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ATM CMG
BEARING
FAILURE ANALYSIS
FINAL REPORT
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CENTER

HUNTSVILLE,
ALABAMA
6401-75-R06

S. SEEB
DIRECTOR OF ENGINEERING

E. C. REUTTER
DIRECTOR OF MARKETING

B. J. O'CONNOR
GENERAL MANAGER
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SECTION 1.0
INTRODUCTION

The primary mechanisms used to orient and maneuver the Skylab Space Laboratory were three Control Moment Gyros (CMGs) manufactured at Bendix Guidance Systems Division.

Each CMG used two 107 H size angular contact ball bearings to support a 150 pound wheel rotating at a speed of 9100 rpm and developing an angular momentum of 2300 ft-lb-sec. Each of the gyros were tested at Bendix GSD and at NASA, Huntsville, Alabama, for approximately 1500 hours prior to their launch in the Skylab Space Laboratory on May 14, 1973.

On November 23, 1973, after 195 days of continuous operation in space, ATM CMG S/N 5 (Skylab #1) ceased to function due to a probable bearing failure. During the remaining 75 days of the mission, close monitoring of speed, current and temperature data from the remaining two functioning ATM CMGs revealed periods of slightly unstable behavior in ATM CMG S/N 6 (Skylab #2).

The unstable behavior manifested itself by slight speed, current, and bearing temperature fluctuations about the norms but did not affect the functioning of the gyro. However, the trend of the data indicated a possible deterioration in the performance of one of the two bearings paired in this gyro.

Both of these remaining ATM CMGs completed the mission of approximately 6500 hours in space and were run down
after the egress of the Skylab IV astronauts.

Task Order 25-74 of Contract No. NAS 8-20661 was issued to the Bendix Corporation Guidance Systems Division to study the ATM CMG data from Skylab, to initiate a complete design review and to assess the cause or causes for the failure of ATM CMG S/N 5 and for the anomalies experienced with ATM CMG S/N 6. The enclosed reports address (see Appendices) each of the areas investigated under the task order.
SECTION 2.0
DISCUSSION

The investigation into the cause or causes for the failure of ATM CMG S/N 5 (Skylab #1) and the anomalies associated with ATM CMG S/N 6 (Skylab #2) was divided into the following four areas:

1. Review of the Skylab data
2. Investigations into specific areas by outside consultants and a literature search
3. Bendix tests and simulations
4. Bendix design reviews

The review of the Skylab data was divided into eleven periods. These periods were selected to try to isolate and/or correlate various factors possibly related to the failure and the anomalies observed. The periods were selected to determine long term trends in wheel speed, currents and temperatures and how these parameters were related to various maneuvers, temperature cycles, gravity gradient desaturations, docking and undocking and periods of manned and unmanned operations.

The primary source for the data reviewed was the Skylab telemetry data. Other data sources were also used where necessary. A computer program was utilized to present this data in the form of parameter distributions. The report on the details of the data review is presented in Appendix A.
Because of the size of the report generated during this task, we have not been able to include it within this volume. This data review report and its appendix is on file at NASA and is available for review there.

The investigations by outside consultants were performed to develop more fully specific areas of interest, and to reinforce the initial theory that the problems associated with ATM CMGs S/N 5 and S/N 6 were caused primarily by marginal bearing lubrication.

Other areas of investigation included the possible occurrence of bearing ball speed variation (BSV), the conditions necessary for its initiation, and the effects if any. Also investigated was the effect of centrifugal forces on the lubricant present on the bearing rotating inner race and bearing balls. The effects of thermal gradients and surface conditions (specifically the presence of retainer transfer material) on the migration of oil from the bearing outer race land to the contact area was investigated. The dynamics of retainer instability, particularly under the conditions of high lubricant viscosity and marginal lubrication, were also studied. The reports submitted by the consultants and a summary of their efforts appear in Appendix B.

A literature search was conducted into phenomena associated with the affects of orbital conditions, particularly hard vacuum and zero "g", on lubricants. A total
of forty-five documents were reviewed some of which were helpful in further understanding certain areas relevant to the ATM CMG situation on Skylab. The results of this search appears in MT 15,785 in Appendix BV.

The Bendix testing was concerned primarily with various bearing tests to investigate the effect of lubricant or lack of lubricant in the ATM CMG bearings and the dispersion and migration of the lubricant. The tests were all conducted to simulate, to a certain extent, the vacuum and weightless conditions of space. Four life tests were conducted with bearings containing a redesigned bearing retainer for the purpose of proving the effectiveness of the design change. The reports on these tests may be found in Appendix C VII and C VIII. In addition, three tests were conducted in which minimally lubricated bearings were monitored and run to imminent failure. The purpose of these tests was to attempt to simulate and correlate the bearing conditions and the recorded variations in speed, current and temperature with the data telemetered from the Skylab ATM CMGs. The report on these tests is presented in Appendix CIII.

To determine the degree of dispersion and migration of oil in a bearing with standard retainers, a test was conducted utilizing a minimally lubricated bearing with its spin axis oriented such that the migration of any resupplied oil would be against 1g. The test was conducted using red dyed oil in the resupply system such that the dispersion and migration of the oil was readily visible. It is assumed that if the resupplied oil does
migrate or is pumped into the race against $lg$, then an equal or greater amount would similarly enter the race in a $0g$ space environment. The report of this test appears on Appendix CV. Appendix CIV contains a report of a test to determine the life expectancy of an ATM CMG bearing which has sufficient lubrication initially but is not resupplied with lubricant.

Other tests were conducted to determine the effect of temperature, particularly lower temperatures, on lube nut flow rate and ATM CMG bearing torque. The reports covering these tests appear in Appendices CI and CII, respectively. Appendix CVI contains a report on bearing lubricant hydrodynamic film tests.

The Bendix design reviews included investigation into areas most of which had been reviewed previously. These areas included the lube nut design and flow rates, bearing stresses and loads, wheel stresses, electric motor design, and bearing fits and the bearing retainer. The fact that the retainer feed holes were positioned to direct the supply oil to the land adjacent to the race way rather than directly to the ball-race way control angle was previously noted and was not considered to be a problem. However, upon further investigation and review, Bendix GSD now considers this to be the major failure mechanism. A modified retainer was then investigated and has corrected this condition.
SECTION 3.0
RESULTS AND CONCLUSIONS

Analyses of the results of the tests conducted in this task and the consensus of opinion of the consultants commissioned, points to inadequate lubrication as the predominant factor causing the failure of ATM CMG S/N 5 (Skylab CMG #1) and the anomalies associated with ATM CMG S/N 6 (Skylab CMG #2).

The problem of inadequate lubrication was caused by the inability of the makeup supply oil to migrate from the outer race land to the immediate bearing contact area. This was caused by the original positioning of the makeup oil on the land and the oil wetting restriction which inhibited migration. This migration restriction was due to thermal gradient effects and possibly retainer debris and oil polymerization after the failure mechanism was established and the problem was compounded.

Retainer loads resulting from Ball Speed Variation during maneuvers could have added to the retainer debris for an inadequately lubricated bearing.

This problem did not manifest itself during exhaustive testing on earth. In five years of testing prior to launch, a total of 480,000 hours were accumulated on 22 units including bearing tests in life test fixtures and wheel qualification tests. We believe the failure mechanism was masked by the pooling of oil in a 1 g field which would supply oil whenever the spin axis was horizontal. Although it now appears possible that the failure of ATM CMG S/N 11 during qual tests was precipitated by the same mechanisms, (the cause assignment was low rate of flow from the lube replacement system) it was not designated such by the investigative
team of Bendix, NASA, and various consultants and contractors at the time of its failure.

As a means of showing that the original Skylab retainer did not supply sufficient lube, a series of tests were run (MTs 15,781, 15,784 and 15,786) with marginally lubed bearings in the SA vertical position. In all cases, with the holes supplying oil to the lands (as in the Skylab CMGs), we were able to induce failure in less than 200 hours. Since the design modification was made we have been able to run marginally lubricated bearings for thousands of hours. These tests with both new and old retainers were performed several times and always yielded the same results (the old retainers failed and the new ones survived).

As a result of all of the efforts discussed, Bendix GSD believes that the failure mechanism associated with the bearing retainer holes caused the problem in the Skylab CMG and was probably slowly starving the second CMG. If allowed to run further, we also believe that the third CMG would have eventually suffered problems when it depleted the oil supply on its retainer. It is also felt that the P-11 failure in Houston with the S.A. vertical can also be attributed to this same failure mechanism.

Further testing of the new and old retainer designs have been proposed to further validate the above premise.
Other than retainer hole relocation, the design reviews and studies did not disclose any areas where further design changes are necessary.

There is nothing in the data search that implies that external events were responsible for the failure of Skylab CMG #1 and the anomalies associated with Skylab CMG #2.

From the data reviewed it was observed that normal changes in speed, current and bearing temperature did occur with coincident bearing heater cycles, periods of high solar intensity, and with changes in electronics assembly temperature. There were also changes related to docked and undocked modes. There were small changes in bearing $\Delta$ temperature related to bearing heater cycle and change in EA temperature. However, the primary sign denoting anomalies in the bearings of CMG S/N5 and CMG S/N 6 and the impending failure of CMG S/N 5 were changes in the bearing $\Delta$ temperature in excess of normal changes due to the above effects. It also appears that bearing number one (the failed bearing) in CMG S/N 5 always had slightly higher heat generation than its opposite number. This temperature generation appears to trend upward with time particularly during the weeks prior to its failure. There is no such consistent trend in $\Delta$ temperature associated with the bearings on CMG S/N 6 or CMG S/N 7.

The gravity gradient desaturations and other maneuvers do not appear to affect the CMG wheel speeds or currents
since the expected small changes are within the granularity of the telemetry equipment. However, there appears to be some induced gyro thermal variations related to some of the CMG S/N 6 anomalies. Although the effect of the gravity gradient dumps were not noted during the earlier phases of the mission, there was some correlation noted during the later phases between the anomalies and the occurrence of "g-g" desaturations.

From the results of MT 15,774 (Appendix CI) the flow of the lube nuts in the Skylab CMGs was sufficient to adequately relubricate the bearings. The results of the thermal vacuum test (MT 15,783, Appendix CII) indicated an expected change in bearing torque of approximately 25-33% in the Skylab CMG bearing temperature cycle range of 60°F to 80°F. This was due to the change in viscosity of the KG 80 oil used to lubricate the bearings and accounts for the changes in speed and current associated with the temperature cycle and periods of high solar intensity.

The results of MT 15,775 indicate that an ATM-CMG bearing would run for at least 1500 hours and probably in excess of 3000 hours without additional lubricant other than the lubricant in the ball bearing retainer and the residual oil on the bearing races and the balls.

The results of tests to determine whether the resupplied oil would migrate against lg is documented in MT 15,776 and indicates some oil does migrate or is pumped against lg into the bearing outer race. However, most of the
oil does not and the quantity and rate of migration into the race is insufficient for adequate relubrication. However, this would support the supposition that resupplied oil could reach the bearing race from the land in a "0" g space environment. However, the minimal amount moving this way against the thermal gradient flow was probably not enough to sustain bearing life.

The results of MT 15,782, an investigation to determine the adequacy of the bearing lube hydrodynamic film indicated that the speed at which the hydrodynamic film is developed in a normally lubricated bearing is 66-84 rpm. Therefore, an adequate film is available if the bearing is adequately supplied with oil.

The results of life tests utilizing bearings with redesigned retainers (MT 15,781, MT 15,784) indicate that the modified retainer redesign described in MT 15,777 does deliver all of the resupplied oil directly to the bearing outer race and at a rate that is more than adequate to properly relubricate the bearing.

The results of tests which attempted to simulate the failure of CMG S/N 1 and the anomalies of CMG S/N 2 (MT-15,786) by running marginally lubricated bearings in a life test fixture were vague. Although some similarities appear, both for the failure of CMG S/N 1 and the "glitches" of CMG S/N 2, overall correlation between minimal lubrication and the Skylab anomalies was not accomplished.
The fact that all of the CMG's lasted at least 195 days (4680 hours) agrees with preliminary earth tests that predicted at least 1500 to 3000 hours of running with no supplemental lubrication. The additional time beyond the 3000 hours can probably be attributed to the reduction in load associated with zero G operation (i.e. removal of the 150 lb. wheel weight).

Concurrent with the writing of this report GSD has been awarded a follow-on contract to further validate our failure mechanism premise. As part of this new program we will continue running our three life test fixtures and unit E-2 (all with the modified retainer design). We will also do further testing to compare the old and new retainer designs by utilizing two more life test fixtures and the IGRA of Unit S/N 4.
SKYLAB ATM/CMG DATA REVIEW

DUE TO THE SIZE OF THE ATM-CMG DATA REVIEW AND ITS APPENDIX, IT IS NOT INCLUDED IN THIS VOLUME. THIS REPORT AND ITS APPENDIX IS ON FILE AT NASA, HUNTSVILLE AND IS AVAILABLE FOR REVIEW THERE.
ATM CMG BEARING FAILURE

CONSULTANT REPORT

Prepared by: T. Wheelock

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608
1.0 DEFINITION OF CONSULTANT EFFORT

The nature of the failure of Skylab CMG #1 (P-5) and the anomalies of CMG #2 (P-6) had been discussed and analyzed by several organizations: (Draper Lab, Battelle); NASA facilities; (MSFC, Lewis Research Center, Langley Research Center) and Industrial organizations; (Bendix, Barden Corp., IBM). Some preliminary agreement as to probable failure modes had been established. Certain aspects of the problem had not been fully developed and it was agreed the other sources of bearing technology should be contacted for more complete analysis of these areas.

These consultant organizations and individuals, to be subcontracted through BGSD included Battelle (Columbus Lab.,) Tom Barish (consulting engineer) and A. B. (Bert) Jones (consulting engineer). These three consultants had been involved with an earlier CMG (P-11) failure analysis. A basic program outlining the areas of investigation for each consultant was proposed. (See Table 1).
TABLE 1

B. Consultant Support

B.1 Battelle

1.1 Lube migration phenomenon in 0 g and hard vacuum.
1.1.1 Surface tension - surface energy equilibrium at lube vapor pressure.
1.1.2 Surface adhesion.
1.1.3 Circulation and flow in 0 g's.
1.1.4 Effect of surface temperature gradients.
1.1.4.1 Lube flow.
1.1.4.2 Lube evaporation.
1.1.5 Long term effects at hard vacuum and radiation environment.

1.2 Ball speed variation.
1.2.1 Expansion of preliminary report critical range etc.
1.2.2 Transient loading. Torque pulses.

1.3 Labyrinth seal analysis.
1.3.1 Analysis of current labyrinth seal and housing orifice design using temperature and pressure data profile.
1.3.2 Theoretical analysis 0 g effects (if any).
TABLE 1 (Continued)

1.4 Retainer Instability
1.4 General testing, possibly with unstable retainers, continuous monitors, power, torque temperature vibration, etc.

1.5 Tests of outer race control bearings - (LTF) may be done at BGSD.

B 2. A. B. Jones

2.1 Ball speed variation and computer analysis for 0 g conditions.
2.2 Computer run - orbit.
2.2.1 Orbit steady state.
2.2.2 Orbit Torqueing.
2.2.3 Orbit transient torqueing.

B 3. Tom Barish

2.1 BSV analysis, orbit conditions.
2.2 Transient torque.
2.3 Normal torqueing in orbit.
2.4 Possible experience with lube in space environment.
Although several phone conversations with Bert Jones were made it was determined that due to his current work load he would not be able to contribute meaningfully to the analysis, and suggested that Battelle's computer programs could produce results comparable to those from his.

2.0

1ST REPORT FROM T. BARISH - APRIL 23, 1974

The initial report from Tom Barish was received at the end of April 1974. He had been furnished with all available reports from the meetings held at MSFC. He was also furnished a complete copy of NASA Report 50M23157 which included the analysis and report of the failure of CMG P-11 as well. The first report contained the following analysis and conditions:

1. The #1 CMG failure was a result of overheating and retainer breakup in bearing #1.

2. The cause of this condition was a combination of marginal lubrication, Ball Speed variation, and a possible hangup of the outer race in the slider producing high preload.

An interesting aspect of the analysis (section 10 of report) is the effect on oil migration due to centrifugal forces within the bearing. The analysis suggests the net effect would be a depletion of available lubricant at the inner race - ball and ball-pocket contact areas.
Battelle had agreed to conduct some testing to evaluate this effect (see section 5.3). The effects of Ball Speed variation (BSV) had been studied as a possible failure mode for P-11 CMG earlier. Under a specific range of radial loads (for a given preload) a maximum in contact angle variation occurs, and in turn produces the maximum BSV and resulting forces in the retainer ball pockets. An analysis can be made to determine the excursions of the individual balls, however there is some question as to the magnitude of the forces generated on the retainer. This force is dependent on the coefficient of friction at the ball race contacts and the value can vary by at least an order of magnitude depending on the lubrication situation. BSV is also considered as a "glitch" generation mode as described in para. 19. A modified ball retainer with increased pocket clearance is recommended. This would allow more ball excursion and reduce the possibility of damage from BSV. It should be noted that this modification will necessitate reduction of the ball complement from 15 to 13. Also recommended are modifications to increase the amount of lubricant available.

3.0 MEETING BGSD, (T. Wheelock) BATTELLE, (J. Kannel) and T. Barish - MAY 7, 1974

The preliminary analysis conducted by Tom Barish and at Battelle had resulted in some areas of confusion and disagreement as to certain aspects of the problem.
Numerous phone conversations has been necessary; however, it was effectively impossible to arrange a three way discussion. As Tom is unable to travel, it was decided to have a meeting at his office in Van Nuys, Calif. Areas to be resolved were:

4.3.1 Critical radial/axial load ratios producing max. BSV.

4.3.2 Friction forces producing BSV forces on the retainer.

4.3.3 Oil loss from inner and balls due to centrifugal forces.

4.3.4 Comparison of failure scenarios for CMG-1 and CMG-2 anomalies.

There was agreement as to the critical radial/axial load ratios for maximum BSV. This range was determined to be 1.5 to 2.0 which corresponds to a slew rate of 3 to 4 degrees per second. The ball retainer forces due to BSV were discussed at length and it was generally agreed the friction forces would be greatly modified by the presence of an EHD lubricating film. BSV however was still to be considered a factor in the failure of CMG #1.

The lubrication problem was also discussed as results of preliminary tests at Battelle to determine effects of centrifugal forces did not indicate this was a
significant factor. It was agreed that testing and analysis of the phenomenon would continue however. Other aspects of the analysis such as race curvature and the relative merits of inner and outer race "control" in this application, thermal consideration such as primary heat generation being at the ball retainer contact, lubrication starvation mechanisms, and design considerations for the "lube nut" reservoir were discussed.

In connection with the redesign of the bearing in order to withstand larger ball excursions in similar application (NERVA) with a design life goal of 40,000 hrs. for a 105 size bearing was cited. The bearing life improved from approximately 500 hrs. to greater than 15,000 hrs. with a similar modification. The modification consisted of removing 1 ball from the assembly and increasing pocket and land clearances. The unit operated at 10,000 RPM.

4.0 SUPPLEMENTARY REPORTS FROM T. BARISH

Shortly after the meeting on May 7, Tom Barish issued two additional reports relating to the discussions and conclusions reached. A graph developed from Battelle's computer program and illustrating the critical radial loads, and resulting ball excursions had been included in their preliminary analysis and in NASA document 50M23157. This analysis in effect supported the BSV failure mode and Tom's report strongly suggests BSV was a prime factor in the failure of P-11,
Skylab CMG #1 and a likely responsible for glitches in CMG #2. He again recommended the modified retainer design described in his first report as a solution. The third report concerned the effects of inner and outer race ball control on the BSV situation. A modified bearing with reduced outer race curvature had been proposed (principally by Bert Jones) at the time of the P-11 failure analysis. This would in effect maintain all balls in a loaded condition during periods of high radial loads.

This would in turn reduce the tendency for abrupt changes of contact angle during each revolution and subsequent "impacts" between the balls and the retainer. The report suggested that the actual situation is one of partial inner race control and partial outer race control or an oscillation or shift between the two conditions. Included was an approach to determine the actual degree of "control" at the inner and outer races under various loading conditions.

5.0 BATTELLE REPORT - JULY 5, 1974

5.1 Lubricant Migration

This report contains detailed analyses and test results for several important areas in the failure scenarios. Of particular importance is the lubricant migration tests which evaluated the effects of temperature gradients and surface conditions. BGSD had also
conducted investigations into the effects of the orbital environment on lubricant migration (see Appendix 1). The question of what is the fate of oil deposited on the lands of the outer race from the lube nut were of prime importance as this lubricant availability could well determine the life of a marginally lubricated bearing. There is a good chance that a transfer film of phenolic material will occur at the inner race, as the retainer is inner land riding. Any material at the outer race, however, must be wear debris, probably generated in the ball pockets. The possibility of the transfer film combined with the temperature gradient effect in the absence of gravitational forces would indicate lube would not tend to migrate from the lands into the ball grooves.

5.3 Other Lubricant Loss Mechanisms

The results of lube film measurement under various centrifugal loads indicates this is not likely to be a significant factor at the levels which occur in the operation of the CMG. Lubricant loss through the labyrinth seal was found to be negligible. This had also been determined at BGSD during the literature study (ref. MT-15,785). The possibility of lubricant from the retainer being directed onto the lands of the outer race rather than in the ball groove was another important factor in determining the available lubricant quantity. The retainer design and oil distribution are discussed in MT-15,777.
5.4 BSV and Retainer Instability

Reference is made to the computer analysis for critical loading for maximum BSV. It is emphasized that for significant ball pocket forces to occur, a very marginal lubrication situation must exist. The retainer stability analyses indicates no significant instability mode.

It is interesting to note the ball impacts occur in situations of marginal lubrication, which may also produce a degree of race control and additional pocket forces. This suggests that the lubrication condition can produce instability as "starved" ball race contacts occur.

5.5 Weightlessness

The effects of lack of gravitational forces were discussed briefly. However some question remains as to the magnitude of the various factors involved. The detrimental effects of lubricant drainage during vertical operation are eliminated as are similar forces which cause flow in the bearing contact areas. Spreading of accumulated oil, dispersion of oil droplets and the behavior of oil in the grooves will also be affected. The operation of the lube nut oil reservoir systems will generally be unaffected, since the operation is dependent primarily on centrifugal forces. It is suggested that further study of the gravitational effect on bearing lubricating mechanisms be conducted.
6.0 Conclusions

1. The most likely primary cause of the failure of CMG #1 was lubricant starvation.

2. Ball speed variation was probably a factor in the failure of CMG #1 in combination with condition 1.

3. The critical loading for BSV results from an input rate of nominally $3^0/sec$.

4. Lubricant deposited on the lands of the races will tend not to migrate into the contact area.

5. Retainer instability is not a primary factor of the failure mechanisms involved.

6. The net effect of the lack of gravitational forces remains speculative and further investigation of this phenomenon is recommended.
APPENDIX I

LUBRICANT SPREADING IN A TEMPERATURE GRADIENT

1.0 FACTORS AFFECTING SPREADING

The distribution of lubricant due to migration in the bearing is affected by a number of factors. They include, among others: (Ref. 1, 2, 3, 4)

1. Temperature
2. Materials
3. Contaminants and Coatings
4. Adsorbed gasses and oxide layers
5. Chemical reactions at the solid, liquid interface
6. Surface finish
7. Porosity
8. Gravity
9. Atmosphere
10. Electrostatic phenomena

The relative effects of these factors have been studied and some of the more significant analyzed for specific cases but little quantitative data is available.

2.0 ANALYSIS OF SPREADING

The phenomenon of spreading of a lubricant away from a heated area has been observed previously (Ref. 1, P 379). However, no specific explanation was provided for this effect. The spreading process may be studied by the following considerations:
In general a liquid-solid-vapor interface in equilibrium is defined by

\[ \gamma_{SV} - \gamma_{SL} = \gamma_{LV} \cos \theta \]  \hspace{1cm} \text{(Young and Dupre' eq.)}

where

\[ \gamma_{SV} = \text{Surface energy of solid} \]
\[ \gamma_{SL} = \text{Surface energy at solid liquid interface} \]
\[ \gamma_{LV} = \text{Surface tension of liquid} \]

The spreading of this liquid on the solid is defined by a spreading coefficient

\[ S = \gamma_{SV} - (\gamma_{SL} + \gamma_{LV}) \]

If the surface is wettable \( S \) will be \( > 0 \) and the liquid will spread.

The effect of temperature on \( \gamma_{LV} \) is described by

\[ \gamma_{LV} = \gamma_{LV_0} \left(1 - \frac{T}{T_c}\right)^n \]

\( T = \text{observed Temperature} \)
\( T_c = \text{critical temperature} \)
which shows a decrease in $\gamma_{LV}$ with increasing temperature (incidentally is approximately = 1.23 for most substances (Ref. 2). This suggests from that spreading should occur more readily in the area of higher temperature unless $\gamma_{SV}$ increases, and/or $\gamma_{SL}$ increases at a higher rate than $\gamma_{LV}$ decreases.

This characteristic change in $\gamma_{SV}$ and $\gamma_{SL}$ has not been specifically observed and may be considered unlikely. In the case of a temperature gradient, the surface tension of the liquid may also have a gradient with the lower surface tension at the high temperature edge. This gradient would produce a force tending to draw material from the area of lower surface tension (higher temperature) to that of higher surface tension (lower temperature).

Other factors could also influence spreading in the area of higher temperature. The formation of a preliminary spreading monolayer (ref. 4) would be inhibited due to higher evaporation rates. Evaporation of more volatile components or additions in the area of higher temperature could also produce a lower average surface tension. Tricresyl Phosphate is known to have a high surface tension. If the temperature of the atmosphere is sufficiently below the surface temperature particularly severe gradients may exist in the edge of the lubricant drop. A retraction or negative spreading has been observed in some fluids.
tested on heated surfaces (Ref. 4).

3.0 REDUCTION OF SELF INDUCED TEMPERATURE GRADIENTS

The type of temperature gradient in a bearing is typically one which causes lubricant to flow away from the contact areas. In order to counteract this condition a source of heat remote from the constant region may be introduced. Since the ATM CMG bearing now has heaters, it may be expedient to relocate them in order to generate an opposing axial gradient which would produce flow toward the bearing races.

4.0 REFERENCES


THOMAS BARISH
CONSULTING ENGINEER
Bendix Navigation and Control Div.
Teeterboro, N. J. 07608
Copy To - Battelle Memorial Institute
505 King Ave.
Columbus, Ohio 43201
Attn: J. W. Kannel

Sky Lab - Bendix Gyros
Bearing Failures
Fourth Report - April 23, 1974

1. References: (a) NASA report number 50M23157; etc. (821 pages)
(b) Three previous reports by TB November 7, 1972, December 31, 1972, and January 22, 1973.

2. Present Problem: To determine as far as possible the specific cause of failure of CMG-no. 1: also the anomalies shown by CMG-no. 2: similar to the ones that occurred before failure of no. 1.
We have available for this analysis the data about the land failure on unit CMG-no. 11, the very extensive and thorough reviewing and testing of this failure. To this we are adding telemeter data from no. 1 and no. 2. This data shows characteristics that must be matched up with the type of failure as follows:
(a) Torque readings indicated by current which give positive measurement of the size of the additional friction.
(b) The speed with which the heat builds up. This is important as mechanical action would be rapid, and heat effects would be much slower.
(c) Periodic variations both high frequency and low frequency.

