg-mm level by 1159 hours gives some evidence that the rapid wear of the PTFE-MoS_2 retainers was caused by the presence of oxygen and/or water vapor.

No definite conclusions regarding the short lifetimes obtained in air can be drawn at this time, but some subsequent tests (not complete enough to report) corroborate the short lifetimes obtained for the 2 oz - 3600 rpm air tests. The premature stoppings of Air Test II and the Dry Nitrogen Test illustrate some of the problems encountered in getting all five 30 degree data points from a given test setup. Similarly, the 22,960 hour run of the 4-1/4 oz - 600 rpm air test without any 30 degree indications shows the difficulty in terms of required running time in obtaining the needed statistical data from low-speed tests.

CONCLUSIONS

The objective of this task was to make it possible to gather statistical quantities of comparative data on instrument sized bearings under carefully controlled load, speed, and vacuum conditions. The initial results from the tests discussed herein indicate that this goal has been achieved with the radial load tester and associated test system developed at GSFC. Within a single cabinet a complete and self-contained test system can be used to gather statistical samples of data simultaneously in air and vacuum at three different radial loads.

1. MOST TEST CONDITIONS REASONABLE

Nine data points (3 loads at each of 3 speeds) are needed to construct the design curves of Load vs. Life or Speed vs. Life for a given environmental condition.

Thus a total of 18 data points are needed to construct design curves for a given lubricant being evaluated in vacuum and air. This requires three setups of the complete system of six fixtures. The 2738 hour mean life obtained for the 2 oz - 3600 rpm vacuum test of PTFE-MoS_2 lubricated SR2-6 bearings indicates that for bearings lubricated with similar solid retainer material a reasonably obtainable running time can be expected. The 20,000 hours plus without a single 90,000 mg-mm indication at the 4-1/4 oz - 600 rpm condition indicates that an unrealistically long running period would be required to obtain the full statistical sample. It may be necessary to cut off the low speed tests at some arbitrary point - say at a two-year lifetime. This would be justifiable on the basis of present mission requirements.

Following the completion of the five initial tests discussed herein, a decision was made to continue the LOAD-LIFE TEST PROGRAM at the plant of the vendor which supplied the initial lot of test bearings. This method of operation has the advantage of utilizing specialists to make the initial setups and monitor the test equipment. Also, specialized equipment is readily available if needed and the post-test autopsies can be performed more efficiently. The complete series of tests with PTFE-MoS_2 solid lubricant retainer will be performed twice each in vacuum and in air to obtain a set of design curves.

2. SEVERAL IMPROVEMENTS NEEDED

Some improvements in the equipment and procedures are presently being made and others are being considered. A circuit utilizing an SCR and relay has been incorporated to shut off the tester drive motor upon the occurrence of a 90 degree "failure". This prevents the occurrence of false 30 degree indications caused by the excessive vibration which usually follows the flip-over of a test weight. It also prevents damage to the equipment. To assure that the effects of dust are completely eliminated from both air and vacuum tests, the entire test cabinet has been moved into a clean room where all assemblies are made.

Two other improvements are in the planning stage of this time.

The first improvement is to try some tests with wider spacing between the test bearing pair. A slight wobble of the test weights has been detected during operation with the present back to back mounting of the test bearings. A spacing of approximately 1/2" between the test bearings can be achieved by modifying the test weights and reducing the number of test weights on the shaft to three. In the original design of the test fixture the possible advantages of this wider spacing was anticipated and extra space was provided adjacent to the outermost test weights. The three redesigned weights can be centered on the reed switches presently used by the first, third, and fifth test weights, eliminating the need for any redesign other than that of the 2, 4-1/4, and 9 ounce test weights themselves.

The same eccentric moments would be retained and the total weights would be kept as close to the present values as possible.

It is recognized that the elimination of two test weights (four test bearings) considerably reduces the statistical sampling desired. However, as noted in the discussion of the preliminary tests, all five data points were not obtained from every test. Premature wear of the bearing components may have been caused by the small dynamic loading associated with this unsteadiness in running. The increased spacing would provide greater dynamic stability in operation and reduced retainer wear. This in turn should improve the chance of getting all three 30 degree indications before a single 90 degree indication. Each 3-weight test could be run twice to obtain the data points needed for statistical analysis.

A second planned improvement is the development of a means to continue a test after a 90 degree indication has shut the fixture off before all test weights have given 30 degree indications. A visual inspection would be made to see that the weight hadn't flipped over. If it hadn't, attempts would first be made to simply restart the fixture. This would show whether or not the 90 degree indication was a random occurrence. If shutoffs persist, the only way to continue a test would be to remove the test weight causing the shut offs. Consideration of this possibility indicates that new bearings would have to be installed and the weight replaced to keep the shaft loading the same. There are some delicate problems to be solved such as restoring the same axial loading that existed before disassembly. (It is known to be less than the original 28 oz-in.) Some experimental work will have to be accomplished before it can be determined whether or not this removal and replacement of a test weight is possible. Such a capability would add greatly to the objective of obtaining statistical data.

3. SUMMARY

The LOAD-LIFE radial load bearing test system has been designed, fabricated, and assembled. The objective of running tests at several different loads and speeds with a common "failure" criterion has been achieved and several preliminary tests have been completed. These tests tentatively show that the PTFE-MoS_2 material chosen as solid lubricant for the first series of tests offers promise for space applications, but the series must be completed before definite conclusions can be made. Initial results show that there may be a need to eliminate or limit running time in a normal atmosphere with this type of lubricant. The exhaustive tests planned for the future should provide the answers to these and other questions space designers have today.

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3. Haltner, A. J., "Friction and Wear of Solid Materials Sliding in Ultrahigh Vacuum and Controlled, Gaseous Environments," AFML-TR-68-76, April 1968.

4. Winer, W. O., "Molybdenum Disulfide As A Lubricant: A Review of the Fundamental Knowledge," Wear, Volume 10, 1967.

APPENDIX I

CALIBRATION PROCEDURE FOR TEST WEIGHTS

Each of the three different eccentric weights (2, 4-1/4, and 9 ounce) had to have an eccentric moment (maximum static torque) calibration to 180,000 ± 2000 milligram millimeters (mg-mm). With this 90 degree calibration, the desired 30 degree eccentric moment of 90,000 ± 1000 mg-mm is attained without a direct calibration.

