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Experimental Test Program for Evaluation of Solid Lubricant Coatings as Applied to Compliant Foil Gas Bearings to 315 °C

Robert C. Wagner
Case Western Reserve University

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CHAPTER 1

INTRODUCTION

The gas bearing is an ideal mechanism for supporting high-speed, high performance turbine driven rotors. The main advantages of gas bearings over conventional rooling element and hydrodynamic bearings are the longer bearing life and the elimination of liquid lubricant systems. Problems with lubricant vaporization, coking and complex external lubricating systems are eliminated by the gas bearing system.

Two types of gas bearings are used: hydrostatic and self-acting. The hydrostatic gas bearing is dependent upon an external pressure source to lift it away from the rotor prior to operation. The additional complexity and expense of the pressure source favors self-acting bearing systems for turbomachinery applications. Self-acting bearings induce pressure within the bearing cavity by relative motion between the rotor and bearing. A converging wedge is formed to provide conventional hydrodynamic lubrication, eliminating the need for external pressurization.

Traditionally, self-acting bearings are fixed in a rigid or tilting pad configuration. Since gases do not have the damping capabilities of liquid lubricants, self-acting bearings can be subject to destructive whirl instabilities, the unsteady journal motion when the shaft orbits with angular eccentricity to the bearing axis. These instabilities are the unwanted result of random excitations when force or displacement variations occur, Gross et al. [1]. Another limitation of rigid gas bearings is that bearing seizure at high temperatures can occur when thermally induced distortions reduce the minute radial clearances.

Compliant foil gas bearings represent a technological advancement which overcomes these deficiencies. In a foil bearing, a thin flexible surface replaces the rigid boundary. As load and rotational speeds vary, this surface deflects correspondingly. A means of minimizing whirl instabilities is produced by maintaining a constant gas film thickness to damp out load fluctuations. Deflection of the surface is also useful in retaining radial clearances at elevated temperatures, reducing the possibility of bearing failure. Because of the ability for this type of bearing to deform, other inherent advantages are realized: accommodation of misalignment, geometric shaft imperfections, and thermal distortions, and a higher tolerance for dirty environments, by allowing particles to pass through the clearance, Beercheck [2].

Compliant foil bearings can be classified as tension-dominated, bending-dominated, or cantilevered. Tension-dominated bearings consist of flexible metal strips under tension wrapped around a journal and connected to the bearing housing. The strips are stretched as the bearing speed increases expanding the clearance. Cantilevered foil bearings consist of overlapping foils which deflect as individual beams under load providing damping to the bearing system. Bending-dominated gas bearings utilize a corrugated foil base under a thin foil surface. Under varying load conditions, this corrugated foil will deflect circumferentially within the bearing housing increasing bearing clearance. Figure 1 illustrates these types of compliant foil gas bearings. The bending-dominated gas bearing shows great promise due to ease of fabrication as compared to the others. For this reason, the testing program utilizes this type of foil bearing.

Although compliant foil gas bearings require no liquid lubrication, it is necessary to provide lubrication since sliding contact occurs between the bearing and journal during liftoff and coastdown phases of operation. Solid lubricant coatings are used to protect the surfaces from wear and decrease the Coulomb sliding friction. Current state of the art for the lubrication of foil bearings has the maximum temperature capability of about 260°C (500°F) using polytetrafluoro ethylene (PTFE) coatings. Work has been done to develop coatings with capabilities to 650°C (1200°F), Suriano et al. and Bhushan [3-4]. The coatings for high temperature applications are plasma-sprayed, sputtered, or ion-plated materials. While such coatings can give acceptable performance at high temperatures, they are not usually as effective at room temperature to 315°C (600°F) as are the layer lattice class of solid lubricants. Materials of interest at this time are coatings which effectively lubricate to 315°C in intermediate temperature applications, specifically, turbomachinery driven by compressor bleed air from turbojet engines.

Polyimide-bonded graphite fluoride (PBGF) is selected as the primary coating material for evaluation in this study. PBGF has been shown to achieve along wear life and low friction coefficients in pin on disk experiments in the temperature range of interest here, Fusaro and Sliney [5]. Silicate-bonded graphite/cadmium oxide (SBGC) is used to lubricate foil bearings to 420°C (800°F) Bhushan and Gray [6]. This composition is selected as a primary baseline material for judging the comparative effectiveness of PBGF as an intermediate temperature foil bearing lubricant.

The scope of this study is to determine the frictional characteristics and durability of PBGF and SBGC coatings on Inconel X-750 foil bearings from 25°C to 315°C. In addition, some preliminary experiments with MoS_2 and other graphite coatings are performed.

CHAPTER 2

MATERIALS

2.1 Polyimide Bonded Graphite Fluoride (PBGF)

The polyimide used in this coating serves as a binder for graphite fluoride and itself has good tribological properties at elevated temperatures. Fusaro [7] has shown, using a pin on disk apparatus, that polyimide coatings have low wear rates and good friction properties at temperatures above 100°C (212°F). At lower temperatures, polyimides exhibit poor lubricating qualities.

Polyimides belong to a class of thermally stable organic polymers which are capable, by tailoring the monomeric starting materials, of achieving acceptable mechanical properties, Fusaro [7]. The basic structure shown in Figure 2 represents the typical aromatic polyimide where R represents a thermally stable group. Acceptable mechanical properties are important due to the adhesion difficulties that can occur on a flexible foil bearing during operation. The coating must be ductile enough to remain on the substrate during bending. Simple flexure of the foil must not cause the polyimide to crack or spall.

At elevated temperatures, the effectiveness of the polyimide, both mechanically and thermally, is evident. They have good lubricating properties in this environment. However, there exists a transition temperature, between 25°C and 100°C, below which friction and wear rates increase considerably. Above this temperature the properties of the polyimide change, allowing the surface layer of the film to flow plastically. Below the transition temperature, the polyimide tends to become brittle, producing high wear and friction. The effects of this transition temperature are illustrated in Figure 3. As compared to polyimide-bonded molybdenum disulfide, polyimide with no solid lubricant additive, and unlubricated surfaces, the polyimide-bonded graphite fluoride exhibits a much higher wear life and lower friction below 100°C (212°F). The effect of the transition temperature becomes evident above 100°C with the polyimide-bonded graphite fluoride performing only marginally better than the polyimide with no additive. The effect of wear on the sliding surface is also improved with the polyimide-bonded graphite fluoride as compared to the others, as shown in Figure 4, Fusaro and Sliney [5]. Other factors, such as relative humidity and polyimide type are involved in determining this transition temperature. However, this determination is beyond the scope of these experiments.

To counteract the tribological instabilities of polyimide below the transition temperature, graphite fluoride is added. Graphite fluoride is used as an additive in greases, mechanical carbons, and polytetrafluoroethylene (PTFE) fibrous carbon composites. It is found to provide increased load-carrying capacity and good friction properties, Fusaro and Sliney [5]. Graphite fluoride has the ability to readily undergo plastic flow during sliding contact. The crystal structure is shown compared to graphite in Figure 5, Sliney [8]. It exhibits as much as 6 times longer wear life than molybdenum disulfide and graphite. It is thermally stable to 420°C (800°F). Graphite fluoride is a hydrophobic material. Polyimide is a good solid lubricant above 100°C, but being hygroscopic, will attract water molecules to its hydrogen bonds at lower temperatures. This tendency decreases the ability of the polyimide molecules to plastically flow. Graphite fluoride inhibits association with water molecules providing good lubrication at lower temperatures.

Preparation and application of the polyimide-bonded graphite fluoride is presented according to Fusaro [7].

Preparation and Application

A thick precursor solution of pyralin polyimide (PI-4701) is formed into a sprayable mixture by adding a thinner consisting of N-methylpyrolidone and xylene. Graphite fluoride, $(CF_x)_{1.1}$, with a fluorine-to-carbon ratio of 1.1, is mixed in equal parts by weight of polyimide solids. The precursor solution contains 43% weight solids. The coating is applied to the Inconel X-750 foil by means of an artist's airbrush. Only a thin layer at one time is applied to prevent 'running'. Each layer is then cured completely by heating at 100°C for 1 hour, then 300°C for 2 hours.

2.2 Silicate-Bonded Graphite/Cadmium Oxide (SBGC)

Graphite has served as a solid lubricant for many years. It has effective lubricating properties at low (25-100°C) and high

(>450°C) temperatures but loses this ability at temperatures within that range. This phenomenon results because the slipperiness of graphite is not inherent in the crystal structure alone but must depend on absorbed gases or intercalated impurities to provide a surface of low cohesion, Savage [9]. Graphite is a layered lattice type material having hexagonal layered crystal structures. It has anisotropic shear properties within preferred planes allowing it to shear easily parallel to the basal planes of the crystallites, Sliney [8]. To assist in this shear effect, absorption films must be present between the basal planes. The most common film is water. At 25-100°C (75-212°F), graphite will slide effortlessly due to adsorbed water. When the temperature exceeds 100°C (212°F), the water is desorbed and the lubricating properties deteriorate. The effectiveness returns at extremely high temperatures when oxides, formed on the metal substrate, combine with the graphite and aid in adherence to the substrate. Figure 6 illustrates the temperature dependence of the coefficient of friction from Peterson and Johnson [10].

By combining metallic oxides with the graphite, adherence is improved. Interstitial or intercalation compounds are formed by the reaction of these materials with graphite. Peterson and Johnson have shown that adding cadmium oxide (CdO) to graphite in a 2:1 weight ratio mixture lubricates most effectively to 540°C (1000°F) [10]. Figure 7 shows this improved lubricating ability versus the degradation at high temperatures for graphite alone.

This experiment was conducted using a loose powder at the rubbing interface. It is believed that a coating bonded to the sliding surface of a compliant foil bearing will achieve similar low friction results throughout a wide temperature spectrum.

To bond the graphite/CdO material to the bearing surface, sodium silicate (water glass) is selected as a binder material. This hard oxide is oxidation resistant and has good wear resistance. It has poor friction characteristics but in combination with the graphite/CdO powder produces a low friction, highly wear resistant durable coating.

The following preparation and application of the SBGC coating is performed according to Bhushan, Ruscitto and Gray [17].