3. General Conclusions:
(a) The final catastrophic failure of the bearing including very severe overheating and cage breakup in bearing no. 11 and probably in bearing no. 1 were certainly produced by loss of internal clearance to the point where the thermo expansion of the balls and the inner race being greater than that of the
3. **General Conclusions:** (continued)

- Outer race developed very large loads. This "runaway" type of failure occurs when the rate of heat generation exceeds the rate of heat dissipation or equalization.

(b) Those bearings operate near the brink of this type of failure because of extremely low oil supply and unusual heat transfer conditions.

(c) To begin the runaway internal-tightness failure requires some triggering mechanism: candidates -

  - Reduction of already marginal lubrication,
  - Sudden large persistent increase in load,
  - Ball speed variation (BSV) resulting from the load combination which produces this, and
  - Radial bind of the outer race preventing axial motion plus over heating of the shaft.

The most likely causes indicated by the evidence are the first and third, and perhaps a combination of them. Each is discussed in turn below.

4. The **internal radial clearance** of the bearing is too small and not well controlled making the bearing especially sensitive to runaway heat failure. The specifications call for a close control of contact angle but no internal clearance control or control on the race curvatures and tolerances.

<table>
<thead>
<tr>
<th>Curvature</th>
<th>Initial Contact Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner</td>
<td>Outer</td>
</tr>
<tr>
<td>Avge.</td>
<td>51.6</td>
</tr>
<tr>
<td>Min.*</td>
<td>51.1</td>
</tr>
<tr>
<td>Max.*</td>
<td>52.5</td>
</tr>
</tbody>
</table>

*Tolerance estimated

A thorough test was made heating the inner race to determine how the drag increased when the inner race became hotter than the outer race. The test reported .0017 radial clearance, expected and measured. The test gave the critical temperature difference as over 100 degrees F. With the minimum clearance indicated in the table above, this temperature difference would be about 40 degrees.
Furthermore, in operation the balls are likely to be hotter than the inner race and hence would require even less temperature difference. This occurs because the heat transfer does not have any convection, the radiation is small, and the conduction is limited by the small contact area. Therefore, the likelihood of runaway heat failure from loss of clearance is far greater than indicated by the test. This brinkmanship could be avoided by using a minimum radial play of .0015, by controlling the radial play to avoid excessive top play, and by permitting a slightly larger tolerance on the final contact angle.

5. Clearance Measurements on Critical Bearings

(From Records, by phone from Wheelock, 4/24/74)

<table>
<thead>
<tr>
<th>End Shake</th>
<th>CMG No. 1 (Space Failure)</th>
<th>CMG No. 2 (with anomalies)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>End Shake: &quot;0140&quot;</td>
<td>End Shake: &quot;0144&quot;</td>
</tr>
<tr>
<td></td>
<td>Make &amp; S/N: Brg 1 Brg 2</td>
<td>Make &amp; S/N: Brg 1 Brg 2</td>
</tr>
<tr>
<td></td>
<td>Barden 17 Barden 16</td>
<td>SBB 009 SBB 002</td>
</tr>
<tr>
<td>Radial Play*</td>
<td>.00091 - .00129</td>
<td>.000873 - .001165</td>
</tr>
<tr>
<td></td>
<td>.000969 - .001299</td>
<td>.000719 - .000950</td>
</tr>
<tr>
<td>Retainer Loose-</td>
<td>.014</td>
<td>.013</td>
</tr>
<tr>
<td>ness on race in ball</td>
<td>.0055-.0060</td>
<td>.0060-.0065</td>
</tr>
<tr>
<td>pocket</td>
<td>.0060-.0065</td>
<td>.0060-.0060</td>
</tr>
<tr>
<td>Total</td>
<td>.0195-.020</td>
<td>.0190-.0195</td>
</tr>
<tr>
<td></td>
<td>.0200-.0205</td>
<td>.0190</td>
</tr>
<tr>
<td>Clearance on Brg. O. D.</td>
<td>.000602</td>
<td>.000425</td>
</tr>
<tr>
<td></td>
<td>.000646</td>
<td>.000565</td>
</tr>
</tbody>
</table>

* Calculated from measured contact angles and race curvature specifications.

From the above radial play readings, it is quite evident that the two critical units could have had much less radial play than that used in the check test, .0017. The temperature differential necessary to produce a bind would be 40 - 50 degrees instead of over 100 degrees estimated by test.

ORIGINAL PAGE IS OF POOR QUALITY.
This would be further reduced by the fact the test produced heat inside of the inner race but not at the ball. Note that "delta T" in this case means difference between inner race temperature and outer race temperature and will be used as such in the following. The main report uses this delta T for difference between bearing no. 1 and bearing no. 2 which has been invariably small and not very significant.

When delta T begins (temperature differential, inner to outer ring), the first thing that takes place is a reduction in internal clearance, a reduction in contact angle without actual binding. This is equivalent to having a bearing initially made with less contact angle. The table below indicates that the load per ball roughly doubles before binding begins. Hence there should be a gradual increase in friction to about double; an increase in wattage of about 30 to 40 per cent since the bearing friction is not all of it and since you would ordinarily involve one bearing first. The Table below shows how a bearing with average tolerances would change. It becomes tight with 78.5° delta T., inner hotter than outer. With the balls tending to hold the heat, this estimate should drop to about 50°.

<table>
<thead>
<tr>
<th>Change in Loading with Reduced Clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Angle</td>
</tr>
<tr>
<td>Angle with 40 lb. Thrust</td>
</tr>
<tr>
<td>Normal Load/ball</td>
</tr>
<tr>
<td>Initial Looseness</td>
</tr>
<tr>
<td>Δ T (est.)</td>
</tr>
</tbody>
</table>

Further delta T will increase the radial load very rapidly.

<table>
<thead>
<tr>
<th>Further Delta T Increase</th>
<th>10°</th>
<th>20°</th>
<th>40°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Added Squeeze (on diam.)</td>
<td>.000125</td>
<td>.00025</td>
<td>.00050</td>
</tr>
<tr>
<td>Total added load (normal)</td>
<td>28 lb.</td>
<td>60</td>
<td>138</td>
</tr>
</tbody>
</table>
Telemetry

The telemetry from CMG-1 during failure (next page) showed first the difficulty started about one minute after a vigorous manœuvre. Then the following actions are noted:

(a) The amperage rose from 1.0 to 1.8 in twelve minutes,
(b) The amperage leveled off for twelve minutes,
(c) A short peak, two minutes to 2.4 amps and immediate settling back,
(d) Leveling off at 2.0 for about one hour

The above indicates that the jump in power is much greater than would obtain with a normal reduction in clearance.

The sudden peak might have occurred from the beginning of runaway failure but dropped off too fast for that and is more likely an actual cage breakage. The following high friction would be continued ball-cage or ball-ball friction.

Page 7 shows the telemetry for CMG-1 in one of the prefailure anomalies. One very significant condition in this picture and which occurred in the similar anomalies on CMG-2 is the pulsing of the current: maximum + or - .0045 amps. The size and frequency of the change makes it possible that this was a reduction in internal clearance with the contact angle changed as per above. Each peak and drop-off takes two-four minutes and this would be the type of thermal delay that would be evidenced. However, the drop-offs are too sudden and difficult to explain.

These pulses could be very small drops of oil entering the ball path. Then the drop-off could be sudden as indicated; however, it is very unlikely to be regular and persistent.

The third possibility is cage bind from BSV which is relieved as soon as heating occurs, changes the contact angle and reduces the increase speed of the unloaded balls. (See discussion later on BSV)

The burst (page 6) to 1.066 amps and the fall off, totaled three hours, is most likely a straight lubrication effect. If there were internal clearance buildup it would show difficulty much more rapidly. This implies a sparse lubrication effect. The bearing temperatures confirm this in that the amperage drops off when the bearing temperature increases which as proved by test increases the oil flow.

The variation in bearing temperature does not help much in determining internal radial bind. In the latter case, it is the delta T (inner to outer ring) which counts. We can only say that an increase in outer race temperature would require a preceding considerable increase of inner race temperature. After this delta T rise, the temperature would equalize in minutes.
CMG - 1 PREFailure Anomaly

REP-1200
NUM-0071 VIEWIN-MSF2 SEIS-0002 TO 0002 OF 0003
CMG 1 4A CURRENT VS CIRCUIT 1 & 2 TEMPERATURES

GMT

CH51 4A CUR
M04-762 41CP C0000-102 =2(L) C0007-702 =3(L)

00:00.000 06:00.000 06:00.000 00:00.000 06:00.000 06:00.000

0.1035 0.1055 0.1035 0.1035

0.1055 0.1035 0.1035 0.1035

50 60 70 80

Brq 1 Temp
Brq 2 Temp
10. **Lubrication**

The centrifugal force effects need more attention. The inner race is subjected to 2200 "G"s. Hence the oil creep will be far more rapid than under one "G" and directed definitely towards the inner race shoulder. The race can be expected to be bone dry in less than a minute after start of operation. A more exact analysis: the surface tension does not vary with the film thickness. Hence the film thickness can be determined as that which would just equal the surface tension at 2200 "G"s. However, there are too many questions involved. It would be better to make a simple test and find out how much film remains and how long it remains under the high CF.

Note that this oil would be thrown off the edge of the inner race and would tend to provide a certain minimal lubrication for the bore of the cage.

11. The oil delivered by the nut should definitely be much greater at the first of the run because the pressure developed by CF on the metering device varies with the depth of the oil in the nut. However, this need not modify previous conclusions since we actually tested the oil flow at different depths.

12. The oil leaving the nut and entering the cage lip has a CF of 390 "G"s hence the cage is practically bone dry in the pockets except for whatever the ball transfers from the outer race to the cage and the inner race. There is also possibly slight lubrication from the inner race throwing into the cage but even this will mostly go through the cage pockets.

Hence the unusual squealing under severe loading because of the dry contact. However, producing a squeal requires positive forces on the contact and these are most likely from ESV.

13. The high CF on both the nut and the cage means that oil flow from them will take place strictly in a tangential direction at high speed and at the very first available opening. The location of the nut lip with respect to the cage was determined as .006 minimum overlap. However, this figure did not include an allowance for the axial motion of the cage (.004 off-center possible) and the fact that the ball center is not in the same position axially as the groove of the outer race (.002). This indicates that with some tolerance variation and wear the oil from the nut could skip the cage entirely.

The oil leaving the cage oil holes would move outward radially at the first small edge and this would clearly hit the outer race bore outside of the ball groove. It would take an appreciable creep in the outer race under zero "G" to reach the ball path and some of the oil would definitely creep in the opposite direction.
14. The sparsity of oil on the inner race (practically none) will make the friction at that contact considerably higher than normal and definitely higher than the outer race. The A.B. Jones computer setup uses .07 friction as carefully determined by tested calculations for normal oil and normal bearings. This inner race will develop more friction and more heat. The outer race can have an elastohydrodynamic (EHD) film. There will be two problems: 1. the removal of oil by the balls and feed to the inner race and the cage. Also creep of oil away from the contact will starve it unless the creep brings the oil back relatively rapidly. Both of these effects will leave so little oil in the race that its film thickness may be less than required for an EHD film. A rough calculation assuming the average oil flow and one minute for removing the oil by CF and creep leaves a film thickness of .0004. This should be ample for an EHD film but the calculations are very rough. Furthermore, the radial load and the load per ball becomes 5-10 times as large under violent manoeuvres. We should check the EHD film capabilities at these conditions. This is one likely candidate for the pulsing of the friction.

15. The enormous amount of thorough testing of the metering system, the amount of flow and what it does to the bearing leaves little to add.

We add one note: the difference in oil flow that exists from the maximum to the minimum temperature allowing at the bearing outer race, can be quite critical in view of the fact that the remaining oil film is extremely small. It is quite possible that at the low rates, the inner race very soon becomes bone dry with much higher friction and much heat added to it.
Ball Speed Variation (BSV)

(Ref.: "BSV In Ball Bearings and Its Effect on Cage Design" ASLE Paper 69 AM6C-2)

Thrust load only
No appreciable BSV

Thrust and radial
Approximately equal
Major BSV occurs

Radial load mainly
No BSV binding
Upper balls loose (except for CF)

A.B. Jones Computer setup for earth conditions gave a contact angle variation (center figure above) of 27 degrees.
(T.E. 3rd report, Jan. 22, '73)

It is urged that computer runs be made for space conditions. This should be 40 lb. thrust and radial varying from 20-60 lb. in small steps. We need to define the BSV which exists only for a narrow band and also how large it will be. Indications are that it will be maximum at about 40 lb. radial load; 1/20 per second gimble motion.

BSV Forces:

The BSV makes the cage operate eccentrically (see figure). For small BSV the cage floats in its most comfortable position and the increase in friction is practically nothing. However, when the ball excursion exceeds the total looseness of the cage (cage-race and cage pockets), the forces between the cage and balls and races rise very sharply. With our flexible cage there is a small band when the cage yields and this determines the size of the forces but they soon reach a magnitude of 5-20 times the normal bearing friction. With metal cages, this is sudden.
In these gyro bearings, the total available looseness is smaller than usual:

\[ 0.0190 - 0.0205 \]

for the critical space bearings. (see page 3)

The computations for earth conditions showed a maximum ball excursion \(0.032"\) for outer race control. This would give positive binding. We do not yet know what the computation will give for the space conditions.

The BSV is sharply different in the critical areas when the outer race controls the ball spin than when the inner race controls the ball spin. The computer setup gives the ball excursion for the early conditions, but in this critical area where it is not clear which race controls, the computer does not give the ball excursion but it does give the contact angle. T.B. report no. 2 and 3 computed the ball excursion for both conditions.

Furthermore, the computer assumes \(0.07\) constant coefficient of friction for both conditions. The inner race, when it becomes bone dry due to CF will surely have a higher coefficient of friction.

Cage Instability would not normally be noticed with the very light and flexible bakelite compound cages unless there was a very definite bouncing with considerable force behind it. It is the author's belief (not confirmed by tests) that this occurs during the unstable condition when the bearing may be rapidly changing from inner-race control to outer-race control. This would give a considerable forced vibration that could be perceived, and would explain this squealing.

If the BSV produced cage bind, the current pulsing of CMG-2 would have occurred as follows: the onset of rapid heating would first reduce the internal clearance. This would happen even more rapidly than from normal load because the heat is generated at cage-race contact and cage-ball contact. The balls would heat up more and more rapidly. The reduction in clearance would quickly reduce the effective contact angle (table, page 4). This would sharply cut the BSV since the high angle at the top of the bearing (middle figure, last page) would come down rapidly. The elimination of BSV bind would then permit an equalization of the temperature and a drop in friction. This would also explain the "glitches" as the speed would rapidly drop and come back more slowly.

Outer Race Bind

The housing around the outer race of the bearing has \(0.12\) thickness of beryllium bronze and \(0.93"\) of aluminum. Any changes with temperature would closely follow that of the aluminum. The clearance here is \(0.0005\) nominal. Actual clearances in the critical bearings were \(0.00043 - 0.00055\) (page 3).

If the unit were measured and assembled at \(87^\circ\), operation at \(50^\circ\) would completely remove the minimum looseness above and make the unit bind. This would happen much more quickly if the aluminum were colder than the outer race. Such binding would quickly prevent any expansion axially of the shaft.
and build up large thrust loads: 50° delta T gives .003" expansion and would increase the thrust load from 40 lb. to 700 lb.
This difficulty is unlikely: first, because both bearings would have to bind. Secondly, the aluminum housing would expand also so we would need a differential. Nevertheless, steps should be taken to prevent its possibility on future designs. Suggestions: slightly increased radial play perhaps .0003. Also the measurement of the aluminum housing should be specified at a particular temperature.

21. Temperature Differences on Outside of Gyro Housing.
If one side of the gyro housing is in sunlight (not the end-bells) and the other side in the shade, the difference in axial expansion would produce a tilt of one end-bell with respect to the other, and a misalignment on the two ball bearings. As explained above, these bearings are especially sensitive to misalignment because of the rapid increase in ESV. This problem is very remote because the gimble housing is shielded. Even a 100° difference in temperature would give only .0004"/" misalignment on each bearing.

Summary
22. This entire analysis is only of value for assuring better performance on future gyros. The following actions have been suggested:
(a) Compute the BSV for space conditions.
(b) Increase the bearing looseness a small amount but control the looseness rather than the contact angle. Also specify a control on the race curvature. A flatter race curvature on the inner race particularly would permit an increase in looseness without much change in the contact angle.
(c) The cage needs many adjustments:
The cage pocket should be elongated circumferentially about .05 This will require cutting from 15 to 13 balls. The cage-race clearance can be cut to the original .005 - .005 Increase width to better its oil catcher. The oil holes in the cage should come nearer to the center. The scallops on the bore should leave an appreciable flat between each of them since the pointed condition now gives poor bearing and easy removal of cotton slivers.
(d) The OD looseness should be slightly increased.
(e) Changing the curvatures should be checked carefully. Any attempt to flatten the race curvature on the inner race and leave the outer race controlling the ball spin, would also mean more heat generated at the inner race and greater delta T. Nevertheless,
Summary (continued)

(f) The oil flow should be checked to avoid very low figures as occurred in some of the space bearings. Also perhaps, closer temperature control.

(g) The nut OD can be increased about 1/3". This would permit about 1/16" larger oil space and would require moving the metering out also.

(h) Retest to confirm no additional trouble from all the above changes.

(i) It may be desirable to again attempt to reproduce the symptoms obtained in space by running a unit with all the unfavorable condition discussed.

Thomas Barish
Ball Speed Variation in Ball Bearings and its Effect on Cage Design

THOMAS BARISH, ASLE
Consulting Engineer
Van Nuys, California
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THOMAS BARISH, ASLE
Consulting Engineer
Van Nuys, California

A review is presented of the common causes of Ball Speed Variation (BSV), with emphasis on misalignment, the most common cause, and including typical calculations.

Three very different stress regimes result, depending upon the extent of BSV. When large, rapid cage failure is produced by the large forces at the cage-ball and cage-race contacts.

In many cases, the trouble can be cured by small changes in the cage design, particularly increases in the looseness at the ball pocket and the race. Typical field experience is quoted.

INTRODUCTION

Ball Speed Variation (BSV) refers to the speed of the ball along the pitch circle. A. B. Jones aptly names it "orbital velocity" and gives a complete set of equations (1) (2). Variation occurs in practically every operating ball bearing, but fortunately, rarely large enough to be serious. When large, it produces rapid failure at the cage-ball and cage-race contacts.

BSV has been known for a long time; i.e., Fig. 1, a production cage design used by D.W.F. (Germany) about 1920; clearly the designer expects changes in the ball-to-ball distance. The author's first contact with this problem was in 1924: a 420 size Conrad type bearing under heavy thrust and radial load at moderate speeds showed excessive temperature rise and roughness. A bearing without a cage was put in a testing machine under the same loads and turned slowly by hand. The distance between the balls was measured with an inside micrometer, and varied up to 1/16 in. Removing the cross-sectioned area A in Fig. 2, without any other change in the bearing, eliminated the roughness and reduced the temperature rise about 75F.

Cause and Types of Stress

The orbital velocity of the ball is one-half the velocity of the contact point B on the inner race, Fig. 3 (one exception at high speeds where the centrifugal force is described later). The only factor in the equation which can vary in operation is the contact angle. Hence all BSV results from contact angle changes.

The cage rpm is controlled by the balls and equals the average ball speed. The balls advance or retard from the average speed and crowd the cage pockets and the cage. This is the major source of severe loading at cage-ball and cage-race contacts; possible other source, severe unbalance.

These stresses are not controlled by the BSV itself, but by the change in the ball position, the displacement variation. Most variations in speed are cyclical: the ball will travel faster than average in one-half the bearing and slower in the other half. The advance of the ball accumulates over half a revolution. To determine the ball position and the distance it departs from the average requires an equation or curve for the ball speed variation, and an integration for the ball position variation from average.

The ball-cage forces can have three radically different

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Presented at the 23rd ASLE Annual Meeting in Cleveland, Ohio, May 6–9, 1968

Fig. 1—Spring cage—circa 1920 (D.W.F.—Germany)
regimes: (a) **When the ball displacement is less than the looseness** in the cage pocket and no cage-ball bind results. Only two small forces remain: the relatively negligible cage weight, and the unbalance force appreciable only at the highest speeds. For the 7204 bearing the cage weight is only 0.013 lb for a suitable lightened machined cage (0.009 lb for a stamped cage). The unbalance force at 75000 shaft rpm \((1.5 \times 10^6 \text{ DN})\) will be 4.7 lb for 0.015 eccentricity (0.030 loose), and drops to 0.60 lb at 27000 shaft rpm. These unbalance forces are small and occur only when the cage eccentricity rotates.

If the balls are not initially spaced in the middle of their travel, considerable forces arise to push them into median position, but these exist only for the first few revolutions. The cage should be sturdy enough to avoid failure under these momentary forces. (b) **Moderate Cage-Ball Forces** arise when the ball displacement exceeds the clearances. The cage must yield by an amount equal to the difference between displacement and clearance. The force developed varies with this difference and the spring rate of the cage circumferentially.

The cage revision of Fig. 2 sharply reduced the spring rate and the cage-ball forces. Most cages have high spring rates circumferentially and large cage-ball forces in this regime of moderate BSV. Hence rapid failures occur especially above moderate speeds. Stamped cages are better than machined cages, and there is room for much ingenuity in making them soft circumferentially but sturdy as a ring. A spring cage like Fig. 1 or those used in gyro gimbal bearings keeps these forces low, but such cages are usable only for low speeds.

The usual cage materials have low fatigue limits and fail soon under these conditions. Hence the range of BSV that can be handled by flexing is quite narrow.

Cage materials, especially the bronzes, are selected so they can spread or coin or wear at the cage-ball contact without immediate failure under average conditions and without transferring metal from the cage to the balls or races. For small BSV, the cage pocket wear and increased looseness can soon exceed the BSV and the large forces disappear. The same action takes place at the cage-race contact but to a much smaller extent, i.e., Fig. 4. (c) **When ball displacements are large and cages stiff**, the balls are forced to skid or slide on the races. Cage-ball forces are 10 to 100 times larger and failure very rapid. The force size is indicated by Palmgren data (3) of Fig. 5, where a ball was forced at right angles to the direction of rolling. With BSV, the ball is forced in the direction of rolling, either faster or slower than the rolling speed; the forces will be nearly the same. Above 0.4% sliding-to-rolling, the friction is large enough to induce gross sliding. Indicated coefficient of friction can be up to 0.25, probably sliding on one race first. With the 2 to 1 leverage, this gives ball-cage force up to 50% of the ball load.

At very small amounts of skid, actual sliding does not occur. Instead, the ball and race materials yield in

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**Fig. 2**—Removing metal A dropped temperature about 75F (1924).

**Fig. 3**—Ball speed formula:

\[ n = \frac{1}{2} N \left[ 1 - \left( \frac{d}{P} \right) \cos \alpha \right] \]

where

- \(n\) = orbital velocity, rpm
- \(N\) = shaft rpm
- \(d\) = ball diam. in.
- \(P\) = pitch diam. in.
- \(\alpha\) = contact angle

**Fig. 4**—Cage wear from BSV.

**Fig. 5**—Ball rolling between flat plates with side force.

\[ M = \tan \frac{\alpha}{1 - \frac{d}{P}} \]

\(M\) = angle ball travels from pure rolling,

\(\tan\) = ratio cross motion to rolling

\(\alpha\) = Tangential stress/parallel stress
shear parallel to the surface and below it. This absorbs much less power than sliding. For a more complete discussion of this effect, called "surface shear", see reference (4).

One Large Ball

Most BSV effects are cyclical and repeat exactly each revolution. One rare exception occurs when one or more balls are appreciably different in size from the rest of the balls in the bearing. The oversize ball will have a smaller contact angle and lower speed, and will lag continuously. In a 7204 bearing, a 0.0001 in. oversize ball reduces the contact angle from 35° to 33.5°, and causes the ball to lag 0.0022 in. per cage revolution. At 3600 rpm, this would accumulate to 3.2 in per minute. Hence the greater importance of ball size equality in angular contact type ball bearings. Failures produced by this condition have one pocket with severe ball-cage stress on one side of the pocket (Fig. 6), and all other pockets will have lesser stress on the opposite side of the pocket; also, mild rubbing on the inner race, localized on the cage, not on the race.

THRUST LOAD PLUS MISALIGNMENT

Misalignment is the most common cause of BSV and cage failures. Under pure radial load no unusual cage-ball distress occurs because the balls readjust their position in the unloaded area. BSV problems arise only with pure thrust load, or enough thrust to keep all the balls in contact (also in very high speed bearings, discussed later).

Figure 7 shows ball path pattern for thrust load plus misalignment in an angular contact bearing with inner race off-square. Fig. 8 with outer off-square. Figure 9 shows an inner race after operation. No real failure occurs in the race contact. The indicated contact angle change does not cause failures in the races or the balls, no more than they would operating in a bearing designed to such angles. The only difference at the contact is a small reduction in fatigue life, because the load distribution among the balls is modified. The failures are primarily in the cage.

The balls travel faster in the upper half of the bearing, and slower in the lower half, (Fig. 10), and the ball position displacement accumulates as shown. This forces the cage downward to an extent equal to the maximum amount of ball displacement from average. The displacement of the ball from the average position is approximately sinusoidal. If the cage is allowed to operate in an eccentric position as shown in Fig. 10, the ball pockets also provide a sinusoidal displacement from average. Since the cage is not fixed except by its clearance from the race, it will adjust itself to this position and no unusual cage-ball stresses occur, only the weight of the cage. If the eccentricity rotates with the inner race, there will also be an unbalance.
so, sinusoidal motion. Large pocket clearances would give cage instability axially. Hence most of the increased clearance should be at the cage-race contact. Also satisfactory would be cage pockets elongated circumferentially, but these are difficult to make except in a two-piece riveted cage where temporary spacers between the halves will give elongated pockets.

Stamped ribbon-type cages, riding the balls, automatically leave room for liberal clearance on both the inner and outer race. However, they are frequently made with the inner and outer clearances unequal limiting the alignment capacity to that provided by the smaller opening.

The same type of BSV but of lesser magnitude results from: (a) excessive angular deflection of shaft or housing, (b) moderate radial loads with larger thrust loads, large enough to keep all of the balls in contact, and (c) off-center loads or moment loads on a single bearing mounting. In all of the above cases, the ball displacements rapidly increase with closer curvatures and with higher contact angles.

Table 1 gives typical figures for ball position variation under misalignment.

It is evident that the misalignment capacity of such a bearing can be materially increased by increasing the looseness of the cage, both in the pockets and on the inner race. If the total looseness in both locations were increased from the normal of about 0.015 in. to 0.04 in., the aligning capacity would be increased from 0.11° to 0.29°. The cage-pocket and cage-race loads would then be in regime (a) above, just cage weight and unbalance for inner off-square: no heavy loads as in regime (b) or (c).

