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# EXPERIMENTS IN HYDRODYNAMIC GREASE BEARINGS

John L. Burch and Peter H. Broussard, Jr.
Astrionics Laboratory

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16. ABSTRACT

Gyro bearings used on extended space missions should have low friction, be relatively stiff, be capable of multiple starts and stops, and operate in a smooth stable manner without attention for a year or longer. Ball bearings are commonly used for gyro bearings but their life expectancy has been difficult to predict.

The suitability of using hydrodynamic conical- and spherical-shaped spiral-grooved grease bearings as a replacement for gyro ball bearings has been investigated. Grease bearings have been built and tested for rotors which vary in diameter from a few centimeters to 30.5 centimeters and in weight from a few grams to 22.7 kilograms. Speeds from 12 000 to 50 000 rpm have been maintained for hundreds of hours in both air and vacuum environments. Generally, these bearings have operated satisfactorily and retained their lubricant.

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# EXPERIMENTS IN HYDRODYNAMIC GREASE BEARINGS


## Summary

Gyro bearings used on extended space missions should have low friction, be relatively stiff, be capable of multiple starts and stops, and operate in a smooth stable manner without attention for a year or longer. Ball bearings are commonly used for gyro bearings but their life expectancy has been difficult to predict.

The suitability of using hydrodynamic conical- and spherical-shaped spiral-grooved grease bearings as a replacement for gyro ball bearings has been investigated. Grease bearings have been built and tested for rotors which vary in diameter from a few centimeters to 30.5 centimeters and in weight from a few grams to 22.7 kilograms. Speeds from 12 000 to 50 000 rpm have been maintained for hundreds of hours in both air and vacuum environments. Generally, these bearings have operated satisfactorily and retained their lubricant.


## Introduction

The Marshall Space Flight Center (MSFC) has space requirements for bearings to support the spin axis of inertial gyros that weigh anywhere from 4.4482 to 444.82 N (1 to over 100 lb force). These bearings must have low torque, high stiffness, and long life. Ball bearings have been used for this purpose. Longer missions now require the use of higher precision bearings that have their lubricants closely metered at rates which insure that the balls run on elastohydrodynamic films of only a few millionths of a centimeter thickness. Only in this mode of operation can the bearing operate successfully for over 2000 hours. [1]

MSFC has promoted the improvement of precision ball bearings and has applied the laws of probability theory[2] in an effort to shorten the time required for selecting reliable long life gyro bearings. Other types of bearings have been investigated for this purpose in the hope of developing a type which meets performance requirements and is less marginal in operation and less susceptible to fatigue failure. Of particular interest to MSFC was the research on hydrodynamic, spiral-grooved, oil and grease bearings done by Dr. E. A. Muijderman, [3] where a considerable discussion is devoted to the conical- and spherical-shaped spiral-grooved bearings. These are single, dead-ended, inward pumping bearings that can be machined on the end of a shaft. In 1966 several test gyro rotors supported by spiral-grooved grease bearings were designed, built, and tested. In general, the theory and nomenclature of Dr. Muijderman were used. However, special performance requirements were applied for bearings to be used in gyros designed for long life, and a new method was developed for grooving these small bearings.

A typical conical-shaped spiral-grooved bearing (SGB) and nomenclature are shown in Figure 1. The

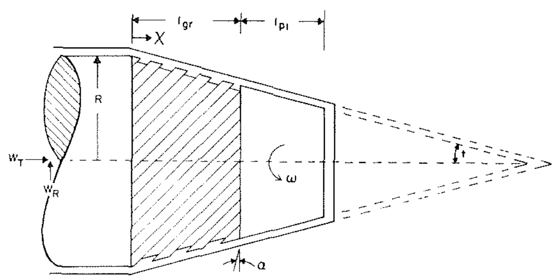


Figure 1. Conical-Shaped Spiral-Grooved Grease Bearing

spherical-shaped SGB is similar to the conical-shaped SGB. Bearing performance equations are shown in Table I. Thrust and radial loads and moments are proportional to the dynamic viscosity, $\eta$, of the lubricant. The viscosity of all lubricants is temperature sensitive and can also be load sensitive in certain bearing applications, as shown in Figure 2.[4,5] The viscosity of all lubricants is more or less sensitive to the rate of shear within a bearing, and greases are more sensitive in this respect than oils. Figure 3 shows the relationship between increase in cone dimension and reduced apparent viscosity of a typical lubricant.[5]

MSFC bearings supporting a gyro spin axis are also required to operate in all positions, to have isometric load carrying capability, and to maintain lubrication with minimum transfer of mass to the rotor for periods of up to 1 year of intermittent service. When the stiffness of the conical SGB in thrust and radial directions is plotted versus half cone angle, t, the intersection and half cone angle required for isometric load-carrying capability is about 29 degrees.

TABLE I. BEARING PERFORMANCE EQUATIONS

$$W_T \approx \frac{1.06\,\eta\,\omega\,R^4}{h^2_{min}}\,,$$

Thrust coefficient, $W_T$, varies from 1.40 for half cone angle t = 10 degrees to 1.06 for t = 15 degrees. The ratio of load to torque is maximun at t ≈ 15 degrees.

$$W_R \approx \frac{1.1\,\eta\,\omega\,R^4}{h^2_{min}} \quad @ \quad \epsilon_R = 0.5$$

$$Q_f \approx 2\,W_T\,f\,r\,\omega$$

where:

$W_T$ = Thrust load-carrying capacity (N)

$W_R$ = Radial load-carrying capacity (N)

$\eta$ = Dynamic viscosity of the lubricant $(\frac{Ns}{m^2})$

$\omega$ = Angular velocity $(\frac{rad}{s})$

R = Radius of conical bearing (m)

$h_{min}$ = Minimum film thickness (m)

$\epsilon_R$ = Radial eccentricity

$Q_f$ = Power consumed by conical bearing (W)

f = Coefficient of friction $\dfrac{2.5\,h_{min}}{R}$

NOTE: These equations are approximate. More exact solutions require that a computer program be used.