Preparation and Application

The mixture consists of 3 parts graphite to 1 part CdO with sodium silicate as a binder and wetting agents for dispersion of the solution. The graphite is 99.9% pure electric-furnace synthetic graphite. Synthetic rather than natural graphite is used because it exhibits a higher temperature tolerance. A very fine powder with 95% of the particle sizes finer than 325 mesh is used. Cadmium oxide is 99.9% commercially pure. It has particle sizes of which 95% are finer than 200 mesh. Sodium silicate is 99% pure, having a composition of 8.9% Na_2O, 28.7% SiO_2, and the balance water. Excluding the water, 30% weight sodium silicate is used to give adequate bonding. Higher contents have been found to be too abrasive. A wetting agent with a cloud point of 65°C is used.

The mixture of graphite and CdO is dissolved in distilled water and ball milled for 4 hrs. Sodium silicate and one drop of the wetting agent are added and the solution is stirred vigorously. The solution is heated to 65°C (150°F) prior to application to the foil. It is sprayed by an airbrush about 25μm (1 mil) thick and left at room temperature for 30 minutes. The coating is baked in an oven at 65°C (150°F) for 2 hrs. and then baked at 150°C (300°F) for 8 hrs. After curing, the coating thickness is 12–18 μm, and is subsequently burnished to a thickness of 10 μm.

2.3 Foil and Journal Substrate Materials

An extensive research program previously selected the materials to be used for the foil bearing and journal construction, Bhushan et al. [11]. The current experimental program utilizes the same materials of construction.

Foil Bearing Material

Inconel X-750 is chosen for the foil bearing. This nickel-chromium alloy exhibits excellent physical properties to 650°C (1200°F). The ease of heat treatment, formability, availability in proper thicknesses and cost make it an excellent candidate for mass production. It retains good spring properties at high temperatures, important since foil bearing operation is dependent on conformability. A disadvantage of this material is its tendency to gall during sliding contact. For this reason, the bearing coating must be successful in resisting penetration. The thermal properties of Inconel X-750 closely match those of the candidate coatings, thereby reducing the possibility of coating separation during temperature transitions.

Journal Material

The material used for the test journal specimens is A-286 stainless steel. It is commonly used in high temperature gas turbine applications. It has the desired thermal and mechanical properties at intermediate (315°C) and high temperatures (650°C): high strength and a low coefficient of thermal expansion, important since in many applications the journal is coated and much of a dissimilarity in thermal expansion coefficients results in separation of the coating from the base metal. Other factors favoring the selection of A-286 stainless steel is its ease of machinability, abundant availability and low cost.

Listed below are some properties of the foil and journal substrate materials by Bhushan et al. [11]:

	Inconel X-750	A-286 Stainless Steel
Temperature °C	650	650
1000 Hour Rupture Strength MPa (ksi)	469 (68)	317 (46)
Modulus of Elasticity GPa (10^6 psi)	176 (15.5)	153 (22.2)

Coeff. of Thermal Expansion 21°C to Temp. x 10^{-6}/°C (70°F to Temp. x 10^{-6}/°F)	15.14 (8.41)	17.78 (9.88)
Thermal Conductivity W/(m.°K) (BTU/ft^2/hr/°F/in)	20.6 (143)	24.8 (172)

CHAPTER 3

EXPERIMENTAL TEST RIG

3.1 Compliant Foil Bearing Test Facility

Start/Stop Test Apparatus

An existing test rig at NASA Lewis Research Center is used for these experiments. Figures 8 to 9 show the entire test facility. The journal support shaft is supported on two preloaded angular contact ball bearings. These ball bearings are standard class 7 bearings. The outboard bearing is a 107H (35 mm bore) and the inboard bearing at the drive motor is a 105H (25 mm bore). Both bearings are enclosed in an insulated housing. The test journal is a light interference fit onto the shaft and held in place with a tie bolt which is threaded to the shaft.

Lubricating oil for the support ball bearing is supplied by an external oil pump system. Figure 10 is a schematic of the lubricating system. This system consists of an oil supply pump for oil feed to the bearings, and a scavenge pump to prevent oil accumulation within the bearing cavities. A water-cooled heat exchanger in the oil supply loop removes heat from the oil. A water jacket in the support housing assists in cooling the support ball bearings. The flow of oil in the supply line is measured by an in-line flowmeter.

The test spindle is driven by a 1 hp. induction electric motor at 3450 RPM. The motor is attached to the main vertical support plate and connected to the spindle with a flat drive belt. A pulley ratio 4:1 is used to generate a 13,800 RPM spindle speed. The spindle is activated for 13 seconds and switched off for 7 seconds (total cycle time is 20 seconds). This time allows the bearing to fully lift off during the start cycle and the spindle to completely stop during the stop cycle. A fiber optics impulse counter is used to record each cycle.

The test apparatus is designed to operate unattended. Two timer switches run each start/stop cycle at the 20 second intervals. One timer activates the running cycle for 13 seconds after which the second timer controls the stop cycle for 7 seconds. Program time switches oversee the amount of time that the test is conducted by deactivating the electric motor at a preprogrammed time.

A heater box is used to heat the test journal and bearing to the desired operating temperature. This box is split into two halves, for ease of mounting around the test journal/bearing. These halves are bolted together after assembly. Each half contains four 500 Watt quartz lamps connected to a manual-controlled rheostat. The rheostats vary the voltage to the quartz lamps which increase the temperature. A temperature controller is used to maintain a constant temperature within the heater box. This is a simple on-off device which activates the heaters when the temperature falls below a preset condition. It deactivates the heaters as the desired temperature is reached. A chart recorder continuously monitors the test chamber temperature.

The test bearing is mounted in a floating housing as shown in Figure 11. The housing is retained from rotation by a torque arm that bears against a calibrated flexure plate. The bearing is inserted into a keyway, locked into position with two tapered pins, and mounted around the test journal. The torque arm is placed against the flexure plate and a small wire spring is pressed against the arm to maintain position.

Measurement and Instrumentation

Rotational speed is measured by a tachometer probe which responds to the once per revolution passing of an unpainted band on the black shaft of the spindle. This is mounted on the end opposite the test journal/bearing assembly directly above the electric drive motor. The output of this sensor is displayed on one channel of each of the two strip chart recorders. The test bearing and chamber temperature are monitored by thermocouples. These are mounted to the bearing housing, extending through to the edge of the foil itself, and suspended directly in the chamber. The outputs are registered on a multi-point chart recorder and continuous strip chart recorder.

Figure 12 shows the arrangment for measuring the frictional drag of the test bearing during operation. The torque arm restrains the floating bearing housing from rotation by bearing against a calibrated flexure plate. Frictional drag causes deflection of this flexure which is measured by a capacitance proximity probe. The range of the probe used is 0-.254 mm (0-.01 in.). The output of the probe is amplified and recorded by the two strip chart recorders.

A conversion chart is formulated to correspond with the coefficient of friction at lift off, running, and touchdown. This is calculated during capacitance probe calibration in Appendix B.

Journal velocity and bearing torque are plotted simultaneously by the recording oscillograph and strip chart recorder. Two recorders are used. The strip chart recorder operates continuously, measuring at every start/stop cycle. The oscillograph is used on a sampling basis, giving a much finer start/stop cycle profile for calculating liftoff and touchdown velocities. Typical velocity and torque profiles for one 20 second start/stop cycle are given in Figure 13.

The test bearing loading is accomplished by the addition of calibrated dead weights suspended from the bearing housing. An application of 1022 gms. (2.25 lbs.), including the weight of the housing itself, corresponds to the 14 kPa (2 psi) radial unit load.

Other instruments are used in the operation of the test rig. These are mainly safety shutoff devices to monitor and deactivate the system should a critical parameter exceed its limit. The shutdown devices are furnace temperature gauge and high shutdown, high torque shutdown, oil temperature gauge and high shutdown, and annuciator system. They prevent damage to the equipment should an uncontrolled situation occur.

3.2 Test Bearing and Test Journal

The test bearing is a partial arc 38.1 mm (1.5 in.) diameter by 19.05 mm (.75 in.) wide journal bearing. The test bearing and test journal are shown in Fig. 14. The bearing diameter and mechanical design are the basis of work accomplished by Bhushan [12]. The bearing consists of a bump foil overlaid with a smooth foil. The solid lubricant coating is applied to the smooth foil surface. A spacer shim separates the two foils which are attached to a key by spot welding. The key is fitted into a keyway slot on the floating bearing housing and secured in place by two tapered dowel pins. This method of attaching the foils to the housing is not typical for foil bearing applications but greatly facilitates the changing of test specimens while having the fewest number of test components.

A partial arc bearing is used instead of a complete bearing to simplify bearing fabrication. This bearing has one bump more than one half the total number of bumps in a complete circular bearing which results in a 186° pad arc. Through earlier testing it was found that a pad of less than 180° results in unstable bearing operation, Bhushan [12]. Rotation of the journal is into the free end of the bearing. This partial arc design is specifically for coating evaluation experiments. It is capable of lift-off at about 3000 rpm (6 m/sec.) at a radial unit load of 14 kPa

(2 psi). However, it is not intended for use as a functional journal bearing. A functional bearing is a full circular single or multi-segment bearing with a larger length over diameter ratio, typically 1.0. Assembly details of the partial arc bearing are shown in Figure 15.

The test journals are uncoated A286 stainless steel, measuring 38.1 mm (1.5 in.) diameter by 44.5 mm (1.75 in.) long. Two foils can be tested against one clean journal by moving the location of the bearing along the axis of the journal. The surface of the journal is machine-finished to a surface finish of 0.2 μm RMS. Figure 16 depicts test journal details.

CHAPTER 4

TEST PROCEDURE

The tests are run at a maximum surface velocity of 28 m/sec at 13,800 rpm with 14 kPa (2 psi) radial unit load. This load is typical of the radial loads encountered at start-up for foil bearings in turbomachinery, Bhushan et al. [11]. The tests terminate when solid lubricant failure is indicated by a sharp rise in bearing starting torque or the completion of 9000 start/stop cycles. The choice of 9000 start/stop cycles as a satisfactory coating life is the number of start/stop cycles that would be experienced on the average of five times per day over a five year period. In practice, some starts can be expected to be cold, others at intermediate bearing temperatures, and others at the maximum bearing temperature depending on the length of time the machine is shut down before restart. The test procedure accounts for this by mixing start/stop cycles at ambient, two intermediate, and a maximum temperature of 315°C. The procedure is conducted in the following sequence:

I. 500 start/stop cycles at ambient

II(a) 250 start/stop cycles at 120°C (250°F)

II(b) 250 start/stop cycles at 230°C (450°F)

III. 500 start/stop cycles at 315°C (600°F)

The above sequence is repeated six times for a total of 9000 start/stop cycles.