The total clearance can be divided between the ball pocket and the cage-race contact, except that a minimum is needed in the pocket for departure from true sinusoidal motion. Large pocket clearances would give cage instability axially. Hence most of the increased clearance should be at the cage-race contact. Also satisfactory would be cage pockets elongated circumferentially, but these are difficult to make except in a two-piece riveted cage where temporary spacers between the halves will give elongated pockets.

Stamped ribbon-type cages, riding the balls, automatically leave room for liberal clearance on both the inner and outer race. However, they are frequently made with the inner and outer clearances unequal limiting the alignment capacity to that provided by the smaller opening.

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**HIGH SPEED WITH COMBINED LOAD**

BSV becomes serious at high speeds when the centrifugal force of the balls is appreciable (usually above 500,000 DN), and when radial loads are combined with small thrust. With pure radial load, the upper balls are held against outer race (Fig. 11a) and the cage force is only that needed to drive these balls. The looseness C increases to initial looseness + deflection of bottom ball + deflection of top ball at outer contact. The first small added thrust load makes little change until the axial motion brings the top ball into contact with the inner race (Fig. 11b). Because of greater looseness this occurs at a high contact angle and produces the peculiar ball path shown. The top ball rides the outer with a low angle, 5° to 10°, and the inner angle can easily reach 30° to 40°. The higher angle increases the ball speed.
It is now more than half the speed of the driving contact (Fig. 11b). Different angles occur also with pure thrust at high speeds but they do not change: hence no BSV. Formulas and experiments are reported by Hirano (6). The BSV becomes large under combined load because the lower balls still have a small contact angle. A. B. Jones' equations (2) and computer programming give the change in angles and orbital velocity.

Details are given for an early case of trouble from this source (1954), the turbine bearings on an aircraft alternator drive. No trouble arose in extensive testing under load with 80 lbs thrust, but there were repeated failures in qualification test at no load with only 20 lbs spring thrust. The higher operating thrust moved the outer axially and reduced the BSV by reducing the inner contact angle at the top ball. The problem was solved quickly by increasing the spring load from 20 lb to 40 lb. The calculated BSV for both loads are given in Fig. 12. The total ball position variation became less than the total cage clearances (Table 2). Increasing the cage clearances would also have eliminated the trouble (as was done in later cases), but would have taken more time.

Table 3 gives data for a later similar case (1964) with angles and orbital velocities by computer, and corrected by increasing clearances.

<table>
<thead>
<tr>
<th>TABLE 3—BSV IN TURBO-PUMP</th>
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<tr>
<td>Bearing 305 Conrad—9 13/64&quot; balls, 25,000 RPM</td>
</tr>
<tr>
<td>17° Initial angle: 51.6 and 53° curvatures</td>
</tr>
<tr>
<td>Loads: 359 lb radial; 175 lb thrust; 32 lb C.F./ball</td>
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Figure 13 shows BSV and position variation from average, and calls for ±0.055 cage looseness, total for pocket and race contacts. Only part of this can be compensated by eccentric operation of the cage; that part

<table>
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<th>TABLE 2—BSV IN TURBO-ALTERNATOR DRIVE AT NO-LOAD</th>
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<tr>
<td>Bearing 204 Conrad—8 3/4&quot; Balls</td>
</tr>
<tr>
<td>51.6 and 53° Curvatures—0.0006&quot; Loose</td>
</tr>
<tr>
<td>Cage Looseness: Pockets 0.008—Outer Race 0.010</td>
</tr>
<tr>
<td>Conditions: 25,000 rpm—Radial Load 50 lb</td>
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<tr>
<td>Centrifugal Force—6.5 lb per ball</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>SPRING THRUST</th>
<th>19.2 LB</th>
<th>38.4 LB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max.</td>
<td>Min.</td>
<td>Max.</td>
</tr>
<tr>
<td>Contact Angles degrees—Inner</td>
<td>36.3</td>
<td>13.7</td>
</tr>
<tr>
<td>Outer</td>
<td>0</td>
<td>13.3</td>
</tr>
</tbody>
</table>

| Ball Position Variation, in. | ±0.019 | ±0.0068 |
| Total | 0.008 | 0.0176 |

Fig. 12—BSV for conditions of Table 2.

Fig. 13—BSV for bearing of Table 3. Speed is variation from average. Position from arbitrary starting point.

Fig. 14—Determination of cage clearances. G is minimum needed at ball pocket. H is maximum usable at race.
which gives a true sinusoidal ball position variation. The remainder requires pocket clearance. To find the nearest sine curve for Fig. 13 is time consuming and inaccurate. It is better to use polar coordinates Fig. 14, plotting the circumferential ball position variation as the radial coordinate. The nearest circle shown gives:

(a) minimum acceptable pocket clearance as variation from curve to circle, \( G = \pm 0.015 \)
(b) maximum useful cage-race clearance, as off-center of circle, \( H = \pm 0.040 \)

Since race clearance might cause unbalance troubles, the pocket clearance was set above the minimum \( \pm 0.015 \) and the cage-race clearance set for the balance of \( \pm 0.055 \) total.

Minor misalignment or angular shaft deflections will change BSV under above conditions rapidly. They can add or subtract according to angular relation between the radial load and the misalignment.

Bearings that have operated under these high BSV do not show the ball path pattern of Fig. 11b. The rotation changes this pattern to an extremely wide path with fuzzy edges. The pattern was evident in only one case where the radial load rotated with the inner race.

**CONCLUSIONS**

Appreciable BSV arise in ball bearings, notably under (a) misalignment plus pure thrust load or predominant thrust load, and (b) high speeds with radial load and small thrust. Examples show the extent of BSV and ball displacements: also methods of determining them, and corrective measures used. Increased cage clearances can permit much larger ball displacements from these sources without generating high cage forces and early failure.

**REFERENCES**

Subject: Sky Lab Gyro Bearings

1. **Reference**: The useful and satisfying conference at this office with both of the above men on May 7th.

2. The curves developed at Battelle for the ball excursion under space conditions (next page) are exactly what the author requested in T.B. 4th report, Apr. 23, 1974 (also suggested in 1st report, Nov. 7, 1972, 2nd report Dec. 31, 1972 and 3rd report, Jan. 22, 1973). I regret having overlooked the considerable importance of this set of curves in the first review of the long report. These curves give the following conclusions:
   (a) A considerable BSV (ball speed variation) occurred under space conditions:
   (b) The resulting ball excursion was approximately twice the total looseness available in the cage for this movement.
   (c) Relatively large forces were generated, enough to produce the "squeal" and the "glitsch".
   (d) The heat generated would be enough to rapidly reduce the clearance, and the high contact angle for the ball opposite the radial load (ball no. 1). The reduced angle would immediately remove the BSV bind, and the expected period for this heat action would be of the same order of magnitude observed in the "anomalies"; about 4 min. cycles:
   (e) The continued heat generation would trigger a complete thermal bind failure.

3. It is still desirable to investigate what happens to the size of the BSV with outer-race-control instead of inner-race-control. Just before the radial load becomes large enough to leave ball no. 1 completely unloaded, the upper balls will certainly be outer-race-control because the load on the inner-race will be very small whereas the outer race load is still the full centrifugal force. This means the lower part of the bearing will be inner-race-control contact and the upper part the opposite.
FIGURE 2. SUMMARY OF COMPUTER PREDICTIONS OF BALL TRAVEL IN CMG BEARING
4. We can now draw the positive conclusions that the initial cause of the failure in CMG-11 and CMG-1 and the anomalies in CMG-2 was the ball speed variation. The final complete failure was induced by the subsequent elimination of internal clearance and the considerable thermal bind: though it is still possible that the BSV actually broke the cage and this induced the final failure. It would have been helpful if we had run this additional check over a year ago.

5. The cage stability papers by Kannel and Walters are a truly admirable and a thorough treatment of the cage motions: a virtuoso computer performance. The reports do not indicate it, but I presume it includes a full calculation of BSV. For our work there are only two of these motions and forces that are appreciable, the cage-race bind and the cage-ball bind. It is enough to know that these forces are large enough to produce the "squeal", the speed-drop, the heating, and ultimately the binding failure. The BSV bind forces would be changed by variation from inner-race-control to outer-race-control and by probable variations in the coefficient of friction. However, this does not change the conclusions appreciably as long as there is a failure. The forces are largely limited by what it takes to make the ball slide on the race.

Fortunately the problem can be completely solved by the suggested redesign of the retainer: circumferentially elongated pockets with reduced clearance at the cage-race contact and axially in the pockets; so that the stability motions are sharply reduced. This will eliminate all of the BSV binds, except for the one revolution of the cage (15 milliseconds) it takes to slide the ball into the available free space.

6. A further reduction in the already sparse lubrication could have been an independent source of failure. The timing makes this very unlikely: the start in CMG-1 at violent motion and the 4 min. cycling. However, the lubrication problem may have been definitely contributory as did happen with the reduced radial clearance.

[Signature]

Thomas Baurick
SUMMARY REPORT

on

TECHNICAL SUPPORT TO NCD OF BENDIX CORPORATION ON ANALYSIS OF SKYLAB CMG ANOMALIES

to

BENDIX CORPORATION

July 5, 1974

by

J. W. Kannel, D. K. Snediker
J. W. Kissel, and S. S. Bupara

Contract No. PO 88-451350

BATTELLE
Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201
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SUMMARY AND CONCLUSIONS

The project was conducted in four tasks directed at clarifying the failure of CMG No. 1 and the abnormalities seen in CMG No. 2 of the Skylab spacecraft. Task I involved experimental evaluations of lubricant migration due to creep against a thermal gradient and in the presence of wear debris and polymeric transfer films. Task II involved evaluations of lubricant leakage through the labyrinth seal. Task III was concerned with elimination of discrepancies between ball speed variations (BSV) predicted at BCL and those predicted by a consultant, Mr. Tom Barish. Finally, Task IV involved an evaluation of the propensity of the retainer to become unstable.

In general, the research substantiated the earlier conclusion that the primary cause of failure was a result of lubricant starvation. Other effects such as BSV could well have contributed to the demise of the starved bearing. The specific conclusions from the four tasks are as follows.

Task I. Lubricant Migration Evaluations

The prime lubricant in the CMG will not creep efficiently across surfaces prerubbed with impregnated synthane. Synthane wear debris also acts as a creep barrier. This suggests that the ball track area cannot be replenished by creep.
Analyses indicate a temperature difference of 6 F exists between the ball-track area and the shoulder of the CMG bearing. Our tests show that KG-80 will not creep from cold to hot against as little as a 4 F gradient. Thus, a relatively hot ball track will cause oil to migrate away and lubricant replenishment by creep will be retarded.

Task II. Labyrinth Seal Investigation

The estimates made of lubricant flow rate through the labyrinth seals indicate that the loss by this means is negligible with respect to the lubricant supply rate to the bearing. Therefore, it is not a factor in lubricant depletion from the bearing.

Task III. Investigation of Ball Speed Variation

Under lubricated contact, BSV does not loom as a significant problem. However, when lubricant starvation occurs, BSV can be an instigator of retainer distress. The worst case for BSV is under the CMG slewing load, which produces a radial load of about 70 lb (>30/sec slew) for an assumed axial load of 40 lb. The discrepancies between Barish and BCL on this subject have been clarified and we are in essential agreement on this subject.

Task IV. Analyses of Separator Stability

Analysis of bearing retainer motions was performed by use of a bearing-dynamics computer program (BASDAP). For the cases run, the results show that the retainer operates in a stable mode. Estimates made on the basis of a retainer stability criterion indicated that unstable motions may arise if the ball-race lubricant film thicknesses are less than 6 μin. This magnitude of film thickness may result under starved lubrication conditions. The computer data also indicated that poor lubrication conditions at the ball-retainer and retainer-race contacts can cause a large change in bearing torque. Hence, such a condition may be manifested in the form of torque glitches as observed from the telemetry data.
FAILURE SCENARIO

The scenario proposed by BCL of the distress of CMG No. 2 and the failure of CMG No. 1 is as follows.

The bearing was operating essentially on the initial charge of lubricant and any additional lubricant that had migrated into the contact region from the lube-nut during ground testing. During space operation of the bearing little, if any, lubricant was fed into the bearing as a result of the following.

1. The lube feed system directed the lubricant to the bearing shoulder rather than into the ball-race contact zone, and
2. The lubricant could not migrate into the contact region due to
   a. An unfavorable thermal gradient
   b. Synthane debris and transfer films in the vicinity of the wear track.

As a result of the combination of factors in the lube supply system, the bearing eventually starved as a result of squeeze-film leakage, oil migration, and, possibly, evaporation into the bearing cavity.

As the lubrication condition deteriorated, secondary factors such as BSV under slew load produced wear debris which

1. Modified the bearing torque and temperature.
2. Further aggravated the migration problem due both to the unfavorable thermal gradient and to inadequate lubricant wetting of the ball-track region.

The torque-temperature glitches abated under conditions of increased external temperature as a result of a shift of the hot region from the ball-race contact region to the bearing housing (i.e., temporary reduction of the axial gradient). This thermal shift produced condensation of the lubricant and favorable lubricant migration. The increased temperature also reduced the propensity of retainer instability and associated torque glitches. The glitches also abated simply as a result of disintegration and dispersion of wear debris.

The remedial action of increasing the bearing cavity temperature and controlling the CMG slew rates doubtlessly enhanced the life of the bearing of CMG No. 2.
INTRODUCTION

On November 23, 1973, Skylab CMG No. 1 incurred a terminal failure. Careful analysis of the available telemetry data indicated that the most probable cause of this failure was lubricant starvation of at least one of the support bearings. At about the same time as the CMG No. 1 failure, the precursors for that failure were also in evidence in CMG No. 2. In an effort to salvage CMG No. 2, remedial action was taken based on a lubricant starvation hypothesis. This action included the following.

1. Increasing the temperature of the bearing cavity to encourage lubricant circulation within the bearing.
2. Minimizing CMG slewing to reduce bearing damage due to a combined Ball Speed Variation (BSV) and lubricant starvation.

Fortunately, CMG No. 2 did satisfactorily survive the remaining portion of the mission and was eventually commanded dormant at the conclusion of the Skylab program. However, in order to design future spacecraft to avoid the problems incurred by CMG's Nos. 1 and 2, research to elucidate the failure hypotheses has been required.

The purpose of the research program reported herein has been to evaluate possible mechanisms and manifestations of lubricant starvation and the probability that the remedial action did in fact extend the bearing performance. Specifically, the following four tasks have been conducted.

1. Oil migration studies aimed at evaluating the circulation (creep) of the lubricant in the bearing cavity.
2. Labyrinth seal analyses aimed at determining lubricant loss through the seal.
3. Ball speed variation evaluations to clarify some minor discrepancies in earlier postulations.
4. Retainer dynamics calculations to determine the possible role of retainer instability on torque glitches.

This report summarizes the results of these tasks.
TASK I. LUBRICANT MIGRATION EVALUATIONS

Lubricant creep and wetting phenomena have long been of interest to the gyro designer. Considerable technology has been accumulated with regard to creep as a depletion and redistribution mechanism in high-speed ball bearings. Most of this work, conducted at the Naval Research (1) and C. S. Draper Laboratories (2), has been devoted to the creep of oils away from a bearing (NRL) and the wettability of new, clean bearing surfaces (C. S. Draper Lab.). The effects of debris and transfer films from non-metallic retainer has been largely neglected. Since it appeared that under certain circumstances, the Skylab CMG oiler could deposit lubricant on the bearing shoulder rather than into the ball track area, it has been vital to evaluate the transport of lubricant by creep, or surface migration, from the shoulder area into the ball track. In addition, we evaluated the creep of oil in the presence of a modest thermal gradient in order to determine if the possible gradient between ball track and shoulder would seriously affect oil replenishment by surface migration.

Estimation of Axial Temperature Gradients

The purpose of this analysis is to estimate the axial temperature gradient in the bearing. It is assumed that continuous heating occurs at the ball-race contact zone as a result of ball-spin friction and that this heat is conducted outward through a "semi infinite" steel medium. The magnitude of the heat generation is assumed to equal the mechanical energy dissipation associated with bearing torque. The coordinate system for the analysis is given in Figure 1.

The two-dimensional (conduction) heat flow equation can be written

\[ \nabla^2 T(x,y) = 0 \]  

(1)

where T is temperature. The assumed boundary conditions are
FIGURE 1. COORDINATE SYSTEM FOR AXIAL THERMAL ANALYSIS
\[ T(x, \infty) = 0, \quad T_x(0, y) = T_x(L, y) = 0 \quad (2) \]

and

\[ T_y(x, 0) = \begin{cases} \frac{Q}{k} & \text{for } |x| < 1 \\ 0 & \text{for } |x| > 1 \end{cases} \]

where \( Q \) is the heat flow per unit area into the race-contact region and \( k \) is the conductivity of the steel.

If it is assumed that \( \frac{b}{L} \ll 1 \) then an approximate solution can be written

\[ T = \frac{Q}{k} L \theta \]

where

\[ \theta = \sum_{n=1}^{\infty} \frac{2}{n^2} \sin \left( \frac{n \pi b}{L} \right) \cos \left( \frac{n \pi x}{L} \right) \exp \left( -\frac{n \pi y}{L} \right) \]

Values for \( \theta \) as a function of \( x \) are given in Figure 2 for \( y = 0 \). It should be noted that this solution is only an approximation and that the negative temperature predictions are an artifact of the analysis. The level of the approximation is illustrated by the comparison between the actual and the approximate thermal gradients.
FIGURE 2. APPROXIMATE SOLUTIONS FOR AXIAL THERMAL GRADIENT ANALYSIS
Numerical Example:

Assume:

\[ q \approx \frac{\text{Torque}}{2} \cdot \frac{\text{rev}}{\text{sec}} \cdot 2\pi \approx 0.5 \text{ (in.-lb)}(150)(\pi) = 235 \text{ in.-lb/sec} \]

\[ A \approx \text{Ball track area} \approx 10 \text{ b (in.)} \]

\[ K = 5.4 \text{ lb/sec} \]

\[ L = 10 \text{ b} \]

\[ \frac{QL}{K} = \frac{43.5}{5.4} = 43.5 \]

Then

\[ AT_{\text{max}} = \frac{\theta}{\text{max}} \cdot \frac{43.5}{5.4} = (0.14)(43.5) = 6.1 \text{ F} \]

Thus, for a torque of 0.5 in.-lb, an axial thermal gradient on the order of 6 F is possible. Realistically, this gradient is most likely on the order of say 2 to 10 F.

Experimental Method

Two general types of vacuum creep tests were carried out in the apparatus shown in Figure 3. All tests were run at a pressure of \(10^{-5}\) torr. A liquid nitrogen trap was used to prevent back-streaming of pump oil.

To determine the effects on creep of synthane transfer films and wear debris, the highly-polished circular disk shown in Figure 4 was employed. The disk was prewetted with KG-80 and the excess was wiped-off after setting over night. This rest period with a relatively thick oil film was used as a wettability test for the surface of the plate. The excess oil was then wiped-off and the disk installed on the spindle in the vacuum chamber. A
FIGURE 3. OVERALL VIEW OF CREEP EXPERIMENTAL APPARATUS
Figure 4. Detail of the circular plate specimen used in transfer film wetting experiments (center) and the plates used in the thermal gradient experiments (right).
A block of synthane vacuum impregnated with KG-80 was installed in the rider shown just behind the spindle in Figure 4. A 500-gram dead-weight load was applied and the disk was rotated at 2000 rpm for 12-16 hours in vacuum. This procedure produced a transfer film that was barely discernible with the naked eye. The system was opened and spots of KG-80, SRG-60, and SRG-160 were placed on the disk such that in creeping outward the oils would intercept the synthane transfer film and creep inward could proceed uninhibited. The system was repumped to $\sim 10^{-5}$ torr and the position of the top edge of the drops, (the edge nearest the spindle and farthest from the transfer film) and bottom edge (the edge nearest the transfer film) determined using a system of mirrors (Figure 4) and a precision cathetometer. Position readings were taken twice daily for over 400 hours.

The thermal gradient experiment was carried out using the two polished plates shown to the right of the disk assembly in Figure 4. Two identical polished plates were fastened on one end to a massive heat sink attached to the vacuum chamber base plate. A small resistance heater was installed in the other end. This arrangement made it possible to achieve and sustain a temperature gradient of 4 to 5 F between the two ends of both plates. The plates were coated with KG-80, checked for wettability and wiped in the same manner as previously described in connection with the large disk. A spot of KG-80 was placed on each plate—on the hot end of one and the cold end of the other. Baseline spot position readings were taken using the cathetometer. The system was pumped for $\sim 24$ hours before the heater was turned on to produce conditions of 87 F on the cold end and 91 F on the hot end. Migration readings in the form of drop edge (top and bottom of each drop) positions were taken twice daily.

**Results**

Figures 5 and 6 summarize the results of the experiments to determine the oil migration in the presence of a transfer film. The data from two replicate experiments with KG-80 are summarized in Figure 5 in terms of the position of the top of each spot, and the bottom of each spot relative to a reference point as a function of time in vacuum. Recall that the top edge of each drop is the edge that is away from the transfer film track and is creeping more or less unimpeded over the surface prewetted with
FIGURE 5. RESULTS OF CREEP EXPERIMENTS IN THE PRESENCE OF A SYNTHANE TRANSFER FILM - KG-80 OIL
FIGURE 6. RESULTS OF CREEP EXPERIMENTS IN THE PRESENCE OF A SYNTHANE TRANSFER FILM - SRG-60 AND SRG-160 OILS
KG-80. Note that the bottom edge of the drop intercepted the edge of the track within 20 hours and did not cross this barrier in the 340 hours of the experiment. (The rather rapid migration rate for the first 20 hours is probably due to spreading rather than creep and might not be expected to occur in zero g.) As seen in Figure 4, the sister oil, SRG-60, was likewise impeded by the transfer film. The SRG-160 did not reach the critical region during the test. After 40-50 hours it was noted that the KG-80 from the prewetting in the transfer film track had beaded up, indicating a severe nonwetting condition brought about by a very thin transfer film of synthane.

A separate, subjective experiment was carried out wherein the block of synthane, was replaced by a ball riding in a impregnated synthane pocket which was dead-weight loaded to 600 grams. Rotation of this assembly on the disk surface produced a credible appearing ball track having a narrow transfer film band flanked by some debris build up. None of the oils would cross this barrier in ~400 hours of vacuum testing. A posttest Talysurf revealed no wear of the steel plate. Hence, the barrier consisted only of synthane debris and transfer film. Other cursory tests indicate that porous polyimide likewise produces nonwettable surface films and debris barriers.

The results of the thermal gradient experiments are shown in Figure 7, 8, and 9. In Figures 7 and 9 the drop migration is expressed as the position of the top and bottom of the drops as a function of time. Figure 7 shows the motion of the drop initially placed on the hot end of one of the plates. Figure 8 shows a sketch of the drop migration in the course of the 340-hour experiment. Note that the drop migrated to the cold end of the plate, leaving only a residual film on the hot end (bottom curve in Figure 7). The drop placed on the cold end of the other plate showed only 1.6 cm of migration at 87 F (bottom curve; the top of the drop went off the plate)--a figure that compares favorably to the 1.2 cm migration observed for KG-80 at 75 F in the transfer film experiment.
Plate A - Drop on Hot End - (91 F)
Total Travel in 340 hours - 5.6 cm

FIGURE 7. RESULTS OF THE THERMAL GRADIENT CREEP EXPERIMENTS - KG-80 ON HOT END
FIGURE 8. RESULTS OF THE THERMAL GRADIENT CREEP EXPERIMENTS - KG-80 ON COLD END
FIGURE 9. SKETCH SHOWING MIGRATION PATTERNS OF KG-80 IN THE PRESENCE OF A 4 F THERMAL GRADIENT
Evaluation of Lubricant Loss Due to Centrifugal Force

One additional possibility for lubricant loss which has been postulated by Barish\(^3\) is associated with the centrifugal-slinging of the lubricant on the balls and races. Since the bearing rotates at moderately high speeds (~9,000 rpm) this possibility does loom as a mechanism for lubricant starvation. However, there are a considerable number of bearings operating at speeds significantly higher than 9,000 rpm that do not incur starvation failure. Nonetheless, the unique lubricant circulatory system used in the CMG poses new questions pertaining to role of centrifugal effects.

To evaluate lubricant loss due to centrifugal action, some cursory experiments have been conducted using the BCL rolling-disk apparatus. This apparatus consists of two counter-rotating disks which can be loaded together into lubricated contact (Figure 10). The apparatus is equipped with various types of instrumentation including a device for monitoring the extent of lubrication using an electrical conductivity technique as shown in the figure. The disks in the apparatus are 1.42-inch in diameter and can be rotated at speeds to 20,000 rpm. For the experiment reported here, the contact load was set so as to produce a maximum Hertz contact stress of 150,000 psi. The instrumentation allows for an assessment of the percentage of lubrication in this contact. (I.e., 0 percent implies a dead electrical short across the disks where 100 percent implies an open circuit.)

The sequence for the centrifugal starvation experiments was as follows.

1. The disks were solvent cleaned and were coated with a minimum of 150 mg of KG-80 lubricant.
2. The disks were rotated at the desired speed for 1 minute out of contact to sling off the lubricant.
3. The disks were brought into contact and the extent of lubricant film was measured. This was invariably (initially) 100 percent film.
FIGURE 10. BLOCK DIAGRAM FOR CONTACT CONDUCTIVITY EXPERIMENTS
(4) The disks were rotated together under load until a 70 percent film indication occurred and a record of the time for this occurrence was recorded.

(5) The sequence was repeated for other disk speeds. In all, speeds of 5,000, 7,500, and 10,000 rpm were evaluated. These speeds should be indicative of range of rotational speeds in a bearing.

A plot of speed (rpm) versus time for the film to collapse is given in Figure 11. At no time did instantaneous collapse occur. Further, the collapse time is primarily related to the number of times the lubricant passes through contact. That is, lubricant starvation is more related to the squeeze film effect due to contact pressure than due to centrifugal force effect. Any speed phenomena beyond this number of cycles was within the scatter of the data.

Our general conclusion is that centrifugal slinging of the lubricant is not a uniquely significant factor for the CMG bearing. This conclusion is heavily supported by the fact that many life-test units as well as extraterrestrial units have operated successfully under centrifugal slinging conditions that are identical with those experienced by the failed bearing.

Conclusions

On the basis of these simple experiments it is possible to draw several conclusions.

(1) Nonmetallic retainer materials give rise to oil migration barriers in the area of the ball track. These barriers are (a) transfer films that are not wetted by the lubricant, and (b) debris piles that act as physical barriers.

(2) Under conditions of operation wherein bearing contact angles are changing due to load changes, this nonwetting area can be somewhat broader than the nominal ball track and can effectively prevent replenishment of oil in the critical ball-track region by surface-migration mechanisms.
FIGURE 11. TIME TO COLLAPSE OF EHD FILM IN A CENTRIFUGALLY - DEPLETED CONTACT
(3) Oil tends to creep from hot to cold\(^{(1)}\). A gradient as low as 4 F, shown by analysis to be reasonable, can not only prevent oil from creeping into the ball track region, but can also force oil to migrate away from the ball-track region.