This is shown in Figure 4 and described in Reference 5. Design parameters other than half cone angle, t, are radius R; ratio of grooved to ungrooved length, $\frac{l_{gr}}{l_{pl}}$ ; angular speed, $\omega$; and groove angle, $\alpha$, versus the velocity vector. The principle of operation of the SGB, loaded in the radial direction, is similar to that of all hydrodynamic bearings. The relative motion of the bearing surface sets up shear stresses within the lubricant which generate a positive pressure gradient. This gradient is proportionally higher in the circumferential area of minimum bearing gap and over the land area as compared to the grooved area. When the conical SGB is loaded in the axial direction, a positive pressure gradient is generated by the inward pumping action of the grooves.

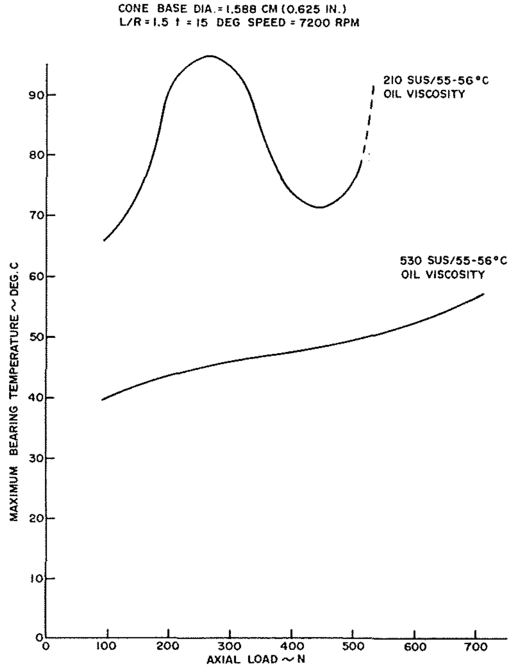


Figure 2. Maximum Bearing Temperature Versus Load With Two Different Greases

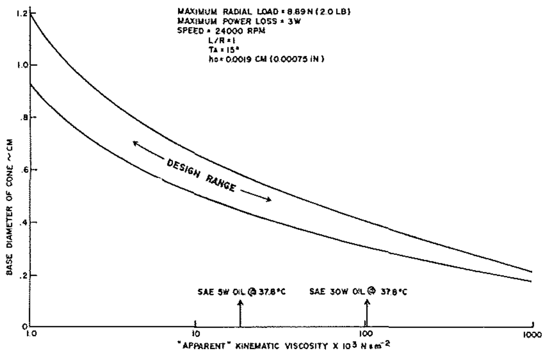


Figure 3. Relationship Between the Cone Dimensions and the Lubricant "Apparent" Viscosity

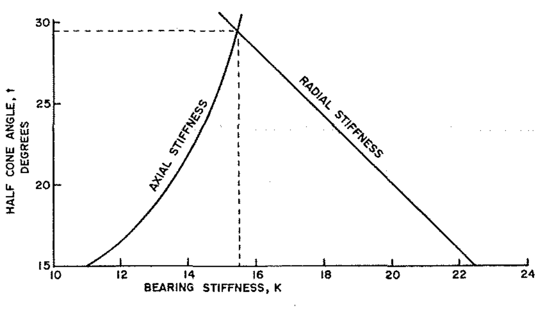


Figure 4. Half Cone Angle Versus Bearing Stiffness

## Initial Work

Our first effort in the area of spiral-grooved oil and grease bearings was to develop an acceptable method for grooving small bearings; preferably a method that would work on stainless steel shafts as small as 0.318 cm (0.125 in.) diameter. Two general methods were considered; etching the grooves and cutting them. It became apparent from the beginning that either method would be made unduly complicated by trying to make each groove a perfect loxodrome, crossing the velocity vector at a constant angle, as assumed by theory. Because the length of the grooved area is only 57% of the total bearing length, the grooves could be made spirals whose angle with the circumferential velocity vector increases toward the small end of the cone or sphere. An average effective groove angle, $\alpha$, can then be assumed with small error. Photo-etching specialists and machinists worked on this problem. Because the machinists were first with a workable solution, all of our bearings have had machined grooves; however, grooves can also be accurately etched in small conical and spherical surfaces by using a transparent epoxy for the mask material.[6]

A shop setup for cutting spiral grooves in conical- and spherical-shaped bearings is shown in Figure 5. The bearing is chucked in a rotary fixture with the indexing head and hydraulic tracer attachment carried on a high precision lathe which is run at 200 rpm. Tungsten carbide cutting tools are used which have points ground to about $5.08 \times 10^{-3}$ cm (2 mils) diameter and a 30-degree included angle. The hydraulic tracer attachment supplies constant pressure between tool and conical or spherical bearing surface. Each new spiral groove is started by

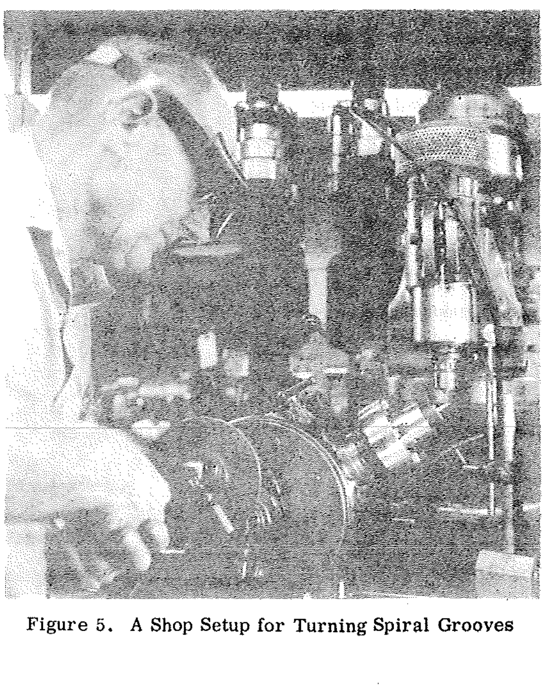

Figure 5. A Shop Setup for Turning Spiral Grooves

rotating the indexing head to a new setting. All grooves are started at the large end of the bearing and faded out at the end of the grooved section by relieving pressure in the hydraulic tracer.