Operating procedures for a test sequence are presented in Appendix A.

CHAPTER 5

EXPERIMENTAL RESULTS AND DISCUSSION

5.1 Friction

Relatively short duration experiments were performed early in the test to generate baseline data with:

1) Commercially available, dry film lubricants

2) Polyimide bonded graphite fluoride (PBGF)

3) Silicate bonded graphite/cadmium oxide (SBGC)

These experiments were conducted to obtain quantitative torque profiles and apparent friction coefficients. The friction coefficients are calculated directly from bearing torque measurements. The starting friction coefficients (μ_1) will be termed 'apparent' because they are higher than the generally accepted values of coulomb friction for the particular coating. In addition to sliding friction, other factors contribute to the friction at start. These may include dynamic elastic deformation forces acting on the highly-conformable foils. Liftoff occurs at about 3000 rpm (6 m/sec surface velocity). Assuming proper bearing clearance during normal 'airborne' operation at 13,800 rpm (28m/sec), the computed friction coefficient (μ_2) is due to viscous shear of the lubricating gas film. A summary of friction coefficients at various operating temperatures is given in Figure 17.

There is little variation in the room temperature starting torque for the various coatings. The computed values for μ_1 from starting torques are about 0.3 for molybdenum-disulfide (MoS_2) and graphite coatings. For the as-sprayed PBGF, this friction coefficient μ_1 is higher, about 0.4. This is reduced to 0.3 by the application of a burnished (rubbed-on) overlay of graphite fluoride to the PBGF surface. Significant differences do occur at elevated temperatures.

The value of μ_1 for the SBGC coating remains at about 0.3 at temperatures from 25°C to 315°C. But this coefficient is observed to decrease for the as-sprayed PBGF coated bearing, from 0.4 to 25°C to 0.2 at 315°C. A similar behavior occurs for the burnished PBGF coating, decreasing from 0.3 to 25°C to 0.1 at 315°C.

Friction coefficients while airborne (μ_2) are uniformly low (less than 0.1) in all of the tests. They tend to be lower with the thinner coatings, which indicates that the increase in clearance provided by thin, smooth coatings is beneficial in achieving a satisfactory gas film thickness at the relatively low speed (for gas bearings) of 13,800 rpm (28 m/sec).

Figures 18 and 19 give representative steady-state friction coefficients as a function of bearing temperature for SBGC and PBGF coatings. The friction coefficients plotted are the apparent starting friction (μ_1), friction while airborne (μ_2), and maximum apparent friction during coastdown (μ_3). Values of the starting and running friction are of concern, since they contribute to the power requirements for starting the machine and the effect on its operating efficiency.

Early in the endurance tests, considerably higher values of μ_1 are observed as compared to the steady-state values for the PBGF coatings. This may be attributed to a 'running-in' mechanism at the coating/journal contact. Figure 20 illustrates the starting friction coefficients for a PBGF coating as a function of test duration for the first two programmed temperature sequences (3000 start/stop cycles). Friction decreases steadily from 0.53 to 0.40 during the first 500 start/stop cycles at room temperature. Erratic behavior is demonstrated during the next 500 cycles at the intermediate temperatures, then becomes steady at 0.25 during 315 C operation. Steady-state behavior prevails during the next programmed 1500 start/stop cycles with μ_1 of about 0.3 at room temperature, 0.25 at intermediate temperatures, and 0.23 at 315 C.

There are several factors that contribute to the reduction in bearing starting torque as the number of start/stop cycles accumulate: (1) a preferred orientation of solid lubricant crystallites occurs, aligning low shear strength crystal planes parallel to the sliding direction, (2) as-sprayed PBGF surfaces are relatively rough and sliding smoothes the surface asperities which is more favorable to efficient bearing operation, (3) this smoothing action also increases the effective radial bearing clearance, a factor which is conducive to lower bearing torque.

5.2 Coating Endurance

Coating endurance is here defined as the number of start/stop cycles accumulated by a test bearing before the coating wears through to the foil metal substrate. Failure is determined by a substantial increase in starting torque and verified by visual inspection of the foil and journal surfaces. Results of the endurance tests are summarized in Figure 21.

The heat-cured MoS_2 coating survive 175 to 400 start/stop cycles at room temperature before failure. Heat-cured proprietary graphite coatings survive approximately 1000 start/stop (500 at room temperature and 500 at 230 C). The relatively short lives of these coatings are probably due to the fact that the Inconel X-750 foils are not pretreated to enhance coating adherence except by lightly sanding the surface with #200 sandpaper. (Roughening the surface by the customary procedure of sand blasting tends to distort the thin foils severely.) However, even with this mild surface preparation, SBGC and PGGF coatings are very durable and adherent. Their performance in long duration endurance testing is described below:

Polyimide-bonded graphite fluoride (PBGF) - Endurance tests of four PBGF-coated bearings were conducted at temperatures from ambient to 315 C. All four coatings survive 9000 start/stop cycles of the standardized endurance test procedure. In fact, one PBGF-coated bearing was subjected to an extended period of start/stop cycles. After 32,000 start/stops at ambient conditions, the bearing coating still performed with acceptable friction characteristics. This durability is achieved despite the fact that the smooth foils are not pretreated (other than light sanding) prior to coating application. Sliding contact during starts and stops polishes the coatings to a reflective finish, and a very thin transfer film of coating is deposited on the journal. The coatings are especially glossy in the areas of highest pressure contact, that is, over the bumps of the supporting smooth foil and at the edges of the foil which are areas of minimum film thickness. Photomicrographs and surface profiles of the foil bearing and journal after 9000 start/stop cycles are shown in Figures 22(a) and (b).

In one endurance test, it was noted that extraordinarily high torque was present at the beginning of the experiment. The bearing fit was suspected to be tight and total liftoff never occurred. The coating was given a light sanding to reduce its thickness, thus increasing the radial clearance between the bearing and journal. A thin film of graphite fluoride $(CF)_{1.1}$ was burnished over the finished surface, resulting in normal torque characteristics.

Silicate-bonded graphite/cadmium oxide (SBGC) - These coatings also demonstrate good durability but they do not survive the programmed 9000 start/stop cycles. The two coatings tested failed at 7500 and 4500 start/stop cycles. Failure was indicated for both bearings by excessive bearing torque, coating wear to the substrate metal, and by scuffing of the A286 steel journal.

The surfaces of the foil and journal are periodically inspected and are in very good condition at all times prior to coating failure. After 500 start/stop cycles at ambient, the foil bearing is in very good condition with polished areas over the bumps and at the foil edges. A light film of coating transfers to the journal. After one complete heating sequence from 25 C to 315 C, and an accumulated 1500 start/stop cycles, the journal surface is oxidized and also coated with a transfer film. The foil is highly polished except for a few areas between the bumps.

Inspection of the surfaces after coating failure revealed severe coating wear to the substrate metal over much of the foil surface. The oxide film on the journal is worn away and the steel surface is circumferentially scored. There is a great deal of coating transferred to its surface. Figures 23(a) and (b) are photomicrographs and surface profiles of the journal and foil bearing surfaces.

Discussion of Comparative Endurance

These results demonstrate that both polyimide-bonded graphite (PBGF) and silicate-bonded graphite/cadmium oxide (SBGC) coatings are effective solid lubricant coatings for foil bearings to 315 C. The PBGF coatings have superior durability, none of these coatings fail in 9000 start/stop cycles. The SBGC coatings are less durable but nevertheless have respectable endurance lives, since they survive thousands of start/stop cycles before failing.

Graphite-coated bearings have a more consistent friction coefficient than do PBGF coatings. A characteristic of the SBGC coatings is an apparent starting friction coefficient (μ_1) of about .03 throughout all of the operating temperatures. The same coefficient for PBGF coatings varies considerably with temperature, ranging from 0.4 at 25°C to 0.2 at 315°C for a well run-in bearing. Coefficients as high as 0.5 are observed at 25°C during the period of run-in at the start of the experiment. Friction can be appreciably reduced by burnishing graphite fluoride powder ($CF_{1.1}$[2]) onto the bearing and journal surfaces prior to testing. The effects of this burnished film persist over thousands of start/stop cycles.

CHAPTER 6

CONCLUSIONS

Polyimide-bonded graphite fluoride (PBGF) coatings are evaluated to determine their suitability as dry film lubricants for compliant foil gas bearings at temperatures from 25 C to 315 C. A silicate-bonded graphite fluoride (SBGC) coating which is a known foil bearing solid lubricant, is evaluated as a baseline for comparison. The following conclusions are obtained from the experimental program:

(1) Both coatings are effective solid lubricants from 25 C to 315 C. At ambient, bearing torques at startup are about the same for both coatings. However, while starting torque remains relatively constant for SBGC at all temperatures, it decreased with increasing temperature for PBGF.

(2) Both coatings are durable. They are capable of surviving thousands of starts and stops over a 25 C to 315 C operating temperature range. PBGF coatings can survive in excess of 9000 start/stops. SBGC is less durable, surviving 4500 to 7500 start/stop cycles before failure. Failure occurs when the coating wears to the substrate metal in the minimum gas thickness areas over the bumps and at the edges of the support foil.

(3) Friction coefficients computed from bearing torque profiles at low sliding velocities (before liftoff) are higher than the values typically observed for the dry film lubricants employed. Other factors, such as elastic deformations of the foil, may be adding to the Coulomb friction of the sliding surfaces.

(4) Starting torques tend to decrease with accumulated start/stop cycles and then level to a steady-state value. This can be attributed to a normal 'run-in' effect, but in gas bearings, the smoothing of the coating surface during 'run-in' has the additional effect of increasing the effective radial clearance which contributes to this behavior.

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APPENDIX A

OPERATING PRCEDURES

The foil bearing test rig is designed to run unattended. These are procedures that are followed in preparing the foil bearing and journal prior to testing.

Preparing the Test Specimen

The journal is inspected for wear marks and discoloration. Any such defects can be removed by polishing the journal surface with #400 and #600 aluminium oxide sandpaper. The bearing, housing, and journal are thoroughly scrubbed with ethyl alcohol to remove dust and dirt particles. The foil bearing is attached to the bearing housing by a slotted key. It is secured in place with two tapered dowel pins. Bearing orientation in the housing is such that the journal rotates into the free end. The journal is installed on the support spindle. An alignment is made for a maximum runout of .005 mm (.0002 in) at a distance of 6.35 mm (.25 in) from the free end of the journal. The locking nut is tightened with a torque wrench to 80-100 in. lb. The bearing housing assembly is placed onto the journal shaft. The top of the torque arm must be in contact with the flexure plate. This arm penetrates the guide slot nearest the support bearing housing. A small spring rod is placed on the opposite side of the torque arm and tightened.