(4) Lubricant loss due to centrifugal force does not appear to represent a significant mechanism for the Skylab CMG.
TASK II. LABYRINTH SEAL EVALUATIONS

The purpose of this task was to analyze lubricant loss rate through the labyrinth seals in order to assess the possibility that such a mechanism could be responsible for lubricant depletion within the bearing leading to bearing failure. The approach taken was to estimate, analytically, the seal loss rate based on classical flow theory and to compare the computed values with the lubricant feed rate to the bearing.

It is assumed that molecular flow is the governing mechanism for flow of lubricant through the labyrinth seals. The supply of lubricant potentially available for loss through the seals will depend on the equilibrium condition established by the fluid within the closed chamber. Here, equilibrium refers to the balance between molecular addition and removal of lubricant to and from various surfaces of the sealed chamber. The balance is a function of the vapor pressure and temperature of the fluid which may vary throughout the chamber. The estimates of lubricant loss rate based on the foregoing assumptions are summarized below. In the determination of the estimate, the assignment of parameter values has been conservative so that lubricant-loss predictions are believed to be somewhat higher than those which would actually obtain.

Molecular Flow Analysis

Vapor Pressure of KG-80

The driving force for molecular (mass) flow through the labyrinth is the number and energy level of the molecules of lubricant on the bearing-chamber side of the annulus (labyrinth). In turn, the number and energy level are dependent on the state of the lubricant at the bearing surfaces and at any other of the inner chamber surfaces at which lubricant has collected. Initially, the potential source for loss through the seals is small; however, due to creep and to vaporization from the bearing areas and subsequent condensation on the surrounding walls, an equilibrium condition will be established wherein the pressure of the lubricant vapor in the
bearing-chamber side will be the vapor pressure of the lubricant at the temperature of the lubricant.

Thus, the knowledge of the vapor pressure of the lubricant as a function of temperature is a critical parameter for estimating lubricant loss through the seals. Unfortunately, very little data concerning KG-80 are available for the conditions of interest here. Vapor-pressure data for KG-80 have been estimated by extrapolation of data from other studies\(^{(4,5)}\). The results of this extrapolation are shown in Figure 12. For example, it is estimated that the vapor pressure of KG-80 at 100°F is \(1 \times 10^{-7}\) torr. This value agrees with that quoted by Singer at MIT\(^*\), and hence, is taken to be representative of KG-80. Of note from Figure 12 is the significant change in vapor pressure over a very short range of temperature. Also, concerning the question of the assumption of molecular-flow regime, Apt and Wiederhorn\(^{(6)}\) calculated a mean-free path of 300 cm (118 in.) for molecules of an oil having a vapor pressure of \(10^{-6}\) torr at a temperature of 290 K (63°F). Based on this, the mean-free path for molecules in the present situation should be somewhat greater; consequently, the criterion for the molecular-flow regime—a mean-free path that is greater than any linear dimension within the chamber—would appear to be justified.

**Estimate of Equilibrium Condition within the Sealed Chamber**

In molecular flow, the number of molecules that strike a unit area per unit time is given by

\[
\nu_n = n \frac{\bar{C}}{4} \tag{3}
\]

where \(n\) is the number of molecules of gas per \(\text{cm}^3\) and \(\bar{C}\) is the arithmetical average velocity of those molecules. \(\bar{C}\) is given by

\[
\bar{C} = \sqrt{\frac{8RT}{\pi M}} \tag{4}
\]

Here, \(M\) is the molecular weight of the gas, \(R\) is the gas constant \((8.31 \times 10^7 \text{ ergs/g-mole/K})\), and \(T\) is the absolute temperature. Substitution

\*Personal communication.
FIGURE 12. VAPOR PRESSURE OF KG-80
of Equation 4 into Equation 3 and use of the gas law yields

\[ v_n = \frac{Np}{(2\pi RT)^{\frac{1}{2}}} \]  \hspace{1cm} (5)

where \( p \) is the pressure and \( N \) is the total number of molecules. Equation 5 is multiplied by \( M/N \), the mass per molecule, to obtain the total mass of material (in grams) per second striking a unit area of \( 1 \text{ cm}^2 \), i.e.,

\[ v_g = p \left( \frac{M}{2\pi RT} \right)^{\frac{1}{2}} = 1.60 \times 10^{-3} \left( \frac{M}{p} \right)^{\frac{1}{2}} \]  \hspace{1cm} (6)

Here it is assumed that KG-80 has a molecular weight of 598\(^*\), and the operating temperature is 100 F or \( \sim 310 \text{ K} \). Applying these values to Equation 6 there results

\[ v_g = 2.22 \times 10^{-10} \text{ g/cm}\text{/sec} \]  \hspace{1cm} (7)

The result shown in Equation 7 indicates the evaporation rate of the lubricant at 100 F. If the vapor contained in the sealed region of the CMG is in equilibrium and if all of the surfaces within the sealed region are at uniform temperature, then Equation 7 also represents the rate of condensation. Hence, the net transfer from any surface is zero. However, a temperature gradient exists between the bearing surfaces and the walls of the enclosed chamber. The critical areas for lubricant loss within the bearings are those of the races and balls. It is estimated that this area amounts to \( 50 \text{ cm}^2 \). If the surrounding areas of the enclosure are at 60 F the net loss rate from the bearing based on an exposed area of \( 50 \text{ cm}^2 \) could be rather significant (of the order of 40 \( \mu \text{g/hr} \)) compared to the supply

\*H. Singer, MIT.
rate to the bearing. However, lubricant depletion from the bearing-race region by molecular flow is impeded by the annuli formed by the bearing races and the cage. The total area of the annuli on both sides of the bearing is estimated to be 2 cm$^2$. On this basis, the loss rate to the surrounding enclosure at 60 F from the bearing-race region at 100 F is estimated to be approximately 1.6 μg/hr. Estimates of the net loss from the bearing-race region to the surrounding walls of the enclosure at 60 F have been made for a range of bearing surface temperatures. The results are shown in Figure 13. It is noted that the net loss rates by molecular flow are small compared to the supply rate to the bearing.

Estimate of Lubricant Loss Through Labyrinth Seals

Equation 7 is taken as the basis for calculating the amount of lubricant per unit area striking the inner wall in the vicinity of the labyrinth seals. This is an overestimate inasmuch as it is expected that the temperature of the lubricant in the vicinity of the seal annulus will be at a lower temperature than the 100 F level on which Equation 7 is based. The annular shaft-clearance gap, an integral part of the labyrinth seal configuration, may be regarded as removal of an area of the wall. To determine the flow of lubricant vapor through the gap, account is also taken of the length of the clearance. This produces an additional reduction to molecular gas flow.

This area of the annular gap clearance may be estimated by application of the formula

$$A = 2.03 \times 10^{-2} \text{DC}$$

where $D$ is the shaft OD (inches), $C$ is the seal clearance (mils), and $A$ is the area (cm$^2$). From drawings of the CMG unit the dimensions involved are

$$D = 1.812 - 0.001 \text{ in.}$$

ID of seal = 1.843 + 0.003 in.

Clearance of seal gap is therefore

$$C = \sim 18 \text{ mils (max)}$$
Bearing Cavity Wall Temperature = Assumed = 60 F
Estimated Annular Area Between Cage and Rings = 2 cm²

FIGURE 13 KG-80 LUBRICANT LOSS FROM BALL-RACE AREA OF CMG BEARING (SIZE 107)
and

\[ A = 0.662 \text{ cm}^2 \]

The correction to be made to molecular flow rate through the annulus is defined by the following:

\[ K = (L + \frac{3}{8} \frac{t}{d})^{-1} \]  

(9)

where \( t/d \) is the ratio of the thickness of the aperture to its diameter.

The factor is applied as a modification of Equation 6 as follows:

\[ \nu_g = 1.60 \times 10^{-3} pK \left( \frac{M}{T} \right)^{1/2} \]  

(10)

The labyrinth seal is composed of seven straight segments, actually annular lands, and five annular grooved segments—each having a semicircular cross section of 0.031-inch radius. These annular grooves provide expanded "pocket" spaces which effectively "turn back" a sizeable fraction of the escaping lubricant molecules. For simplicity, this special geometry will be disregarded in the first approach to estimating seal leakage. For this consideration the clearance gap will be treated as a straight-walled tube of 0.765-inch length. Also, the \( t/d \) ratio for this simplified geometry is

\[ \frac{t}{d} = 0.415 \]

By applying the annular clearance area and the value for the annulus length correction, Equation 9, to the lubricant loss estimate, Equation 7, we obtain

\[ \nu_g = 0.46 \mu g/\text{hr} \]
The effect of seal clearance on loss through the labyrinth seal based on the above considerations is shown in Figure 14. The loss rate is estimated to be negligible with respect to the rate supplied to the bearing.

Another method of estimating loss rate through labyrinth seals is based on gas volume considerations. General conductance is defined as

\[ F = K \sqrt{\frac{T}{M}} \]  \hspace{1cm} (11)

where

- \( F \) = Conductance, liters/sec
- \( M \) = Mass of gas molecules, amu
- \( K \) = Area factor depending on geometry.

For several conductances in series, the overall conductance is

\[ F_T = \frac{1}{\frac{1}{F_1} + \frac{1}{F_2} + \frac{1}{F_3} + \ldots} \]  \hspace{1cm} (12)

This treatment further separates the entrance conductance, \( F_0 \), from the length consideration, \( F_a \). The entrance formula is

\[ F_0 = 3.63 A \sqrt{\frac{T}{M}} \]  \hspace{1cm} (13)

where \( A \) = Area of entrance in cm\(^2\). The formula given for the annulus length is

\[ F_a = 9.7\pi (a_1 - a_2)^2 (a_1 + a_2) \frac{1}{l} \sqrt{\frac{T}{M}} \]  \hspace{1cm} (14)

where

- \( a_1 \) = Radius of outer cylinder, cm
- \( a_2 \) = Radius of inner cylinder, cm

For a shaft of 1.812 in. - 0.001 diameter and a housing of 1.843 in. + 0.003 diameter.
Assumptions:
Smooth Annulus, 0.765-in. (1.945 cm) long
External Pressure = 0
Internal Pressure = $1 \times 10^{-7}$ torr

FIGURE 14. KG-80 LUBRICANT LOSS BY DIFFUSION THROUGH SHAFT LABYRINTH SEAL FOR TEMPERATURE OF 100 F (310 K)
\[ a_1 + a_2 = 4.64 \text{ cm} \]

Also, 
\[ a = 0.765 \text{ inches} \ (1.945 \text{ cm}) \]
\[ T = \sim 310 \text{ K} \]
\[ M = 598 \text{ amu.} \]

Substitution of these values in Equation 14 yields:
\[ F_a = 8.38 \times 10^{-2} \text{ liters/sec} \]

Also, from Equation 13
\[ F F = 3.63 A (0.518)^{\frac{1}{2}} \]

With 0.662 cm²
\[ F_o = 1.73 \text{ liters/sec} \]

Finally \( F_T \) is estimated by applying Equation 12
\[ F_T = \frac{1}{12.48} = \sim 0.08 \text{ liters/sec} \quad (15) \]

To convert to mass flow rate,
\[ Q = pF \quad (16) \]

where
\[ Q = \text{Quantity, torr liters/sec} \]
\[ p = \text{Differential pressure, torr} \]
\[ F = \text{Conductance, liters/sec}. \]

In terms of mass,
\[ G = \frac{pF M}{1.7 \times 10^4} \quad (17) \]
By substituting the values found above it is calculated that

\[ G = \sim 1 \, \mu \text{g/hr} \quad (18) \]

This value for mass-flow conductance is about two times as large as the rate calculated by the previous method. In any case the loss rate estimated above is negligible when compared with the supply rate to the bearing.

**Conclusion**

Lubricant loss rate through the labyrinth seal has been estimated analytically on the basis of molecular flow. First, vapor pressure data for KG-80 were determined as a function of temperature by extrapolation from existing data on lubricants in the range of interest. Then, the equilibrium condition of the lubricant within the sealed chamber was determined in order to obtain an estimate of the rate of lubricant molecules striking the labyrinth seal annular area on the inlet side. Estimates indicated that at 100 F and 10^{-7} torr vapor pressure, the loss rate through the seal by molecular flow is 0.5 to 1 \, \mu \text{g/hr}. This is an overestimate based on a conservative choice of parameters. Since the predicted loss rate through the seal is small with respect to the supply rate to the bearing (of the order of 40 \, \mu \text{g/hr})\text{(7)} it is concluded that lubricant depletion by molecular flow is not likely to occur.
The spin-axis bearings for the Skylab CMG's represent a reasonable configuration for the application. Under well-lubricated conditions, the bearings should operate indefinitely without failure. However, the bearings do experience rather severe radial/axial load situations which, under poor lubrication conditions, can trigger bearing distress. Specifically, under certain combinations of radial and axial loads considerable variations in the ball speeds can occur; when this speed variation is large, the variation in the gap between the balls can be sufficient to "take up" the pocket clearances. Under extreme BSV, significant ball-pocket loading will occur which (under starved lubrication conditions) can result (hypothetically) in retainer tear or fracture.

Ball-Pocket Load

During the course of the evaluation of the P-11 test failure, as well as the Skylab CMG failure, considerable attention has been given to evaluation of (1) the conditions which promote excessive BSV, and (2) the effect of BSV on retainer distress. Such BSV studies have been conducted both at BCL(8,9) and by Barish(3,10,11).

The early conclusion reached in the BCL studies(8) was that under adequate lubrication conditions BSV could not cause retainer failure. That is, if the ball-race interface contains a complete lubricant (EHD) film, then the ball can slide as well as roll on the lubricant without generating severe forces that would be transmitted to the retainer. Thus, even if severe BSV occurs, the lead and lagging balls would simply slip and accommodate the excessive excursions. Forces of only about 1 lb would be developed even under the very severe BSV condition experienced in the P-11 fixture. Further, these forces would be considerably less in space due to the low level of axial loading. Nevertheless, lubricant starvation occurs when a dry contact situation at the ball-race interfaces could produce severe retainer loading.
Some consensus seemed to evolve in the P-11 failure reports pertaining to the fact that a combination of poor lubrication coupled with BSV would be required to promote bearing distress. However at the demise of Skylab CMG No. 1 some discrepancies appeared to exist between Barish and BCL on the conditions promoting excessive BSV. Before discussing the nature of the discrepancy it is helpful to discuss first the effect of loading on BSV.

Under pure axial load, all the balls in a perfect bearing will have the same contact angle and hence will rotate at the same velocity. Under a combined radial and axial load, however, some balls tend towards unloading whereas some tend toward an overload condition. Under this load combination, all balls experience different contact angles; therefore, they rotate at different velocities. That is, the velocity of the ball is directly related to the distance $R_i$ (Figure 15) which is the velocity of the inner race at the point of contact. $R_i$ is, in turn, related to the inner contact angle $\beta_i$.

In the BCL report$^9$ on the Skylab CMG's, a graph of BSV as a function of radial load was presented. This graph Figure 16 was based on bearing dynamics calculations at BCL of the type discussed in Reference 8. The analyses indicated that radial loads in excess of 70 lb (assuming an axial preload of 40 lb) could produce excessive BSV which could lead to bearing distress under poor lubrication conditions. The 70-lb load represents a slew rate of nominally 30/sec. The graph was intended to serve as a guideline to develop policy to limit CMG slew to the 30/sec level. A policy to limit slew rate was adopted and this policy along with the temperature control policy did enhance the performance of the CMG at least for the first few weeks following the observance of initial indications of failure.

Some discrepancies between Barish and BCL existed concerning the level of the critical radial load. To reconcile these discrepancies a meeting was held on May 7, 1974. As a result of the meeting, Barish issued an addendum$^{11}$ to his initial report which implied reasonable agreement with the BCL predictions.
FIGURE 15. CONTACT ANGLES IN A BALL BEARING
FIGURE 16. RELATIONSHIP BETWEEN BALL TRAVEL, CONTACT PRESSURE AND RADIAL LOAD FOR CMG BEARING
Inner-Race Versus Outer-Race Control

A belief which is common among bearing technologists is that one race controls the ball spin. This hypothesis is an outgrowth of the assumption of a constant coefficient of friction at the inner and outer races of contact. That is, the inner and outer contact ellipse (Figure 17) are somewhat different as a result of different ball-race conformities—both axially as well as circumferentially. With the coefficient of friction assumption, the spin torque generated on one race would be different than the torque on the other and would produce a nonequilibrium situation. Hence the race with the larger spin torque is classically assumed to "control" the spin.

In lubricated contact, the ball-race tractions are slip dependent by virtue of the basic laws of lubrication. As a result, if a ball spins more than the race, a torque producing counter rotation is generated to impede the spin. Thus, in lubricated contact, an equilibrium condition can be achieved which allows for spinning relative to both races and, therefore, race control is not operative. Nevertheless, under starved-lubrication conditions, race control can still occur and this will affect the operation of the bearing and the level of BSV.

The maximum BSV occurs just prior to radial unloading of the balls (which occurs at around 70 lb)—regardless of the race-control situation. Therefore, Figure 17 is, at least, qualitatively valid despite the particular prejudices one may have toward the race-control concept.
FIGURE 17. INNER AND OUTER RACE CONTACT ELLIPSES
TASK IV. RETAINER STABILITY ANALYSIS

The principal purpose of this task was to appraise, analytically, the likelihood that retainer instabilities may arise in the bearing which would contribute to the problem of torque glitches. To accomplish this task, use was made of (1) a bearing dynamics computer program to compute bearing motions, and (2) a stability criterion for retainer-ball interactions.

Description of BASDAP and Input Data

For the investigation of possible retainer instabilities, the motions of the various elements of the gyroscope bearing were analyzed using a bearing dynamics computer program named BASDAP. BASDAP couples elasticity, rigid body dynamics, and hydrodynamic calculations for the balls, races, and the retainer in the bearing.

BASDAP computes ball-, retainer pocket-, and race-contact conditions as a function of time, operating speed, temperature, load, bearing design parameters, dimensional tolerances, and lubricant properties. The retainer is considered as a 6-degree-of-freedom rigid rotator with two principal axes of inertia in a plane perpendicular to the axis of the cylinder forming the inner surface of the retainer. Computer outputs include positions of the balls and retainer, lubricant-film thickness, ball-race, ball-pocket, and retainer-race tractions, stresses, and total bearing torque.

The input data required for the computer program which describe the bearing and lubricant parameters for the Skylab CMG bearing are shown in Table 1.

In the computer program, stepwise (in time) calculations are made to determine the motions and positions of the bearing elements. The convergence or divergence of the numerical computations is subject to an appropriate choice of time step which in turn depends on the magnitude of the forces acting within the bearing. For this case, the tractive forces at the ball-race contacts can be very large because of the high viscosity of the lubricant (\(-10^6\) cp) in the contact region. It was found that very small time steps of the magnitude of \(3 \times 10^{-5}\) secs were required to produce satisfactory numerical results.
TABLE 1. INPUT DATA FOR COMPUTER EVALUATIONS OF BEARING PERFORMANCE

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Numerical Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of balls</td>
<td>15</td>
</tr>
<tr>
<td>Ball diameter, in.</td>
<td>0.3125</td>
</tr>
<tr>
<td>Ball mass, lb</td>
<td>0.00448</td>
</tr>
<tr>
<td>Pitch radius, in.</td>
<td>0.92</td>
</tr>
<tr>
<td>Race curvatures, percent</td>
<td></td>
</tr>
<tr>
<td>Outer</td>
<td>0.53</td>
</tr>
<tr>
<td>Inner</td>
<td>0.516</td>
</tr>
<tr>
<td>Speed (inner race), rpm</td>
<td>9000</td>
</tr>
<tr>
<td>Cage mass, lb</td>
<td>0.0088</td>
</tr>
<tr>
<td>Cage width, in.</td>
<td>0.423</td>
</tr>
<tr>
<td>Cage land width, in.</td>
<td>0.05</td>
</tr>
<tr>
<td>Pocket clearance (diametral), in.</td>
<td>0.0165</td>
</tr>
<tr>
<td>Cage/inner race clearance</td>
<td></td>
</tr>
<tr>
<td>(diametral), in.</td>
<td>0.006</td>
</tr>
<tr>
<td>Contact angle, degrees</td>
<td>15</td>
</tr>
<tr>
<td>Preload, lb</td>
<td>40</td>
</tr>
<tr>
<td>Lubricant viscosity (for KG-80), cp</td>
<td>250</td>
</tr>
<tr>
<td>Viscosity at ball-race contact conditions, cp</td>
<td>$10^6$</td>
</tr>
<tr>
<td>Pressure coefficient of viscosity, psi$^{-1}$</td>
<td>$0.77 \times 10^{-5}$</td>
</tr>
</tbody>
</table>
To assess the possibility of unstable cage operation, suitable plots of cage motion with time have been made. These are described as follows.

1. The axial position of the outside edge of the retainer, XSV, as viewed from a spatially fixed point.
2. The radial position of the outside edge of the retainer, ZSV, as viewed from a spatially fixed point.
3. The in-plane center-of-mass position of the retainer, i.e., ZS versus YS. The point (ZS = 0, YS = 0) corresponds to the geometric center of the bearing.

These plots are shown and discussed in the following section.

**Evaluations of Retainer Motions**

In order to perform an evaluation of the retainer motions three principal cases were run. These are characterized as follows:

**Case 1.** Establishment of a baseline condition for bearing operation using the inputs described in Table 1.

**Case 2.** Assessment of the effect of temperature on the bearing operation by reducing by an order of magnitude the ball-race contact-zone viscosity.

**Case 3.** Assessment of the effect on retainer motions of poor lubrication conditions at the ball-retainer and race-retainer contacts.

For all of the cases considered, the ball-race lubricant film thickness was nominally 45 μin. which is taken to be representative of the lubrication condition in the actual bearing.

For the conditions of Case 1, the nominal bearing torque was computed to be 1.5 in.-oz. Graphs of the resulting retainer motions for Case 1 are shown in Figures 18 through 20. The quantities XSV, ZSV, and ZS and YS which describe the retainer motions have been defined above. The total time spanned is 1.86 msec which is equivalent to 43 degrees rotation of the ball group. It is noted that, with respect to the fixed spatial position, there is a low-amplitude wobble in the retainer and its frequency is approximately 410 Hz. The computed data did not indicate any abnormally large changes in torque level as a result of the retainer wobble.
Figure 18. Predicted axial position of retainer edge versus time.
FIGURE 19. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME
In Case 2, the effect of temperature on bearing operation was assessed by assuming that the contact-zone viscosity of the lubricant in the ball/race contacts was lower by at least an order of magnitude than that in the previous case (contact zone viscosity \( \approx 6 \times 10^4 \) cp). As a result of the increase in temperature (lowering of the contact-zone viscosity), the bearing torque is lower than that for Case 1 and is nominally 0.5 in.-oz. The retainer position plots comparable to those shown for Case 1 are given in Figures 21 through 23. The plots for Cases 1 and 2 are quite similar, the main difference being that in Case 2 the amplitude of the wobble is less than that in Case 1. The wobble frequency remains essentially the same for both cases. The computer data showed also that the torque tended to be more uniform over time for the higher temperature condition than for the previous case.

For Case 3, it was assumed that conditions of very poor lubrication existed at the ball-retainer and retainer-race contacts. For this condition, the computer output data indicated that the balls were intermittently impacting the retainer. The retainer position curves for this case are shown in Figures 24 through 26. Of interest is the ZSV-versus-time plot (in-plane radial motion). The overall frequency of the wobble is approximately 1200 Hz. The small-scale step changes in the retainer position are attributed to the ball-retainer impacts. The significant factor determined in this case is that large variations in bearing torque are noted in the computer data (ranging in the order of 9 in.-oz). From this it is concluded that poor lubrication at the various retainer contact locations can result in an indication of a large change in torque time in the bearing over a relatively long period of time.

In general, the results obtained from the three principal cases run do not show any propensity for the retainer to operate in an unstable node—even when an initial condition was assumed in which the rotational speed of the retainer was greater (by 10 percent) than that of the ball group velocity. The retainer quickly reached a velocity close to that of the ball group and wobble amplitudes were generally quite low. In a separate study being performed for the U.S. Government, a stability criterion which establishes the conditions under which the retainer may become unstable is being investigated. The criterion is based on the notion of the
Case 2
(Effect of Increased Temperature)

FIGURE 21. PREDICTED AXIAL POSITION OF RETAINER EDGE VERSUS TIME
Case 2
(Effect of Increased Temperature)

FIGURE 22. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME
Case 2
(Effect of Increased Temperature)

FIGURE 23. IN-PLANE ORBIT OF RETAINER CENTER OF MASS
Case 3
(Poor Lubrication at Retainer Contacts)

FIGURE 24. PREDICTED AXIAL POSITION OF RETAINER EDGE VERSUS TIME
Case 3
(Poor Lubrication at Retainer Contacts)

FIGURE 25. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME
FIGURE 26. IN-PLANE ORBIT OF RETAINER CENTER OF MASS

Case 3
(Poor Lubrication at Retainer Contacts)
capability of the retainer to damp ball-retainer impacts. The stability
criterion takes the form

\[
\frac{4\mu C^2}{MCs} > 1,
\]

where \( C \) is a measure of the viscous damping at the ball-race contact (dependent
on lubricant viscosity and ball-race film thickness and contact pressure),
\( C_s \) is a measure of the spring rate of the retainer and ball, and \( M_c \) is the
mass of the retainer. By using this criterion, it is computed that the
film thickness between the ball and race must be of the order of 7 \( \mu \text{in.} \) or
less. Such a condition could be obtained under very starved lubrication
conditions and may cause retainer instabilities. These, in turn, may give
rise to large torque variations as noted for the case of poor lubrication at
the ball-retainer and race-retainer contacts.
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October 7, 1974

Mr. Richard Abramowitz
Navigation and Controls Division
Bendix Corporation
Teterboro, New Jersey 07608

Dear Rick:

In connection with the Skylab CMG investigation, the question of gravitational effects on oil feed and distribution was raised. We have studied this problem from the standpoint of the performance of the oil feed/distribution process, and offer the following comments for your consideration.

The dominant forces in the oiler itself are centrifugal and are of such a magnitude that the presence of gravity is probably insignificant. That is, it is unlikely that oiler feed rates, measured at the oiler lip, will be influenced by gravity. After leaving the oiler lip, a drop of oil is slung across a short gap and is caught under a lip on the retainer. Since the gap is short compared to the kinetic energy of the droplet, this step in the feed process is probably not appreciably different in zero-g. It is conceivable, however, that an aerosol could be formed, either by slinging very small droplets off the lip or by larger drops breaking up upon impact with the retainer and not sticking. I would expect the critical drop size for aerosol formation to be somewhat larger in zero-g than in one-g. This, however, appears to be a second order effect and is likely of little consequence.