## Preliminary Tests

The first SGB test rig, shown in Figure 6, consisted of a separate rotor, shafts with spiral-grooved male bearings, cup-shaped plain female bearings, frame, and threaded rod and knurled thumb wheel. Threads of 15.75 per cm (40 per in.) were machined on the rod for micrometer lateral adjustment of the grease gaps. The

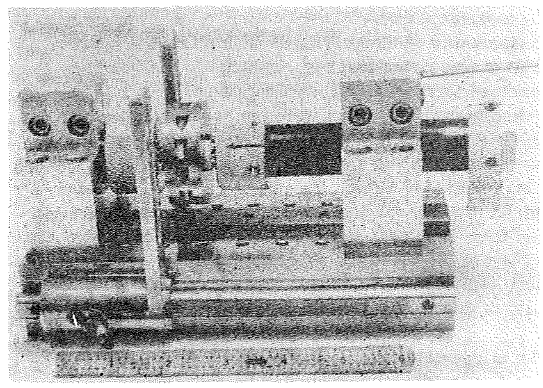

Figure 6. Spiral-Grooved Bearing Test Rig

rotors were designed for rotation by air turbine. Buckets were cut in the center outer circumference of the rotor and two air jets were mounted on the frame, 180 degrees apart. Several sets of brass rotors were made in sizes varying from 3.18 to 6.35 cm (1.25 to 2.50 in.) in diameter. Shafts, including male bearings, were made of aluminum 0.635 cm (0.25 in.) in diameter (Fig. 7). The shop could easily gain experience grooving the relatively soft aluminum, and we could judge bearing performance by observing any tendency of the grooves to smear under load. Both conical- and spherical-shaped bearings were made and tested. The shop had no more difficulty making one shape than the other. The rig was run in all positions

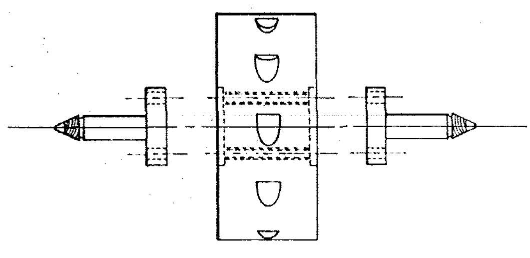

Figure 7. Brass Rotor and Aluminum Shaft Assembly

but principally in the vertical position, identifiable as the thrust axis. Numerous oils and greases were tested, few of whose viscosities were known. Because no provision had been made for torque measurements, estimates of torque caused by bearing drag were judged by rundown time. After testing various lubricants, one gyro grease, a type BRB-2 (green grease), was chosen. A set of conical-shaped spiral-grooved grease bearings and shafts were attached to a 5.08 cm (2.0 in.) diameter rotor and the bearings were lubricated with type BRB-2 grease. Nominal grease gap was set at $5.08 \times 10^{-3}$ cm (0.002 in.). The test rig was placed in a protective nose cone for safety purposes, and the rotor was brought up to a speed of 50 000 rpm and left running. The rotor was maintained at this speed for 1118 hours before noticeable vibration was observed.

Disassembly of the bearing revealed that the source of vibration was in the test rig, external to the bearings. The bearings themselves showed no sign of wear. The grease had turned somewhat dark but apparently had not lost its efficiency as a lubricant. Several high speed tests were run with the same rig, using the 3.175 cm (1.25 in.) diameter rotor. Speeds of 100 000 rpm were maintained for several minutes during each test. Bearings retained their lubricant and showed no wear after these tests. One of the principal shortcomings of the micrometer head test rig was the flexibility of the shaft and the difficulty of assuring alignment of the bearings.

The micrometer rig was used only to gain experience, and an improved test rig and bearings were designed. To improve alignment, the frame was designed to be hogged out of a single aluminum block with ends line-bored to receive the bearings, as shown in Figure 8. The rotor was turbine driven for reasons of simplicity. The female bearings were made of bronze, and the one piece rotor and shafts were made of stainless steel. The female bearings were assembled by means of a press fit into the cube. As before, the grooves were cut into the bearing surfaces with the shop rig shown in Figure 5. Two identical 5.08 cm (2.0 in.) diameter rotors were made; one had conical-shaped and the other had spherical-shaped bearing surfaces. An innovation to the bearing

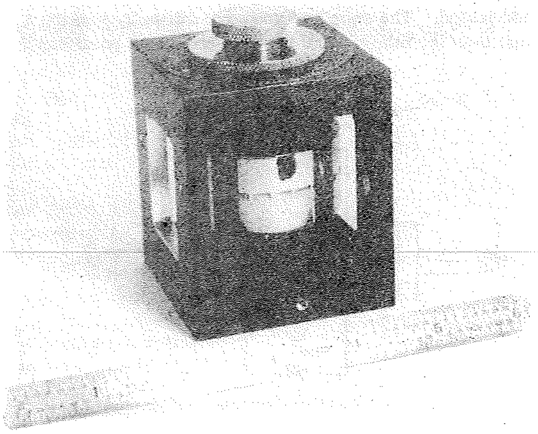

Figure 8. Improved Spiral-Grooved Bearing Test Rig

design was the machining of an inclined surface just inboard of the grooved area, as shown in Figure 9. The nonoperating bearing tends to have its lubricant squeezed out by thrust loads. The indentation and inclined surface inboard of the grooved area provides a reservoir for the squeezed-out lubricant.* The component of centrifugal force returns the lubricant along the inclined surface to the grooved pumping area when rotation begins. The

