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Evaluation of Dry Lubricants and Bearings for Spacecraft Applications

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Instrument-size ball bearings lubricated with bonded dry films or transfer films have been successfully operated for 130 million revolutions during a 9000-h vacuum test at 10^{-9} torr. This test simulated conditions anticipated in unsealed mechanisms of an interplanetary spacecraft. The conditions included prolonged idle periods, as well as extended periods of continuous rotation, with reversals, while the bearings were exposed to hard vacuum. The performance of several bearing–lubricant combinations surpassed that required to complete the projected spacecraft mission. Extensive quality control in the preparation of the bearing–lubricant combinations was necessary to attain the lifetimes achieved. This paper describes the unique test procedures and techniques developed and the test results, which include some new insights into the behavior of transfer film lubricants.

I. Introduction

During preliminary design studies of an interplanetary spacecraft, it became apparent that severe mechanism lubrication problems would have to be overcome to ensure mission success. These included long-term (up to 2 years) exposure of the mechanisms to hard vacuum and intermittent operation interspersed with protracted static dwell periods. Low outgassing would be required both to conserve lubricant during the 2-year mission life and to minimize contamination of optical and thermal control surfaces on the spacecraft. Anticipated temperatures ranged from possible prelaunch thermal sterilization at 250°F (minimum) to extended subzero conditions during interplanetary cruise and subsequent orbital operations. In addition, low and uniform bearing frictional torque would be needed to minimize mechanism design weight and power consumption and to ensure reliable operation.

These requirements were judged too severe for the liquid lubricants used on a majority of earlier space vehicles, unless all mechanisms were provided with hermetic sealing and active thermal control. In hope of avoiding the penalties of increased mechanism weight, complexity, and power consumption which these measures would entail, attention was turned to the possibility of using dry (solid) lubricants. These lubricants generally exhibit extremely low vapor pressure, their lubrication mechanisms are essentially independent of ambient pressure, and they are only slightly affected by temperatures within the range of interest. In addition, operating life is predictable since it is primarily a function of the number of operating cycles, rather than time. Because of these desirable properties, the suitability of a group of commercially available dry lubricants was studied.

A detailed literature search identified several potentially suitable lubricants, but the use of conditions and measurement techniques from which the available data derived had varied widely. In addition, performance of a given lubricant sometimes varied significantly in the different applications or tests reported. Besides the variability of use conditions and batch-to-batch variations in the lubricant and substrate, handling and quality control differences were suspected of having a significant effect on previously observed performance. These factors combined to make uniform comparison and determination of relative life expectancy extremely difficult. Therefore, this test program was established to provide comparable performance evaluation of several of the most promising dry lubricants operating under conditions closely simulating those expected in the planetary exploration spacecraft being studied.

The lubricants selected for evaluation are described in Table 1. They fall into two major categories: (1) lubricants bonded or plated onto the surfaces to be lubricated, and (2) lubricants transferred to the balls and races from the retainer surface bearing operation.

Table 1. Lubricants selected for evaluation

Code	Description
	Group 1: Bonded or plated lubricants
A	Molybdenum disulfide and graphite bonded with sodium silicate
B	Molybdenum disulfide and antimony trioxide bonded with polyimide resin
C	Molybdenum disulfide diffused into a soft, multilayered, plated metallic film
D	Molybdenum disulfide applied by an electrophoretic process
E	Molybdenum disulfide applied by an *in situ* process
F	Tungsten disulfide (modified) applied by a diffusion process
G	Electroplated silver
	Group 2: Transfer film lubricants
X	TFE and molybdenum disulfide, reinforced with glass fibers
Y	TFE reinforced with ceramic filler
Z	TFE coating on metallic retainer