When the assembly is complete, the electric drive motor is manually rotated several turns to verify that the bearing is correctly installed. The motor is started to determine that the bearing floats freely. If it does not, there most likely is no radial clearance between the bearing and journal. The bearing coating may be too thick and requires treatment, such as sanding to insure a looser fit. Calibrated weights are added to the bottom of the bearing housing to attain the required bearing unit load.

During heating operations, the quartz heater box is placed around the bearing/journal assembly. It is necessary to insure that the heater box thermocouple is not in contact with the bearing housing. The two halves of the heater box are bolted together and positioned with guides at the bottom. The temperature controller is set to the desired temperature. Both rheostats are simultaneously activated to the desired voltage level to insure a uniform temperature distribution around the bearing. The temperature recorder is operated to monitor the temperature within the heater box. When the experiment is operating in the heating mode, it is desirable to run the bearing continuously until the temperature is reached so as to minimize any thermal effects that may exist between the journal and foil bearing.

Start-up Procedure

For safety considerations, the rotation area is inspected. The strip chart recorders are verified to contain an adequate paper supply and proper inking function. The program and heater duration timers are set for the length of the experiment. The impulse counter is set to zero. A push button activates the total system power. The scavenge pump then oil supply pump are activated with verification of oil supply operation from the digital flowmeter readout. Annuciating system is acknowledged and furnace overtemperature, high torque shutdown, and oil out temperature gauges are reset. The motor control switch is set to hand (continuous) operation and the motor power pushbutton is pressed. The start/stop cycle timers are verified to be set for the proper sequence - 13 seconds for the starting and running cycle and 7 seconds for the shutdown cycle. The strip chart recorder is activated. Finally, the motor control switch is set to automatic to induce start/stop operation.

APPENDIX B

CALIBRATION OF FRICTION MEASUREMENT DEVICE

The apparatus used to measure the friction force of the foil bearing requires calibration for proper interpretation of the coefficient of friction values. The friction force is measured as a function of the deflection of an elastic beam which is caused by the torque of the test bearing. The flexure and schematic of the physical arrangement of the torque measuring device are shown in Fig. 24. Deflection of the beam is measured with a proximity capacitance probe. The capacitance of the probe varies with the distance between the end of the probe and the deflecting beam. The probe is one leg of a balanced capacitance bridge. Any change in probe capacitance unbalances the bridge and produces an output voltage which is measured by a recording potentiometer (millivoltmeter). Capacitance change occurs due to beam deflection, decreasing the distance across the dielectric air gap between the probe and flexure. The measurement range of the capacitance probe is 0-.254 mm (0-.01 in.). The bearing/housing combination, when placed on the test journal, is restrained from rotation by a torque arm resting against the beam which is a thin, metal strip. The force produced to deform this metal strip is converted into a friction coefficient value. The task is to arrive at a friction coefficient value from the measurement of the amount of pen deflection on the potentiometer chart.

A schematic of the flexure and simple beam theory equations are shown in Fig. 25. From these equations, the amount of force required to deflect the metal strip a given amount is known. In practice, this is also easily arrived at experimentally. This is done by using a pulley arrangement. Calibrated weights are hung to apply a controlled or calibration force, (F_{cal}.) to the flexure as shown in Fig. 24. After the force is applied, a measurement of pen displacement is taken. Since the bearing within the pulley is not frictionless, two measurements are taken, one by lowering the weight very gently and another by slightly pushing the weight downward, realeasing and measuring its equilibrium position. An average of the two measurements is used. A refinement of this device would be to replace these pulley bearings with hydrostatic air bearings. This would virtually eliminate the need for correctin due to pulley friction.

When a sampling is taken at various weights, a calibration chart is devised to determine friction coefficient as a function of pen deflection. This is calculated by using simple force-moment equations and the dimensions of the bearing and housing configuration. The torque arm to the level of the calibration pulley has a moment arm of 10.16 cm (4 in.) and the foil bearing radius constitutes a moment arm that is 1.91 cm (.75 in.) from the axis of rotation. The torques are the products of these moments and the corresponding calibration and friction forces and are equal to maintain a balance of forces.

The balance of forces is shown schematically in Fig. 26. By applying the calibrated weight, F_{cal}, to the pulley, a deflection is produced in the metal flexure. The capacitance probe transmits the signal to the measurement device and the amount of beam deflection is shown on the meter. A corresponding value on the potentiometer is produced by the amount of pen deflection on the chart. These values are shown in Fig. 26. The values of beam deflection using simple beam theory are shown in Fig. 27. The tabulated values of pen deflection are plotted vs. the amount of calibrated applied, F_{cal}, as seen in Fig. 28. The theoretical results are also plotted. From the slope of the curve, a conversion factor can be computed as in Fig. 29. The friction coefficient, μ, is a constant for a given normal load multiplied by chart pen deflection, μ = (C)(MM), which then can be plotted in terms of μ vs. pen deflection, MM. Fig. 30 allows the friction coefficient to be read directly from the measured pen deflection.

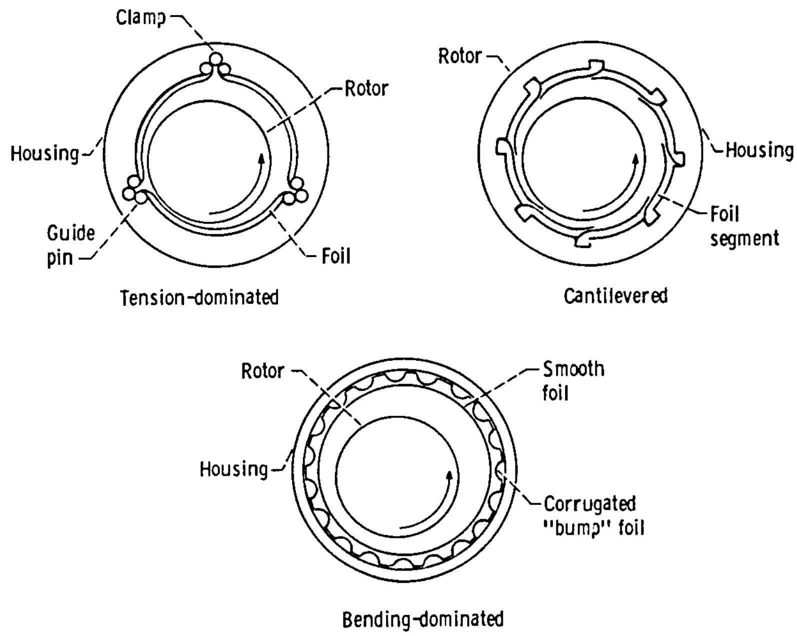


Figure 1. - Types of self-acting compliant foil gas bearings.

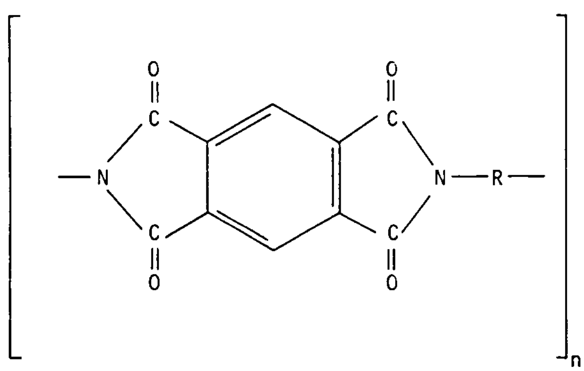


Figure 2. - Basic structure of aromatic polyimide.

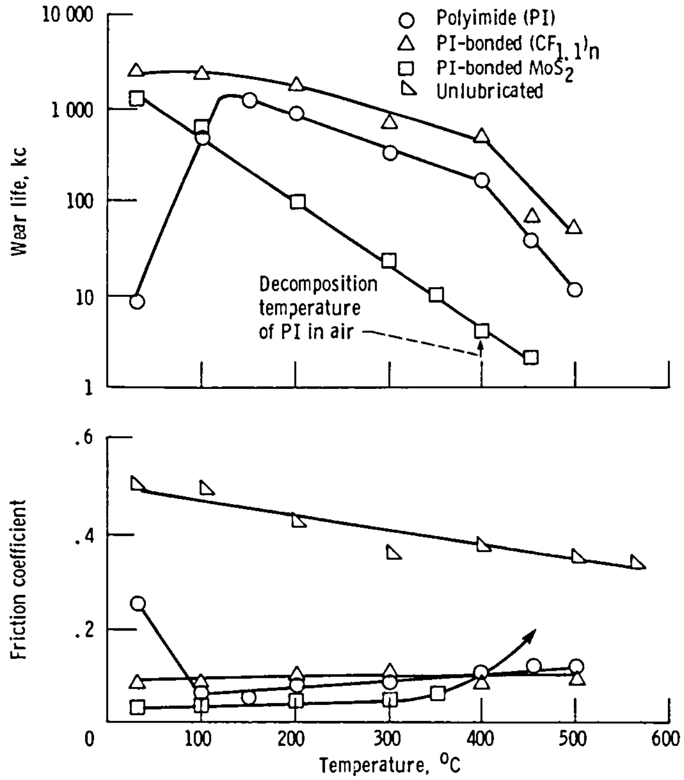


Figure 3. - Friction coefficient and wear life as a function of temperature for three solid lubricant films run in dry air (moisture content, 20 ppm).