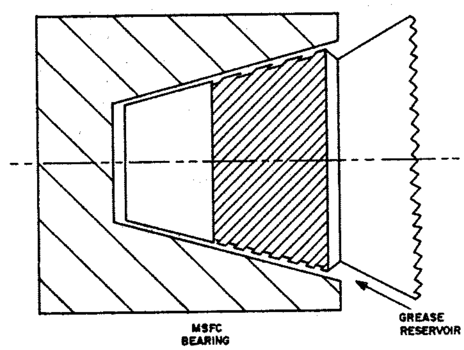


Figure 9. Method for Returning Squeezed-Out Grease to the Bearing Surfaces

designs of the conical and spherical bearings met two specifications: The bearings were optimized for low torque rather than for isometric load carrying and the bearing cone diameter was made small enough that a heavy oil or light grease would be required as a lubricant. The following design parameters were used:

Rotor

Diameter = 5.08 cm (2 in.)

Rim width = 2.54 cm (1 in.)

Weight = 2.224 N (0.5 lb)

Material = stainless steel

Conical Bearings**

Base diameter of cone, 2R = 0.318 cm (0.125 in.)

Half cone angle, t = 15 deg

Number of leads, N = 9

Land to groove ratio, $\gamma \approx 3$

Groove angle, $\alpha \approx 15$ deg

Groove depth, $h_o \approx 1.27 \times 10^{-3}$ cm ($5 \times 10^{-4}$ in.)

Lubricant viscosity, $\eta$ (minimum) $\approx 3.45\times10^{-6}$ Ns cm$^{-2}$ ($5\times10^{-6}$ lb in.$^{-2}$)

*A fluorochemical coating (Nyebar Film Barrier) was applied to the rotor shaft to prevent lubricant migration inboard of the reservoir.

**Bearings were made on opposite ends of the shaft; rotor and shaft were made from one piece of stainless steel.

Preliminary tests were run on both the conical- and spherical-shaped bearings with nominal grease gaps as small as $2.54 \times 10^{-3}$ cm (0.001 in.). Rundown times (bearing torques) were very nearly equal, indicating that alignment of properly designed conical-shaped bearings should no longer be a problem. The conical shape offers more flexibility than the spherical shape in designing for different ratios of axial to radial bearing stiffness, as can be seen by examining the bearing performance equations in Table I. Tests with the spherical-shaped bearing confirmed the comments by Dr. Muijderman that this bearing behaves excellently as a thrust bearing with full fluid-film lubrication but has limited use as a journal bearing. Therefore, the remaining gyro tests were for conical-shaped SGBs. Test results obtained with this rig were very good. Plots of rotor displacement versus rotor speed and actual bearing stiffness versus theoretical bearing stiffness, with the rotor in the axial loaded position, are shown in Figure 10. The

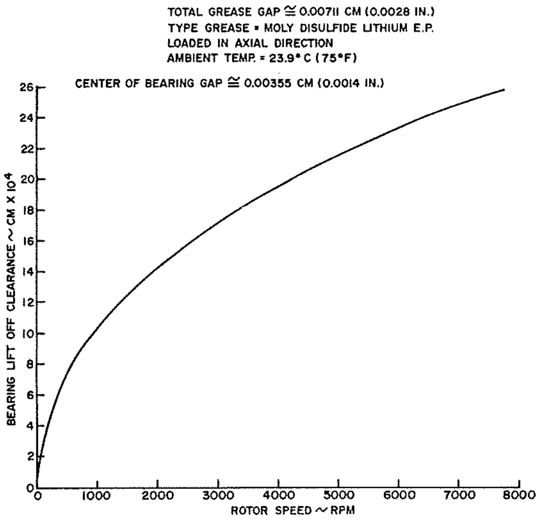


Figure 10a. Turbine-Driven Gyro Rotor in Cubic Housing Supported by Two Conical-Shaped Spiral-Grooved Grease Bearings

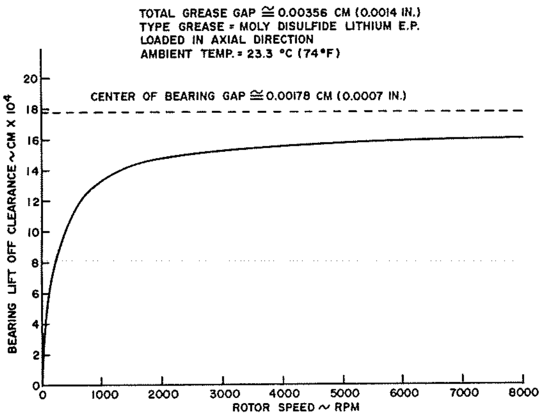


Figure 10b. Turbine-Driven Gyro Rotor in Cubic Housing Supported by Two Conical-Shaped Spiral-Grooved Grease Bearings

grooved male bearing lifts off the smooth surface of the female bearing immediately after rotation begins and its rate of lift-off is much faster for the nominal grease gap of $3.566 \times 10^{-3}$ cm (0.0014 in.) than for the gap of $7.112 \times 10^{-3}$ cm (0.0028 in.) (Fig. 10).

## Conversion of LEV-3 Gyros to Grease Bearings

The next SGB was designed for electric motor-driven rotors. One requirement was that the shaft rotate. Precision motors fitting the requirements were available from a stock of LEV-3 autopilot gyros formerly used in a missile attitude reference system. As shown in Figure 11, the entire assembly is a two-gimbaled configuration using a three-phase, 400 Hz motor. Design speed of the rotor is about 22 000 rpm and power consumption is about 18 watts. These gyros were designed to have a natural resonant frequency between 6900 and 7800 rpm.