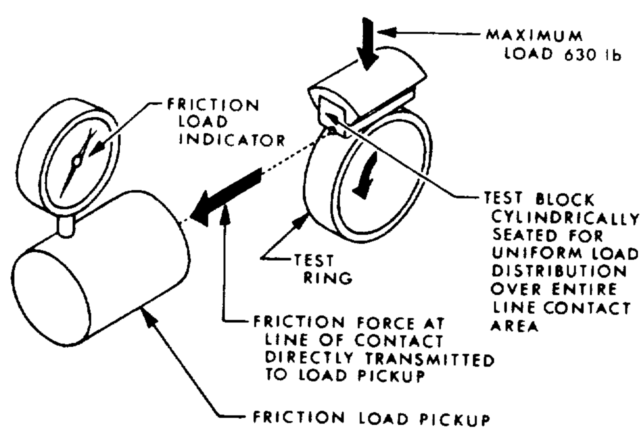
II. Lubricant Analysis Techniques

To ensure future repeatability and usefulness of the lubricant performance data obtained in this test program, techniques for determination of lubricant constituency and uniformity were evaluated. X-ray, emission-spectrographic, electron-probe, and electron and optical microscopic techniques were investigated to determine their usefulness in the quantitative and qualitative evaluation of these lubricants. Such investigations identified areas of usefulness of several techniques. These analytic techniques were supplemented with conventional sliding friction tests in air to determine adhesion and wear resistance of the bonded and plated lubricant films. Results of these tests are shown in Table 2.

Table 2. Sliding wear test results in air using the Alpha Model LFW-1 sliding friction and wear test machine[a]

Lubricant	Lifetime[b], h	
	Range[c]	Average
A	11 to 17	13
B	27 to 35	31
C	13 to 228	77
D	1.4 to 38	18
E	26 to 104	78
F	0.06 to 0.08	0.07

[a]Functional diagrammatic view of the Alpha Model LFW-1 machine is shown below. Friction force sensed by the load cell is shown on the load indicator.



Radial load: 630 lb
Peak Hertz stress: 110 kpsi
Rotational speed: 72 rpm
Ambient conditions: 70°F, 1 atm, 60% RH

[b]Tests were ended when coefficient of friction exceeded 0.1.

[c]Four samples of each lubricant were tested, two from each of two different batches.

X-ray emission techniques have been useful in determining the distribution of molybdenum disulfide by determining the quantity and distribution of sulfur in several of the lubricant coatings. X-ray diffraction techniques have proved relatively ineffective in the determination of the uniformity of lubricant film constituency and consistency, because of the amorphous character of some lubricants and the multilayered nature of others. Emission spectrographic analysis has given useful data for batch constituency determination, but has not proved well-suited to bearing parts.

The electron probe has also shown itself to be an effective tool in quantitative and qualitative determination of the composition and uniformity of solid lubricant coatings. Photographs of each lubricant constituent have been made from the characteristic X-ray spectra generated by scanning the sectioned lubricant sample with a focused high-energy electron beam. This technique has been particularly effective in the examination of multilayered lubricants such as the Code C material.

Electron and optical microscopy were both used in the examination of lubricant coatings. Because of the extremely high magnification (10,000×) of the electron microscope, slight compositional variations in good coatings have widely different appearances and may give the impression of extreme nonuniformity. Fortunately, the ultramicroscopic nonhomogeneity indicated in observed specimens does not necessarily preclude acceptable lubricant performance.

Optical microscopy (400×) proved to be extremely useful in the examination of lubricant coating characteristics. This technique has obvious value for the nondestructive inspection of lubricant coatings to determine surface uniformity and to detect inclusions, porosity, and voids. In addition, it is possible to observe the uniformity and apparent adhesion of fluorocarbon transfer films.

In the inspection of the inner ring of a Code Y bearing, which had previously been run-in in air and disassembled for inspection, evidence was found that significant quantities of gases were entrained within the film. The microscope used had a small vacuum chamber with optical ports surrounding its specimen stage. The lubricant was first examined at atmospheric pressure and room temperature. It had a whitish, translucent, gelatinous appearance. The specimen chamber was then evacuated in preparation for examining the surface with the electron probe. As the pressure was reduced to the 10^{-5} torr level in 3 min, it was noted that the transfer film coating on the bearing race surface began to craze and flex, although the part was not being subjected to thermal or mechanical disturbances. As the crazing process continued, the coating lifted from contact with the race in several areas and ultimately attained a somewhat shriveled appearance.