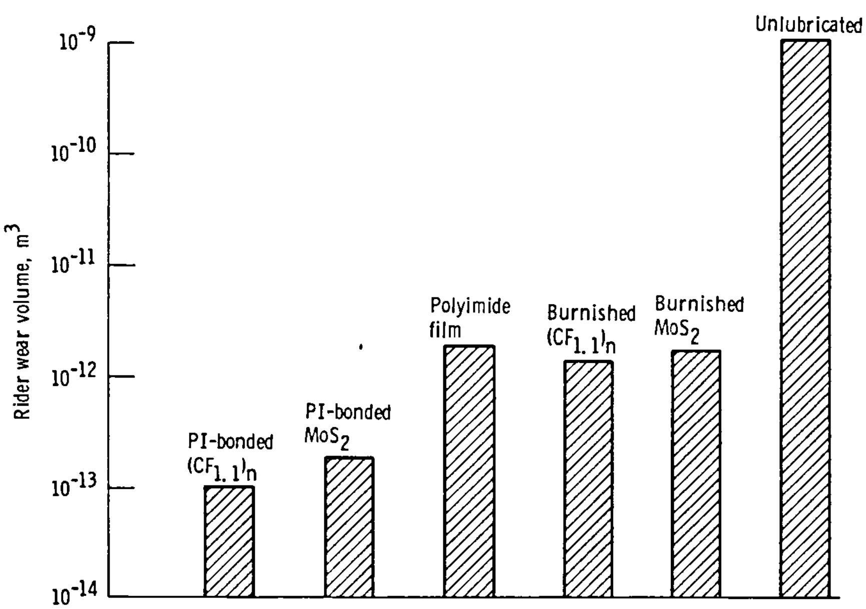
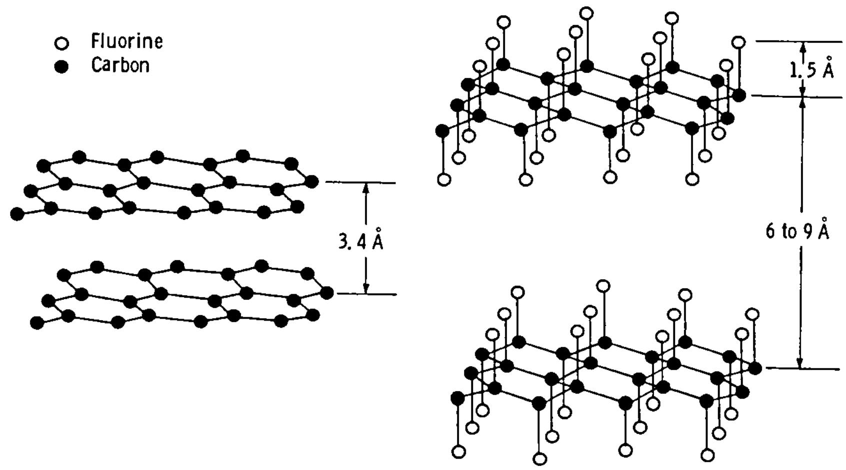


Figure 4. - Comparison of wear to 440° C stainless steel riders which slid for one hour (60 kilocycles) against 440° C stainless steel disks coated with the above solid lubricant films.



(a) Graphite.

(b) Graphite fluoride.

Figure 5. - Structure of graphite and proposed structure of graphite fluoride illustrating the expansion of the carbon layer planes due to the intercalation of fluorine atoms.

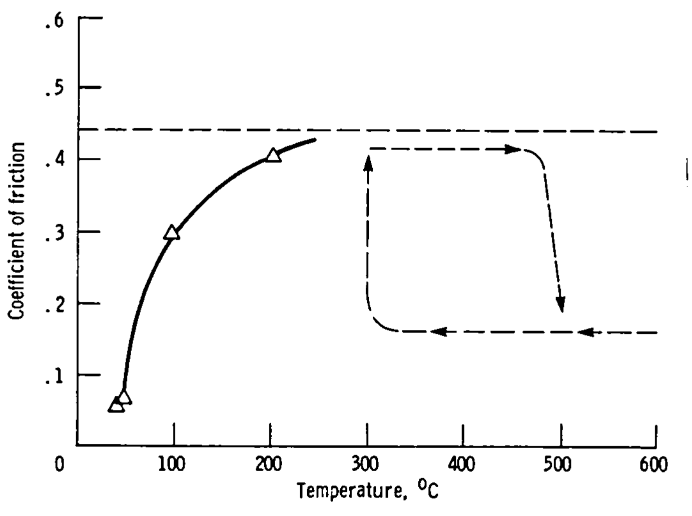


Figure 6. - Lubrication with graphite alone. Sliding velocity, 29 mm/s; load, 180 N.

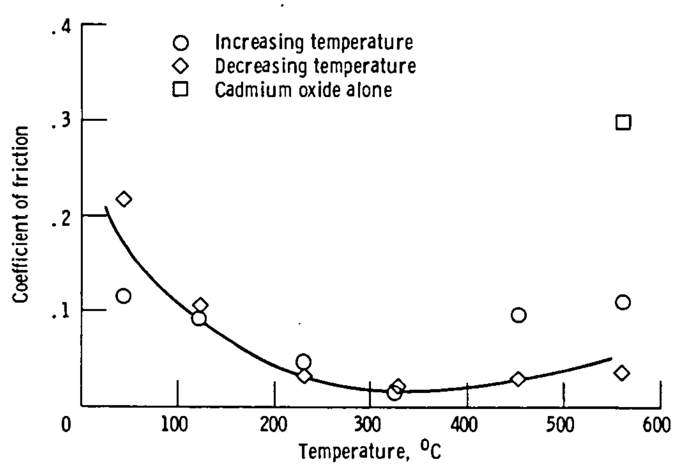


Figure 7. - Lubrication with cadmium oxide-graphite mixture. Sliding velocity, 29 mm/s; load, 180 N.

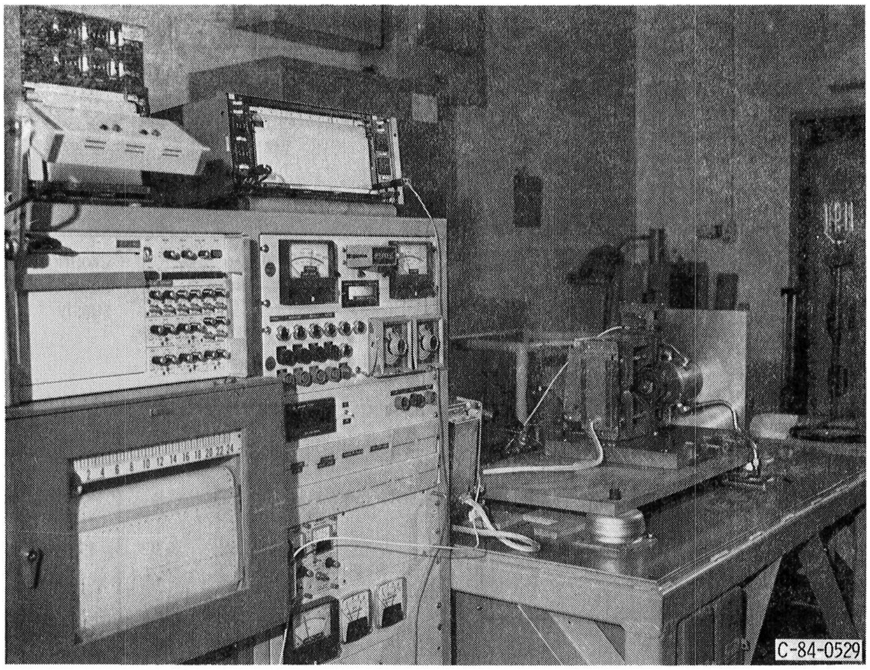


Figure 8. - Foil bearing coating evaluation test apparatus.

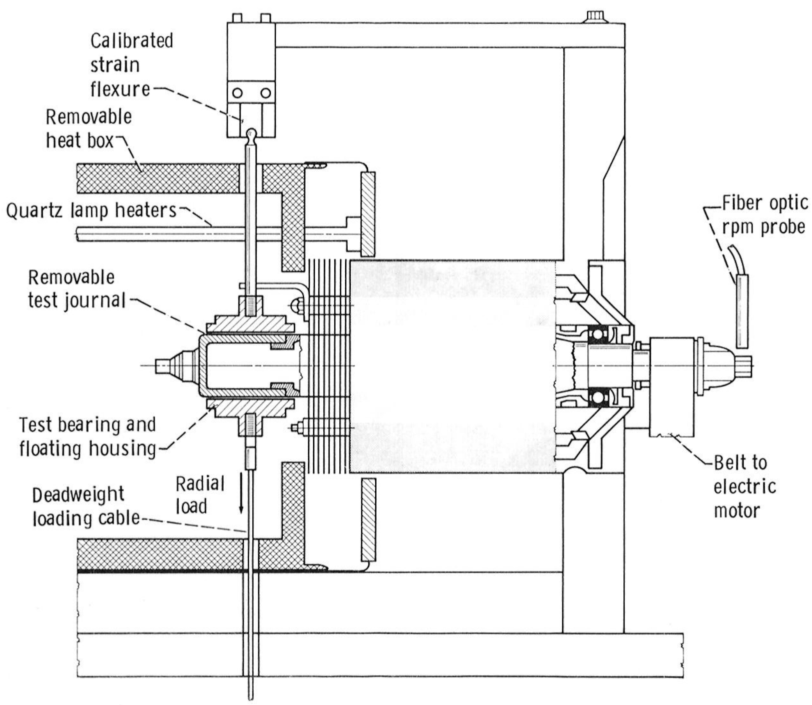


Figure 9. - Foil journal bearing materials test rig.

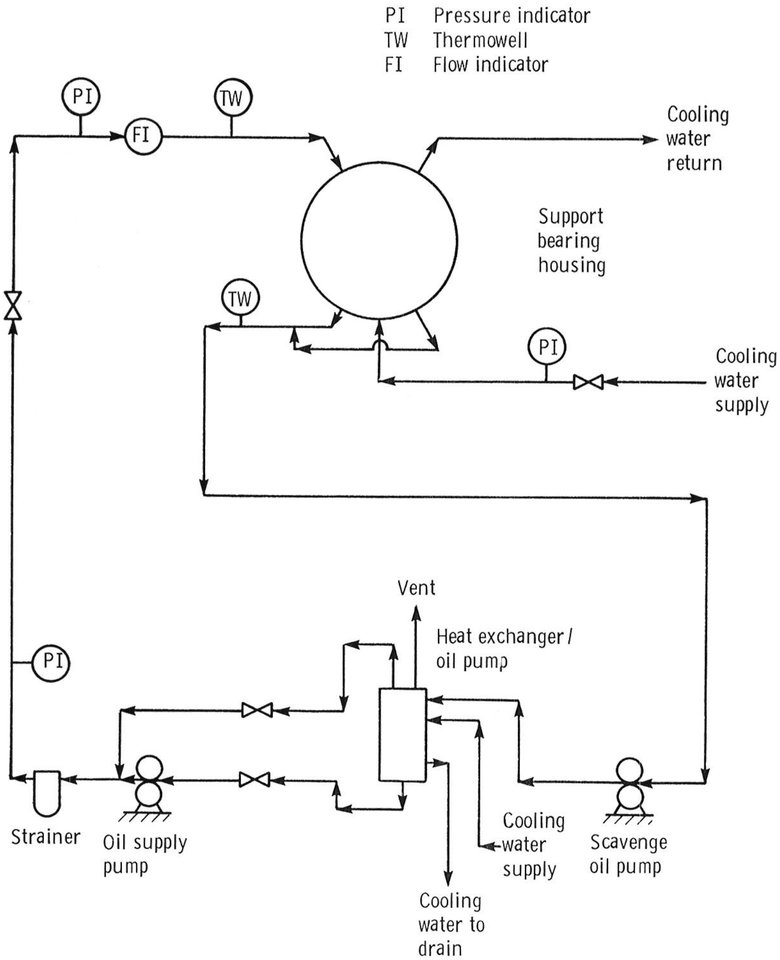


Figure 10. - Schematic of foil bearing test rig lubrication system.