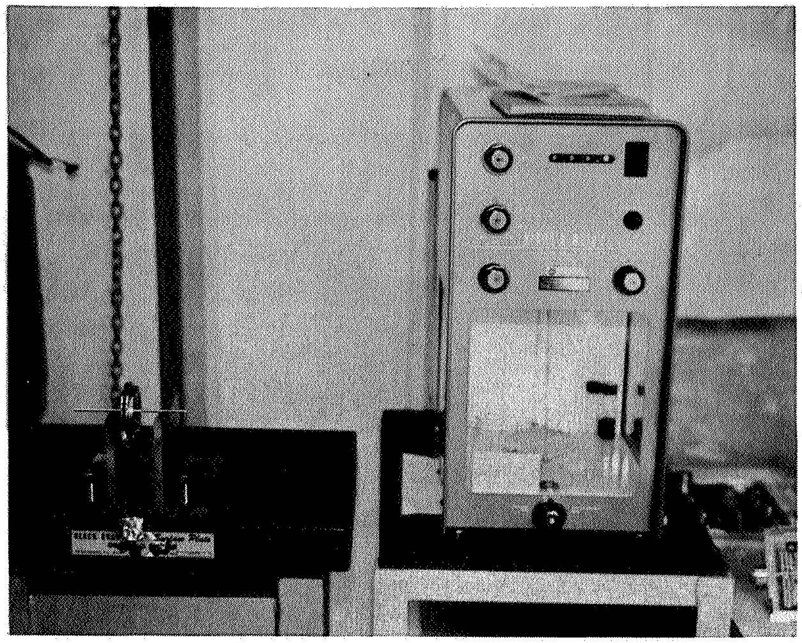
The equipment necessary to accomplish this task consists of a surface plate, precision level-adjusted parallel knife edges, support shaft with dummy bearings, a balanced spool, weight bucket and line, and a scale that weighs accurately in grams to the fourth decimal place. A view of this equipment may be seen in Figure 16a.

The calibration procedure is accomplished in the following steps:

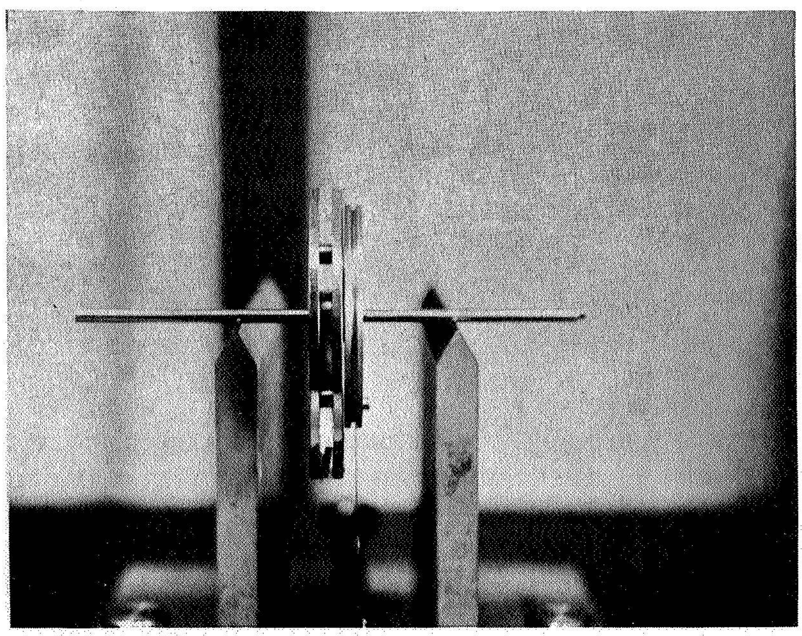
(i) The test weight to be calibrated is assembled, taking care to mount the dummy bearings (inner and outer races glued together) properly. Using double backed tape (cut in a circle with a center hole) attach the 1.000" radius balanced spool to the test weight. One end of a 5 pound test nylon string is attached tangentially to the spool and a hook is attached to the other end. The 1/8" diameter shaft is inserted through the assembly.

(ii) The surface plate is leveled using a level with each calibration division equal to .0005 inches per foot. The parallel knife edges are then leveled through the use of the same level in conjunction with two gauge blocks.

(iii) The test weight, spool, and shaft assembly is placed on the parallel knife edges as shown in Figure 16b. The nylon string is wrapped around the spool and the small bucket is attached by means of the hook on the end. Weight in the form of fine lead shot is added to this bucket until the vertical centerline of the test weight is rotated to the 90 degree position. Because 90 degrees is a point of unstable equilibrium, it must be approached very slowly – practically one shot at a time. Each operator has to establish a particular technique in making this balance, but it is recommended that the test weight be able to withstand a very small disturbance without either falling below 90 degrees or flipping over.



(a) View of work area showing level surface plate, parallel knife edge fixture and accurate scale



(b) Close-up view of knife edges with test weight in the calibrate position. The balanced spool is attached to the test weight with double backed tape.

Figure 16–Equipment for eccentric moment calibration of the test weights

(iv) Following the 90 degree balance, the weight bucket is removed from the hook and weighed on the gram scale to fourth place accuracy.

(v) Two more calibrations are made according to the procedures of (iii) and (iv) above to obtain a total of three independent readings. (The bucket is emptied each time before the next reading.)

(vi) The three weights obtained are averaged and the resulting value used to compute the eccentric moment as illustrated in the following sample calculation:

$$\left.\begin{array}{r} 6.9445 \\ 6.9402 \\ +\ 6.9419 \end{array}\right\} \text{Three independent weight readings}$$

$$20.8266$$

$$\frac{20.8266}{3} = 6.9422 \text{ gm } \} \text{Average reading}$$

6.9422 gm.	
+.15 gm	Wt. of string and hook
7.0922 gm	Total tangential force
× 25.4 mm	Radius of spool

equals 180.1418 gm-mm

or

180,141.8 mg-mm Eccentric moment

The final calibration values for the 2, 4-1/4, and 9 ounce test weights are summarized in Table II. Note that the 150,000; 180,000; 200,000; 250,000; 300,000; and 350,000 mg-mm positions were calibrated on the 9 ounce weight. Thus 6 discrete settings are available by placing the Heavy-Met balance weight at the desired hole positions. For additional information on the features of this weight, refer back to Figure 9 in the main body of this report.

Table II
Final Calibration Values of Test Weight Eccentric Moments

Test Weight	Torque mg-mm	Test Weight	Torque mg-mm
2 ounce			
A	180,604	R	178,760
B	180,083	S	178,224
C	180,136	T	177,878
D	180,337	U	180,555
E	180,642	V	178,770
(Completed 9/6/66)		*(Completed 8/15/66)*	
4-1/4 ounce			
6	180,429	16	179,588
7	181,610	17	179,301
8	180,558	18	179,557
9	180,652	19	181,097
10	180,919	20	180,538
(Completed 10/3/68)		*(Completed 2/28/68)*	

9 ounce

Position	A	B	C	D	E	F
Vacuum Test Group						
L	149,672	179,474	200,060	251,699	303,035	352,486
F	150,046	180,386	200,213	249,073	300,144	351,574
M	150,119	180,094	202,227	252,595	303,088	352,190
G	148,826	179,702	200,947	250,340	300,728	350,258
N	150,012	180,142	199,621	252,089	301,823	353,659
Air Test Group						
K	150,017	180,134	200,080	251,383	300,933	350,504
H	150,274	179,735	201,534	250,033	301,904	351,711
P	148,206	177,768	199,740	253,121	304,348	354,835
J	149,529	180,258	201,021	250,708	301,808	351,732
Q	152,275	181,336	198,603	250,624	301,861	354,020

Weights L, M, N, K, P, Q completed 5/9/66
F, G, H, J completed 8/15/68

APPENDIX II
CONTROL OF PRELOAD

A. BEARING STICKOUT DETERMINATION

Definition: Bearing stickout is defined as the axial distance the face of the inner race will extend beyond the face of the outer race upon the application of axial pressure. The stickout is a measure of approximately one-half of the axial freeplay.