Once on the retainer surface, a moiety of oil is conducted to the critical friction interfaces by a combination of centrifugal and surface forces. None of these transport mechanisms is strongly affected by gravity, unless oil builds up in sufficient quantity (or thickness) to begin to flow under the influence of gravity. Such a critical flow point might be just adjacent to the ball track, where the ball-pass squeeze effect, acting in concert with wear debris dams and nonwetting due to transfer films from the phenolic retainer, brings about a quantity of oil that can develop a sufficient meniscus to respond to a gravitational force. Such menisci have been observed in our laboratory experiments. If this situation develops (most probably on the outer race where centrifugal forces are absent), gravity could have an effect on the quantity of oil in the ball track.

For axis-vertical operation in a one-g field, the bottom oiler is feeding upward against the gravitational field. The slump effect, adjacent to the ball track, would tend to deplete the lubricant supply to the outer race of the bottom bearing. Furthermore, the oil feed on the bottom bearing is dependent upon surface transport from the shoulder to the groove. Very thin
films will creep under the influence of surface forces and, thus, thin film transport will tend to be independent of gravity. For thick films having a sufficient meniscus, gravity will tend to retard feed to the lower bearing and enhance feed to the top, in an axis-vertical configuration. The position of the retainer (thought to be critical with regard to the alignment of the oil transport holes relative to the outer-race groove) will be influenced by gravity. In an axis-vertical, one-g test, the retainer of the lower bearing will tend to drop down into its extreme outboard position and the oil holes will probably be located over the shoulder. The primary oil feed to the outer-race ball groove will then be by means of the surface transport and flow, viz., the mechanisms just discussed.

Of the two possible areas of gravitational influences considered, aerosol formation is likely to be most detrimental in zero-g. Flow effects, on the other hand, will tend to degrade the oil-supply process in one-g under certain conditions. Of the two processes, I think the flow effects is most important and, therefore, the one-g tests are conservative relative to orbital performance. Aerosol formation, and oiler droplet size is an interesting area for potential study in the refinement of the oiler concept for long missions. However, I believe this was a relatively unimportant influence on the Skylab CMG performance.

If you have any further questions, please feel free to call me on Extension 2942.

Very truly yours,

David K. Snediker
Senior Project Manager

DKS:1k
LITERATURE SEARCH
SKYLAB CMG BEARING INVESTIGATION

Prepared by: T. Wheelock

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
1.0 REQUIREMENT FOR FURTHER INVESTIGATION OF BEARING OPERATION IN ORBIT

The limited amount of information as to the nature of the bearing failure in CMG 1 indicated that failure modes in orbit could be produced by conditions not readily duplicated by on-earth testing.

In order to further investigate phenomena connected with orbital conditions of weightlessness and hard vacuum, a study of published material was conducted.

The NASA Scientific and Technical Information Facility, College Park, Maryland was contacted in order to obtain a bibliography of applicable documents. They were to provide a literature search concerning the topic, "Lubricants in Zero Gravity and High Vacuum" (NASA Literature Search No. 25393, Mar. 23, 1974). This literature search was reviewed and 59 documents were selected as most likely to contain information relative to the specific application. Some were duplications issued under various organizations. The documents reviewed are listed in Appendix 1. Most reports originated in the mid-1960's as this was the period of maximum activity (and minimal actual experience).

2.0 REVIEW OF DOCUMENTS

The documents contained results of investigation analysis and testing of various lubricants and apparatus. Many concerned testing and comparison of various types of oils, greases, solid lubricants, metal films, transfer films, and combinations of these in hard vacuum ($10^{-5}$ to $10^{-10}$ torr). *Ref 20, 25, 27, 30, 32, 33. Some contained results

*Circled no's. refer to documents listed in Appendix I.
of life testing under various conditions simulating the space environment \(12, 15, 36, 30, 34, 35\). The effects of other factors of the space environment such as primary and secondary radiation \(5, 33, 40\), micro meteoroid impacts \(23, 40\) and weightlessness \(3, 14, 31\), were also discussed. A few documents actual results of operation in orbit \(17, 42\). Topics of particular interest included coefficient of friction investigations \(9, 17, 27, 42, 22\), oxide formation, surface chemistry and adhesion \(15, 16, 40, 22, 20\), evaporation loss \(19, 13, 33, 16, 40, 24\), test procedures used \(19, 33, 54, 26, 23\), and labyrinth seals \(20, 3, 15, 25\).

The entire concept of simulation of orbital conditions for life testing on Earth was questioned in one report \(19\). One report described high speed 6000 RPM life testing with greases \(16\).

There was little information concerning the operation of large preloaded bearings with oil lubrication only at high RPM in orbital conditions (which is the case for the ATM CMG). What data was applicable did not cover areas actually evaluated with operating CMG's in testing which simulated space conditions. An evaluation of the ATM CMG bearings and lubricants was made by Bendix Research Lab (Ref \(43\)).

Some papers had useful discussions and analyses of lubricant failure modes. The evaporation loss of KG-80 under various conditions of vacuum and temperature was determined from Ref. \(1, 17, 19, 24\). The effects of radiation were referred to, and judged not to be a significant consideration.
The effects of weightlessness ("0" gravity) were covered only briefly and affects were limited to feed from lubricant reservoirs, changes in bearing loads, convective cooling and suggestions that migration and circulation of lubricant could be affected. The theoretical effects of weightlessness on bulk fluids was thoroughly investigated in Ref. 43. These results do not directly apply to thin films or droplets of lubricant which occur in the bearings. Ref. 44 gave some insight into formation and adhesion of films.

3.0 CONCLUSIONS
The results of the literature search indicate that while generalized testing programs for conditions to be expected in a space application are useful, data for each application must be determined under specific conditions of the actual application. While many test programs were documented, there were none actually dealing with a comparable CMG application. These documents probably exist (the Bendix Document Ref. 43 is an example); however, they are not readily available for general information. There is surprisingly little information generally available for the behavior of lubricants (for any specific fluids) in a weightless condition. This is considered (possibly rightly so) a minor factor in the lubrication mechanisms; however, the quantitative effect on oil migration with the system remains in question. The actual behavior of the lubricant and lubrication system will only be completely defined when actual behavior in orbit is observed or bearings which have been in orbit are recovered.
4.0 APPENDIX I - PAPERS REVIEWED

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3. A70-34158 - Controlled Leakage Sealing of Bearings for Fluid Lubrication in a Space Vacuum Environment
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LUBRICANTS IN ZERO GRAVITY AND HIGH VACUUM

March 29, 1974

SCOPE: References pertinent to the above subject.

PERIOD: 1962 to date shown above

FORMAT: Citations arranged by Accession Number

NUMBER OF CITATIONS: Machine Search - 21

This Literature Search was prepared in response to an individual's specific request, and contains references selected to meet the requester's needs.

NASA SCIENTIFIC AND TECHNICAL INFORMATION FACILITY

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LUBRICATION IN SPACE ENVIRONMENTS

A/ADAMCZACK, R. L.; B/BENZING, R. J.; C/SCHWENKER, H.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS DIV., WRIGHT-PATTERSON AFB, OHIO.

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FORMAT AND CONTENT OF LIMITED DISTRIBUTION REFERENCES

This part of the NASA Literature Search is based on a computerized records search of the technical reports and other documents in the NASA scientific and technical information system which either bear a security classification or are not publicly available.

Many limited distribution documents received prior to 1973 were announced in NASA's classified announcement journal Classified Scientific and Technical Aerospace Reports (CSTAR). Publication of CSTAR was discontinued at the end of 1972.

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X---70001 through X---89999—documents with distribution limitations that are relatively old at the time of processing or contain preliminary information. These references have not
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- GOVT & CONTR. ONLY = U. S. Government Agencies and U. S. Government Agency contractors only (including NASA and NASA research and development contractors)
- DOMESTIC ONLY = U. S. organizations and citizens only

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Department of Defense documents accessioned by the Defense Documentation Center (DDC) (identified by "AD" number in the report number position in the citation) are available to registered DDC users as follows:

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NASA Literature Search Number

25393

LUBRICANTS IN AERO GRAVITY AND HIGH VACUUM

March 29, 1974

SCOPE: References pertinent to the above subject.

PERIOD: 1962 to date shown above

FORMAT: Citations arranged by Accession Number

NUMBER OF CITATIONS: Machine Search - 138

This Literature Search was prepared in response to an individual's specific request, and contains references selected to meet the requester's needs.

NASA SCIENTIFIC AND TECHNICAL INFORMATION FACILITY

FF NO. 862 Rev. Sept. 67
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Other comments, suggestions, or criticism which you have to offer will be appreciated. Finally, you will note that a form entitled Optional Form for Requesting NASA Literature Searches is included with this search for your use in any subsequent request. We believe use of the form will result in preparation of a more satisfactory search.
RESEARCH ON BEARING LUBRICANTS FOR USE IN A HIGH VACUUM

(WORK, DURING THE RESEARCH PROGRAM, WAS CONCENTRATED ON LUBRICANT
DEVELOPMENT, GEAR LUBRICATION AND EVALUATION, PROVIDING COATED TEST
SPECIMENS, ADVISING NASA CONTRACTORS ABOUT SOLID LUBRICATION SPECIFIC
APPLICATIONS, AND INVESTIGATION OF NEW METHOD OF ATTACHING LUBRICATING
SOLIDS TO BEARING SURFACES BY SPUTTERING TECHNIQUE.)

A/CAMPBELL, M. E.; B/HASS, H. A/(MIDWEST RES. INST.); B/(MIDWEST RES. INST.)

/*BEARINGS*//*BINDERS (MATERIALS)*//*GRAPHITE*//*HIGH
VACUUM*//*LUBRICANTS*//*MELTING POINTS*//*MOLYBDENUM DISULFIDES*//*NOBLE
METALS*//*PHOSPHATES*//*RESEARCH AND DEVELOPMENT*//*SILICATES

IMPROVED MOLECULAR SORBENT TRAP FOR HIGH-VACUUM SYSTEMS

(CLOSED CYCLE REFRIGERATION LOOP IN WHICH TRAYS HOLDING MOLECULAR
SORBENT ARE MADE TO SERVE AS COOLING BAFLES IMPROVES THE PERFORMANCE
OF HIGH VACUUM SYSTEMS. HIGH PERFORMANCE IS OBTAINED WITH ALMOST NO
DECREASE IN PUMPING SPEED.)

A/KNECHTEL, F. D.; B/PITTS, H. C.

/*BAFFLES*//*COLD WATER*//*COMPRESSING*//*CONDENSERS (LIQUIFIERS)*//*COOLING
SYSTEMS*//*FEEDBACK CONTROL*//*FREQUENCY*//*HIGH VACUUM*//*INDUCTORS*//*LIQUID
NITROGEN*//*LUBRICATING CILS*//*MOLECULAR
ABSORPTION*//*REFRIGERANTS*//*SORBENTS*//*STAINLESS
STEELS*//*TRAPS*//*WELDING

MACHINE TESTS SLOW-SPEED SLIDING FRICTION IN HIGH VACUUM

(TESTING MACHINE THAT OPERATES WITHOUT ANY LUBRICATION OF THE
MACHINE ELEMENTS WITHIN THE VACUUM CHAMBER MEASURES STATIC FRICTION AND
SLIDING FRICTION AT VERY LOW SPEEDS. MOVING PARTS ARE HELD TO A
MINIMUM TO SIMPLIFY OPERATION IN THE VACUUM CHAMBER.)

A/SKYRUS, J.; B/WILKINSON, C.

/*AMPLIFIERS*//*FRICTION MEASUREMENT*//*HIGH VACUUM*//*KINETIC
FRICTION*//*LUBRICANT TESTS*//*MACHINERY*//*MEASURING INSTRUMENTS*//*METAL
STRIPS*//*MOLYBDENUM DISULFIDES*//*PINS*//*PLATES (STRUCTURAL
MEMBERS)*//*RECORDING INSTRUMENTS*//*SHAFTS (MACHINE ELEMENTS)*//*SIGNAL
ANALYZERS*//*SLIDING FRICTION*//*STATIC FRICTION*//*STRAIN GAGES*//*VACUUM
APPARATUS*//*WHEATSTONE BRIDGES
66B10679* CATEGORY 5 LEWIS-359 66/12/CC UNCLASSIFIED DOCUMENT
DOMESTIC

IMPROVED ROLLING ELEMENT BEARINGS PROVIDE LOW TORQUE AND SMALL TEMPERATURE RISE IN ULTRAHIGH VACUUM ENVIRONMENT.

(ROLLING ELEMENT BEARING WITH STAINLESS STEEL RACES AND ROLLING ELEMENTS AND A POROUS BRONZE CAGE SUCCESSFULLY OPERATES IN ULTRAHIGH VACUUM ENVIRONMENTS AT A LOW TORQUE AND WITH SMALL TEMPERATURE RISE. ALL COMPONENTS ARE BURNISHED IN MOLYBDENUM DISULFIDE.)

A/GLENN, D. C.

/*BALL BEARINGS/*CONTAMINANTS/*COPPER ALLOYS/*ELECTRONIC EQUIPMENT/*HIGH VACUUM/*LUBRICANTS/*MOLYBDENUM DISULFIDES/*POLYTETRAFLUOROETHYLENE/*STAINLESS STEELS/*TEMPERATURE GRADIENTS/*TIN ALLOYS/*TORQUE

66B10165* CATEGORY 3 LEWIS-245 66/04/CC UNCLASSIFIED DOCUMENT
DOMESTIC

GALLIUM ALLOY FILMS INVESTIGATED FOR USE AS BOUNDARY LUBRICANTS.

(GALLIUM ALLOYED WITH OTHER LOW MELTING POINT METALS HAS EXCELLENT LUBRICANT PROPERTIES OF FLUIDITY AND LOW VAPOR PRESSURE FOR HIGH TEMPERATURE OR VACUUM ENVIRONMENTS. THE ADDITION OF OTHER SCFT METALS REDUCES THE CORROSIVITY AND FORMATION OF UNDESIRABLE ALLCYS NORMALLY FOUND WITH GALLIUM.)

/*BEARING ALLOYS/*COATINGS/*CORROSION PREVENTION/*FRICTION REDUCTION/*GALLIUM ALLCYS/*HIGH TEMPERATURE ENVIRONMENTS/*HIGH VACUUM/*INDIUM ALLOYS/*LUBRICANTS/*SLIDING FRICTION/*STAINLESS STEELS/*TIN ALLOYS/*VAPOR PRESSURE/*WEAR INHIBITORS/*WEAR TESTS

65B10366* CATEGORY 3 JPL-SC-075 65/12/CC UNCLASSIFIED DOCUMENT
DOMESTIC

UNIQUE GEAR DESIGN PROVIDES SELF-LUBRICATION.

(COMPOSITE GEAR CONFIGURATION PROVIDES A RELIABLE AUTOMATIC MEANS FOR REPLENISHING GEAR MECHANISM LUBRICANTS THAT DISSIPATE IN THE HARSH ENVIRONMENT OF SPACE. THE CENTER OR HUB SECTION OF THE GEAR CONSISTS OF A POROUS, OIL IMPREGNATED MATERIAL, AND THE OUTER OR TOOTHED SECTION HAS RADIALLY DRILLED PASSAGES TO CAUSE THE OIL TO GRADUALLY FLOW TO THE GEAR TEETH SURFACE.)

A/WINIARSKI, F. J.

/*FLOW VELOCITY/*GEAR TEETH/*GEARS/*LUBRICATING CILS/*POROUS MATERIALS/*SELF LUBRICATION/*SPACECRAFT LUBRICATION

63B10453* CATEGORY 3 M-FS-54 64/11/CC UNCLASSIFIED DOCUMENT
DOMESTIC

MOLYBDENUM DISULFIDE MIXTURES MAKE EFFECTIVE HIGH-VACUUM LUBRICANTS.

(FIVE DIFFERENT MIXTURES OF MOLYBDENUM DISULFIDE ARE FOUND TO BE EFFECTIVE BEARING LUBRICANTS WHEN TESTED AT VERY LOW PRESSURES AND HIGH TEMPERATURES.)

/*BEARINGS/*ENVIRONMENTS/*HIGH VACUUM/*LUBRICANT TESTS/*LUBRICANTS/*MOLYBDENUM DISULFIDES
GALLIUM USEFUL BEARING LUBRICANT IN HIGH-VACUUM ENVIRONMENT
(SOLID GALLIUM IS USED AS A LUBRICANT ON BEARINGS MADE OF COMPATIBLE
MATERIALS. SUCH LUBRICANTS PERFORM WELL IN A HIGH VACUUM AND UNDERS LOW
TEMPERATURE.)

A/BUCKLEY, D. H.

BEARINGS/COSTS/GALLIUM/HIGH VACUUM/LOW TEMPERATURE
ENVIRONMENTS/LUBRICANTS/STEELS

GREASE LUBRICATION OF ROLLING BEARINGS IN SPACECRAFT. I
A/MAHNCKE, H. E.; B/SCHWARTZ, A. J. B/ (SKF ENGINEERING AND
RESEARCH CENTER, KING OF PRUSSIA, PA.) MEMBERS, $1.50; NONMEMBERS,
$2.00

AMERICAN SOCIETY OF LUBRICATION ENGINEERS AND AMERICAN SOCIETY OF
MECHANICAL ENGINEERS, JOINT LUBRICATION CONFERENCE, ATLANTA, GA., CCT.
15-18, 1973, ASLE 8 P. RESEARCH SPONSORED BY THE SPERRY RAND CORP.

EVAPORATION RATE/GREASES/LUBRICANT TESTS/ROLLER
BEARINGS/SPACECRAFT LUBRICATION/ACCELERATED LIFE TESTS/SEALS
(STOPPERS)/TEMPERATURE EFFECTS/VACUUM EFFECTS/VACUUM PRESSURE

SOLID LUBRICANTS FOR SPACE TECHNOLOGY --- HIGH TEMPERATURE AND
VACUUM ENVIRONMENTS
A/MUKHERJEE, M. K. A/INDIAN SPACE RESEARCH ORGANIZATION, VIKRAM
SARABHAI SPACE CENTRE, TRIVANDRUM, INDIA) $1.00

WORLD CONFERENCE IN INDUSTRIAL TRIBOLOGY, 1ST, INDIAN INSTITUTE OF
TECHNOLOGY, NEW DELHI, INDIA, DEC. 11-18, 1972, PAPER. 13 P.

HIGH TEMPERATURE LUBRICANTS/LUBRICANT TESTS/SOLID
LUBRICANTS/SPACECRAFT LUBRICATION/VACUUM EFFECTS/BEARINGS
COMPOSITE MATERIALS/TECHNOLOGY ASSESSMENT/ THERMAL VACUUM TESTS
MODELING THE EFFECT OF AIR AND CIL UPON THE THERMAL RESISTANCE OF A SPHERE-FLAT CONTACT.

A/YOVANOVICH, M. M.; B/KITSCHA, W. W.; C/WATERLOO, UNIVERSITY, WATERLOO, ONTARIO, CANADA; D/WATERLOO, UNIVERSITY, WATERLOO, ATOMIC ENERGY OF CANADA, LTD., SHERIDAN PARK, ONTARIO, CANADA) MEMBERS, $1.50; NONMEMBERS, $2.00

AMERICAN INSTITUTE OF AERONAUTICS AND ASTROPAUTICS, THERMOPHYSICS CONFERENCE, 8TH, PALM SPRINGS, CALIF., JULY 16-18, 1973, 13 P.

RESEARCH SUPPORTED BY THE NATIONAL RESEARCH COUNCIL OF CANADA.

AIR/CONTACT RESISTANCE/LUBRICATING CILS/SLIPFACE PROPERTIES/ THERMAL RESISTANCE/BEARING/GAS-SOLID INTERFACES/LIQUID-SOLID INTERFACES/SPACECRAFT INSTRUMENTS/SPACECRAFT LUBRICATION/THERMAL VACUUM TESTS.

USAGE OF THE VNII NP-300A LUBRICANT IN GAS-LIQUID CHROMATOGRAPHY AND HIGH-VACUUM TECHNOLOGY

A/NAZAROVA, L. I.; B/NIKONOROV, E. M.; C/LULLY, K. I.; D/VOLOBOEV, N. K.; E/KUZ'MINA, A. V. E/VSESOIUZNYI NAUCHNO-ISSLEDOVATELSKII INSTITUT PC PERERABOTKE NEFTI I GAZA I POLUCHENII UHSSKUSTVENNOGO ZHIDKOGO TOPLIVA, USSR)

KHIMIA I TEKHNOLOGIIA TOPLIV I MASEL, VOL. 18, NO. 2, 1972, P. 53-56. IN RUSSIAN.

GAS CHROMATOGRAPHY/HIGH TEMPERATURE LUBRICANTS/HIGH VACUUM/HYDROCARBONS/LUBRICATING CILS/GAS-LIQUID INTERACTIONS/LUBRICANT TESTS/ THERMAL STABILITY/VACUUM APPARATUS.

FRICTIONAL BEHAVIOUR OF MOLYBDENUM DISULPHIDE IN HIGH VACUUM.

A/MATSUNAGA, M.; B/HOSHIMOTO, K.; C/UCHIYAMA, Y. E/ITCHY, UNIVERSITY, TOKYO, JAPAN; B/NATIONAL RESEARCH INSTITUTE FOR METALS, TOKYO, JAPAN; C/KANAZAWA UNIVERSITY, KANAZAWA, JAPAN)

WEAR, VOL. 22, NOV. 1972, P. 185-192.

COEFFICIENT OF FRICTION/CONTACT RESISTANCE/DISULFIDES/LUBRICANT TESTS/MOLYBDENUM SULFIDES/SOLID LUBRICANTS/ADHESION/ABSORPTION/HIGH VACUUM/METAL BEARING/PROTECTIVE COATINGS/SLIDING FRICTION/TEST EQUIPMENT/TRANSIENT RESPONSE/VACUUM EFFECTS/WEAR.
THE EVAPORATION OF VARIOUS LUBRICANT FLUIDS IN VACUUM.

A/HAMILTON, D. B.; B/OGDEN, J. S.; B/BATTELLE COLUMBUS LABORATORIES, COLUMBUS, OHIO

MEMBERS, $1.50; NONMEMBERS, $2.00

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND AMERICAN SOCIETY OF LUBRICATION ENGINEERS, INTERNATIONAL LUBRICATION CONFERENCE, NEW YORK, N.Y., OCT. 9-12, 1972, ASLE 6 P.

/*EVAPORATION/*GREASES/*LUBRICANT TESTS/*PERFLUORO COMPOUNDS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/*ETHERS/*MATERIALS TESTS/*OUTGASSING/*TEMPERATURE EFFECTS/*VISCOSITY

LUBRICATION FOR LONG-TERM AEROSPACE APPLICATIONS.

SPACECRAFT LUBRICATION SYSTEMS DESIGN FOR LONG MISSION LIFETIMES, DISCUSSING MECHANICAL DESIGN PARAMETERS AND LIQUID AND SOLID LUBRICANT CHARACTERISTICS

A/HINRICKS, J. T.; B/LORAN, T. J.; C/FRIEBEL, W. R.; C/(BALL BROTHERS RESEARCH CORP., BOULDER, COLO.)


/*LONG TERM EFFECTS/*LUBRICATION SYSTEMS/*SERVICE LIFE/*SPACECRAFT LUBRICATION/*AEROSPACE ENVIRONMENTS/*LUBRICATING OILS/*SOLID LUBRICANTS/*SOLID-SOLID INTERFACES/*SPACECRAFT ENVIRONMENTS/*SYSTEMS ANALYSIS

SPACE RESEARCH PHYSICAL AND TECHNICAL PRINCIPLES.

RUSSIAN BOOK ON PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH COVERING NEAR EARTH AND INTERPLANETARY ENVIRONMENTAL FACTORS AND EFFECTS ON SPACECRAFT DESIGNS AND MATERIALS

A/KROSHKIN, M. G. $12

(TRANSLATION OF FIZIKO-TEKHNIKESKIE OSNYCHKOSMICHESKOGO ISSLEDOVANII, MOSCOW, IZDATELSTVO MASHINOSTRUCENIE, 1969.) JERUSALEM, ISRAEL PROGRAM FOR SCIENTIFIC TRANSLATIONS, LTD., 1972. 316 P.

/*AEROSPACE ENVIRONMENTS/*AEROSPACE SCIENCES/*EARTH ATMOSPHERE/*INTERPLANETARY SPACE/*SPACECRAFT DESIGN/*AEROSPACE INDUSTRY/*GEOMAGNETISM/*RADIATION DAMAGE/*SPACECRAFT COMPONENTS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS
SELECTING MATERIALS AND HARDWARE.
(HIGH AND ULTRAHIGH VACUUM EQUIPMENT AND COMPONENTS SELECTION,
DISCUSSING GAS-SURFACE INTERACTIONS, CONTAMINATION AND CLEANING
PROBLEMS)
A/WHEELER, W. R. (VARIAN ASSOCIATES, PALO ALTO, CALIF.)
PHYSICS TODAY, VOL. 25, AUG. 1972, P. 52-58.
/*CLEANING*/DECONTAMINATION/*HIGH VACUUM/*SURFACE
REACTIONS/*ULTRAHIGH VACUUM/*VACUUM SYSTEMS/ ENVIRONMENTAL CONTROL/
GAFFETS/ JOINTS (JUNCTIONS)/ LUBRICANTS/ OUTGASSING/ VACUUM PUMPS/
VALVES

TRANSFER FILM FORMATION BY LUBRICATIVE COMPOSITES.
(SELF LUBRICATING POLYTETRAFLUOROETHYLENE AND POLYIMIDE COMPOSITES
TRANSFER FILM FORMATION TESTS, STUDYING FILM THICKNESS AND UNIFORMITY)
A/JONES, J. R.; B/GARDOS, M. N. (HUGHES AIRCRAFT CO., CULVER
CITY, CALIF.)
IN INTERNATIONAL CONFERENCE ON SOLID LUBRICATION, 1ST, DENVER,
COLO., AUGUST 24-27, 1971, PROCEEDINGS. (A72-18583 C6-15) PARK RIDGE,
ILL., AMERICAN SOCIETY OF LUBRICATION ENGINEERS, 1971, P. 185-197.
RESEARCH SPONSORED BY THE HUGHES AIRCRAFT CO.
/*FILM THICKNESS*/LUBRICANT
TESTS/*POLYIMIDES/*POLYTETRAFLUOROCETHEYLENE/*SELF LUBRICATING
MATERIALS/*SPACECRAFT LUBRICATION/ BALL BEARINGS/ COMPOSITE MATERIALS/
CONFERENCES/ MOLYBDENUM DISULFIDES/ PHOTOMICROGRAPHY/ SLIDING FRICTION/
THICK FILMS

FRICTION AND WEAR CHARACTERISTICS OF LUBRICATIVE COMPOSITES IN AIR
AND IN VACUUM
(FRICTION-WEAR CHARACTERISTICS OF SELF LUBRICATING COMPOSITES UNDER
SLIDING CONDITIONS IN AIR AND VACUUM)
A/GARDOS, M. N.; B/JONES, J. R. (HUGHES AIRCRAFT CO., CULVER
CITY, CALIF./) MEMBERS, $0.75, NONMEMBERS, $1.50.
CHICAGO, ILLINOIS, AMERICAN SOCIETY OF LUBRICATION ENGINEERS,
AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND AMERICAN SOCIETY OF
LUBRICATION ENGINEERS, LUBRICATION CONFERENCE, CINCINNATI, OHIO, COT.
/*COMPOSITE MATERIALS*/LUBRICANT TESTS/*SELF LUBRICATION/*SLIDING
FRICTION/*VACUUM EFFECTS/*AIR/ CONFERENCES/ SPACECRAFT LUBRICATION/
WEAR
STRUCTURAL MODIFICATIONS OF FLUoro-ALKYL S-TRIAZINES AND THEIR LUBRICANT PROPERTIES

A/SNYDER, C. E., JR. (AA/USAF, MATERIALS LAB., WRIGHT-PATTERSON AFB, OHIO/.) MEMBERS, $0.75, NONMEMBERS, $1.50.