Observation of this phenomenon led to the conclusion that the transfer film, which had been established on the bearing during run-in in air, contained a quantity of entrained gases and/or moisture which were desorbed in vacuum causing mechanical damage to the film. This observation may indicate a method for attaining a higher degree of reliability in transfer film-lubricated bearings. It is likely that an initial transfer film of greater mechanical integrity, which would not be subject to vacuum-shock deterioration, could be established through initial run-in of the bearing in vacuum.

III. Test Bearing Preparation

For vacuum performance testing of the selected lubricants, the R-4 instrument bearing was chosen as the standard test specimen. Bearing rings and balls were made of AISI 440C steel; the retainer type and material depended on the lubricant used. The Code X and Y bearings (see Table 1) had one-piece retainers machined from the lubricating material. All other bearings had two-piece stainless steel ribbon retainers. The Code Z bearing lubricant was bonded to the retainers only. The other lubricants were applied to the races and the retainers.

By means of a thorough visual inspection of the lubricated parts with a 40× microscope, we eliminated all parts that had incomplete coverage, nonuniformity of surface texture, and apparent inclusions on some parts.

The accepted parts were returned to the bearing vendor for assembly. Balls were selected to achieve a radial clearance of 0.0008 to 0.0011 in. in each bearing. This relatively large clearance was specified to lessen the danger of bearings being jammed by lubricant wear debris.

Visual inspection of the assembled bearings revealed several defects which could have caused failures if they had not been corrected. On two of the ribbon retainers, a holding tab had not been crimped; consequently, it scraped against the outer race during rotation. Also, on several of the Code X and Y bearings, the retainers were not completely deburred. Either of these conditions could

Table 3. Running torque[a] after run-in of dry-lubricated R-4 bearings

Lubricant	Number of samples	Torque, 10^3 mg-mm			
		Avg torque, range	Peak torque, range	Torque for entire sample, avg	Peak torque, avg
A	5	8 to 14	26 to 36	11	32
B	12	2.5 to 11	10 to 42	6	23
C	5	8 to 16	28 to 48	11	37
D	5	6 to 15	20 to 80	10	44
E	4	10 to 30	38 to 50	16	45
F	10	3 to 16	11 to 50	9	30
G	5	6.5 to 35	25 to 125	17	66
X	10	3 to 13	8 to 25	6.3	19
Y	10	2 to 4	7 to 15	3.2	12
Z	10	6 to 14	25 to 44	8.4	34
None[b]	4	1.5 to 2	5 to 25	1.8	10
Grease[c]	5	50 to 200	80 to 200	144	150

NOTE: Dry- and grease-lubricated bearings shown for comparison only, not run or tested further.

[a]Method: MIL-STD 206, 400-g axial load, 0.5 rpm.

[b]Steel ribbon retainer, no lubricant.

[c]GE G-300 silicone grease, phenolic retainer, 25% fill.

have resulted in a jammed bearing through generation of excessive debris in operation.

Prior to subsequent testing, all bearings were run-in in a uniform cycle to ensure a smooth ball track in the case of the bonded or plated lubricants and to ensure that a transfer film had been established on the ball and race surfaces in the case of the transfer lubricants. Each bearing was then tested on a MIL Standard 206 torque tester to determine its running torque. The results of these tests, summarized in Table 3, were used both to determine the range of torque levels which might be expected from dry-lubricated bearings with various lubricant types and to provide screening data for the subsequent selection of bearings for vacuum testing.

IV. Vacuum Test Fixture Design

A vacuum test fixture was designed for endurance testing of bearings under combined radial and axial loads. There are six identical test shafts in the fixture; a diagram of one test shaft is shown in Fig. 1. On each shaft there are two pairs of test bearings, H and L. These are loaded radially, by rotationally balanced cylindrical weights, and axially, by calibrated compression springs concentric with the shaft. The H pair of test bearings supports a 3-lb weight, while the L pair supports a 1.5-lb weight. The axial loading is 1.0 lb on the H pair of bearings and 0.75 lb on the L pair.