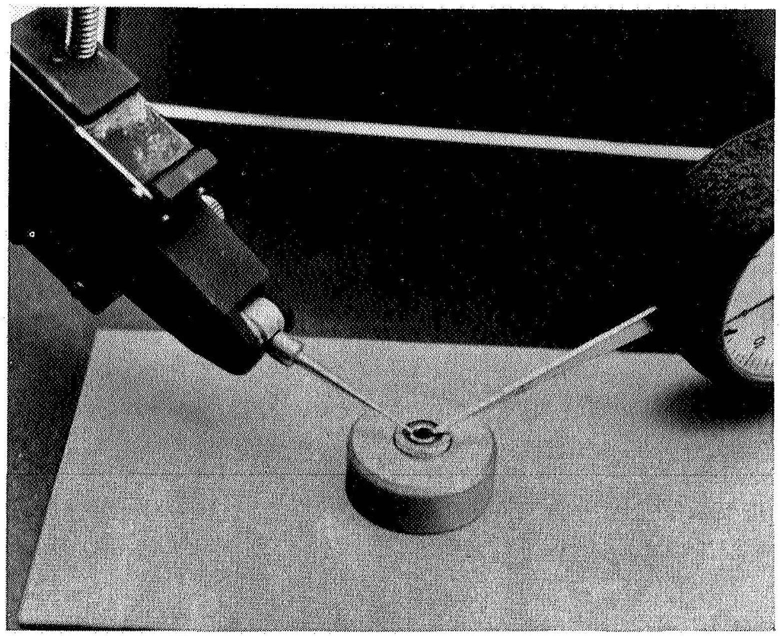
This axial displacement occurs for two reasons. First, the radius of curvature of the raceways is slightly larger than that of the balls. Second, radial freeplay (0.0006 to 0.0008" for the PTFE-MoS_2 retainer bearings used in the initial series of tests) is provided in order to have running clearances.

The manufacturer of the bearings evaluated in the preliminary tests describe herein utilizes a special apparatus to measure stickout. Its operation is describe as follows: The outer race of the bearing is held vertically fixed while a center shaft which is connected to a mechanical indicator is attached to the inner race. The gauge is zeroed when the bottom faces of the inner and outer races (on the cutaway side of the inner race) are parallel. Then an 8 ounce vertical load is added to the inner race from above to displace the cutaway side downward. This downward force must overcome 4 to 5 ounces of indicator force so the stickout reading obtained represents the effect of 3 to 4 ounces of net displacing force. This is considered to be a very light force; very close to the same displacement occurs when the inner race is subjected only to gravitational force.

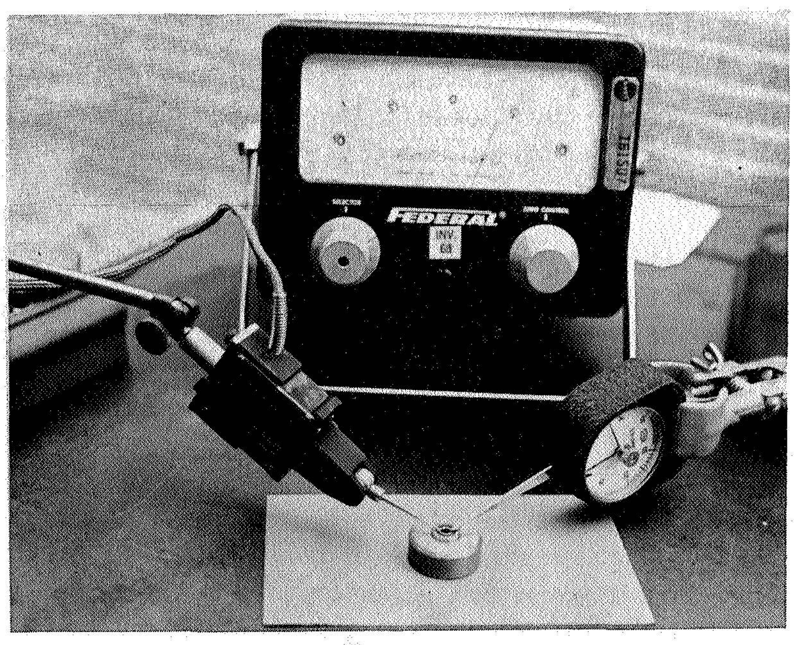
One may refer back to Figure 4 under SELECTION OF FAILURE CRITERION to see how the sum of the two stickouts is taken up by the outer race spacer. The manufacturer's method gives the "true" stickout – and this is the one used in honing down the spacer by the formula discussed in Part B of this appendix. However, it is deemed advisable to recheck all the stickout values supplied to make sure that no errors have occurred.

To accomplish a recheck of each test bearing's stickout, a small circular fixture was made (see photographs in Figure 17) to hold the outer race of the bearing parallel to the surface plate. A center bore in the fixture allows the bearing's inner race to be unrestrained. A precision electronic indicator gauge is used in conjunction with a gram force scale to make the actual measurements.

The procedure is to place the tip of the electronic indicator gauge arm on the inner race of the bearing. This applies 15 grams of force. The top of the



(a) View showing bearing in holding fixture with electronic and mechanical gauges in place on the inner race



(b) Close-up view of the above setup

Figure 17—Apparatus for checking bearing stickout

gram scale arm is placed on the inner race 180 degrees opposite the indicator arm's point of contact and pressure is applied until the scale reads 15 grams. While the inner race is in this level position, the electronic gauge is zeroized. The holding fixture is then moved radially outward from the electronic gauge arm until the tip rests on the outer race. The outward displacement of the arm's tip is read from the electronic gauge dial to the nearest 0.0001". This same procedure is repeated at angular positions 120 and 240 degrees from the original radial position and the three readings averaged to obtain the "stickout" value.

The word stickout is placed in italics because this in-house procedure leads to a "stick-in" value instead. The two values are the same if thicknesses are the same and raceways are exactly centered in their respective races. Because of the small tolerance zones permitted in manufacture, stick-in and stick-out are generally slightly different. However, these differences are insignificant and the above described method of rechecking the factory supplied values of stickout is good enough to detect any significant error in the original readings, box mis-labeling, etc.

These rechecks which are referred to as MSB stickout readings are included on most of the summary data sheets for the Preliminary Tests in Appendix III.

B. HONING AXIAL SPACER

The formula for thickness and the general techniques employed have been discussed under Section B of PRELOAD CONTROL. The only additional information provided here concerns a description of the laboratory techniques used to hone down axial spacers to a uniform width.

A standard laboratory Handi-Met wet sanding fixture is utilized with silicon-carbide paper to hone down the standard brass shims (0.003, 0.004, or 0.005"). If several ten-thousandths of an inch have to be removed, #320 grade wet paper is used initially. The paper is wet down and the circular shim moved across the paper in a circular motion on the tip of the second or third finger. Periodically the thickness is checked around the circumference at three points approximately 120 degrees apart with a precision micrometer. Additional polishing is accomplished at the thicker places until the thickness is relatively uniform. Final polishing is accomplished with #600 grade wet paper until the total thickness variation around the circumference is less than 0.0001".

For support bearings the same techniques are used to produce a differential width between the inner and outer race spacers. In fact, on the magnet end the bearings are flanged and there is no outer race spacer. Precision measurements must be made to determine the distance between the outer races and then the inner race spacer is made shorter by the sum of the stickouts less 0.0004".