ENERGY TRANSFER, FLUOROCARBONS, HIGH TEMPERATURE LUBRICANTS, SPACECRAFT LUBRICATION, WORKING FLUIDS, AEROSPACE SYSTEMS, CHEMICAL PROPERTIES, CONFERENCES, PHYSICAL PROPERTIES, THERMAL STABILITY

PERFORMANCE OF SOLID LUBRICANT COATINGS

A/KURILEV, G. V.

RUSSIAN ENGINEERING JOURNAL, VOL. 49, NO. 12, P. 28-31. /VESTNIK MASHINOSTROENIIA, NO. 12, 1969./

INORGANIC COATINGS, LUBRICANT TESTS, MOLYBDENUM DISULFIDE, PERFORMANCE TESTS, SOLID LUBRICANTS, HIGH VACUUM, HUMIDITY, VACUUM EFFECTS

DESIGN CRITERIA FOR SPACECRAFT FLUID MECHANICAL SYSTEMS

A/HUDSON, J. T. (AA/MARTIN MARIETTA CORP., ADVANCED CONCEPTS DEPT., BALTIMORE, MD./.) MEMBERS, $1.00, NONMEMBERS, $1.50.

NEW YORK, SOCIETY OF AUTOMOTIVE ENGINEERS, INC., SOCIETY OF AUTOMOTIVE ENGINEERS, NATIONAL AERONAUTIC AND SPACE ENGINEERING AND MFG. MEETING, LOS ANGELES, CALIF., CCT. 5-9, 1970.

FLUIDICS, MECHANICAL DEVICES, SPACECRAFT DESIGN, SYSTEMS ENGINEERING, CONFERENCES, SPACECRAFT CONSTRUCTION, SPACECRAFT ENVIRONMENTS, SPACECRAFT LUBRICATION, WEIGHTLESSNESS
70A43325# ISSUE 22 PAGE 4C47 CATEGORY 15 7C/09/00 16 PAGES
UNCLASSIFIED DOCUMENT
AEROSPACE LUBRICATION FOR ADVANCED VEHICLES
(LUBRICATION TECHNOLOGY FOR SPACE SHUTTLES, DISCLOSING VEHICLES AND
AIRBREATING AND ROCKET ENGINES)
A/JOHNSON, R. L.; B/MANGANIELLO, E. J. (AB/NASA, LEWIS RESEARCH
CENTER, CLEVELAND, OHIO/)
GERMAN SOCIETY OF TRIBOLOGY, ANNUAL MEETING, ESSEN, WEST
GERMANY, SEP. 22, 23, 1970, PAPER.
/LUBRICATION SYSTEMS/*SPACE SHUTTLES/*SPACECRAFT LUBRICATION/ AIR
BREATING ENGINES/ CONFERENCES/ ROCKET ENGINES

70A35247 ISSUE 17 PAGE 3060 CATEGORY 11 69/00/00 10 PAGES
UNCLASSIFIED DOCUMENT
THE PUMPING SYSTEM OF THE ULTRA-HIGH VACUUM SPACE ENVIRONMENTAL
CHAMBER
(CRYP Demping SYSTEMS OF ULTRAHIGH VACUUM SPACE ENVIRONMENTAL
CHAMBERS)
A/ASAI, J.; B/NAKAGAWA, H.; C/SASAKI, T.; D/TSLNCDA, R.
(AC/MINISTRY OF POSTS AND TELECOMMUNICATIONS, RADIOS RESEARCH
LABS., ITOKYO, JAPAN/, AD/JAPAN OXYGEN CO., LTD., KAWASAKI, JAPAN/)
TOKYO, AGNE PUBLISHING, INC., IN- INTERNATIONAL SYMPSIUM ON
SPACE TECHNOLOGY AND SCIENCE, 8TH, TOKYO, JAPAN, AUG. 25-30, 1969,
PROCEEDINGS. P. 481-490. /A7C-35241 17-30/
/CRYOPUMPING/*SPACE ENVIRONMENT SIMULATION/*ULTRAHIGH
VACUUM/*VACUUM CHAMBERS/ CONFERENCES/ ION PUMPS/ LUBRICATION/ VACUUM
PUMPS

70A34166# ISSUE 16 PAGE 2923 CATEGORY 15 7C/00/00 8 PAGES
UNCLASSIFIED DOCUMENT
EVALUATION OF DRY LUBRICANTS AND BEARINGS FOR SPACECRAFT
APPLICATIONS
(INSTRUMENT SIZE BALL BEARINGS LUBRICATED WITH BONDED DRY CR
TRANSFER FILMS IN SIMULATED INTERPLANETARY SPACECRAFT TESTS)
A/KIRKPATRICK, C. L.; B/CLING, H. C. (AB/GENERAL ELECTRIC CO.,
VALLEY FORGE SPACE TECHNOLOGY CENTER, KING OF PRUSSIA, PA./)
SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
8 - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A7C-34154 16-15/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 107-114.
/BALL BEARINGS/*INTERPLANETARY SPACECRAFT/*SOLID LUBRICANTS/*SPACE
ENVIRONMENT SIMULATION/*SPACECRAFT LUBRICATION/ BONCING/ PERFORMANCE
TESTS/ SERVICE LIFE/ VACUUM EFFECTS

PAGE 8 (ITEMS 23- 25 OF 138)
CONTROLLED-LEAKAGE SEALING OF BEARINGS FOR FLUID LUBRICATION IN A SPACE VACUUM ENVIRONMENT

(CONTROLLED LEAKAGE SEALING OF HYDRODYNAMIC BEARING LUBRICATION SYSTEMS FOR SPACE VEHICLES IN SYNCHRONOUS ORBIT)

A/SILVERSHER, H. I. (AA/LOCKHEED MISSILES AND SPACE CC., SUNNYVALE, CALIF. /)
SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A70-34154 16-15/
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/*BEARINGS/*LUBRICATION SYSTEMS/*SEALING/*SPACECRAFT LUBRICATION/*SYNCHRONOUS SATELLITES/*LEAKAGE/*LONG TERM EFFECTS/*SPACE MISSIONS/*VACUUM EFFECTS

LUBRICATION OF DC MOTORS, SLIP RINGS, BEARINGS, AND GEARS FOR LONG-LIFE SPACE APPLICATIONS

(LUBRICATION SYSTEM FLIGHT PERFORMANCE AND LABORATORY TEST DATA FOR SPACE APPLICATIONS, CONSIDERING TORQUE MOTORS, SLIP RINGS, BEARINGS AND GEARS)

A/MAYER, R. W.; B/PERRIN, B. J. (AA/BALL BRCS. RESEARCH CORP., BOULDER, COLO./)
SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A70-34154 16-15/
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/*LUBRICATION SYSTEMS/*PERFORMANCE TESTS/*SPACECRAFT LUBRICATION/*BEARINGS/*GEARS/*ORGANIC COMPOUNDS/TORQUE MOTORS/*VACUUM EFFECTS

LUBRICATION AS PART OF TOTAL DESIGN

(LUBRICATION ROLE IN AEROSPACE ENGINEERING, DISCUSSING LUBRICANT AND COMPONENT SELECTION, ENVIRONMENTAL FACTORS, ETC)

A/CLAUSS, F. J. (AA/LOCKHEED MISSILES AND SPACE CC., SUNNYVALE, CALIF./)
SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
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/*AEROSPACE ENGINEERING/*LUBRICANTS/*SPACECRAFT LUBRICATION/*AEROSPACE ENVIRONMENTS/*BALL BEARINGS/*ELECTRIC MOTORS/*GEARS/*VACUUM EFFECTS
UNIQUE MECHANISM FEATURES OF ATS STABILIZATION BCCM PACKAGES
(ATS ATTITUDE STABILIZATION BOOM PACKAGES, DESCRIBING TORQUE
TRANSMISSION, DRUM SYNCHRONIZATION, ELECTRICAL ISOLATION, LUBRICATION,
ETC)

A/GRIMSHAW, F. R.; B/LOHNES, R. A.; C/MATTEO, C. N. (AC/GE
VALLEY FORGE SPACE TECHNOLOGY CENTER, KING OF PRUSSIA, PA., AA/SPAR
AEROSPACE PRODUCTS, LTD., TORONTO, CANADA/.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
A - GENERAL APPLICATIONS. EDITED BY G. G. HERZL. /A70-34101 16-31/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 399-407.

/*APPLICATIONS TECHNOLOGY SATELLITES/*BOOMS (EQUIPMENT)/*SATELLITE
ATTITUDE CONTROL/ DEPLOYMENT/ GEARS/ MECHANICAL DEVICES/ SPACECRAFT
LUBRICATION/ WEIGHTLESSNESS

DESPINNING THE ATS SATELLITE
(TWO STAGE YO-YO WITH NUTATION CAMPER FOR DESPINNING ATS SYNCHRONOUS
SATELLITES, DESCRIBING MECHANICAL AND THERMAL DESIGN, LUBRICATION,
ASSEMBLY, TESTING, ETC)

A/DALLAS, J. P. (AA/HUGHES AIRCRAFT CO., LOS ANGELES, CALIF./.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
A - GENERAL APPLICATIONS. EDITED BY G. G. HERZL. /A7C-34101 16-31/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 139-145.

/*APPLICATIONS TECHNOLOGY SATELLITES/*SPIN REDUCTION/*SYNCHRONOUS
SATELLITES/*YO-YO DEVICES/ COST REDUCTION/ MECHANICAL DEVICES/
PERFORMANCE TESTS/ SPACECRAFT LUBRICATION/ SPACECRAFT RELIABILITY/
VACUUM EFFECTS

SLIP RING ASSEMBLIES FOR SPACECRAFT DEVICES
(SLIP RING ASSEMBLIES FOR SPACECRAFT DEVICES, EVALUATING SLIDING
ELECTRICAL CONTACT INDUSTRY TECHNOLOGICAL CAPABILITIES)

A/GLOSSBRENNER, E. W. (AA/LITTON INDUSTRIES, INC., POLY-
SCIENTIFIC DIV., BLACKSBURG, VA./.)

BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA
POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL
CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG,
VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 79-90. /A70-33805 16-15/
/*ELECTRIC CONTACTS/RING STRUCTURES/*SLIDING CONTACT/*SPACECRAFT
COMPONENTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ LUBRICATION SYSTEMS/
MOYLBDENUM DISULFIDES/ NIOBIIUM COMPOUNDS
PERFORMANCE OF SOLID LUBRICATED CONTACTS FOR SPACECRAFT
(ISOLID LUBRICATED CONTACTS PERFORMANCE IN NIMBUS AND AAC SPACECRAFT, DISCUSSING SLIP RING ASSEMBLY AND NOISE PROBLEMS IN AG-GRAPHITE BRUSHES)
A/ORBENK, S.
BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG, VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 71-77. /A70-33805 16-15/
*/ELECTRIC CONTACTS/*NIMBUS SATELLITES/*OAO/*SOLID LUBRICANTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ MCLYDENAUM DISULFIDES/ PERFORMANCE/ RING STRUCTURES/ SATELLITE INSTRUMENTS/ STAR TRACKERS

SLIDING ELECTRICAL CONTACTS FOR UNMANNED SCIENTIFIC SATELLITES
(SLIDING ELECTRICAL CONTACTS FOR UNMANNED SCIENTIFIC SATELLITES, DISCUSSING NIMBUS AVCS CAMERA IRIS MOTORS)
A/ORBENK, A. J. (AA/NASA, GODDARD SPACE FLIGHT CENTER, SYSTEMS DIV., GREENBELT, MD. )
BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG, VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 49-70. /A70-33805 16-15/
*/ELECTRIC CONTACTS/*SCIENTIFIC SATELLITES/*SLIDING CONTACT/*SPACECRAFT CONSTRUCTION MATERIALS/ CAMERA SHUTTERS/ CONFERENCES/ NIMBUS SATELLITES/ OAO/ SATELLITE INSTRUMENTS/ SPACECRAFT LUBRICATION/ STAR TRACKERS

LUBRICANTS FOR SPACE. II
(VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES APPLICABLE TO MINIATURE BALL BEARINGS AND INSTRUMENT BALL BEARINGS AND INSTRUMENT GEARING FOR SPACE SYSTEMS)
A/HARRIS, C. L. (AA/ELLIOTT BRCS. /LONDON/, LTD., ELLIOTT SPACE AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND/)
70A17780# ISSUE 6 PAGE 1141 CATEGORY 30 69/CO/00 6 PAGES IN FRENCH IN FRENCH. UNCLASSIFIED DOCUMENT
DESCRIPTION OF SOME IMPORTANT APPLIED RESEARCH IN THE SPACE FIELD (APPLIED SPACE RESEARCH COVERING PHOTOVOLTAIC CELLS, ELECTRIC PROPULSION, ONBOARD DATA STORAGE, VACUUM LUBRICATION AND NASA-EUROPEAN ORGANIZATIONS BUDGETS)
A/BONCHET, J.-C. (AA/ESRO, PARIS, FRANCE)
SCIENCES ET INDUSTRIES SPATIALES, VOL. 5, NO. 7-8, P. 19-23, 51.
/*BUDGETING*/DATA STORAGE/*ELECTRIC PROPULSION/*PHOTOVOLTAIC CELLS/*SPACECRAFT LUBRICATION/*AEROSPACE INDUSTRY/ ENERGY CONVERSION/ EUROPEAN SPACE PROGRAMS/ NASA PROGRAMS/ ONBOARD EQUIPMENT/ TECHNOLOGY UTILIZATION

70A17340 ISSUE 6 PAGE 1075 CATEGORY 15 69/CC/0C 10 PAGES
UNCLASSIFIED DOCUMENT
TESTING SOLID LUBRICANTS (SOLID LUBRICANTS FRICTION AND WEAR BENCH TESTING UNDER SIMULATED SPACE ENVIRONMENT)
A/AZZAM, H. T. (AA/DOW CORNING CORP., TRUMBULL, CTN/.)
/*SOLID LUBRICANTS*/SPACE ENVIRONMENT SIMULATION/*SPACECRAFT LUBRICATION/*WEAR TESTS/ CONFERENCES/ DEEP SPACE/ SERVICE LIFE

70A11693# ISSUE 2 PAGE 36C CATEGORY 30 69/CC/00 288 PAGES IN RUSSIAN IN RUSSIAN. UNCLASSIFIED DOCUMENT
PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH (SOVIET BOOK ON PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH COVERING NEAR EARTH AND INTERPLANETARY ENVIRONMENTAL FACTORS AND EFFECTS ON SPACECRAFT DESIGNS AND MATERIALS)
A/KROSHKIN, M. G.
MOSCOW, IZDATEL'STVO MASHINYSTROENIE,
/*AEROSPACE ENVIRONMENTS*/AEROSPACE SCIENCES/*INTERPLANETARY SPACE/SPACECRAFT DESIGN/ AEROSPACE INDUSTRY/ EARTH ATMOSPHERE/ RADIATION DAMAGE/ SPACECRAFT COMPONENTS/ SPACECRAFT LUBRICATION/ VACUUM EFFECTS

PAGE 12 (ITEMS 35- 37 OF 138)
VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES FOR SPACE-EXPOSED COMPONENTS

(VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES APPLICABLE TO MINIATURE BALL BEARINGS AND INSTRUMENT GEARING FOR SPACE SYSTEMS)

A/HARRIS, C. L. (AA/ELLIOTT BROTHERS /LONDON/, LTD., ELLIOTT SPACE AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND) MEMBERS, $0.75, NONMEMBERS, $1.50.


BALL BEARINGS/GEARS/SPACECRAFT LUBRICATION/VACUUM EFFECTS/LOADS (FORCES)/SELF LUBRICATION/TORQUE/VIBRATION EFFECTS

A FLYWHEEL FOR STABILIZING SPACE VEHICLES.

SPACE VEHICLE ATTITUDE STABILIZATION BASED ON ROTATING FLYWHEEL RESISTANCE TO CHANGES IN ROTATION AXIS ATTITUDE, NATURAL GREASE LUBRICATED HYDRAULIC BEARING SYSTEM)

A/REINHOUDT, J. P. (AA/PHILIPS' GLOELAMPENFABRIKEN, PHILIPS RESEARCH LABS., EINCHÒVEN, NETHERLANDS)

PHILIPS TECHNICAL REVIEW, VOL. 30, NO. 1, P. 2-6.

ATTITUDE STABILITY/FLYWHEELS/ROCKET ORIENTATION/SPACECRAFT STABILITY/ANTIFRICTION BEARINGS/GREASES/JOURNAL BEARINGS/SPACECRAFT LUBRICATION/STABILIZATION

A SHORT SURVEY OF EUROPEAN WORK ON LUBRICATION IN VACUUM.

EUROPEAN RESEARCH ON ULTRAHIGH VACUUM LUBRICATION IN SPACE ENVIRONMENTS, EMPHASIZING FRICTION OF MATERIALS UNDER VARIOUS LOADINGS AND TEMPERATURES)

A/BRISCOE, H. M.; B/DAUPHIN, J. (AB/EURCPEAN SPACE RESEARCH ORGANIZATION, EUROPEAN SPACE RESEARCH AND TECHNOLOGY CENTRE, NOORCWIJ, LONDON, ENGLAND) $7.20 PER SET OF 12.

LONDON, ENGLAND. INSTITUTION OF MECHANICAL ENGINEERS, INSTITUTE OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

AEROSPACE ENVIRONMENTS/EUROPEAN SPACE PROGRAMS/FRICTION/SPACECRAFT LUBRICATION/ULTRAHIGH VACUUM/BEARINGS/CONFERENCES/LOAD TESTS/TEMPERATURE EFFECTS
LUBRICATION OF BEARINGS AND GEARS FOR OPERATION IN A SPACE ENVIRONMENT.

(MINIATURE BALL AND JEWEL BEARINGS AND GEAR LUBRICATION IN ULTRAHIGH VACUUM TESTS FOR SPACE ENVIRONMENT OPERATION)

A/HARRIS, C. L.; B/WARWICK, M. G. (AB/ELLIOTT-AUTOMATION, LTC., SPACE AND WEAPONS RESEARCH LAB., CAMBERLEY, SURREY, ENGLAND/.) $7.20 PER SET OF 12.

LONDON, ENGLAND. INSTITUTION OF MECHANICAL ENGINEERS, RESEARCH SUPPORTED BY THE MINISTRY OF TECHNOLOGY. INSTITUTION OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

/*ANTIFRICTION BEARINGS/*GEARS/*LUBRICATION/*SPACE ENVIRONMENT SIMULATION/*WEAR TESTS/ BALL BEARINGS/ BONDING/ COMPOSITE MATERIALS/ CONFERENCES/ FRICTION REDUCTION/ LIFE (DURABILITY)/ METALS MATERIALS/ SOLID LUBRICANTS/ ULTRAHIGH VACUUM/ VAPOR PRESSURE

FRICTION AND WEAR STUDIES IN ULTRA-HIGH VACUUM AND THE EVALUATION OF ELECTRICAL SLIP-RINGS.

(DRY SLIDING FRICTION AND WEAR IN ULTRAHIGH VACUUM, EMPHASIZING SLIP RINGS AND BRUSHES FOR SPACE APPLICATIONS)


LONDON, ENGLAND. INSTITUTION OF MECHANICAL ENGINEERS, INSTITUTION OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

/*DRY FRICTION/*ELECTRIC CONTACTS/*SLIDING FRICTION/*ULTRAHIGH VACUUM/*WEAR TESTS/ BRUSHES/ CONFERENCES/ FRICTION REDUCTION/ GEARS/ LUBRICATION/ SPACE ENVIRONMENT SIMULATION/ STATIC FRICTION/ VAPOR PRESSURE

ACHIEVEMENTS IN STUDIES OF PROPERTIES OF MATERIALS USED FOR SPACE OBJECTS AND THEIR USE IN INDUSTRY.

(SATELLITE, SPACECRAFT AND ROCKET COMPONENTS COATINGS AND LUBRICANTS, NOTING FIG. RADIATION RESISTANCE OF POLYIMIDES)

A/KONRADI, G. G.; B/KOZELKIN, V. V.; C/NOVITSKII, L. A.; D/SVAREV, V. V.

UNITED NATIONS, CONFERENCE ON THE EXPLORATION AND PEACEFUL USES OF OUTER SPACE, VIENNA, AUSTRIA, AUG. 14-27, 1968, PAPER.

/*POLYIMIDES/*PROTECTIVE COATINGS/*RADIATION TOLERANCE/*SPACECRAFT LUBRICATION/ CONFERENCES/ POLYTETRAFLUROETHYLENE/ POLYVINYL CHLORIDE/ TECHNOLOGY UTILIZATION/ U.S.S.R. SPACE PROGRAM

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SPACE MECHANISM LUBRICATION.

(SPACE LUBRICATION SYSTEM FOR ORBITING SOLAR OBSERVATORY PROGRAM, DISCUSSING THEORETICAL HIGH VACUUM PRINCIPLES AND FLIGHT PERFORMANCE AND ENVIRONMENTAL TEST DATA)

A/MAYER, R. W.; B/PERRIN, B. J. (AA/BALL BROTHERS RESEARCH CORP., BOULDER, COLO. /)


/*ENVIRONMENTAL TESTS/*LUBRICATION SYSTEMS/*OSC/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/*CONFERENCES/*GROUND TESTS/*HIGH VACUUM/*ORGANIC LIQUIDS/*PERFORMANCE TESTS/*SPACE PROGRAMS

A MULTIPLE ENVIRONMENT TEST SYSTEM FOR COMBINED ENVIRONMENT SPACE SIMULATION.

(FLEXIBLE SYSTEM FOR ENVIRONMENT TESTING OF INSTRUMENT BEARING LUBRICANTS)

A/TIERNEY, W. L. (AA/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF./)

NORTH HOLLYWOOD, CALIF., WESTERN PERIODICALS CO. /SYMPOSIUM RECORD. VOLUME 9/, CONVENTION SPONSORED BY THE INST. OF ELECTRICAL AND ELECTRONICS ENGINEERS AND THE WESTERN ELECTRONIC MANUFACTURERS ASSN. IN- WESTERN ELECTRONIC SHOW AND CONVENTION, INTERNATIONAL ELECTRONIC CIRCUIT PACKAGING SYMPOSIUM, LOS ANGELES, CALIF., AUG. 15, 20, 1968, PROCEEDINGS. A968-43935 23-09:

/*ELECTRONIC EQUIPMENT TESTS/*ENVIRONMENTAL TESTS/*SPACE ENVIRONMENT SIMULATION/*TEST FACILITIES/*BEARINGS/*CONFERENCES/*HARDWARE/*LUBRICANTS/*SYSTEMS ENGINEERING/*TABLES (CATA)
PROBLEMS OF LUBRICATION AND BEARINGS IN SPACE VACUUM

LUBRICATION AND BEARINGS IN SPACE VACUUM ENVIRONMENT, EXAMINING GREASE, LIQUID AND SEMISOLIDS EVAPORATION RATE BY LANGMUIR EQUATION

A/BISSON, E. E. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO)

PARIS, INSTITUT FRANCAIS DES COMBUSTIBLES ET DE L'ENERGIE, JOURNEES INTERNATIONALES, 7TH, PARIS, FRANCE, APR. 4-8, 1967, PROCEEDINGS "ETATS DELA MATIERE SOUS LES EFFETS EXTREMES DES TRES HAUTES ET TRES BASSES TEMPERATURES, TRES HAUTES ET TRES BASSES PRESSIONS, INSTITUT FRANCAIS DES COMBUSTIBLES ET DE L'ENERGIE, JOURNEES INTERNATIONALES, 7TH, PARIS, FRANCE, APR. 4-8, 1967, PROCEEDINGS.<P. 529-542.

SELF LUBRICATING MATERIALS/*SLIDING FRICTION/*SOLID LUBRICANTS/*SPACECRAFT LUBRICATION/*ULTRAHIGH VACUUM/*VACUUM EFFECTS/ CONFERENCES/ MECHANICAL ENGINEERING/ SPACECRAFT DESIGN
SOME ASPECTS OF THE SURFACE CHEMISTRY OF ADHESION AND OF FRICTION.
(SURFACE CHEMISTRY, ADHESION AND RELATION TO FRICTION NOTING METHODS
FOR REDUCTION IN HIGH VACUUM ENVIRONMENTS).
A. ADAMSON, A. W. (AA/SOUTHERN CALIFORNIA, U., DEPT. OF CHEMISTRY,
LOS ANGELES, CALIF./.)

PHILADELPHIA, AMERICAN SOCIETY FOR TESTING AND MATERIALS /ASTM
SPECIAL TECHNICAL PUBLICATION NO. 431/, IN-ADHESION OR COLD WELDING
OF MATERIALS IN SPACE ENVIRONMENTS, AMERICAN SOCIETY FOR TESTING AND
MATERIALS AND AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL
MEETING, TORONTO, CANADA, MAY 1, 2, 1967, PROCEEDINGS. <A68-34168
17-15< P. 6-19.

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LUBRICANTS FOR THE SPACE ENVIRONMENT.
(SPACE ENVIRONMENT LUBRICANT SELECTION FOR SPECIFIC SPACE SYSTEMS
AND APPLICATIONS, GIVING TABLES OF LUBRICANTS FOR VARIOUS SATELLITES)
A. FLOM, D. G.; B. H. ALTNER, A. J. (AB/GENERAL ELECTRIC Co.,
AEROSPACE GROUP, MISSILE AND SPACE DIV., SPACE SCIENCES LAB., VALLEY
FORGE, PA./.) $0.50.

NEW YORK, AMERICAN INST. OF CHEMICAL ENGINEERS, AMERICAN INST.
OF CHEMICAL ENGINEERS, MATERIALS CONFERENCE, SYMPOSIUM ON MATERIALS FOR
RE-ENTRY AND SPACECRAFT SYSTEMS - SPACECRAFT MATERIALS, PART 2,
PHILADELPHIA, PA., MAR. 31-APR. 4, 1968.

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FRICTION AND WEAR OF POTENTIOMETER CONTACTS IN VACUUM.
(FRICTION AND WEAR OF WIRE WOUND POTENTIOMETERS SLIDING ELECTRIC
CONTACTS IN VACUUM, EVALUATING LUBRICATIVE POWDERS AND CONDUCTIVE
COMPOSITES)
A. JONES, J. R. (AA/HUGHES AIRCRAFT CO., CULVER CITY, CALIF./.)
WEAR, VOL. 11, P. 355-367.