During test operation, as the motor rotates the shaft, a separate torque transducer arm balances the bearing friction torque on each weight. A strain-gage bridge bonded to the arm provides an electrical output proportional to the resulting deflection of the arm and, therefore, proportional to the running torque of the bearing pair. A pair of test bearings fails, by definition, when its torque exceeds 300,000 mg-mm (30 gm-cm). As this torque is reached, the two pins on the weight that contact the flexure arm rotate to a position where they slip free of the end of the arm. As this occurs, the flexure arm is attracted by one of two magnets on the frame which pulls the arm 0.010 in. clear of the pins, permitting the failed bearing load weight to rotate freely with the shaft. This unique arrangement permits continued operation of either test bearing pair if the other fails.

Other instrumentation includes thermocouples to monitor the operating temperatures of the support bearings and the rear bearing of each motor. A rotation-sensing

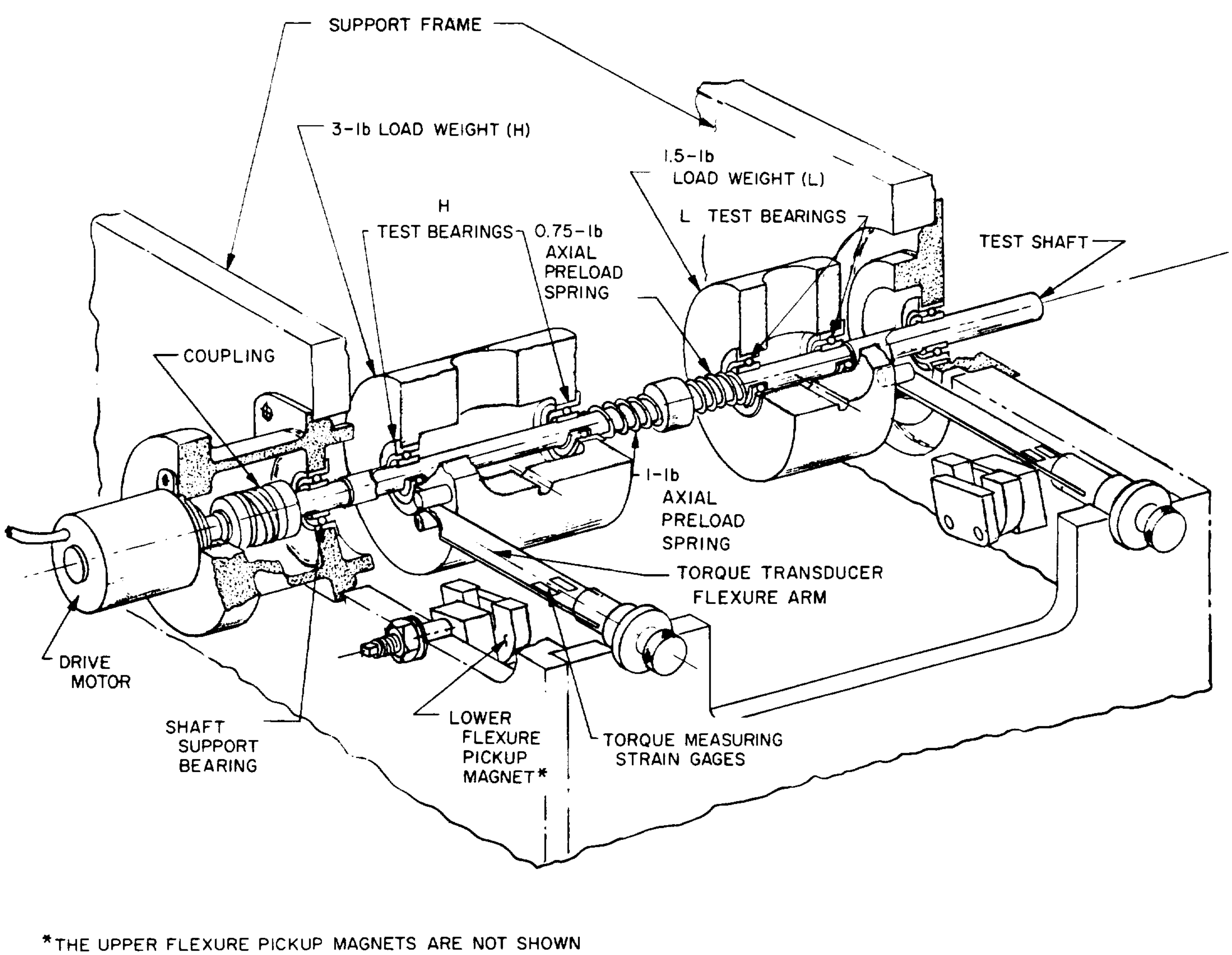


Fig. 1. Typical shaft of bearing test fixture

reed switch is provided for each shaft; a magnet rotating with the shaft actuates the switch once each revolution. Electrical impulse counters outside the vacuum chamber count the revolutions of each shaft. Vibration transducers are applied at three points on the end of the test fixture opposite the motors and are monitored during vacuum testing.

Particular care was taken in designing the fixture to minimize or eliminate potential sources of outgassing in hard vacuum. Materials selection was particularly important. Polyimide or TFE insulation was used wherever uncoated woven glass fiber could not be employed. Strain gages were bonded with a sprayed ceramic material (Al_2O_3); a limited amount of special epoxy was used to bond the vibration pickups and ceramic terminal strips. Most shaft support bearings had the same lubricant as the test bearings on that shaft. The remaining support bearings and all motor bearings are provided with Code X lubricant.

V. Vacuum System