For the support bearings this spacer width adjustment is final unless some unusual looseness or tightness is noted in the turning of the shaft upon applying the specified 28 oz-in. to the fastening nut. For test bearing pairs further adjustments in spacer thickness are made until all five test weights pass the Swing Test which is covered in detail in the next section of this Appendix.

C. SWING TEST

This laboratory procedure was developed to insure that uniform preloading conditions exist on each test weight out of the five assembled in a fixture for a given test and from one test assembly to the next. The procedure described in the succeeding paragraphs begins with the assembly of a single test weight and proceeds to a final adjustment of a complete fixture prior to beginning a test.

(1) Single Weight Test: The selected pair of bearings are installed in a test weight using a spacer of the thickness calculated by the formula:

$$\text{Thickness} = \text{Stickout}_1 + \text{Stickout}_2 - 0.0003$$

where all dimensions are to the nearest 0.0001 in inches.

The screws joining the two side plates of the test weight are tightened with all additional hardware in place. The assembled test weight is placed on a shaft with spacers designed to come in contact with the inner races of the test bearings. The end nut of this shaft is torqued to a 28 oz-in. reading.

With the shaft held in a horizontal position, the test weight is rotated upward to a 90 degree position from the bottom dead center. It is then released. The return to near the original 90 degree position is counted as the fast swing. Upon each subsequent swing, the amplitude decays. Each cycle is counted until the test weight ceases oscillating. (Note – the final swings are of very small amplitude.)

The number of swings established as a standard for beginning a test is 25±2.* However, when making the first try with the spacer sized by the formula, either a greater or lesser number of swings may occur. If the number of swings is too large, the test weight must be disassembled and a spacer several ten-thousandths of an inch thicker installed. If too few swings occur, the present spacer must be honed down another ten-thousandth or two. Following the spacer change, the Swing Test is repeated. This process is repeated until the test weight meets the 25±2 swings standard.

*This standard was adopted following the set-up of Vacuum Test I. The previous standard had been 25±5 swings.

(2) Assembled Fixture Test: When each of the five test weights have passed the swing test individually, then all of them are installed in the test fixture and another swing test performed on each test weight while assembled in the fixture. The assembled axial loading of the inner races takes into account all the particular characteristics of the individual parts. If one or more test weights do not pass the Swing Test, then the given test weights are removed and their spacers modified as required. This process is continued until all five tests weights individually swing an average of 25±2 times while assembled on the test shaft. Upon achievement of this goal, the fixture is ready for test.

APPENDIX III
DATA FROM PRELIMINARY TESTS

This appendix consists of copies of the original summary data sheets for the five preliminary tests performed with the LOAD-LIFE bearing test system. These tests were as follows:

4-1/4 oz - 600 rpm Test

2 oz - 3600 rpm Tests:

- Air Test I
- Vacuum Test I
- Air Test II
- Dry Nitrogen Test

GODDARD SPACE FLIGHT CENTER

SERVICE REPORT
STRUCTURAL AND MECHANICAL APPLICATIONS SECTION

ORIGINATOR	PROJECT	JOB ORDER NUMBER	REQUEST NUMBER
B. W. Ward	Mechanical Elements in Space	723-129-03-13-08	1200-84

4-1/4 oz. Air Test – 600 rpm

Started 7/21/65

Disk	Front Brg #	Back Brg #	Factory Stickout Front	Factory Stickout Back			Shim Thick-ness		MSB Stickout After Test Has Been Run Front	MSB Stickout After Test Has Been Run Back
6	1	2	.00175	.0017			.00305		.0012	.0010
7	3	4	.0025	.0022			.0043		.0016	.0015
8	5	6	.0021	.00255			.00425		.0015	.0014
9	7	8	.00205	.0017			.00335		.0020	.0015
10	9	10	.0025	.00195			.00405		.0016	.0013

Swing test not developed when this test was started.

This test ran for 22,960 hours without any failures

720-3 (10/65)

James L. Wall 7-21-65
(Signature) (Date)

GODDARD SPACE FLIGHT CENTER

SERVICE REPORT
STRUCTURAL AND MECHANICAL APPLICATIONS SECTION

ORIGINATOR	PROJECT	JOB ORDER NUMBER	REQUEST NUMBER
B. W. Ward	Mechanical Elements in Space	723-129-03-13-08	1200-84

Two ounce Air Test – 3600 rpm
Fixture No. 1

Air Test I
Started 9/7/66

Disk	Front Brg #	Back Brg #	Factory Stickouts Front	Factory Stickouts Back			Shim Thick-ness		MSB After Stickouts Front	MSB After Stickouts Back
A	21	22	.0020	.0020			.0038		.0060	.0052
B	23	24	.0025	.0031			.0052		.0052	.0039
C	25	26	.00215	.0020			.0038		Apart	.0086+
D	27	28	.0021	.0020			.0036		.0069	.0086
E	29	30	.0022	.0023			.0042		.0056	.0060

Swing Test

Disk	Avg. No. Swings		Individual Trials		Hours to Indicate 90,000 mg. mm		Hours to Indicate 180,000 mg. mm	
A	19		20,20,18,19,19		204			
B	21		21,19,22,21,21		236			
C	27		27,27,27,28,27		163		231	
D	19		19,19,19,19,19		189			
E	19		19,20,20,19,19		219			

720-3 (10/65)

James L. Ward 4-03-67
(Signature) (Date)

GODDARD SPACE FLIGHT CENTER

SERVICE REPORT
STRUCTURAL AND MECHANICAL APPLICATIONS SECTION

ORIGINATOR	PROJECT	JOB ORDER NUMBER	REQUEST NUMBER
B. W. Ward	Mechanical Elements in Space	723-129-03-13-08	1200-84

Two ounce Vacuum Test – 3600 rpm
Fixture No. 2

Vacuum Test I
Started 8/15/66

Disk	Front Brg #	Back Brg #	Factory Stickouts Front	Factory Stickouts Back	MSB Before Stickouts Front	MSB Before Stickouts Back	Shim Thick-ness		MSB After Stickouts Front	MSB After Stickouts Back
R	31	32	.0032	.0025	.00307	.00251	.00515		.0065	.0070
S	33	34	.0023	.00246	.00246	.00233	.0040		.0039	.0060
T	35	36	.0022	.00305	.00216	.00298	.0052		.0060	.0060
U	37	38	.0019	.0024	.00204	.00251	.0040		.0035	.0039
V	77	58	.0019	.0028			.0040		Apart	Apart

Swing Test				Hours to Indicate					
Disk	Avg. No. Swings			90,000 mg. mm		180,000 mg. mm			
R	19			1990					
S	22			2818					
T	28			2865					
U	24			3129					
V	20			2889		3127			
Avg.				2738					

720-3 (10/65)

James L. Ward 4-03-67
(Signature) (Date)

GODDARD SPACE FLIGHT CENTER

SERVICE REPORT

STRUCTURAL AND MECHANICAL APPLICATIONS SECTION

ORIGINATOR	PROJECT	JOB ORDER NUMBER	REQUEST NUMBER
B. W. Ward	Mechanical Elements in Space	723-129-03-13-08	1200-84

Two ounce Air Test – 3600 rpm
Fixture No. 1 in Chamber at Atmospheric Pressure

Air Test II
Started 1/05/67

Disk	Front Brg #	Back Brg #	Factory Stickouts Front	Factory Stickouts Back	MSB Before Stickouts Front	MSB Before Stickouts Back	Shim Thick-ness			
A	41	42	.00205	.0020	.0026	.0026	.0039			
B	43	44	.0021	.0024	.00225	.0024	.0037			
C	45	46	.0021	.0020	.0026	.0027	.0043			
D	47	48	.0019	.0020	.0026	.0025	.0035			
E	49	50	.0023	.0017	.0021	.0025	.0038			

Swing Test						Hours to Indicate			
Disk	Avg. No. Swings		Individual Trials			90,000 mg. mm		180,000 mg. mm	
A	23		25, 24, 21, 22			237		388	
B	28		29, 29, 27, 27			280			
C	23		26, 23, 21, 21			*			
D	27		31, 29, 24, 24			355			
E	23		26, 22, 23, 22			*			

*Still running when test was stopped at 429:30 hr.