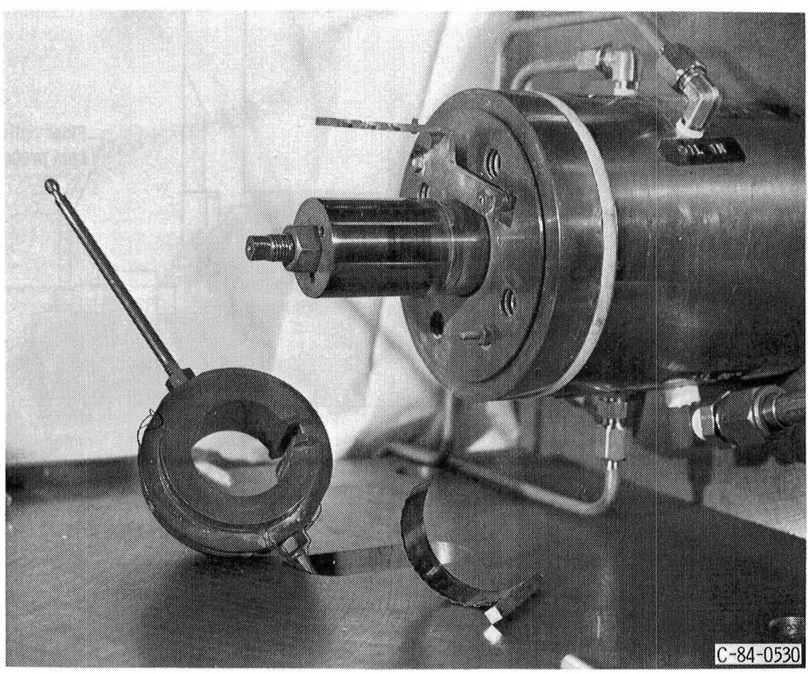


Figure 11. - Test bearing, floating housing, and test journal.

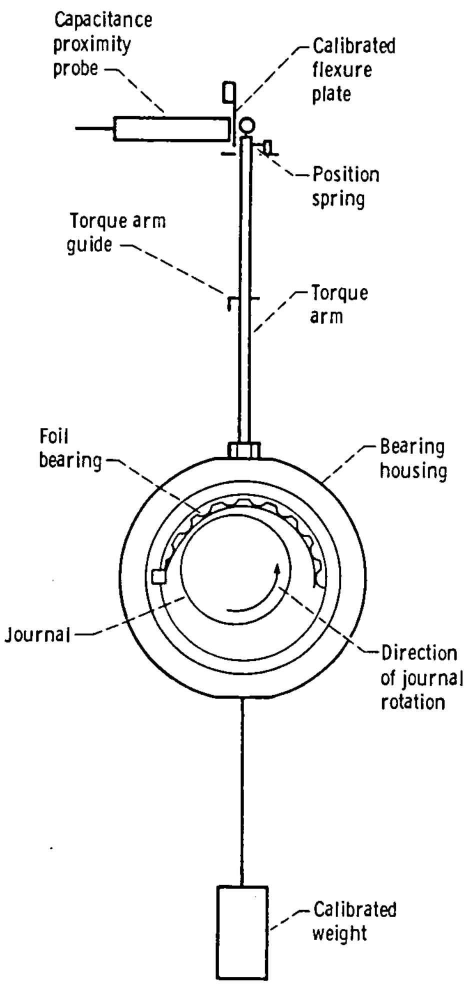


Figure 12. - Arrangement of foil bearing and journal during frictional drag testing.

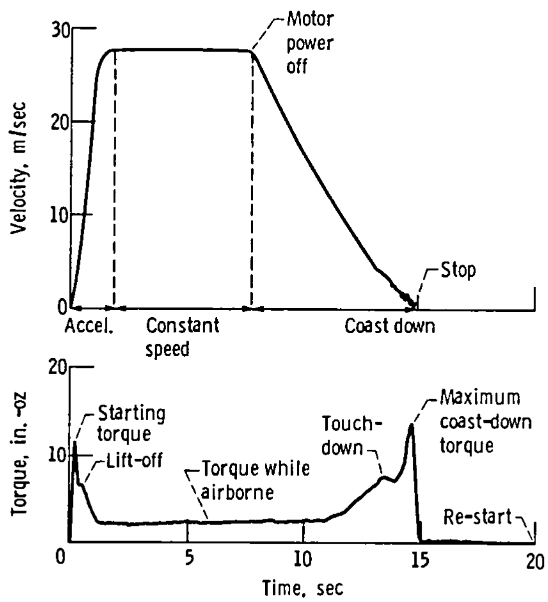


Figure 13. - Typical torque profile of a foil bearing during a single start/stop cycle.

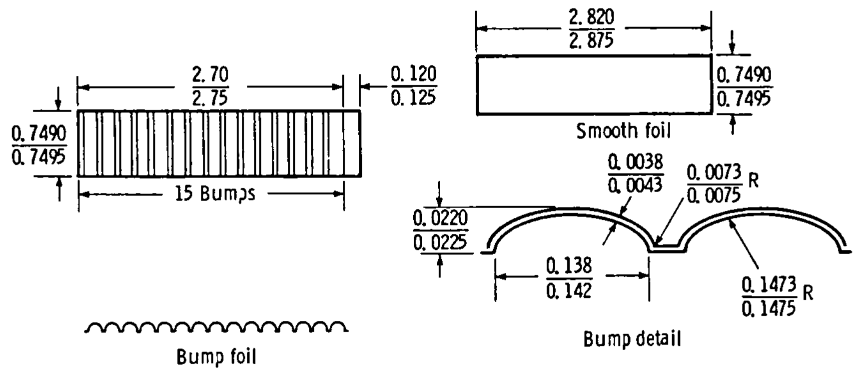


Figure 14. - Details of compliant foil gas bearing. Material, Inconel x-750. Dimensions are in inches.

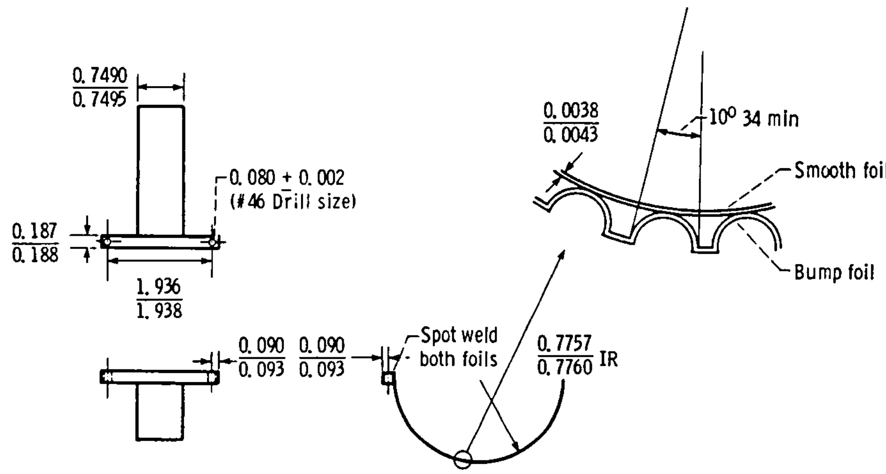


Figure 15. - Compliant foil gas bearing assembly. Material, Inconel x-750. Dimensions are in inches.

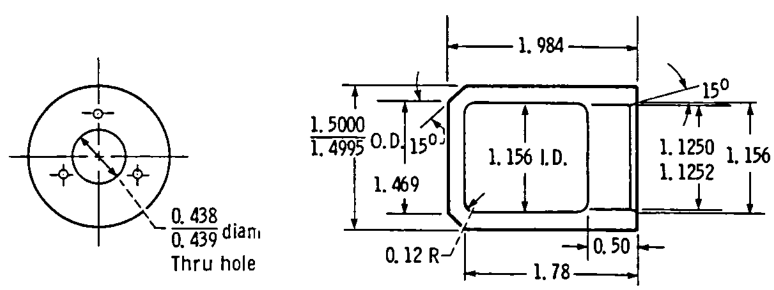


Figure 16. - Journal for compliant foil gas bearing. Material, A286 stainless steel. Dimensions are in inches.

Coating	Temperature, °C	Friction coefficients	
		μ_1	μ_2
		Starting	Running at 28 m/s (13 800 rpm)
Bonded MoS_2 (heat cured)	25	0.27	0 to 0.06
Bonded graphite (heat cured)	25	.30	0 to 0.08
	150	.30	0 to 0.08
	230	.26	0 to 0.08
Silicate-bonded graphite/CdO	25	.30	0 to 0.05
	150	.33	0 to 0.05
	230	.28	0 to 0.05
	315	.30	0 to 0.05
Polyimide bonded graphite fluoride			
- As-sprayed	25	.40	0.06 to 0.08
	150	.26	.06 to 0.08
	230	.23	.06 to 0.08
	315	.20	.06 to 0.08
- Burnished $CF_{1.1}$ overlay	25	.30	.03 to 0.05
	150	.25	.03 to 0.05
	230	.20	.03 to 0.05
	315	.10	.03 to 0.05

Figure 17. - Typical friction coefficients for foil bearings with various coatings.