The two vacuum systems used are each equipped with a 500-liter/s ion pump and a 5000-liter/s titanium sublimation pump. Each chamber is equipped with a mass spectrometer monopole detector for use in partial-pressure gas analysis within the chamber during operation. A trigger ion gage reading to 1×10^{-11} torr is used to monitor chamber pressure. The systems are capable of pumping to and holding approximately 1×10^{-11} torr (if the test fixtures are not operating). Both vacuum systems were purchased new for this test. Therefore, there was no danger of residual contamination from earlier test operations. To produce a hard vacuum in this type of vacuum system, it is first necessary to drive out moisture, entrapped or adhered gases, and organic contaminants from the test fixture and the internal surfaces of the chamber. To accomplish this, each vacuum chamber is equipped with an electrically heated bakeout mantel capable of raising the temperature within the chamber to 750°F. In these tests, a temperature limit of 375°F was established to prevent damage to the soft-soldered electrical connections and the bonding epoxies used in the test equipment. One six-shaft bearing test fixture was installed in each vacuum system.

After a 24-h bakeout at 375°F, each system was pumped to 10^{-11} torr, and test operations were begun. An initial increase in chamber pressure of about three decades was seen when all bearing drive motors were started. After a few hours of operation, chamber pressure decreased to the 10^{-9} torr range and remained in this region throughout subsequent tests. The second chamber has frequently been at approximately 5×10^{-11} torr, although during periods of continued operations the pressure normally rises into the 10^{-9} torr range. The pressure rise during operation can be attributed to two principal causes:

(1) The major source of outgassing (determined by partial pressure analysis of the chamber contents) is the internal gases from the lubricating materials. In bearing operation, continual exposure of fresh molecular surfaces permits evolution of these entrained or adsorbed gases. Evidence of free fluorine has also been observed, indicating a breakdown of the CF bond in the fluorocarbon lubricating compounds. This probably results from localized generation of temperatures in excess of 350°F (the lowest CF dissociation temperature) in the operating bearings.