720-3 (10/65)

4-03-67

(Signature) (Date)

GODDARD SPACE FLIGHT CENTER

SERVICE REPORT
STRUCTURAL AND MECHANICAL APPLICATIONS SECTION

ORIGINATOR	PROJECT	JOB ORDER NUMBER	REQUEST NUMBER
B. W. Ward	Mechanical Elements in Space	723-129-03-13-08	1200-84

Two ounce Nitrogen Atmosphere Test – 3600 rpm
Fixture No. 1

Nitrogen Test
Started 4/13/67

Disk	Front Brg #	Back Brg #	Factory Stickouts Front	Factory Stickouts Back	MSB Before Stickouts Front	MSB Before Stickouts Back	Shim Thick-ness			
A	51	52	.0022	.0021	.00238	.0026	.0043			
B	53	54	.0019	.0017	.0022	.0019	.0031			
C	55	56	.0019	.0025	.0024	.0028	.0036			
D	57	59	.0019	.00185	.0020	.0022	.0030			
E	60	61	.00175	.00225	.0020	.0020	.00345			

Swing Test						Hours to Indicate	
Disk	Avg. No. Swings		Individual Trials			90,000 mg. mm	180,000 mg. mm
A	27		27, 26, 29, 26, 27			694	1159
B	26		27, 26, 26, 27, 26			988	
C	27		26, 27, 28, 26, 27			*	
D	25		25, 25, 24, 25, 25			*	
E	26		27, 26, 25, 27, 28			*	

*Still running when test was stopped at 1159 hr.

720-3 (10/65)

James L. Wall 6-2-67
(Signature) (Date)

APPENDIX IV
AUTOPSY REPORT ON TEST BEARINGS

Included in this section are reproductions of the introductory remarks and summary information on the autopsies of the bearings tested in three of the four 2 oz - 3600 rpm tests. In addition the individual post mortem information, photographs, and torque traces for a representative bearing from Air Test I is included. In the complete report (New Hampshire Ball Bearings, Inc. LWR No. 20K) this information is included for each bearing autopsied. This report is available for loan in the Mechanical Systems Branch for those readers who need complete autopsy information.

The following listing is given to clear up any misunderstandings that may arise between the identifying numerals used in this contractor report and those used in the text of this document.

Text:	Report LWR No. 20K
Air Test I	I - Air Test
Vacuum Test I	II - Vacuum Test
Air Test II	III - Air Test

BEARING EXAMINATION SECTION

INTRODUCTION

The bearings analyzed are from a Life Test series conducted at National Aeronautics and Space Administration, Goddard Space Flight Center to obtain base line data for vacuum application. Five pairs of SR2-6B15K68 bearings (5813 Duroid Retainers) with each pair mounted in plates weighing two ounces were tested on a spindle suspended between two sets of guide bearings and driven at 3600 rpm through a magnetic coupling. This test was conducted twice in an air atmosphere and once in a hard vacuum test chamber. The bearings are set up with outer ring spacers, inners clamped, so that no end play exists. Actually a positive preload is often the case. The outer rings are clamped between two steel plates, each approximately 3/4 × 3 × 1/8" thick. The bearings are mounted off-center of these plates so that torque is indicated when the plate is turned from rest position. These plates are also referred to as discs, due to an early design. Life listed on the detail sheets is the time for a pair to indicate 90,000 mgmm of torque.

ANALYSIS PROCEDURE

The returned bearings were examined as received for general condition, measured for radial play, torque tested on the Mil-Std-206 Running Torque Tester, and then disassembled to determine the cause of failure.

RESULTS

Table No. 1 is a comparison of the stickout and radial play before tests and the radial play after tests. The very high readings are an indication of the wear that has taken place between the rolling elements of the bearings. Tables No. 2 through 4 are a summary of the Mil-Std-206 running torque for Test I through III, respectively. This includes before and after testing and after wash testing. The missing readings indicate that the bearings were not operable in some manner, either they were disassembled or had an excessively high torque.

CONCLUSION

Both air tests (I and III) show the results of loading high enough to swage a wider raceway, resulting in a burr between the raceway and land of a number of bearings. The extent of the misalignment in both air tests suggests the loading occurred as a dynamic pulsing of the test bearing mounting plates.

The vacuum test (II) has some bearings with indications of dynamic loading, but these bearings have far less damage than general in the air tests. Some of the vacuum test bearings actually showed no sign of heavy loading.

The difference in life between the air and vacuum tests appears to reflect a difference in dynamic loading rather than lubrication efficiency.

I - AIR TEST

All bearings are in very poor condition even though only one pair reached a failure torque of 180,000 mgmm. The bearings have a high radial play after tests with the pair in the center of the spindle receiving the most wear.

There is no visible lubricant in the form of a grease or oil in these bearings. They are lubricated by the controlled wearing away of the Duroid retainer which is deposited on the balls and raceways to form a film and prevent metal to metal contact.

Disc A contained Bearings No. 21 and 22, front and rear respectively. The front bearing did not have any appreciable contamination, but the rear contained a small amount of reddish powder and shiny metallic particles on the interior surfaces.

Table No. 1
Radial Play and Stickout

Brq. No.	Before Tests		After Tests
	Radial Play	Stickout	Radial Play
I			
21	.0007"	.002"	.0018"
22	.00075"	.002"	.0017"
23	.0007"	.0025"	.0011"
24	.0006"	.0031"	.0012"
25	.0007"	.00215"	.0036"
26	.0007"	.002"	.0036"
27	.00075"	.0021"	.0025"
28	.00065"	.002"	.0012"
29	.0007"	.0022"	.0025"
30	.0007"	.0023"	.0020"
II			
31	.0007"	.0032"	.0016"
32	.0007"	.0025"	.0017"
33	.0007"	.0023"	.0010"
34	.0007"	.00246"	.0015"
35	.0006"	.0022"	–
36	.0007"	.00305"	.0014"
37	.0006"	.0019"	.0010"
38	.0007"	.0024"	.0010"
58	.00065"	.0028"	–
77	.00065"	.0019"	–
III			
41	.0007"	.00205"	.0040+"
42	.0007"	.002	.0040+"
43	.0007"	.0021"	–
44	.0007"	.0024"	.0032"
45	.0007"	.0021"	.0014"
46	.0007"	.0020"	.0014"
47	.0007"	.0019"	.0026"
48	.0007"	.002"	.0020"
49	.0007"	.0023"	.008"
50	.0007"	.0017"	.008"
SFR3-1			.009"
SR3-2			.006"