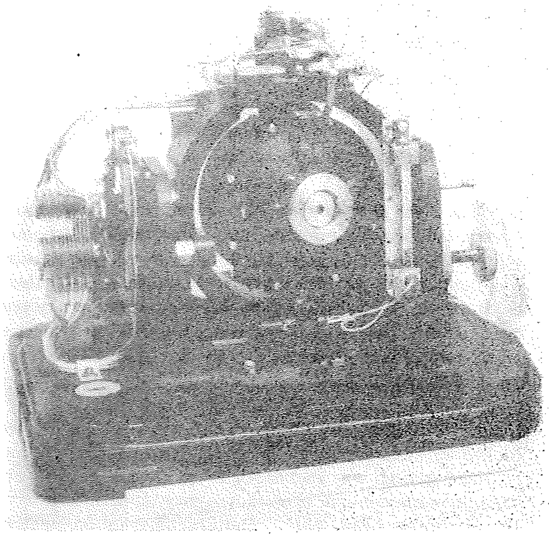

Figure 11. LEV-3 Gyro Converted from Ball Bearing to Spiral-Grooved Bearing Suspension

Conversion of the LEV-3 gyro from a ball bearing spin axis system to an SGB system proved to be very simple. Two units were converted in the following manner. The ball bearing assembly was removed and the threaded SGB female unit was inserted and secured by a lock nut. The ends of the shafts were simply turned down into conical shapes, and spiral grooves were cut using the shop setup shown in Figure 5. Conversions of component parts are shown in Figure 12. The design of the conical-shaped SGBs was the same bearing design used in the previous test. Differences in operation were (1) an increase in the weight of the rotor of almost 2.5 times, (2) an increase in bearing temperatures because of the electric motor, and (3) an increase in running speed. These changes required that a grease be used as the lubricant because of the higher viscosity required for load carrying (Table I). The bearing was also designed for grease lubrication because grease is easier to retain

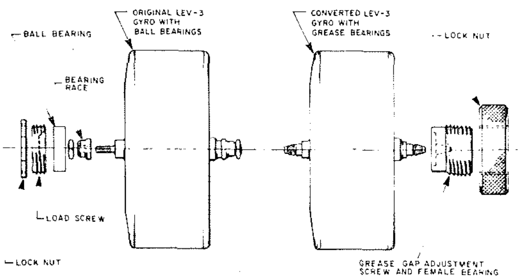


Figure 12. Conversion of LEV-3 Component Parts

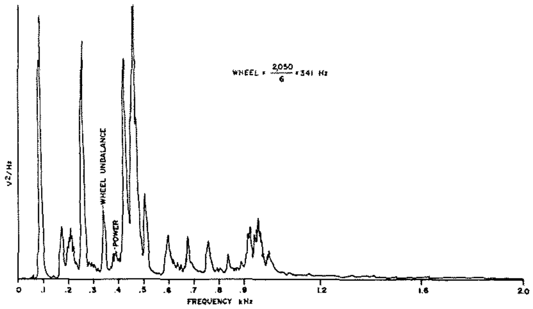


Figure 13b. Power Spectral Density Curve for LEV-3 Gyro Converted to Conical-Shaped SGBs

than oil during intermittent running in different positions, as would be required for operation in space. A description of the LEV-3 rotor is given below:

LEV-3 Rotor

Diameter = 6.985 cm (2.75 in.)

Width = 2.858 cm (1.125 in.)

Weight = 5.427 N (1.22 lb)

Material = mild steel

The first LEV-3 gyro converted to a hydrodynamic grease bearing ran smoothly on the same gyro grease, type BRB-2, used in preliminary tests. Power required was comparable to the original ball bearing suspended configuration, and initial start-up required slightly less current than the ball bearing configuration. An unexpected aspect of performance of the converted LEV-3 gyro was the onset of persisting vibration, a phenomenon not encountered with the ball bearing configuration. This instability disappeared when either the outer gimbal was locked or when the moment of inertia of either the inner or outer gimbal was increased (e.g., by a heavy C-clamp). This phenomenon has been thoroughly investigated by Dr. H. Muira (Fig. 13)[7]. The predominant frequencies are below 500 Hz and are easy to identify. Also seen on the power spectral density curves are the power at 400 Hz and wheel angular velocity. Other spikes in the high power region are judged to be unimportant, but identifiable. While the paper by Dr. Muira correctly describes the instability, we believe that part of the instability was caused by the method of mounting the female bearing to the housing. For example, instability

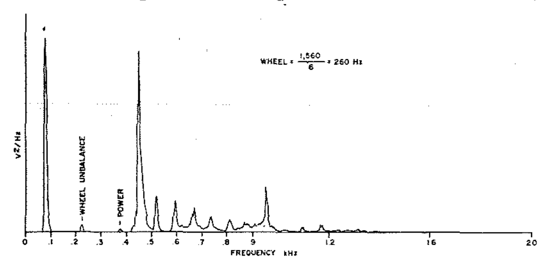


Figure 13a. Power Spectral Density Curve for LEV-3 Gyro Converted to Conical-Shaped SGBs

was also averted by mounting the endplate to the gyro housing with rubber shims.

The outer gimbal of the first LEV-3 gyro was locked and an endurance test of 24 hours was run. An inspection of the bearings showed signs of wear and lubricant starvation. Since the bearing running temperature was about 65°C, we concluded that the lubricant was thinning out too much at this temperature.

The second LEV-3 gyro converted to conical-shaped SGBs was rebalanced on a Schenick type M180-G balancing machine to within about 4 cm-gm before testing. This bearing was lubricated with an extreme pressure type moly disulfide lithium grease. Performance was similar to the first LEV-3 gyro except that there was less instability and the bearings retained their lubricant and showed no signs of wear after about 40 hours of testing.