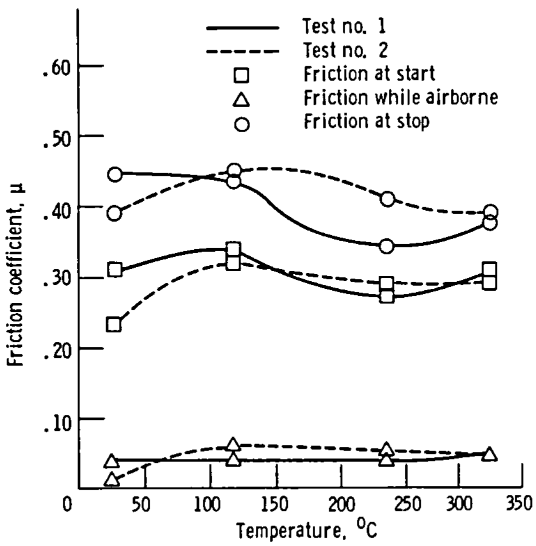


Figure 18. - Friction coefficient versus operating temperature for silicate bonded graphite/CdO coated foil bearing and A286 journal.

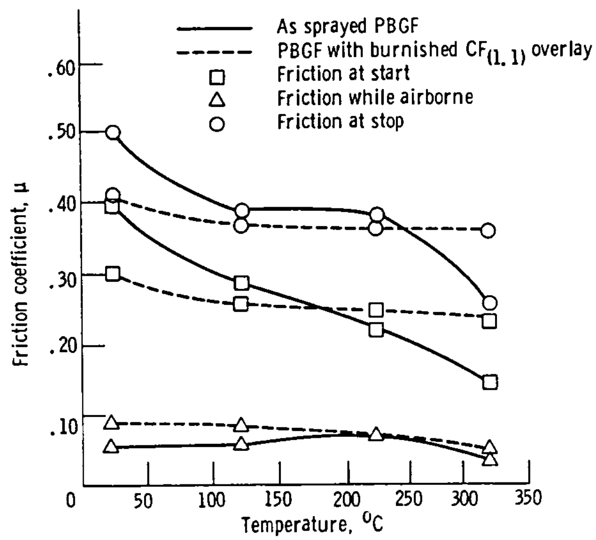


Figure 19. - Friction coefficient versus operating temperature for polyimide graphite fluoride coated foil bearing and A286 journal.

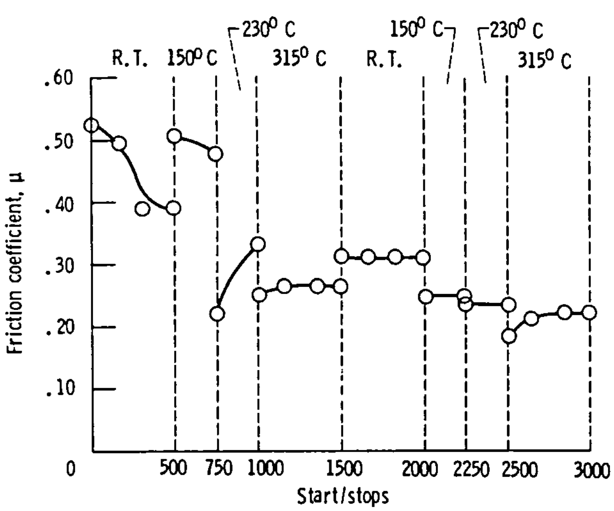


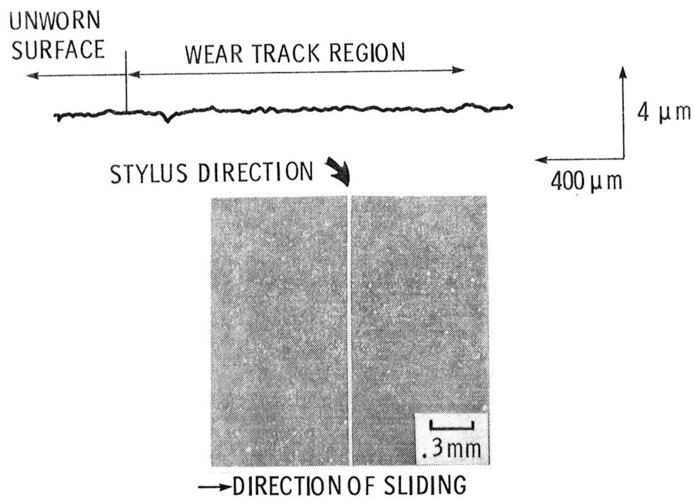
Figure 20. - Effect of run-in on starting friction for PBGF coated foil bearing.

Coating type	Range of bearing temperatures, °C	Endurance life number of start/stop cycles to failure	
		Minimum	Maximum
Bonded MoS_2[a] (heat cured)	25	175	400
Bonded graphite[a] (heat cured)	25 to 230	800	1000
Silicate-bonded graphite/CdO	25 to 315 25 to 430	4500 [b]2070	7500
Polyimide-bonded graphite fluoride	25 to 315	>9000	>9000

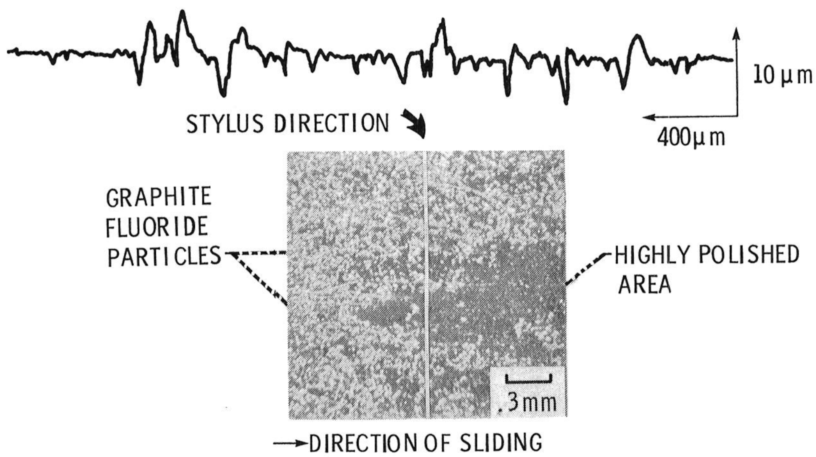
[a]Proprietary coatings bonding agents unknown

[b]Only one test to 430° C

Figure 21. - Coating durability.



(a) Journal.



(b) Foil Bearing.

Figure 22. - Photomicrographs and surface profiles of specimens coated with polyimide bonded graphite fluoride after 9000 start stop cycles. (Surface profiles are 90^0 to sliding direction.)

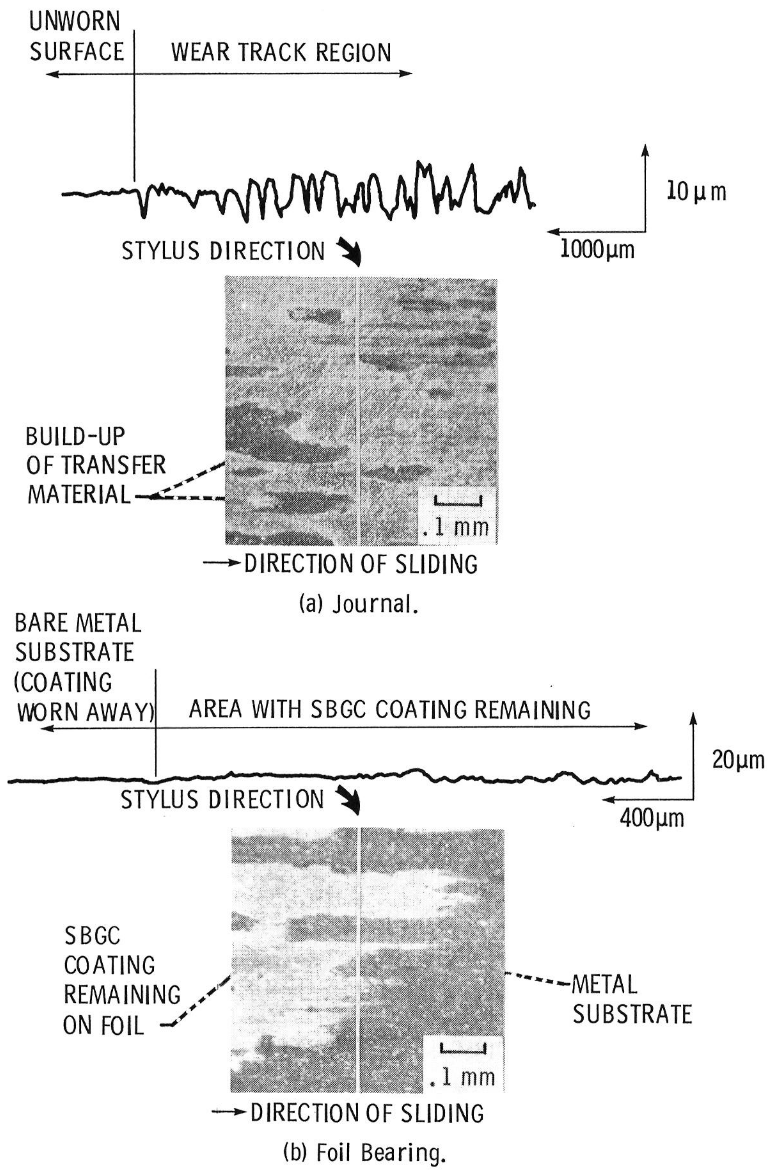


Figure 23. - Photomicrographs and surface profiles of specimens coated with silicate bonded graphite cadmium oxide after 7500 start stop cycles. (Surface profiles are 90° to sliding direction.)