Table No. 2
Mil-Std-206 Running Torque

I - Air Test

Bearing	Before Tests	After Tests Before Wash	After Tests After Wash
A-Front No. 21	ART 2000 MRT 4000	12,000 56,000	– –
A-Back No. 22	ART 1500 MRT 4000	7,000 27,000	2,750 8,750
B-Front No. 23	ART 1500 MRT 5500	10,000 32,000	2,000 8,500
B-Back No. 24	ART 2000 MRT 4000	10,000 36,000	2,000 6,000
C-Front No. 25	ART 1500 MRT 3500	– –	– –
C-Back No. 26	ART 1250 MRT 4250	10,000 36,000	– –
D-Front No. 27	ART 1500 MRT 5500	– –	– –
D-Back No. 28	ART 2000 MRT 12,000	– –	– –
E-Front No. 29	ART 5500 MRT 7000	10,000 42,000	5,500 18,500
E-Back No. 30	ART 1250 MRT 3750	7,000 22,000	2,000 27,000
		Readings in mgmm	

Table No. 3
Mil-Std-206 Running Torque

II - Vacuum Test

Bearing	Before Tests	After Tests Before Wash	After Tests After Wash
R-Front No. 31	ART 1750	6,000	3,000
	MRT 5250	38,000	27,000
R-Back No. 32	ART 1750	6,000	–
	MRT 5250	62,000	–
S-Front No. 33	ART 1500	6,000	2,000
	MRT 7500	38,000	35,000
S-Back No. 34	ART 2000	3,000	1,500
	MRT 8000	20,000	19,000
T-Front No. 35	ART 2000	–	–
	MRT 6000	–	–
T-Back No. 36	ART 2000	2,000	2,000
	MRT 6000	56,000	39,000
U-Front No. 37	ART 2000	2,000	6,000
	MRT 4000	48,000	50,000
U-Back No. 38	ART 1500	4,000	2,000
	MRT 3500	29,000	28,000
V-Front No. 77	ART 1500	–	–
	MRT 3000	–	–
V-Back No. 58	ART 2000	–	–
	MRT 6000	–	–
		Readings in mgmm	

Table No. 4
Mil-Std-206 Running Torque

III - Air Test

Bearing	Before Tests	After Tests Before Wash	After Tests After Wash
A-Front No. 41	ART 2000 MRT 3500	– –	– –
A-Back No. 42	ART 1750 MRT 3750	– –	– –
B-Front No. 43	ART 1500 MRT 5500	10,000 50,000	– –
B-Back No. 44	ART 1500 MRT 6000	10,000 42,000	– –
C-Front No. 45	ART 2000 MRT 4000	14,000 48,000	3,500 25,500
C-Back No. 46	ART 2500 MRT 8000	15,000 59,000	2,500 9,500
D-Ront No. 47	ART 2500 MRT 4500	16,000 56,000	9,000 29,000
D-Back No. 48	ART 2500 MRT 4500	13,000 44,000	3,000 19,000
E-Front No. 49	ART 1500 MRT 5500	10,000 60,000	2,000 6,000
E-Back No. 50	ART 1500 MRT 6000	20,000 60,000	7,000 35,000
		Readings in mgmm	

Table No. 5
Mil-Std-206 Running Torque

Support Bearings

Bearing	Before Wash	After Wash
SFR3B15		
Inner	ART 18,000 MRT 56,000	5,000 18,000
Outer	ART 20,000 MRT 76,000	3,500 9,500
SR3B15		
Inner	ART 8000 MRT 30,000	3,500 7,500
Outer	ART 3000 MRT 14,000	4,000 7,800
		Readings in mgmm

These bearings are all snap-assemblies so it is normal to find light brinell marks on the relieved land from assembly. The raceways in both the bearings have a frosted finish and what appears to be dirt indents on the surface of the parts. Ball wear is light and the path made by the balls is a frosted band that is misaligned. Flaking has started in the raceway of Bearing No. 22. Fretting corrosion appears in the bore.

The balls are shiny but have light wear and scratches. The retainer ball pockets have received moderate wear which has enlarged the holes in the Duroid. Burrs are visible at the OD of the ball pockets on the front bearing and on the rear, one ball pocket has a dent across the retainer.

Disc B, Bearings No. 23 and 24, had a large amount of reddish powder on all internal surfaces. The ball path in the raceways was worn and misaligned in both bearings. The front Bearing No. 23 had light burrs at the regular land and flaking in the raceway, which appears to be a product of surface fatigue. There is fretting corrosion in the bores of Bearings No. 23 and 24.

The balls are shiny but the surface is lightly frosted from wear and has a few light scratches. Wear on the retainer is moderate and has caused the ball pockets to become enlarged. ID wear on the retainer locating surface is light.

Disc C is at the center of the shaft and contains Bearings No. 25 and 26 which ran for 163 hours. Both bearings are in the same general condition with a reddish-brown powder residue, probably iron oxide, on all internal surfaces. The wear, flaking, and misalignment in this pair of bearings are so severe in the raceway that it has reduced the regular inner lands, both front and rear, to approximately one-half its original width at one area.

The outers also have heavy wear and flaking with the misaligned ball path swaged into one land causing burrs where the metal has been forced over. There is fretting corrosion in the bore.

All of the balls are shiny with light wear and scratches. Those from the front bearing have a hard, brown residue adhered to the surface. Severe wear has taken place in the retainer ball pockets enlarging the hole and leaving burrs on the edges.

Disc D ran for 189 hours with Bearings No. 27 and 28 and at the end of that time, they were filled with metallic wear particles and a reddish powdery debris.

Examination of the components revealed severe misalignment on both the raceways of the inner and outer of Bearing No. 28 with small burrs forming where the land has been rolled back. Bearing No. 27 had misalignment and wear across the raceways but the land was not worn as badly. Fretting corrosion has taken place in the bore of both bearings.

The balls from Bearing No. 27 were frosted and had light wear. Those from Bearing No. 28 were shiny but had light wear. Both retainers had heavy wear which enlarged the ball pockets.

Disc E, the last one on the shaft, was supported by Bearings No. 29 and 30. These ran for 219 hours before they were removed from test and then both front and rear bearings were full of the reddish-brown powdery residue which was in most of the other discs. The front bearing, No. 29, had a misaligned ball path but the rear bearing did not. The raceways were very similar with heavy flaking having taken place.

The balls are shiny with light scratches and wear on the surface. The retainer ball pockets are enlarged with wear on the retainer of Bearing No. 30. Duroid flakes are visible on the ID next to the face and in three ball pockets.

New Hampshire Ball Bearings, Inc.
Peterborough, N.H.