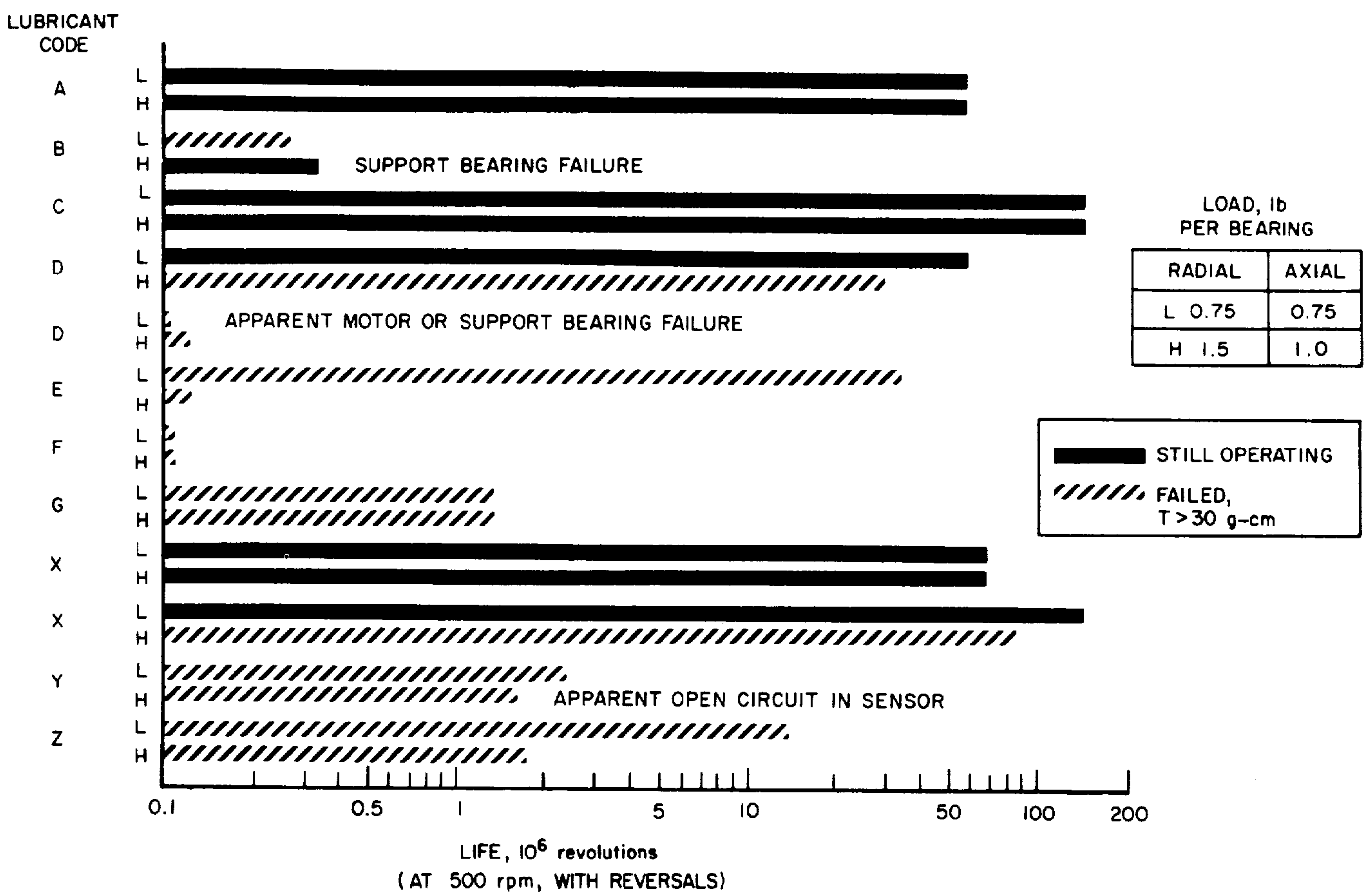


Fig. 2. Vacuum endurance life of dry-lubricated bearings (R4 pairs)

(2) The second source of outgassing is the polyimide insulation in the motors, which reached temperatures in excess of 200°F in normal operation.