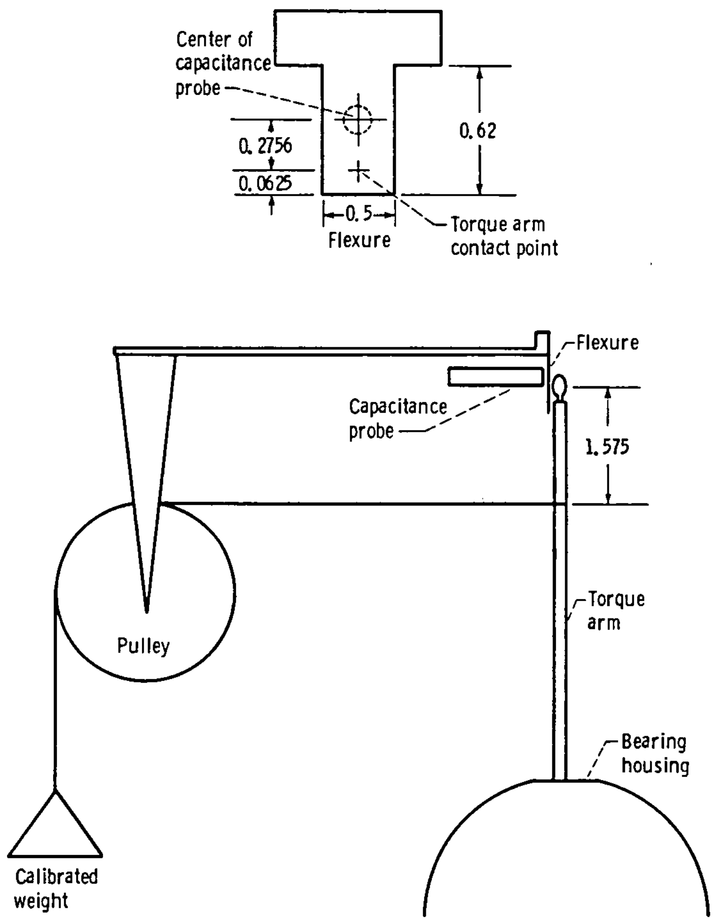
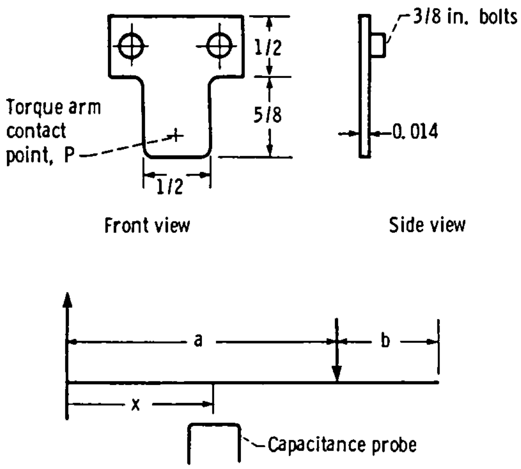


Figure 24. - Arrangement of calibration device. Dimensions are in inches.



$a = 31/64$ in. $= 0.484375$ in. $P = W(1.575$ in.$)$ (W-lb of weight)

$b = 9/64$ in. $= 0.140625$ in. $E = 30 \times 10^6$ psi (stainless steel)

$c = 15/64$ in. $= 0.234375$ in.

Moment of inertio, I: $I = th^3/12 = 0.5(0.014)^3/12 = 1.14 \times 10^{-7}$ in.^{4}

Deflection equation: $EIy = (Px^2/6)(3a - x)$

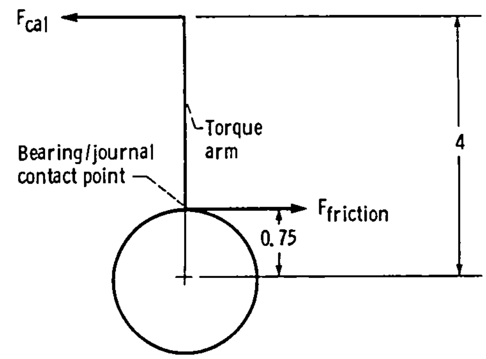
$$(30 \times 10^6)(1.14 \times 10^{-7})y = \frac{W(1.575)(0.234375)^2}{6}(3(0.484375) - 0.234375)$$

$$y = 0.0051385\ W$$

$$W = 454\ F_{cal}\ (g)$$

$$Y = 0.0000113\ F_{cal}.$$

Figure 25. - Deflection of flexure from simple beam theory. Dimensions are in inches.



Calibrated weight, F_{cal}, g	Measured beam deflection, in.	Chart deflection, mm High	Chart deflection, mm Low	Chart deflection, mm Average	Beam deflection per chart deflection, in./mm x 10^{-5}
10	0.00005	2.0	1.5	1.75	2.9
20	.00015	3.5	3.0	3.25	4.6
30	.00025	5.0	4.5	4.75	5.3
50	.00045	8.0	8.0	8.0	5.6
60	.00060	10.0	10.0	10.0	6.0
70	.00065	13.0	12.0	12.5	5.2
100	.00100	18.0	17.0	17.5	5.7
150	.00155	27.0	26.0	27.5	5.6
200	.00200	36.0	35.0	35.5	5.6

Figure 26. - Force balance during calibration and observed values of chart pen deflection during application of calibrated weights. Dimensions are in inches.

Calibrated weight, F_{cal}, g	Calculated beam deflection, in.	*Corresponding pen deflection, mm
10	0.00011	1.97
20	.00023	4.12
30	.00034	6.09
50	.00057	10.21
60	.00068	12.18
70	.00079	14.16
100	.00113	20.25
150	.00170	30.46
200	.00226	40.50

*Using beam deflection per chart deflection: 5.6×10^{-5} in./mm

Figure 27. - Theoretical deflections using simple beam theory.

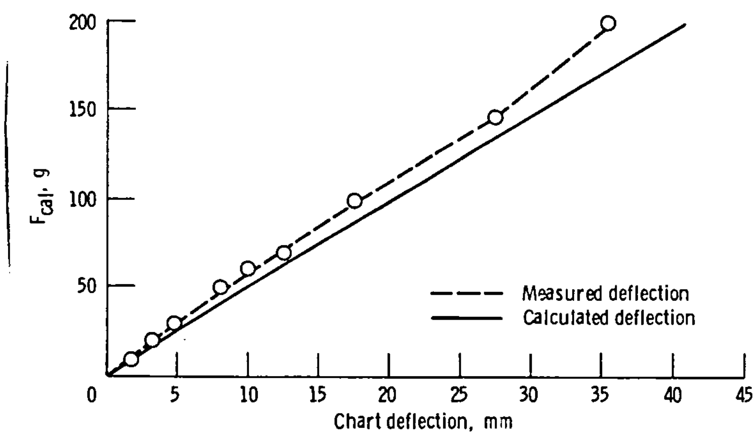
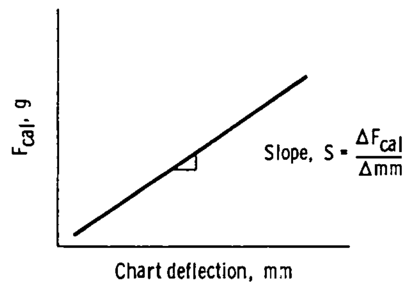


Figure 28. - Conversion factor chart, F_{cal}, g versus chart pen deflection, mm.



Force balance: $0.75\ F_{friction} = 4\ F_{cal}$ (From fig. 26)

$$F_{friction} = 5.33\ F_{cal} = 5.33\ S\ (\Delta mm)_{chart\ deflection}$$

Friction equation: $F = \mu N$

F - Tangential force = $F_{friction}$

N - Normal load (1022 g) (bearing, housing, additional weights)

μ - Coefficient of friction

$$\mu = \frac{F_{friction}}{N} = \frac{5.33\ S\ (\Delta mm)}{1022\ g} = (0.00522\ S)\ (\Delta mm)_{chart\ deflection}$$

Conversion factor: $C = 0.00522S$

Coefficient of friction: $\mu = C\ (\Delta mm)_{chart\ deflection}$

Figure 29. - Computational procedure for conversion factor to plot friction coefficient, μ, versus chart pen deflection.

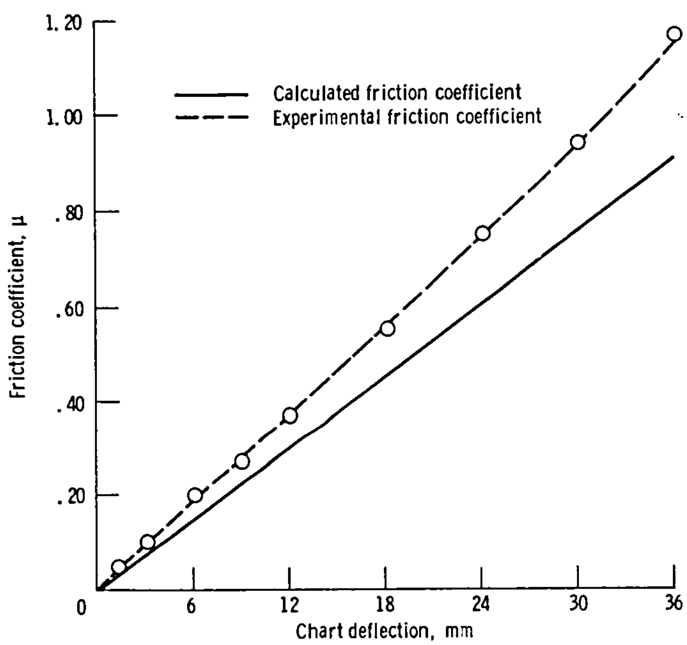


Figure 30. - Coefficient of friction conversion chart.

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16. Abstract

An experimental apparatus and test procedure are developed to compare the performance of two solid lubricant coatings for air-lubricated compliant foil gas bearings in the temperature range of 25 °C to 315 °C. Polyimide-bonded graphite fluoride (PBGF) and silicate-bonded graphite with cadmium oxide additive (SBGC) are tested extensively for durability and frictional characteristics. A partial arc bearing, constructed of Inconel X-750, is coated on the bore with one of these coatings. The solid lubricant protects the bearing and journal surfaces at times when sliding contact will occur. A journal made from A286 stainless steel is used. The foil is subjected to repeated start/stop cycles under a 14 KPa (2 psi) bearing unit loading. Sliding between the bearing and journal occurs during liftoff and coastdown at surface velocities not exceeding 6 m/sec (3000 rpm). Testing continues until 9000 start/stop cycles are accumulated or high starting torque is present indicating coating failure. Failure is defined as sufficient metal to metal contact to cause surface damage. Performance comparisons reveal that although both coatings survive thousands of start/stop cycles, only the PBGF coated bearing achieves the specified 9000 start/stops. There is enough wear on the SBGC coated bearing to warrant termination of the test prior to 9000 start/stop cycles due to coating failure. The frictional characteristics of the PBGF are better at the elevated temperatures than at lower temperatures: a marked increase in sliding friction occurs as the temperature decreases. The SBGC maintains relatively constant frictional characteristics independent of operating temperature.

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