Ref. No. LWR 20K | Date 4/11/67 | Tech EBC | Brg. No. A-Front No. 21
Brg. Size SR2-6B15K68 | Requested By W. W. Ware
Customer NASA -9624 | Application I-Air Test
Speed 3600 rpm | Lube Duroid 5813 Retainer | Amt.
Load | Environment
Criterion for Failure | Measured Radial Play K18
Remarks | Life 204 hours

VISUAL INSPECTION RESULTS
Lubricant None visible | Amt. | Condition
Contamination Numerous white particles on the surface of the bore.
Remarks

INNER: Fit Marks, Bore Axial & circular | Retainer Wear None
Brinell Yes | Location Edge of track and on relieved land | Land Appearance Shiny, several nicks
Ball Path, Location Slightly misaligned, light frosted wear and flaking.
Appearance of Raceway Frosted wear across whole track. Shiny burrs are present on the edge of regular land.
Remarks Fretting corrosion is heavy in the bore

OUTER: Fit Marks, OD Axial - numerous | Retainer Wear Yes, both lands
Brinell None visible | Location | Land Appearance Shiny, grinding marks
Ball Path, Location Slightly misaligned, frosted light wear
Appearance of Raceway Many small nicks on the surface of the track.
Remarks

BALLS No. 6 | Brinell None | Flaking None | Scratches Light
Finish Shiny, frosted, light wear | Dirt Indents None
Remarks

RETAINER: Type Machined | Material 5813 Duroid
Wear, Ball Pockets Heavy wear on sides and bottom
Wear, Locating Surface OD | Finish Dark grey
Deformation Yes, enlarged ball pockets
Remarks Burrs of the material are visible on the OD edge of the ball pockets.

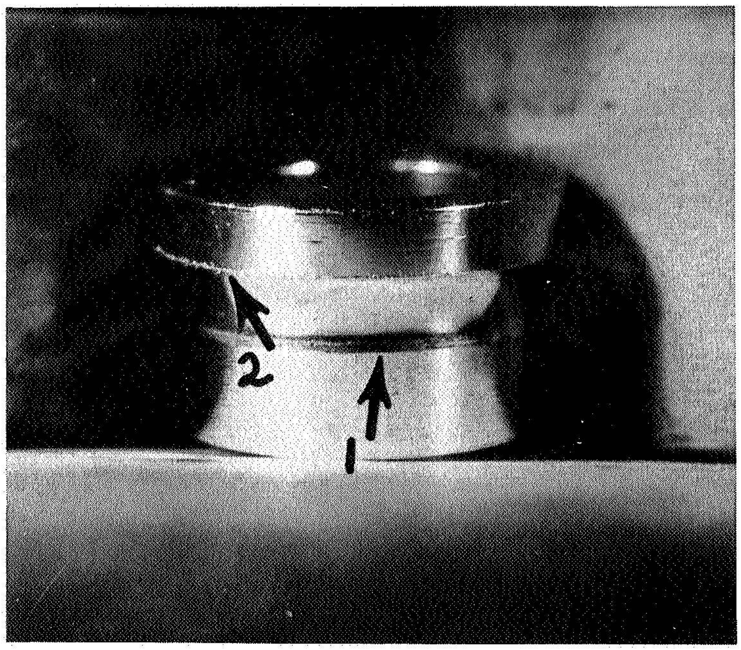


Figure 1

Inner Ring

1. Slightly misaligned, lightly frosted ball path.

2. Shiny, small burrs on the edge of the regular land.

Less than X10

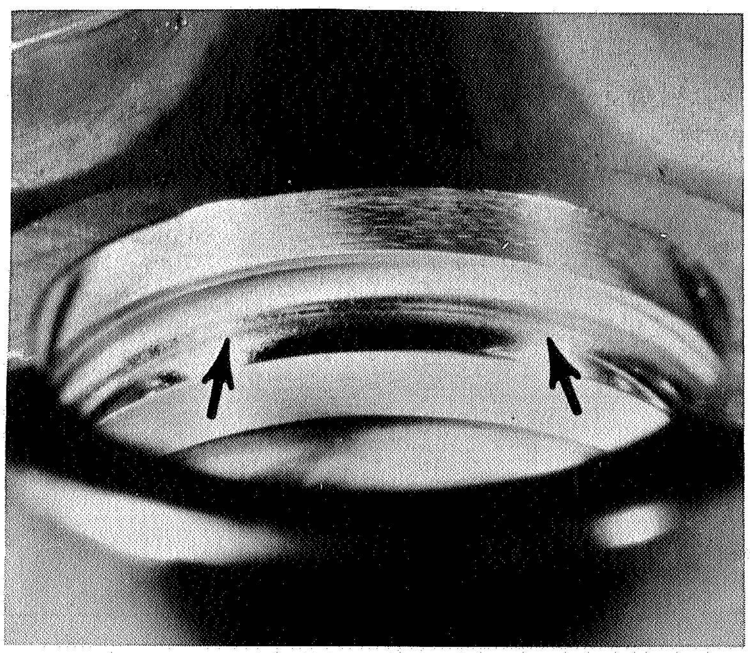


Figure 2

Outer Raceway

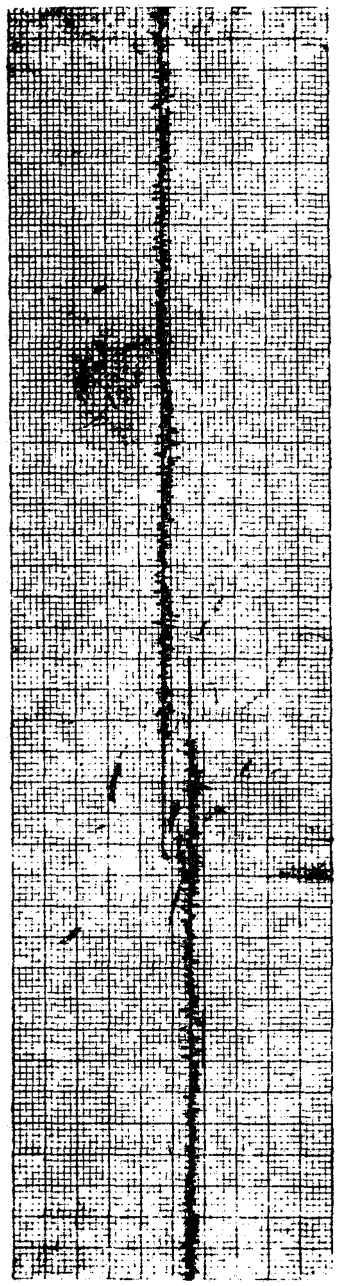
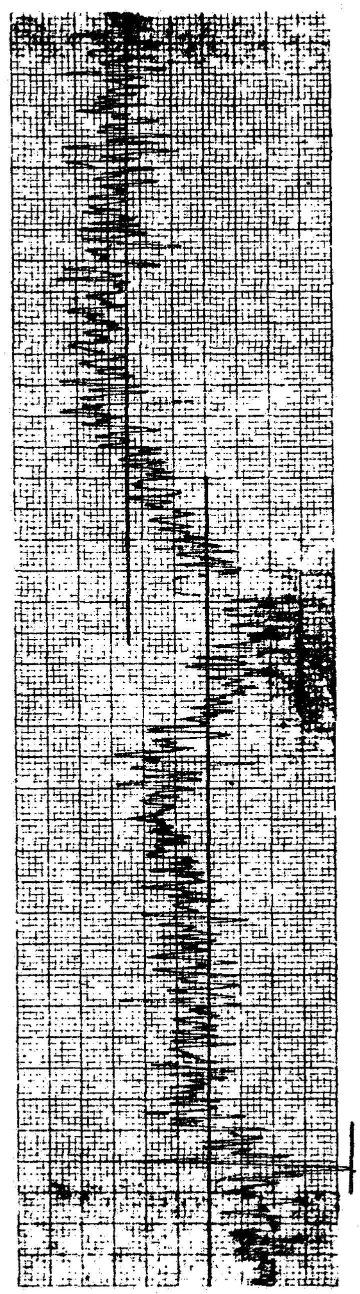
Slightly misaligned, frosted ball path.