## Design and Test of a High Angular Momentum Gyro Wheel

The next design was a high angular momentum wheel supported by conical-shaped SGBs. As mentioned previously, SGBs offer several advantages over conventional ball bearings. Therefore, in addition to adapting the SGB design to conventional gyro-size spin axis bearings, we decided to design, build, and test a high angular momentum wheel suitable for stabilizing a space vehicle. These are often called control moment gyros, CMGs, or reaction wheels.

A study was performed on what might be the optimum size for this CMG. Parameters considered were physical size of the wheel, angular momentum, rotational speed, ease of manufacture, balancing, power required, and bearing design. There were also constraints; e.g., excellent manufacturing facilities exist at MSFC and maximum lathe swing is 38.10 cm (15 in.) at the Beryllium Shop. Thus, to make use of these facilities, we decided to limit the diameter of the rotor to 30.48 cm (12 in.). The design goal for angular momentum was set at about $6.8 \times 10^2$ kg m$^2$ s$^{-1}$ (500 ft-lb-s). Round stock of 30.48 cm (12 in.) diameter was available in 4130 steel; this material was chosen for the rotor because of its inherent strength, heat treatability, etc. Furthermore, since alignment and balance are of critical importance, the rotor and shafts were made as an integral one-piece unit weighing 222.41 N (50 lb). This method of construction was used in preference to shrinking the rim onto the web of the wheel or both onto the center shaft which could have maximized the ratio of angular momentum to gross weight.

With these considerations, the design that emerged is shown in Figure 14. The center case was made of

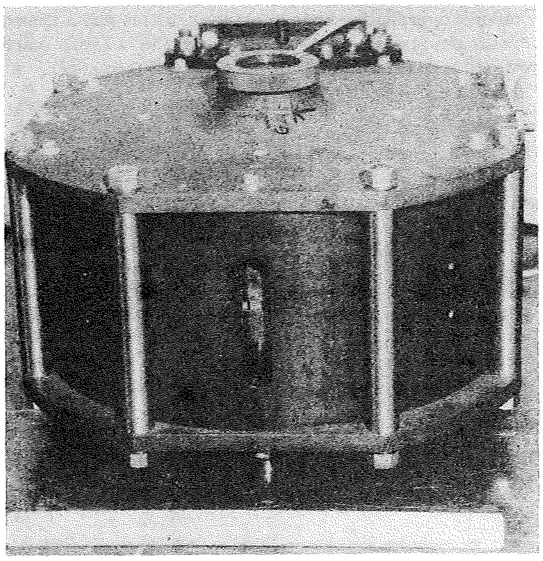

Figure 14a. High Angular Momentum Wheel, Spiral-Grooved Bearings Loaded in Axial Direction, in Air Environment

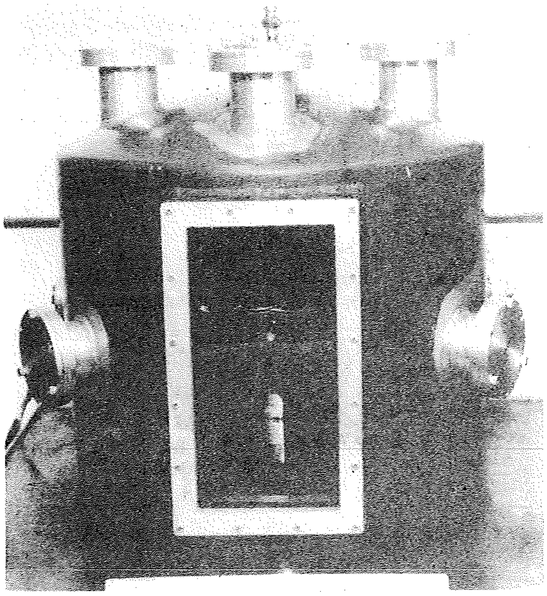

Figure 14b. High Angular Momentum Wheel, Spiral-Grooved Bearings Loaded in Axial Direction, inside Vacuum Chamber

1.27 cm (0.5 in.) thick aluminum pipe section. The two endplates, also made of aluminum, were machined to a tapering thickness from 2.54 cm (1.0 in.) at their edges to 5.08 cm (2.0 in.) at the center hub. The endplates and case were aligned by dowel pins and then screwed together; the two center hubs were line-bored. Additional rigidity and safety were provided by designing the endplates to be secured in place by eight 1.27 cm (0.5 in.) diameter steel bolts. The use of aluminum speeded up machine operations but also caused the case and endplates to have a higher coefficient of thermal expansion than the steel rotor and shafts. This minor drawback was accepted beforehand for the overall advantages of a lighter, cheaper design which could be made without using time-consuming castings and expensive shop procedures. The rotor was machined, hardened to 48C Rockwell hardness, and X-rayed for possible flaws. Grooves were successfully machined in the bearing surfaces after the rotor and shafts were hardened. The same methods were used to machine the grooves as described previously except that greater care was used in lapping the bearing surfaces. The bearing cones were made oversize, grooves were cut, and the bearings were lapped to correct dimensions. Both male and female laps were made by a new process developed at MSFC. These laps are made of anodized aluminum whose surfaces are imbedded with diamond dust before anodizing. The original control moment gyro (CMG) design was for turbine drive by four air jets which impinged on buckets machined into the center of the outer rim surface. This was the cheapest, quickest source of power for proof-testing the bearing design at low angular velocities.

The design of the conical-shaped SGBs was based on the following considerations. As mentioned previously, the rotor is one piece. The basis for this choice is that machining operations are easier and fewer stress calculations are involved. The main object was to develop a grease bearing CMG with an adequately safe rotor, not one with an optimized rotor. Thus, a minumum amount of time was spent in rotor design. The theory used was based on Chree's solution for a rotating disk.[8] Maximum stress at design speed is in the neighborhood of $3.1 \times 10^4$Ns cm$^{-2}$ (45 000 PSI) and occurs near the junction of web and rim. The surface of revolution from this point to the axle is approximately a constant stress surface.