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UNCLASSIFIED DOCUMENT

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LUBRICATION TECHNIQUES FOR VACUUM ENVIRONMENTS OF SPACE VEHICLES INCLUDING OILS, GREASES AND SOLID LUBRICANTS

A/BENZING, R. J.; B/MCCONNELL, B. D. (AA/USAF, SYSTEMS COMMAND, RESEARCH AND TECHNOLOGY DIV., MATERIALS LAB., WRIGHT-PATTERSON AFB, OHIO/ AB/USAF, SYSTEMS COMMAND, RESEARCH AND TECHNOLOGY DIV., WRIGHT-PATTERSON AFB, OHIO/)


*AEROSPACE ENVIRONMENTS/*LUBRICANTS/*SPACECRAFT ENVIRONMENTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ GREASES/ LUBRICATION SYSTEMS/ VACUUM EFFECTS

LUBRICATION IN SPACE VACUUM. III - LIFE TEST EVALUATION OF BALL BEARINGS LUBRICATED WITH OILS AND GREASES.

LIFE TESTS OF BALL BEARINGS LUBRICATED WITH OILS AND GREASES DURING OPERATION IN INDUCTION MOTORS UNDER SIMULATED SPACE VACUUM CONDITIONS.

A/HARRIS, C. L.; B/READ, J. E.; C/THOMPSON, J. E. (AC/ELLICOTT BROTHERS /LONDON/, LTD., SPACE AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND/)

LUBRICATION ENGINEERING, VCL. 24, P. 182-188. RESEARCH SUPPORTED BY THE MINISTRY OF AVIATION.

*BALL BEARINGS/*LUBRICATING CILS/*SERVICE LIFE/*SPACE ENVIRONMENT SIMULATION/*VACUUM EFFECTS/ DEGASSING/ ELECTRIC MOTORS/ ENVIRONMENTAL ENGINEERING/ ENVIRONMENTAL TESTS/ TORQUE

EVALUATION OF SPACE LUBRICANTS UNDER OSCILLATORY AND SLOW SPEED ROTARY MOTION.

SOLID FILM SPACE LUBRICANT METHODS EVALUATED FOR FRICTION SURFACES UNDER OSCILLATORY AND SLOW SPEED ROTARY MOTION.

A/VEST, C. E.; B/WARD, B. W., JR. (AB/NASA, GD/DCR SPACE FLIGHT CENTER, GREENBELT, MD/)


*FRICITION MEASUREMENT/*ROTATING BODIES/*SOLID LUBRICANTS/*SPACECRAFT LUBRICATION/*STRUCTURAL VIBRATION/ BEARINGS/ CONFERENCES/ GEL COATINGS/ MOLYBDENUM DISULFIDES/ POLYTERAFLUOROPOLYETHYLENE

PAGE 18 (ITEMS 51- 53 OF 138)
TEST STAND FOR THE MECHANICAL-DYNAMICAL TESTING OF LUBRICANTS UNDER HIGH- AND ULTRAHIGH-VACUUM CONDITIONS

(TEST STAND FOR CHECKING LUBRICATION OF AIRCRAFT AND SPACECRAFT PARTS UNDER FLIGHT STRESS, WEIGHTLESSNESS AND ENVIRONMENTAL CONDITIONS)

A/SPENGLER, G. (AA/DEUTSCHE VERSUCHSANSTALT FUER LUFT- UND RAUMFAHRT, INSTITUT FUER FLUGTREIB- UND SCHMIERSTOFFE, MUNICH, WEST GERMANY/.)

DVL-NACHRICHTEN, P. 324, 325.

/*AIRCRAFT PARTS/*ENVIRONMENT SIMULATION/*LUBRICATION SYSTEMS/*SPACE FLIGHT STRESS/*SPACECRAFT COMPONENTS/*TEST STANDS/*TEMPERATURE CONTROL/ ULTRAHIGH VACUUM/ WEIGHTLESSNESS

LUBRICATION IN SPACE VACUUM /10 SUPER -10 TORR/. I - TESTING TECHNIQUES FOR BALL BEARINGS.

(MODULAR TEST CAPSULE TECHNIQUE FOR VACUUM TESTING OF LIFETIMES AND BEARING TORQUE PERFORMANCE OF DRY FILM AND OIL LUBRICATED BALL BEARINGS)

A/HARRIS, C. L.; 3/READ, J. E.; C/THOMPSON, J. B.; D/WILSON, C. I. (AD/ELLIOTT BROTHERS /LONDON/, LTD., SPACE AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND/.)

LUBRICATION ENGINEERING, VOL. 24, P. 57-63. RESEARCH SUPPORTED BY THE MINISTRY OF AVIATION.

/*BALL BEARINGS*/LUBRICATING OILS*/MATERIALS TESTS/*TEST EQUIPMENT/*VACUUM CHAMBERS/ BEARING ALLIES/ ENVIRONMENTAL TESTS/ LIFE (DURABILITY)/ LUBRICATION/ MOLECULAR FLOW/ POLYBENE 1 DISULFIDES/ SPACE ENVIRONMENT SIMULATION/ VACUUM APPARATUS

LUBRICATION AND WEAR FUNDAMENTALS FOR HIGH-VACUUM APPLICATIONS.

(LUBRICATION AND WEAR IN HIGH VACUUM, CONSIDERING INABILITY TO MAINTAIN OXIDE FILMS; EVAPORATION OF LUBRICANTS, HEAT TRANSFER AND SLIDING FRICTION)


/*HEAT TRANSFER*/HIGH VACUUM/LUBRICATION/*SLIDING FRICTION/*WEAR/ADHESION/ ALLOY/ BONDING/ CONFERENCE/ CRYSTAL/ DIFFUSION/ EVAPORATION/ FILM/ FRICTION/ METAL/ OXIDE/ SHEAR/ SLIDING/ STRUCTURE

PAGE 19 (ITEMS 54- 56 OF 138)
ADVANCED AEROSPACE GREASES.
(GREASE LUBRICANTS FOR AEROSPACE APPLICATION, DETERMINING PHYSICAL PROPERTIES AND TESTING THEM AT 400 DEGREES F AND UNDER HIGH VACUUM)
/LUBRICATION ENGINEERING, VOL. 23, FEB. 1967, P. 52-56. <FOR ABSTRACT SEE ISSUE 16, PAGE 2711, ACCESSION NO. A66-3C40CS>
/*GREASE/*LUBRICANT/*PHYSICAL PROPERTY/*SPACE ENVIRONMENTAL LUBRICATION/ AEROSPACE/ BEARING/ CONFERENCE/ ENVIRONMENT/ LOAD/
LUBRICATION/ PHYSICAL/ PRESSURE/ PROPERTY/ SPACE/ SFEECE/ TECHNOLOGY/ TEMPERATURE/ TIME/ VACUUM

LUBRICATION AND Bearing PROBLEMS IN THE VACUUM OF SPACE.
(LUBRICATION AND BEARING PROBLEMS IN SPACE ENVIRONMENT, ACTING MOLYBDENUM DISULFIDE AND SILICONE)
A/BISSON, E. E. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/)
/GROUPEMENT POUR L'AVANCEMENT DE LA MECANIQUE INDUSTRIELLE, JOURNEES D'ETUDE DU FROTTEMENT ET DE L'USURE, PARIS, FRANCE, DEC. 5, 6, 1966, PAPER. 32 P. 25 REF. TRANSLATION.
/*BEARING/*MOLYBDENUM SULFIDE/*SILICONE/*SPACE ENVIRONMENTAL LUBRICATION/ CONFERENCE/ ENVIRONMENT/ EVAPORATION/ FRICTION/
LUBRICATION/ MOLYBDENUM/ SPACE/ SULFIDE/ TEFNON

REVIEW OF THE INFLUENCE OF SPACE ENVIRONMENT UPON VEHICLE COMPONENTS.
(SPACECRAFT ENVIRONMENTAL EFFECTS COVERING OUTGASSING IN HIGH VACUUM, DETERIORATION OF MATERIALS BY EVAPORATION, LUBRICATION AND CHANGES IN MECHANICAL AND ELECTRICAL PROPERTIES OF PLASTICS)
A/SHELT, R. (AA/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES AND SPACE CO., SPACE PROGRAMS DIV., SUNNYVALE, CALIF./)
/IN THE FLUID DYNAMIC ASPECTS OF SPACE FLIGHT, PROCEEDINGS OF THE NATO-AGARD SPECIALISTS' MEETING, MARSEILLE, FRANCE, APR. 20-24, 1964, VOLUME 1. AC67-16987 04-12 MEETING SPONSORED BY THE FLUID DYNAMICS PANEL OF AGACE, NEW YORK, GORDON AND BREACH, SCIENCE PUBLISHERS, INC.
/AGARDOGRAPH 87, VOLUME 1/, 1966, P. 141-165. 18 REF.
/*HIGH VACUUM/*LUBRICATION/*RADIATION EFFECT/*SPACE ENVIRONMENT/*SPACECRAFT CONSTRUCTION MATERIAL/*VACUUM EFFECT/ AGENA ROCKET/ BEARING/ CELL/ CONFERENCE/ DAMAGE/ DEGASSING/ DETERIORATION/ ELECTRIC/ ENVIRONMENT/ FLARE/ HAZARD/ MATERIAL/ MECHANICAL/ METEORITE/
RADIATION/ SOLAR/ SPACE/ SURFACE
LUBRICANTS AND MECHANICAL COMPONENTS OF LUBRICATION SYSTEMS FOR SPACE ENVIRONMENT.

(SYSTEM DESIGN AND LUBRICATING MATERIALS FOR SPACECRAFT LUBRICATION)


/BLACK KNIGHT ROCKET/*ENVIRONMENTAL TESTING/*ROCKET ENGINE/SPACE/TESTING/TURBOPUMP/VIBRATION/

THE INFLUENCE OF SPACE ENVIRONMENT ON THE DESIGN AND DEVELOPMENT OF ROCKET ENGINES FOR SPACE APPLICATIONS.

(ENVIRONMENT INFLUENCE ON DESIGN AND DEVELOPMENT OF BLACK KNIGHT ROCKET ENGINE)

A/SUNLEY, H. L. G. (AA/BRISTOL SIDDELEY ENGINES, LTD., LONDON, ENGLAND/.)


/SOLID LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/*CRY/ELEMENT/ENVIRONMENT/*EQUIPMENT/*FRICTION/*LUBRICANT/*LUBRICATION/*MATERIAL/*PARTICLE/RELIABILITY/*SOLID/*SPACE/*SURFACE/*TRANSFER/
THE INFLUENCE OF ENVIRONMENT ON THE DESIGN AND DEVELOPMENT OF ROCKET ENGINES FOR SPACE APPLICATIONS.

(AENVIRONMENT INFLUENCE ON DESIGN AND DEVELOPMENT OF BLACK KNIGHT ROCKET ENGINE)

A/SUNLEY, H. L. G. (AA/BRISTOL SIDDELEY ENGINES, LTD., ROCKET DEPT., COVENTRY, WARWICKSHIRE, ENGLAND/.)

IN SOCIETY OF ENVIRONMENTAL ENGINEERS, SYMPOSIUM ON ENVIRONMENTAL ENGINEERING AND ITS ROLE IN SOCIETY, IMPERIAL COLLEGE OF SCIENCE AND TECHNOLOGY, LONDON, ENGLAND, APRIL 19-21, 1966, VOLUME 5 - SPACE TECHNOLOGY 166-34208 1E-3C 16 P.

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ADVANCED AEROSPACE GREASES.

(GREASE LUBRICANTS FOR AEROSPACE APPLICATION, DETERMINING PHYSICAL PROPERTIES AND TESTING THEM AT 400 DEGREES F AND UNDER HIGH VACUUM)

A/BUNTING, K. R.; B/CHRISTIAN, J. B. (AB/USAF, WRIGHT-PATTERSON AFB, OHIO/ AA/AMERICAN OIL CO., CHICAGO, ILL./.) MEMBERS, $0.60, NONMEMBERS, $1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 21ST, PITTSBURGH, PA., MAY 2-5, 1966, PAPER 66AM 3C2. 16 P.

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LUBRICANTS FOR THE SPACE ENVIRONMENT.

(LUBRICANTS FOR SPACE ENVIRONMENT, NOTING EXPERIENCES WITH SATELLITES AND APPLICATION TO BALL BEARINGS AND ELECTRICAL CONTACTS)

A/CLAUS, F. J. (AA/Lockheed Aircraft Corp., Lockheed Missiles and Space Co., PALO ALTO, CALIF./.)


/*BALL BEARING/*ELECTRIC CONTACT/*SPACE ENVIRONMENTAL LUBRICATION/*SPACE MAINTENANCE/ BALL/ BEARING/ CONFERENCE/ CONTACT/ ELECTRIC/ ENVIRONMENT/ LUBRICATION/ MAINTENANCE/ SPACE
FLUORIDE SOLID LUBRICANTS FOR EXTREME TEMPERATURES AND CORROSIVE ENVIRONMENTS.

(Chemical and Thermal Stability of Fluoride Solid Lubricants Measured and Tested for Aerospace Environment)

A/Allen, G. P.; B/Slaney, H. E.; C/Strgm, T. N. (AA/NASA, Lewis Research Center, Cleveland, Ohio/)


/Nuclear Radiation/*Solid Lubricant/*Space Environmental Lubrication/ Air/ Calcium/ Chemical/ Coating/ Conference/ Environment/ Fluoride/ Friction/ Hydrogen/ Lubricant/ Lubrication/ Metal/ Sodium/ Solid/ Space/ Stability/ Thermal/ Wear

THE EFFECTS OF REACTOR RADIATION ON THREE HIGH-TEMPERATURE SOLID-FILM LUBRICANTS.

(Gamma Ray and Neutron Irradiation and Temperature Effects on Wear-Life of Solid Lubricants for Aerospace Application)

A/Mcdaniel, R. H. (AA/General Dynamics Corp., General Dynamics/Fort Worth, Nuclear Aerospace Research Facility, Fcrt Worth, Tex./)


/Nuclear Radiation/*Radiation Effect/*Solid Lubricant/*Space Environmental Lubrication/*Temperature Effect/ Calcium/ Conference/ Effect/ Environment/ Gamma/ Lead/ Life/ Lubricant/ Lubrication/ Nickel/ Nuclear/ Radiation/ Reactor/ Silicate/ Solid/ Space/ Steel/ Temperature/ Wear
MATERIAL REQUIREMENTS FOR SPACE APPLICATIONS.

A/HAUSNER, H. H.; B/WOLKOWITZ, W. (AB/BROOKLYN, POLYTECHNIC INST., BROOKLYN, N.Y., GRUMMAN AIRCRAFT ENGINEERING CORP., BETHPAGE, N.Y./.)


PROTECTIVE COATING/REFRACTORY MATERIAL/SOAK ENVIRONMENT/STRUCTURAL MATERIAL/AEROSPACE/COATING/CONFERENCE/ENVIRONMENT/FILM/FRICTION/HIGH TEMPERATURE/HIGH VACUUM/LIFE/LUBRICATION/MATERIAL/PROTECTION/REENTRY/REFRACTORY/SPACE/STRUCTURAL/TECHNOLOGY

ROTATING MACHINES FOR EXTREME ENVIRONMENTS.

A/IPANI, D.; B/SMITH, C. S. (AB/GARRETT CORP., AIRESEARCH ENG. CO., LOS ANGELES, CALIF./.)


BEARING/LUBRICATION SYSTEM/RADIATION EFFECT/ROTATING MACHINE/SOAK ENVIRONMENT/TEMPERATURE EFFECT/VACUUM EFFECT/CERAMICS/CONFERENCE/DESIGN/EFFECT/ELECTRIC/ENVIRONMENT/HEAT TRANSFER/INSULATION/LUBRICATION/MACHINE/RADIATION/RECTION/SPACE/SYSTEM/TEMPERATURE/VACUUM

FLUORIDE SOLID LUBRICANTS FOR EXTREME TEMPERATURES AND CORROSIVE ENVIRONMENTS.

A/ALLEN, G. P.; B/SLINEY, F. E.; C/STROM, T. N. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/) MEMBERS, $0.60; NONMEMBERS, $1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C5. 31 P. 11 REFS.

CALCIUM FLUORIDE/SOLID LUBRICANT/SPACE ENVIRONMENTAL LUBRICATION/AIR/CALCIUM/CHLORIC/COATING/CONFERENCE/ENVIRONMENT/FLUORIDE/FRICTION/HYDROGEN/LUBRICANT/LUBRICATION/MENTH/SODIUM/SOLID/SPACE/StABILITY/THERMAL/WEAR
THE EFFECTS OF REACTOR RADIATION ON THREE HIGH-TEMPERATURE SOLID-FILM LUBRICANTS.

(GAMMA RAY AND NEUTRON IRRADIATION AND TEMPERATURE EFFECTS ON WEAR-LIFE OF SOLID LUBRICANTS FOR AEROSPACE APPLICATION)

A/MDANIEL, R. H. (AA/GENERAL DYNAMICS CORP., NUCLEAR AEROSPACE RESEARCH FACILITY, FORT WORTH, TEX./.) MEMBERS, $0.60, NONMEMBERS, $1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C4. 30 P. 11 REFS.

NEW SOLID LUBRICANTS - PREPARATION, PROPERTIES AND POTENTIALS.

(PHYSICAL AND CHEMICAL PROPERTIES OF DICHALCOGENIDES OF GROUP VB AND VIB METALS FOR USE AS AEROSPACE SOLID LUBRICANTS, PRIMARILY FRICITION AND ANTIWEAR CHARACTERISTICS)

A/BOES, D. J. (AA/WESTINGHOUSE ELECTRIC CORP., RESEARCH LAB., PITTSBURGH, PA./) MEMBERS, $0.60, NONMEMBERS, $1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C3. 26 P. 11 REFS.

LOW-LEAKAGE DYNAMIC SEAL TO SPACE.

(HIGH SPEED LOW LEAKAGE SEAL-TO-SPACE USING SLINGER, VISCO AND MOLECULAR PUMP SEALS IN MERCURY RANKINE CYCLE)

A/HODGSON, J. N.; B/LESSLEY, R. L. (AA/AEROJE1-GENERAL CORP., SNAP-8 DIV., AZUSA, CALIF./.) MEMBERS, $0.50, NONMEMBERS, $1.00.

LONG-DURATION LUBRICATION STUDIES IN SIMULATED SPACE VACUUM.

(A LONG DURATION TESTING OF SLIDER AND BALL BEARING LUBRICANTS IN SIMULATED SPACE ENVIRONMENT)

A/BROWN, R. D.; B/BURTON, R. A.; C/KU, P. M. (AC/SOUTHWEST RESEARCH INST., SAN ANTONIO, TEX.)

ASLE TRANSACTIONS, VOL. 7, JUL. 1964, P. 236-248. 13 REFS.

RESEARCH SPONSORED BY THE SOUTHWEST RESEARCH INST.

/*BALL BEARING*/LUBRICATION TESTING MACHINE/*SPACE ENVIRONMENTAL LUBRICATION/ BEARING/ CURATION/ ENVIRONMENT/ LONG/ SLIDING/ TESTING/ VACUUM

SOLID LUBRICATION OF GEARS AND BEARINGS IN A SPACE ENVIRONMENT.

(COMPOSITE SOLID LUBRICANTS FOR SPACE LUBRICATION OF LOADED GEARS AND BEARINGS, INCLUDING TECHNIQUES AND TEST EQUIPMENT)

A/BOWEN, P. H. (AA/WESTINGHOUSE ELECTRIC CORP., RESEARCH LABS., PITTSBURGH, PA.)

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FRICITION VARIATION OF PTFE AND MCS SUB 2 DURING THERMAL VACUUM EXPOSURE.

(FRICTION COEFFICIENTS OF POLYTETRAFLUOROETHYLENE AND MOLYBDENUM SULFIDE-GRAPHITE-SODIUM SILICATE COATINGS FOR SPACE LUBRICATION)

A/CRAIG, W. O., JR. (AA/GRUMMAN AIRCRAFT ENGINEERING CORP., MECHANICAL SYSTEMS SECTION, BETHPAGE, N.Y.)


FRICITION COEFFICIENT/*MOLYBDENUM SULFIDE*/SOLID LUBRICANT/*TEFLON*/COATING/*FILM*/GRAPHITE/*LUBRICATION*/SILICATE/ SPACE ENVIRONMENTAL LUBRICATION/THIN/VEHICLE
GREASE LUBRICANTS AND THEIR POTENTIAL IN AEROSPACE APPLICATIONS.
(GREASE LUBRICANTS FOR AEROSPACE VEHICLES AND THEIR SUPPORT EQUIPMENT)

A/SCHWENKER, H. S. (SCHWENKER, H. /USAF, SYSTEMS COMMAND, RESEARCH AND TECHNOLOGY DIV., AF MATERIALS LAB., FLUID AND LUBRICANT MATERIALS BRANCH, WRIGHT-PATTERSON AFB, OHIO/)


LUBRICANT BEHAVIOR IN HIGH VACUUM.
(HIGH VACUUM EFFECTS ON DRY FRICTION COEFFICIENT, LUBRICATED FRICTION COEFFICIENT AND LOAD CARRYING CAPACITY OF LUBRICANTS)

A/FOSTER, P. G.; B/REICHENBACH, G. S.; C/SHAW, R., JR. (AA/MASSACHUSETTS INST. OF TECH., SURFACE LAB., DEPT. OF MECHANICAL ENGINEERING, CAMBRIDGE, MASS. /)

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, LUBRICATION CONFERENCE, ROCHESTER, N.Y., OCT. 15-17, 1963. / ASLE TRANSACTIONS, VOL. 7, JAN. 1964, P. 82-87. DISCUSSION, D. H. BUCKLEY INST., SAN ANTONIO, TEX./, AND DOUGLAS GODFREY /CALIFORNIA RESEARCH CORP./, RICHMOND, CALIF./, P. 88, 89, AUTHORS' CLOSURE, P. 89, 90. 16 REFS.

LUBRICANTS AND MECHANICAL COMPONENTS OF LUBRICATION SYSTEMS FOR A SPACE ENVIRONMENT.
(SYSTEM DESIGN AND LUBRICATION MATERIALS FOR SPACECRAFT LUBRICATION)

A/BUCKLEY, D. H.; B/JOHNSON, R. L. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/)

AMERICAN SOCIETY OF LUBRICATION ENGINEERS,

SPACE ENVIRONMENTAL LUBRICATION/ DESIGN/ ENVIRONMENT/ LUBRICANT/ LUBRICATION/ MATERIAL/ SPACE/ SPACE VEHICLE/ SYSTEM
ENVIRONMENTAL CONDITIONS AND OPERATING CHARACTERISTICS OF SPACECRAFT LUBRICATION, NOTING SILICONE OILS AND GREASES

A. CLAUSS, F. J.


* LUBRICATING OIL / SILICON COMPOUND / SPACE ENVIRONMENTAL LUBRICATION / SPACECRAFT MECHANISM LUBRICATION / EFFECT / ENVIRONMENT / GREASE / HIGH VACUUM / LUBRICANT / LUBRICATION / OIL / SILICONE / SPACE / SPACECRAFT

(LUBRICATION WITH SILICONE OILS AND GREASES)

A. CLAUSS, F. J.


* LUBRICATING OIL / SILICON COMPOUND / SPACE ENVIRONMENTAL LUBRICATION / SPACECRAFT MECHANISM LUBRICATION / EFFECT / ENVIRONMENT / GREASE / HIGH VACUUM / LUBRICANT / LUBRICATION / OIL / SILICONE / SPACE / SPACECRAFT

MATERIALS FOR SPACE VACUUM. CLARENCE E. JAHNKE /RAYTHEON CO., MISSILE SYSTEMS DIVISION, BEDFORD, MASS./. SPACE/AERONAUTICS, VOL. 40, SEPT. 1963, P. 89-91. 17 REFS.

* MATERIALS SCIENCE / SPACE ENVIRONMENTAL LUBRICATION / SPACECRAFT CONSTRUCTION MATERIAL / APPLICATION / BEHAVIOR / CERAMICS / CRITERION / LUBRICATION / METAL / PLASTICS / PRESSURE / SELECTION / SPACE / SUBLIMATION / VACUUM / VAPOR

COEFFICIENT OF FRICTION UNDER SPACE ENVIRONMENT IN THE RANGER I SPACECRAFT

A. JAFFE, L. D.; B. MARTENS, H. E.; C. NAGLER, R. G.; D. RITTENHOUSE, J. B.


* FRICTION MEASUREMENT / RANGER I LUNAR PROBE / SPACE ENVIRONMENTAL LUBRICATION / COEFFICIENT / ENVIRONMENT / FRICTION / LUBRICATION / MEASUREMENT / METAL / SPACE

PAGE 3C (ITEMS 86-88 OF 138)
(HIGH-SPEED OPERATION OF MINIATURE BALL BEARINGS WITH METALLIC FILM LUBRICATION AND BEST-GEAR MATERIAL COMBINATIONS FOR VACUUM-SPACE OPERATION)

A/EVANS, H. E.; B/FEDERLINE, M. F.; C/FLATLEY, T. H.

AMERICAN ROCKET SOCIETY, NEW YORK.


BALL BEARING/METAL FILM/SPACE ENVIRONMENTAL LUBRICATION/AEROSPACE/BALL BEARING/COMPONENT/ENVIRONMENT/FILM/GEAR/LUBRICATION/Mechanical/Metal/Operation/Space/Vacuum

(Study of Providing Lubrication for Reducing Friction and Wear of Rubbing or Sliding Surfaces of Various Spacecraft Mechanisms)

A/CLAUSS, F. J.; B/COOKE, F. B.; C/DRAKE, S. P.; D/CHARA, C. F.


PROGRAM 461 RELIABILITY MATERIALS RESEARCH AND APPLICATION BEHAVIOR OF SLIP RING IN A SPACE ENVIRONMENT FINAL REPORT.

LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF.