Less than X10

Figure 18–Post mortem raceway condition of test bearing A–Front No. 21 following Air Test I. Reprint from autopsy report, LWR No. 20K.

NEW HAMPSHIRE BALL BEARINGS, INC. - ENGINEERING DEPARTMENT

LWR 20K Brg. A-Front - No. 21 Date 4/17/67 Tech. EBC

Mil-Std-206	Mil-Std-206	Mil-Std-206
Before Tests	Before Wash	After Wash
		
Attenuation Set. X5	Attenuation Set. X10	Attenuation Set.
Load 400 gram	Load 400 gram	Load ____ gram
ART 2000 mgmm	ART 12,000 mgmm	ART ____ mgmm
MRT 4000 mgmm	MRT 56,000 mgmm	MRT ____ mgmm
1000 Scale mgmm/div.	2000 Scale mgmm/div.	Scale mgmm/div.

NOT REPRODUCIBLE

II - VACUUM TEST

Disc R contained Bearings No. 31 and 32 which ran for 1990 hours in a hard vacuum before indicating 90,000 mgmm. These two bearings contained many small metallic particles resembling a shiny wear debris on all internal surfaces. The front bearing also had small white particles which may have come from handling after tests.

Raceway wear is across the whole track and has rolled back the inner land on the front bearing causing burrs to form. Misalignment was noted on the outer of the rear bearing. In both bearings there is a hard, glazed, black, varnish-like residue distributed over the raceways which will flake off when probed with a pin. A shiny band appears on the faces of Bearing No. 32 where something rubbed against it, possibly a shim or spacer.

Heavy wear has taken place on each of the balls and the surface is starting to flake. Retainer wear is moderate to heavy in the ball pockets which are enlarged. There is normal light wear on the OD retainer locating surface.

Disc S had a life of 2818 hours. The two bearings, Nos. 33 and 34, had metallic wear particles throughout. Misaligned raceway surface wear is heavy with a shiny, hard, black residue that was adhered to various components of both bearings. There are also numerous pits in the raceway on the back bearing. A circular line is around the face of both inners, probably from an adjacent spacer.

The balls from the front bearing had a hard, varnish-like residue, dirt indents, and scratches. Those from the rear bearing, No. 34, had light wear, dirt indents, and scratches but the surface still had a high gloss. Retainer wear is moderate to heavy in the ball pockets which are enlarged.

Disc T with 2865 hours life contained Bearings No. 35, front, and No. 36, rear. There were numerous white particles and metallic debris throughout both bearings. Heavy wear has taken place as a misaligned ball path causing burrs to form on the regular land of the front and rear bearing. The raceway surfaces also have a hard, black, glazed residue adhered to them but it will flake when scraped with a pin.

The balls have light wear and scratches. Retainer wear is moderate with enlarged ball pockets.

Disc U contained Bearings No. 37 and 38 which ran for 3129 hours before indicating 90,000 mgmm. Again there was metallic wear debris, but only in the front bearing. The rear bearing had fibers and white particles. Both had a misaligned ball path and a glazed, black, varnish-like residue. In this set of

test bearings, all the bearings from each disc are brinelled on the relieved land, caused by assembly and disassembly methods.

All balls are shiny with light wear. The retainers have moderate wear in the ball pockets causing them to become enlarged. Flakes of retainer material are present on the ID surface between three ball pockets.

Disc V contained Bearing No. 77 in the front and No. 58 at the rear with a life of 2889 hours. There was metallic debris on the interior of both bearings. The inner of No. 77 was fractured in six sections. Wear on this part was across the inner raceway with the misaligned ball path entering the land, leaving heavy rolled burrs. The outer ring of this bearing also had rolled burrs on the land and heavy wear in the raceway.

Bearing No. 58 had heavy wear in the raceway and a wide ball path from one land to the other with burrs formed on the regular land. Metallic debris was pressed into the raceway.

All the balls were missing from the front bearing. Those that were in the rear bearing have severe wear. The retainer ball pockets have severe wear and are enlarged. The retainer in Bearing No. 58 has a split between two ball pockets.

III - AIR TEST

Disc A had an inner rotational speed of 3600 rpm and contained Bearings No. 41 and 42 which ran for 388 hours before the failure torque of 180,000 mgmm was reached; 90,000 mgmm of torque was indicated at 237 hours. A brown powdery residue and metallic debris were on all internal surfaces. The ball path of both bearings was misaligned and flaking is present where metal fatigue has taken place. Burrs have formed at the corner of the regular land and race from this swaging and wear. There is fretting corrosion in both bores.

The balls have wear, scratches, and dirt indents. The retainers have enlarged ball pockets (the retainer from the rear bearing has worn oval shaped) with burrs at the outer edges, bore, and OD. There was slippage on the OD of the front bearing and bore of the rear.

Disc B ran for 280 hours before indicating 90,000 mgmm torque and contained Bearings No. 43 and 44, which were contaminated with metallic debris on all internal surfaces from severe misalignment, flaking, and wear in the raceway. Metallic burrs were at the corner of the land and raceway on the front bearing only.

The balls all had light wear and scratches. Retainer wear in the ball pockets was severe and all had been enlarged. Light burrs were on the edge of the ball pockets of the No. 44 retainer.

Disc C, like disc E, was taken off the test at 430 hours, but had not indicated 90,000 mgmm torque. Bearings No. 45 and 46 were filled with a red powdery residue and metallic debris which was on all interior surfaces. Misalignment has taken place in the moderately worn raceways leaving one area shiny and the remainder frosted. All the rolling surfaces are filled with dirt indents from the balls pressing the generated wear products into the raceway.

Ball wear is light to moderate with indents and scratches on the surface. Moderate to heavy ball pocket wear has taken place and all are enlarged. One pocket from retainer No. 46 was fractured at the face.

Disc D which contained Bearings No. 47 and 48 ran for 355 hours before indicating 90,000 mgmm torque. Both bearings were filled with metallic debris and a brownish powder. The raceways were misaligned as indicated by the ball path wear, which was flaked, and burrs had formed on some of the parts where the lands were forced back. Fretting corrosion has taken place in the bore and on the OD of the front bearing, but only in the bore of the rear bearing. The balls are lightly frosted, some having light dirt indents and others light scratches. The ball pockets from the front bearing, No. 47, have been enlarged more than the rear ones, but they both have rolled burrs.

Disc E was the last one on the spindle and contained Bearings No. 48 and 49. Reddish-brown powdery residue and metallic wear debris choked this bearing. The only trace of misalignment was on the outer raceway of the front bearing where a frosted band could be found adjacent to the finished surface. Dirt indents also cover the track.

The surfaces of the balls range from shiny to a dull, frosted finish of light wear. All ball pockets in the retainers have been enlarged. There is light OD wear from the locating surface. The front bearing, No. 49, has one fractured ball pocket and the rear bearing has flakes and burrs at the OD of the ball pockets.