Preliminary test runs showed that the bearings were performing as designed. They supported the rotor in all positions at rotational speeds as high as 1000 rpm and maintained their lubricity. Because of the inconvenience, noise, and limitations of the air jet system, the CMG was modified for drive by electric motors. Two matched electric induction motors were obtained from the Bendix Company. These motors are 3 phase, 400 Hz, 115 Vac, and each produces a torque of about $5.65 \times 10^{-2}$Nm (8 in. oz.) at its maximun theoretical speed of 8000 rpm. The power was furnished by a Tel Instrument frequency converter with variable frequency control between 100 and 1000 Hz. By increasing frequency above 400 Hz and matching impedances, we were able to run these motors up to double their design speed. Figure 15 shows the component parts of the CMG; the original outline of the rotor is shown on the left side, and the modified outline and electric motor are shown on the right side. The SGB configuration is similar to Figure 9 with one important exception. In the larger CMG with a length of 16.51 cm (6.5 in.) between bearings, closer bearing alignment was needed than a threaded female cup could provide. Therefore, a smooth control section was added ahead of the threaded section. This control section was 3.49 cm (1.375 in.) long and machined and lapped to make an ASA class 4 fit with the line-bored endplate hub. By comparison, the threaded length was made cnly 1.59 cm (0.625 in.) long with an ASA class 2 fit. The only purpose of the threaded section, therefore, is to furnish transverse motion and be secured in place by the large knurled lock nut outside the endplate hub. After reassembly with the electric motors, the rotor was dynamically balanced in one plane on a Schenick balancing machine to within $1.472 \times 10^{-2}$Nm (150 cm-gm).

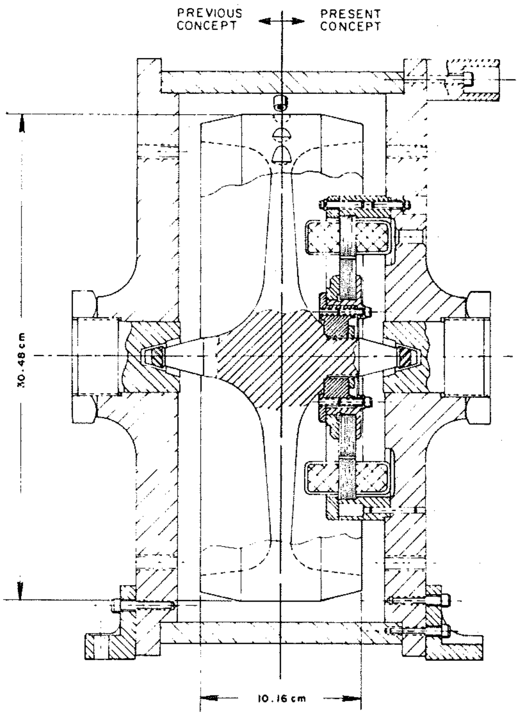


Figure 15. Component Parts of the High Angular Momentum Wheel Showing Conversion from Air Turbine to Electric Motor Drive

The CMG rotor was run up to speeds of 3500 rpm in all positions and the performance of the SGBs was satisfactory. This was found to be the maximum speed for the CMG with the case vented to room air environment. With cold bearings running clearances of $1.524 \times 10^{-3}$cm (0.0006 in.), the rotor was self-starting but it accelerated much faster if allowed to warm up for a few minutes before starting. The motors were used as heaters for this purpose and were turned on two-phase power for about 30 minutes before the gyro was run up to its maximum speed of 7800 rpm at 400 Hz under a vacuum hood. Because of the differences in thermal expansion of the aluminum case and endplates and the steel rotor, the warm bearing opened up to a running clearance of about $2.032 \times 10^{-3}$ (0.0008 in.). The same lubricant, a moly disulfide lithium E. P. grease which proved satisfactory for the LEV-3 gyro bearings, was used in the bearings of the CMG. The viscosity-temperature-degradation characteristics of this and a flouro-silicone grease are shown in Figure 16. These lubricant tests were run on a Haake Rotovisco Viscometer. After

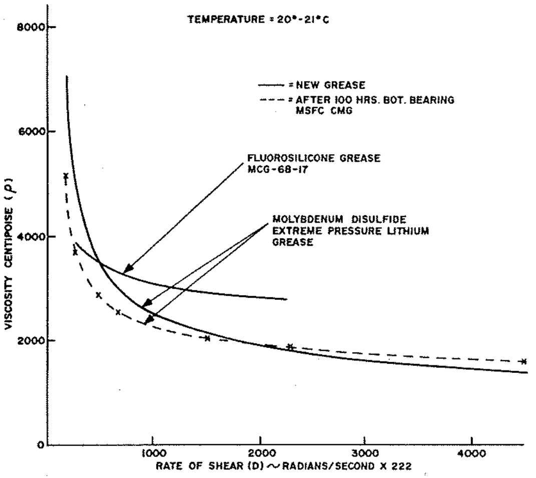


Figure 16a. Viscosities of Greases Versus Shear Rate

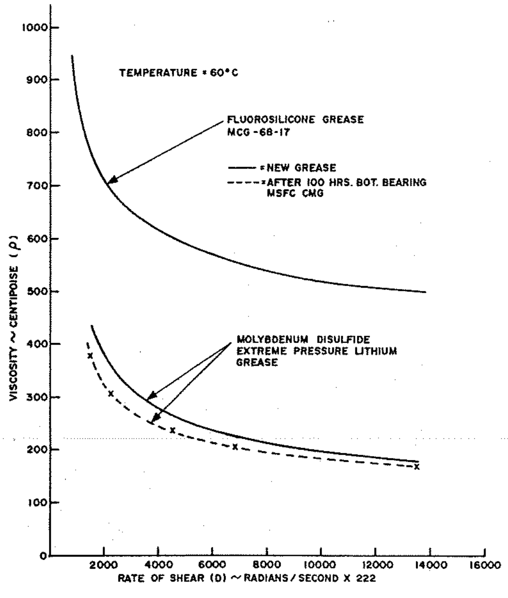


Figure 16b. Viscosities of Greases Versus Shear Rate

tests at various speeds, positions, and ambient pressures, the female bearings were removed and all bearing surfaces were cleaned and inspected. The bearings remained free of scratches, their grooves were not smeared, and they maintained a lubricant coating after running several hours at reduced ambient pressure. Two precautions were taken before running the SGBs at lowered ambient pressure. First, the lubricant was degassed under vacuum before it was applied to the bearing surfaces. Second, the wheel was brought up to about 1000 rpm before the vacuum pump was started to prevent any tendency of the vacuum to suck the lubricant out of the bearings before the pumping action began.