VI. Vacuum Testing: Procedure and Results

Bearing vacuum testing was divided into four phases. The phases included: (A) checkout, (B) space mission simulation, (C) effects of dwell determination, and (D) long-term life testing. Forty-eight bearings – twelve pairs of test bearings, six pairs of support bearings, and six pairs of motor bearings – were operating in each fixture at the beginning of the tests. The cycles attained by the test bearings are shown in Fig. 2; apparent failures of the remaining bearings are also noted.

When operating, all shafts rotated at speeds between 480 and 500 rpm. Bearings were rotated for equal periods clockwise (CW) and counterclockwise (CCW), except in Phase C. Frequency of reversal is noted in the description of each phase, as follows:

A. Phase A

Phase A included a 2-h checkout in air, followed by a 375°F bakeout for 24 h. After pumpdown to 10^{-9} torr, there was a 48-h run, with reversal every 2 h, for a total of 1.5×10^6 total revolutions. All instrumentation and test bearing operation was checked out and carefully monitored during Phase A; several failures occurred during this period.

B. Phase B

Phase B consisted of 2 h of 4-min run, 2-min dwell, with reversal after each dwell, for 0.1×10^6 additional revolutions. Phase B was designed to simulate ten times the total operating cycles in vacuum of R-4 bearings in the motor or gear train of a hypothetical antenna actuator. Although the vacuum exposure of a 2-year space mission was not approached, the total wear of the lubricant was simulated.

C. Phase C

Phase C determined the effects of dwell in vacuum on torque. Dwell durations of 1, 6, 24, 48, and 72 h were employed. Starting and running torque were measured after each of three dwells of each duration. There were 0.1×10^6 additional revolutions.

In Phase C, some of the transfer film lubricants exhibited increased starting torque after periods of dwell, while their running torque level, measured a few seconds after start-up, was not noticeably affected by dwell duration.

As shown in Fig. 3, starting torque of the Code Y(L) bearing pair rose fairly linearly from approximately 10,000 mg-mm for 0 to 1 h dwell to 30,000 mg-mm after 72 h dwell. However, after a few seconds of operation, the running torque decreased and stabilized around 10,000 mg-mm, regardless of prior dwell duration. The