After initial proof tests were completed, the CMG assembly was instrumented for measurement of temperature and vibration. Temperature probes were placed in holes drilled in the ends of the female bearings. These holes extended to within 0.318 cm (0.125 in.) of the inner bearing cone apex. Accelerometers were cemented to the three axes of the CMG assembly. The wheel was brought up to a speed of 7800 rpm, under a vacuum of 50 microns mercury, and allowed to run until the bearing temperatures stabilized. Tests were run with the wheel loaded in axial and radial positions. The CMG was bolted to a test table which provided an excellent heat sink through its 2.54 cm (1.0 in.) thick aluminum top. Maximum bearing temperature in air was 59°C; in vacuum, it was about 65°C. One hour running time was necessary for the bearing temperatures to stabilize. Acceleration measurements, usually too low for concern (at values between 0.1 and 0.2), were useful mostly in detecting the resonance speed of the wheel. Resonance was found to begin at 6000 rpm and produced a mild vibration of 1.0g force within a narrow speed range between 6000 and 6100 rpm.

At this point, we had tested SGBs for about 20 hours in air and vacuum at speeds up to 7800 rpm and were confident that they could be operated continuously for long periods of time at the CMG design speed of 12 000 rpm. However, to prove that the CMG was strong enough for running at this speed, the CMG test rig was placed in a pit test area and the wheel speed was gradually increased to 14 000 rpm. This procedure was repeated several times and the bearings were again inspected and found to be operating satisfactorily.

The CMG test rig was returned to the more convenient test area and the remaining torque and endurance tests were run for over 100 hours on this 222.41 N (50 lb) wheel. Much of this running time was at the design speed of 12 000 rpm. Several torque versus wheel speed curves were run at different ambient pressures. Figure 17 is a typical torque curve at an ambient pressure of 50 microns mercury; at this pressure, windage loss was very low. Torque was calculated by measuring the rate of acceleration. The inflection in the curve between 5500 and 7500 rpm was repeatable; a condition probably caused by the combination of resonance and change in flow pattern of the bearing lubricant. Calculations, however, indicate that the flow should remain laminar at these speeds.

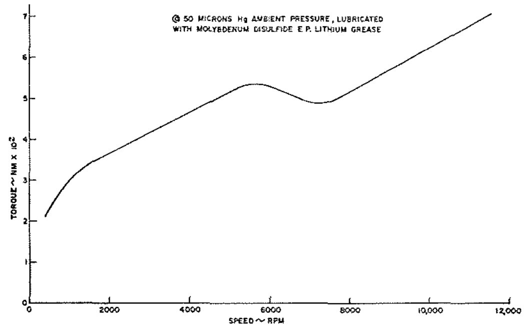


Figure 17. CMG Type, 30.48 (12 in.) Diameter, 222.4 N (50 lb) Rotor Supported by 1.27 cm (0.5 in.) Conical-Shaped Spiral-Grooved Grease Bearings

After 100 hours of successful testing using the moly disulfide lithium E. P. grease as a bearing lubricant, we decided to try other lubricants and discovered that these were not always retained or returned to the bearings after intermittent tests. A good example was the flourosilicone grease, MCG-68-17, whose characteristics are compared with those of the moly disulfide lithium E. P. grease given in Figure 16. The flourosilicone grease was much less temperature-sensitive than the moly grease but, when it was used in the bearings of the CMG, the bearings ran almost dry after 8 hours of intermittent running time. These tests proved the necessity of matching lubricant with bearing and in designing the SGB so that its lubricant is either retained, recirculated, or replenished.

## Conclusions

Gyro rotors weighing from 2.224 to 222.4 N (0.5 to 50 lb) supported by SGBs have been successfully built and tested. Their starting and running frictional (shear) torques were comparable to those of ball bearings, and they ran more quietly than ball bearings. Experience showed that a designer of SGBs should anticipate bearing temperature limits for each installation and match the bearing design with the lubricant. Since the bearing is a single body of revolution, it is dimensionally easier to obtain true hydrodynamic film lubrication with SGBs than with ball bearings.

More analytical and experimental work needs to be done on SGBs in the areas of stability, stiffness, and optimization of design for specific purposes. This work, initially done at MSFC, is being continued under contract at the Franklin Institute Research Laboratories.

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## EXPERIMENTS IN HYDRODYNAMIC GREASE BEARINGS

By John L. Burch and Peter H. Broussard, Jr.

The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

This document has also been reviewed and approved for technical accuracy.

Carl H. Mandel

CARL H. MANDEL
Chief, Guidance and Control Division

F. B. Moore

F. B. MOORE
Director, Astrionics Laboratory

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P. O. Box 33
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Dr. Coda Pan
Mechanical Technology Inc.
968 Albany-Shaker Road
Latham, New York 12110

Mr. Jack McCabe
The Franklin Institute
Friction and Lubrication Laboratory
20th and Race Streets
Philadelphia, Pa. 19103

Dr. Hirofumi Miura
Dept. of Mechanical Engineering
Faculty of Engineering
University of Tokyo
T-Chrome, Hongo, Bunkyo-Ku
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