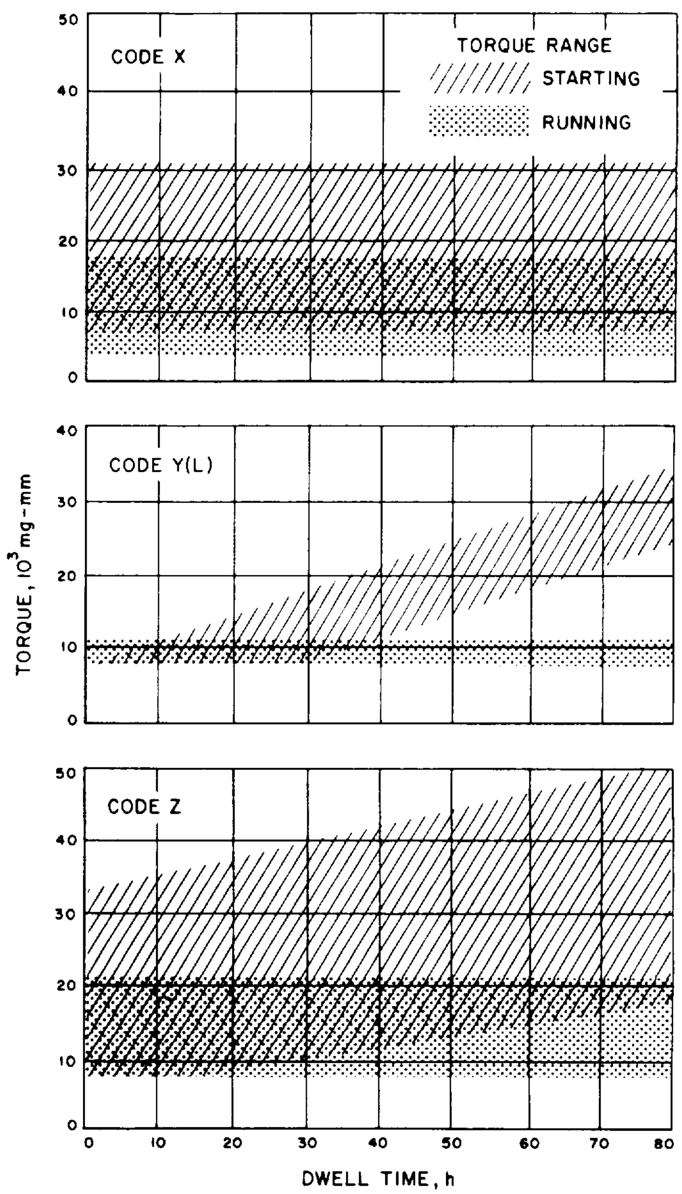


Fig. 3. Effects of dwell on bearing torque

Code Z bearings exhibited a somewhat more pronounced increase in starting torque after dwell. It may be significant that their average starting torque, after even the short duration dwells, was somewhat higher than their average running torque.

If the increase in starting torque with increased dwell duration is attributed to embedment of the bearing balls into the lubricant film under load, this would be expected. The Code Y lubricant, with its glass fiber reinforcement, should have a higher bulk modulus, and hence a higher resistance to indentation, than the Code Z, which is essentially pure TFE.

In contrast to the behavior of the Code Y and Z bearings, dwell appeared to have no significant effect on the Code X bearings. Although both test fixtures contained Code X test bearings, none of the four pairs tested exhibited this phenomenon. In an attempt to determine whether the Code X lubricant would exhibit this behavior after dwells of longer duration, further tests were made (at approximately 45×10^6 revolutions in Phase D). Ultimately, a 10% increase over the starting torque measured after an 80-h dwell was measured after a 240-h dwell. On the basis of the available data, it is not possible to determine whether the Code X lubricant is more resistant to the embedment phenomenon (if this in fact causes this behavior) or whether the MoS_2 present in the Code X lubricant helps to nullify this effect.

D. Phase D

Phase D consists of a sequence of 50 h CW rotation, 50 h CCW rotation, and 68 h dwell, to be repeated until failure occurs, with 3×10^6 additional revolutions each week.

Phase D testing is being continued, to establish relative "life" or cycle capabilities of the lubricants and bearings still operating successfully. Vacuum testing of the second fixture was begun approximately 2 months after the first test series was begun; this is the reason some of the bearings still operating have completed fewer revolutions than others. (The Code X bearings in the second test had previously been run for an additional 8×10^6 revolutions in vacuum in an initial facilities checkout.)

VII. Conclusions

Useful analytic techniques for dry-film lubricant characterization and quality control have been identified or developed. X-ray emission techniques are effective in determining constituent distribution in nonlayered homogeneous lubricants. The electron probe technique proved particularly effective in the analysis of multilayered lubricant coatings. Optical microscopy was found to be extremely useful in the evaluation of all the coatings tested.

Some lubricants have shown definite increases in starting torque as a function of dwell duration. Codes Y and Z bearings exhibited starting torques two to three times their average running torque after 80 h dwell under load in vacuum. While this torque increase is not excessive, consideration should be given to this phenomenon in applications where low excess torque is available and intermittent operation is expected.

Improved transfer lubricant films may result from initial run-in in vacuum. Evidence of "vacuum shock" crazing and reduced film adhesion was seen in a transfer film established in air at atmospheric pressure.

Performance tests have been designed and conducted which have demonstrated the capability of some dry-film lubricants to operate for extended periods in hard vacuum, while other lubricants failed under the same conditions. Two lubricants, Codes C and X, have performed successfully for over 130×10^6 revolutions in 10^{-9} torr vacuum. In a subsequent test, Codes A, D, and X were operating smoothly after 60×10^6 revolutions, also in hard vacuum.