RING/SPACE ENVIRONMENT SIMULATION/BRUSHES/GRAPHS (CHARTS)/LIFE (DURABILITY)/LUBRICATION/CSCILLCGRAPHS/RELIABILITY/SLIDING FRICTION/VACUUM VELOCITY
CURRENT EUROPEAN DEVELOPMENTS IN SCLAR PADDLE CRIVES
A/BENTALL, R. H.
EUROPEAN SPACE RESEARCH AND TECHNOLOGY CENTER, NACRWIJK (NETHERLANDS).
IN NASA, LANGLEY RES. CENTER THE 8TH AEROSPACE MECH. SYMP. P 49-58 (SEE N74-11667 02-31)
*MEECHANICAL DRIVES/*PADDELS/*SATELLITE-BEAKNE INSTRUMENTS/*SCLAR
GENERATORS/ AEROSPACE ENGINEERING/ EUROPE/ LUBRICANTS/ PERFORMANCE TESTS/ SPACE ENVIRONMENT SIMULATION

LUBRICANTS FOR USE IN THE SPACE ENVIRONMENT
(LUBRICANTS FOR USE IN SPACE ENVIRONMENT AT HIGH AND LOW TEMPERATURE AND ZERO GRAVITY - BIBLIOGRAPHIES) REPORT BIBLIOGRAPHY, NOV. 1960 - SEP. 1972
DEFENSE DOCUMENTATION CENTER, ALEXANDRIA, VA. AVAIL.NTIS
*AEROSPACE ENVIRONMENTS/*LUBRICANTS/ BIBLIOGRAPHIES/ HIGH TEMPERATURE/ LOW TEMPERATURE/ WEIGHTLESSNESS

(VAPORIZED OIL BALL BEARINGS FOR SATELLITE ANTENNAS)
A/BAUMGARTNER, G.; B/GRANZOW, M.; C/HOSTENKAMP, R.; D/WEHNHAAS, H.
DORNIER-WERKE G.M.B.H., FRIEDRICHSHAFEN (WEST GERMANY).
(ENTWICKLUNGSABTEILUNG RAUMFAHRT.) AVAIL.NTIS AVAIL-NTIS, ZLDI MUNICH- 12.90 DM
SPONSORED BY BUNDESMIN. FUER BILDUNG UND WISS.
*BALL BEARINGS/*LUBRICATING OILS/*SATELLITE ANTENNAS/*VAPORIZING/
EUROPEAN SPACE PROGRAMS/ EVAPORATION RATE/ OPTIMIZATION/ OUTGASSING/
SPACECRAFT LUBRICATION/ STORAGE TANKS/ TORQUE
71N31664# ISSUE 19 PAGE 3104 CATEGORY 15 8MW-FB-W-71-13
71/03/00 156 PAGES IN GERMAN IN GERMAN, ENGLISH SUMMARY
UNCLASSIFIED DOCUMENT
PROJECT HELIOS - TESTING OF ROTATING MECHANISMS UNDER SPACE
ENVIRONMENTAL CONDITIONS
(SPACE ENVIRONMENT SIMULATION TESTING OF ANTEFFA SPIN REDUCTION
BEARING AND MECHANICAL DRIVES FOR HELICS SOLAR PROBE)
A/BIRKHOLO, E.; B/GRANZOW, M.; C/GROSSNER, P.; D/HAUSER, G.;
E/TREFFEN, H.-R.
DORNIER-WERKE G.M.B.H., FRIEDRICHSHAFEN (WEST GERMANY).
AVAILABLE NTIS AVAILABLE NTIS, ZLDI MUNICH- 32,80 DM
BAD GODESBERG, WEST GER. BLINDES MIN. FUER BILDUNG UND WISS.
SPONSORED BY BUNDES MIN. FUER BILDUNG UND WISS.
/*MECHANICAL DRIVES/*SOLAR PRGBES/*SPACE ENVIRONMENT
SIMULATION/*SPACECRAFT ANTENNAS/*SPIN REDUCTION/ BALL BEARINGS/
ENCAPSULATING/ LUBRICATING OILS/ PARABOLIC ANTENNAS/ ROTATING BODIES/
SPACECRAFT LUBRICATION

71N27158# ISSUE 15 PAGE 2380 CATEGORY 7 ESTEC-1060/70/SL
70/11/00 79 PAGES UNCLASSIFIED DOCUMENT
DEFINITION STUDY OF AN ANTENNA DESPIN MECHANISM FINAL REPORT
(COMMUNICATION SATELLITE DIREC TICAL ANTENNA SPIN REDUCTION
MECHANISM)
DORNIER-WERKE G.M.B.H., FRIEDRICHSHAFEN (WEST GERMANY).
AVAILABLE NTIS AVAILABLE NTIS
/*COMMUNICATION SATELLITES/*DIRECTIONAL ANTENNAS/*SATELLITE
ANTENNAS/*SPIN REDUCTION/ BALL BEARINGS/ ERROR ANALYSIS/ RETAINING/
SHAFTS (MACHINE ELEMENTS)/ SPACECRAFT LUBRICATION/ STEPS/ SYNCHRONCUS
MOTORS/ TEMPERATURE CONTROL/ TORQUE MOTORS

71N27156# ISSUE 15 PAGE 2380 CATEGORY 7 HSO-TP-7213
ESTEC-942/70/AA 70/10/00 232 PAGES UNCLASSIFIED DOCUMENT
DESIGN REPORT TO ESRO FOR A DESPIN ASSEMBLY
(SATELLITE DIRECTIONAL ANTENNA SPIN REDUCTION MECHANISM DESIGN)
A/REES, T. (AACOMP.)
HAWKER SIDDELEY DYNAMICS, LTD., HATFIELD (ENGLAND). (SPACE CIV.)
AVAILABLE NTIS COPYRIGHT AVAILABLE NTIS
/*COST ESTIMATES/*DIRECTIONAL ANTENNAS/*SATELLITE ANTENNAS/*SPIN
REDUCTION/ BALL BEARINGS/ BRUSHES/ SHAFTS (MACHINE ELEMENTS)/
SPACECRAFT LUBRICATION/ STRUCTURAL DESIGN/ TORQUE MOTORS

71N25751# ISSUE 14 PAGE 2352 CATEGORY 33 TASS-70-Y-0731
ESTEC-C0/850/70 70/09/00 83 PAGES UNCLASSIFIED DOCUMENT
MECHANICAL AND STRUCTURAL STUDIES
(ACTUATORS AND BEARINGS FOR SATELLITE LOUVER THERMAL CONTROL ARRAY)
FIAT S.P.A., TURIN (ITALY). (DIV. AVIAZIONE.)
AVAILABLE NTIS AVAILABLE NTIS
/*BEARINGS/*LOUVERS/*SATELLITE TEMPERATURE/*TEMPERATURE CONTROL/
ACTUATORS/ BELLows/ BLINDS/ ELECTRONIC CONTROL/ SPACECRAFT LUBRICATION/
STRUCTURAL ANALYSIS
EFFECT OF OXIDE FILM ON THE FRICTION OF 52100 STEEL IN VACUUM
(COEFFICIENT OF FRICTION DROP IN STEEL IN HIGH VACUUM)
A/TAKAGI, R.
DAYTON UNIV. RESEARCH INST., OHIO. AVAIL. NTIS
/*COEFFICIENT OF FRICTION/*HIGH VACUUM/*STEELS/ BALL BEARINGS/
FRICTION/ LUBRICANTS/ METAL PLATES/ OXIDES/ STRAIN GAGES

ON LUBRICATION PROPERTIES OF MOLYBDENUM DISULFIDE
POLYTETRAFLUROETHYLENE DOUBLE LAYERS IN HIGH VACUUM
(MOLYBDENUM DISULFIDE AND POLYTETRAFLUROETHYLENE DOUBLE LAYERS FOR
COATING SHAFT BEARING SURFACES)
A/HELLWIG, W.; B/SPENGLER, G.
DEUTSCHE VERSUCHSANSTALT FUER LUFT- UND RAUMFAHRT, MUNICH (WEST
GERMANY). (INSTITUT FUER FLUGTRIB- UND SCHMIERSTOFFE.) AVAIL. NTIS
/*BEARINGS/*COATINGS/*LAMINATES/*LUBRICATION/*MOLYBDENUM/
DISULFIDES/*POLYTETRAFLUROETHYLENE/*SHAFTS (MACHINE ELEMENTS)/THIN
FILMS/ ACTIVATION/ COEFFICIENT OF FRICTION/ HIGH VACUUM/ WEIGHT
MEASUREMENT

DESIGN AND DEVELOPMENT OF A RADIAL LOAD BALL BEARING TEST SYSTEM AND
SOME PRELIMINARY RESULTS
(RADIAL LOAD BALL BEARING TEST SYSTEM WITH SPACE ENVIRONMENT
SIMULATION)
A/WALL, J. L.; B/WARD, B. W., JR.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION, GLEDDARC SPACE FLIGHT
CENTER, GREENBELT, MD. AVAIL. NTIS
/*BALL BEARINGS/*LOAD TESTING MACHINES/*LUBRICANT TESTS/*SHAFTS
(MACHINE ELEMENTS)/SPACE ENVIRONMENT SIMULATION/ ANGULAR VELOCITY/
TABLES (DATA)/ VACUUM SYSTEMS
LUBRICATION AND BEARING PROBLEMS IN THE VACUUM OF SPACE
(SUMMARY INFORMATION ON LUBRICANTS AND LUBRICATION FOR USE IN
AEROSPACE ENVIRONMENTS, AND SIMULATION OF SPACE ENVIRONMENTS IN VACUUM
CHAMBER)

A/DISSON, E. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
PRESENTED AT THE SYMP. ON FRICTION DAYS SPONSORED BY GAMI,
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LUBRICATION/*VACUUM CHAMBERS/* BEARINGS/* CRYSTAL STRUCTURE/* FRICTION/
FRICTION REDUCTION/* GALLIUM/* LUBRICANTS/* PRESSURE EFFECTS

PERFORMANCE AND ANALYSIS OF SEALS FOR INERTED LUBRICATION SYSTEMS OF
TURBINE ENGINES
(PERFORMANCE EVALUATION OF NITROGEN INERTED LUBRICATION SYSTEM WITH
CONTACT SEALS FOR BALL BEARING OF TURBINE ENGINES)

A/JOHNSON, R. L.; B/LOOMIS, W. R.; C/LUDIG, L. P.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
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ENGR.

/*BALL BEARINGS/*GAS LUBRICANTS/*LUBRICATION SYSTEMS/*PERFORMANCE
TESTS/*TURBINE ENGINES/* INERTIA/* NITROGEN/* SEALS (STOLLERS)/* SPACE
ENVIRONMENT SIMULATION

DRY FILM LUBRICATED BALL BEARINGS FOR GIMBALS OSCILLATING AT SMALL
ANGLES IN VACUUM
(OPRATIONAL PERFORMANCE TESTS ON DRY LUBRICATED BALL BEARINGS UNDER
SIMULATED GIMBAL OSCILLATION IN VACUUM)

A/DEMORST, K. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. MARSHALL SPACE FLIGHT
CENTER, HUNTSVILLE, ALA. AVAIL. NTIS
/*BALL BEARINGS/*GIMBALS/*PERFORMANCE TESTS/*SLID
LUBRICANTS/*SPACE ENVIRONMENT SIMULATION/*STABLE OSCILLATIONS/
LUBRICANT TESTS/ OPERATIONS RESEARCH
LUBRICATION AND BEARING PROBLEMS IN THE VACUUM OF SPACE (EXPERIMENTS AND ENGINEERING MEASURES DEALING WITH LUBRICATION AND BEARING PROBLEMS IN VACUUM OF SPACE)

A/BISSON, E. E.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH CENTER, CLEVELAND, OHIO. AVAIL.AT NTIS WASHINGTON PRESENTED AT GAMI "FRICTION DAYS," PARIS, 5-6 DEC. 1966

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67N36203** ISSUE 21 PAGE 3937 CATEGORY 31 NASA-CR-87990 JPL-TM-33-355 NAS7-100 67/08/15 197 PAGES UNCLASSIFIED DOCUMENT SECOND AEROSPACE MECHANISMS SYMPOSIUM (CONFERENCE PROCEEDINGS ON DESIGN AND PERFORMANCE OF AEROSPACE MECHANISMS AND MECHANICAL COMPONENTS IN SPACE ENVIRONMENT)

A/HERZL, G. G. (AAED.) JET PROPULSION LAB., CALIF. INST. OF TECH., PASADENA. AVAIL.ATIS SANTA CLARA, CALIF., 4-5 MAY 1967

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A/FRANK, R. G. (AAED.) GENERAL ELECTRIC CO., BURLINGTON, VT. (SPACE POWER AND PROPULSION SECTION.)

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(LUBRICATION, FRICTION AND WEAR CONCEPTS FOR HIGH VACUUM APPLICATIONS)
A. Buckley, D. H.; B. Johnson, R. L.
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A. Frank, R. G. (AAED.)
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REVIEW OF THE INFLUENCE OF SPACE ENVIRONMENT UPON VEHICLE COMPONENTS
ENVIRONMENTAL HAZARDS ENCOUNTERED IN SPACECRAFT FLIGHTS
A. Smelt, R.
Lockheed Missiles and Space Co., Sunnyvale, Calif. (SPACE PROGRAMS DIV.)
IN AGARC THE FLUID DYN. ASPECTS OF SPACE FLIGHT, VOL. 1 1966 P 141-165 165 REFS /SEE N67-13842 04-12/
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(EVALUATION OF PERFORMANCE OF VARIOUS LUBRICANTS ON BALL BEARINGS OPERATING IN SIMULATED SPACE ENVIRONMENT)
A/CLAUSS, F. J.
LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF.
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A/KURITZA, O. M.
IIT RESEARCH INST., CHICAGO, ILL. (TECHNOLOGY CENTER.)
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IN-FLIGHT TEST TO DETERMINE SPACE ENVIRONMENTAL EFFECTS ON FRICTION, WEAR, AND LUBRICATION OF MATERIALS
(IN-FLIGHT TEST TO DETERMINE SPACE ENVIRONMENTAL EFFECTS ON FRICTION, WEAR, AND LUBRICATION OF MATERIALS - PROPOSED MODULAR SYSTEM)
A/DEVINE, E. J.; B/EVANS, H. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. GCOARC SPACE FLIGHT CENTER, GREENBELT, MD. AVAIL.NTIS
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QUALIFICATION TEST
(ADAPTABILITY OF FABROID LOW FRICTION, SELF-LUBRICATING BEARING
MATERIAL FOR OUTER SPACE ENVIRONMENT APPLICATION)
A/WAHLBERG, A.
TRANSPORT DYNAMICS, INC., SANTA ANA, CALIF. (RESEARCH AND
DEVELOPMENT DIV.)

11 FEB. 1964	14 P

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GEARS FOR USE IN AEROSPACE ENVIRONMENTAL CHAMBERS, DECEMBER 7, 1963 -
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(DRY THIN-FILM LUBRICANTS AND SOFT-METAL LUBRICANTS APPLIED TO
BEARINGS AND GEARS FOR USE UNDER HEAVY LOADS AND SLOW SPEEDS IN SPACE
ENVIRONMENTS)
A/RODINGS, T. L.
ARO, INC., ARNOLD AIR FORCE STATION, TENN.
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(FRICTION COEFFICIENTS OF INORGANIC SOLID FILM LUBRICANTS FOR USE IN
SPACE ENVIRONMENTS)
A/GADDIS, D. I. B/HOPKINS, V.
MIDWEST RESEARCH INST., KANSAS CITY, MO. AVAIL. NTIS
<1963> 12 P PRESENTED AT 1963 USAF AEROSPACE FLUIDS AND
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A/WELTONS, F. W.
SKF INDUSTRIES, INC., PHILADELPHIA, PA.

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AEROSPACE LUBRICATION RESEARCH SURVEY
(SURVEY OF RESEARCH CONDUCTED ON PROBLEMS OF SPACE ENVIRONMENTAL LUBRICATION)

A/BROOKS, D. B.
DEFENSE DEPT., WASHINGTON, D.C. (OFFICE OF THE DIRECTOR OF DEFENSE RESEARCH AND ENGINEERING) AVAIL. NTIS

**BEARING**/**ENVIRONMENT**/**FUEL**/**GAS**/**HIGH ALTITUDE**/**LIQUID**/**LUBRICATION**/**OXIDATION**/**RESEARCH**/**SOLID**/**SPACE**/**SPACE ENVIRONMENTAL LUBRICATION**/**SURVEY**/**VACUUM**

SURVEY OF FRICTIONAL PROBLEMS IN SPACECRAFT FINAL REPORT
(FRICTIONAL PROBLEMS IN SPACECRAFT MECHANISMS CAUSED BY SPACE ENVIRONMENTS)

A/IRWIN, A. S.; B/JOHNSON, J. H.
MARLIN-ROCKWELL CORP., JAMESTOWN, N.Y. (RESEARCH AND DEVELOPMENT LABS.) AVAIL. NTIS

**FRICTION**/**SPACE ENVIRONMENT**/**SPACECRAFT MECHANISM**/**LUBRICATION**/**BEARING**/**ENVIRONMENT**/**GEAR**/**LUBRICATION**/**MECHANISM**/**SEAL**/**SPACE**/**SPACE ENVIRONMENTAL LUBRICATION**/**SPACECRAFT**
(Solid Lubricant for Bearings in High Vacuum Environment)
A/GADDIS, D. H.; B/HOLM, F. W.; C/HOPKINS, V.; D/HUBBELL, R. D.
Midwest Research Inst., Kansas City, MO. Available NTIS

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(Lubricants and Mechanical Components of Lubrication System for Space Environment)
A/BUCKLEY, D. H.; B/JOHNSON, R. L.
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A/SALMON, W. A.; B/SCAMMELL, H. (SCAMMELL, H. /LITTLE /ARTHUR D., INC. /ED.)
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DOCUMENT

ADAPTATION OF A MOS SUB 2 "IN SITU" PROCESS FOR LUBRICATING
SPACECRAFT MECHANICAL COMPONENTS

(ADAPTATION OF MOLYBDENUM SULFIDE IN SITU PROCESS FOR LUBRICATING
SPACECRAFT MECHANICAL COMPONENTS)

A/VEST, C. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. GDODARC SPACE FLIGHT
CENTER, GREENBELT, MD.

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DISULFIDE/ ELECTRODEPOSITION/ ENVIRONMENT/ FILM/ OXIDE/ SPACE/ SPACE
ENVIRONMENTAL LUBRICATION/ SPACECRAFT/ SUBSTRATE/ SURFACE

64N17565* ISSUE 9 CATEGORY 17 NASA-RP-146 64/00/CO 10 PAGES
UNCLASSIFIED DOCUMENT

MECHANISM OF LUBRICATION FOR SOLID CARBON MATERIALS IN VACUUM TO 10
TO MINUS 9 MILLIMETER OF MERCURY
(HIGH VACUUM LUBRICATION OF SOLID CARBON MATERIALS)

A/BUCKLEY, D. H.; B/JOHNSON, R. L.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO.

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METAL/ OXIDE/ SOLID/ VACUUM/ HEAT

63N22954# ISSUE 23 CATEGORY 17 AEDC-TDR-63-154 63/10/00 45
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SPACE ENVIRONMENT LUBRICATION REQUIREMENTS

A/RAE, W. F.; B/PINSON, J. D.
ARNOLD ENGINEERING DEVELOPMENT CENTER, ARNOLD AIR FORCE STATION,
TENN.

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TENN. LUBRICATION REQUIREMENTS FOR SPACE ENVIRONMENTS J. D. PINSON AND
W. F. MC RAE OCT. 1963 45P 16 REFS /AEDC-TDR-63-154/
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63N19464# ISSUE 19 CATEGORY 17 ASD-TDR-63-474 AF 33/657/-8240

(HIGH VACUUM AND HIGH TEMPERATURE ENVIRONMENT MAGNETIC BEARINGS)
A/EDGAR, R. F.; B/MC HUGH, J. C.; C/MERCHANT, E. H.; D/TONKS, L.
GENERAL ELECTRIC CO., SCHENECTADY, N.Y. (ADVANCED TECH. LABS.)

63N17683# ISSUE 16 CATEGORY 17 MTI-62TR34 AF 33/657/-8666

(COMPLEX BEARING AND/OR LUBRICATION SYSTEMS)
A/LEWIS, P.; B/MURRAY, S. F.; C/PETERSON, M. B.
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63N17014# ISSUE 15 CATEGORY 33 RM-3164-PR AD-276365 AF 49/638/-700 62/05/00 11 PAGES UNCLASSIFIED DOCUMENT

(ORIGIN OF FATIGUE CRACK AS SURFACE PHENOMENA)
A/SHANLEY, F. R.
RAND CORP., SANTA MONICA, CALIF.

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A/BUGINAS, S. J.

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63N10929 ISSUE 2 CATEGORY 17 62/00/00 12 PAGES UNCLASSIFIED DOCUMENT

(LUBRICATION IN SPACE ENVIRONMENTS)

A/ADAMCZAK, R. L. ; B/BENZING, R. J.; C/SCHWENKER, H. S.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS DIV., WRIGHT-PATTERSON AFB, OHIO. DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS DIV., WRIGHT-PATTERSON AFB, OHIO LUBRICATION IN SPACE ENVIRONMENTS R. L. ADAMCZAK, R. J. BENZING, AND H. SCHWENKER IN SOC. CF AEROSPACE MATER. AND PROCESS ENGR. NATIONAL SYMP. ON EFFECTS OF SPACE ENVIRONMENT ON MATERIALS, ST. LOUIS, MAY 7, 8, AND 9, 1962 12P /SEE N63-10912 02-01/

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62N12809## ISSUE 8 CATEGORY 18 JPL-TR-32-150

NASA-6 61/11/01 124 PAGES UNCLASSIFIED DOCUMENT

(MATERIALS BEHAVIOR IN SPACE ENVIRONMENT)

A/JAFFE, L. D.; B/Rittenhouse, J. B.

JET PROPULSION LAB., CALIF. INST. OF TECH., PASADENA.


CONTENT OF NASA LITERATURE SEARCHES

This NASA Literature Search is based on a computerized records search of the technical reports, journal articles, books, conference papers and other publications stored in the NASA scientific and technical information system. Most of these documents have been announced either in NASA's abstract journal Scientific and Technical Aerospace Reports (STAR) or in International Aerospace Abstracts (IAAA), an abstract journal published by the American Institute of Aeronautics and Astronautics (AIAA) under a NASA contract. No abstracts are included in the literature search.

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ATM CMG LUBRICATION SYSTEM
FLOW RATE TEST

Prepared by: C. Gurrisi

Approved by: R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
PURPOSE
To determine if the flow rate of the ATM CMG Dynamic Lubrication System is sufficient in the temperature range of 50°F to 80°F.

DISCUSSION
As part of the failure investigation for ATM CMG S/N P-5 which failed onboard Skylab, a test program was initiated at Bendix Guidance Systems Division to measure the lubricant flow rate from the control moment gyros inner gimbal and rotor assembly (IGRA) dynamic lubrication system in the temperature range of 50°F to 90°F. Lubrication system flow rate at 90°F was specified to be between .05 and .120 mg/hr with a variance of .030 mg/hr at 90°F, from one test run to the next run. It should be noted that all testing was performed at 8000 rpm, however, the system operates in the IGRA at 9100 rpm. Therefore all measured flow rates will have to be ratioed up by multiplying the measured flow rate at 8000 rpm by \( \left( \frac{9100}{8000} \right) ^2 \) to yield the flow rate at 9100 rpm.

Prior to any testing at low temperatures, three room temperature flow rates were taken on lubrication system S/N 15 to establish a firm baseline set of data. These runs resulted in flow rates of .048, .056 and .051 mg/hr at 8000 rpm and at temperatures of 79.5°F, 82.4°F, and 83°F, respectively. When normalized to 90°F, these flow rates become .075, .077 and .069 mg/hr, respectively, which are well within the specification of .05 to .120 mg/hr and also satisfy the .030 mg/hr allowed variation. When upgraded to 9100 rpm operation, the flow rates are .0975,
The flow rate test fixture with lubrication system S/N 15 was then set up in a standard cabinet cold chamber for testing in the 50°F to 60°F temperature range. The chamber was stabilized at a temperature of 60°F and then a 114 hour flow rate test was performed. At the completion temperature was 67°F. The chamber was then stabilized at 50°F and a 105 hour flow rate test was run, resulting in a lubrication system temperature of 56°F. See Table 1 for the results of these tests.

Once these two temperature exposures were completed, a flow rate test was performed at room temperature to verify the integrity of the lube system. This flow rate test required a normalized flow rate of .072 mg/hr at 8000 rpm. Since this flow rate agreed perfectly with the room temperature baseline tests, the system was not altered by the cold temperature exposures.

Flow rate tests were then repeated at chamber temperatures of 52°F and 45°F with resulting lube system temperatures of 58°F and 52.5°F, respectively, to verify the results of the first two flow rates at the colder temperatures. Correlation of the first set of cold temperature data to the second set of cold temperature exposures was excellent. At the completion of this second set of cold temperature exposures another room temperature flow rate was performed to again verify the integrity of the lubrication system. This verification flow rate also agreed well with the initial baseline tests.
To verify the normalization factors to obtain the flow rates at 90°F, a flow rate measurement was performed with a chamber temperature of 85°F and a lubrication system temperature of 88°F for 98.8 hours.

Table 1 presents all the collected data on the previous mentioned tests. This table presents the flow rates at each temperature, the normalized flow rate for each run to 90°F, and the flow rate for each test normalized to 90°F and 9100 rpm.

CONCLUSIONS
The test data compiled during this test program confirms that the lubricant (KG-80 oil) flow rate from the ATM CMG IGRA dynamic lubrication system does decrease with operation at lower operating temperatures. These tests show an ≈18.5% decrease in flow rate between 88°F and 52.5°F.

The compiled data for flow rate test runs between 50°F and 90°F at 8000 rpm have been plotted in Figure 1. Figure 1 also presents the theoretical curves for flow rates of .05 and .120 mg/hr at 90°F from 90°F to 50°F at 8000 rpm. Since these three plots present parallel straight lines the theoretical flow rate temperature compensations on all previous lube nut testing is valid.

Figure #2 is a plot of the test data for lube system S/N 15 plus the theoretical curves of temp vs flow rate for flow rates of .05 and .120 mg/hr at 90°F, adjusted to 9100 rpm. Again the three curves are parallel straight
lines which further validates the theoretical flow rate temperature compensation factors.
# LUBRICATION SYSTEM S/N 15 FLOW RATES

<table>
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<tr>
<th>CHAMBER TEMP °F</th>
<th>LUBE SYS TEMP °F</th>
<th>COLLECTOR DRY WT GRMS</th>
<th>COLLECTOR WET WT GRMS</th>
<th>QTY OF OIL FLOW MG</th>
<th>RUN TIME HRS</th>
<th>FLOW RATE AT NUT TEMP 8000 RPM MG/HR</th>
<th>FLOW RATE NORMALIZED TO 90°C 8000 RPM MG/HR</th>
<th>FLOW RATE ADJ TO 9100 RPM AT NUT TEMP, MG/HR</th>
<th>FLOW RATE NORMALIZED TO 90°C 9100 RPM MG/HR</th>
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<td>0.060</td>
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</tr>
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</table>
 TEMPERATURE VS FLOW RATE
 AT 3000 RPM

 FIGURE #1
ATM CMG THERMAL VACUUM TEST

Prepared by: J. Pona

Approved by: R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608
ATM CMG THERMAL VACUUM TEST

PURPOSE
To determine the effects of temperature on the drag torque of the CMG spin bearing.

BACKGROUND
The failure of CMG No. 5 (Skylab CMG 1) and the anomalies which occurred with CMG No. 6 (Skylab CMG 2) resulted in the investigation of the effects of temperature on the bearing torque of the CMG spin axis bearings. A life test fixture equipped with two normally lubricated bearings was placed in a thermal vacuum chamber and subjected to temperature changes from 100°F to 1550°F. Current, speed, temperature and torque were monitored throughout the test.

DISCUSSION
Life Test Fixture (BLTF) S/N #2 was equipped with two normally lubricated spin bearings. In order to maintain a fixed quantity of oil (KG-80) in the ball bearing, thus eliminating torque variations as a function of lube quantity, the active lubrication systems were not utilized during this test.

All testing was performed in a thermal vacuum chamber (NRC model 2004), see attached ETS 10152, with the unit securely mounted on a thermal platen. A thin layer of
thermal silicon grease was applied to the mating faces of the BLTF and the thermal platen in order to insure proper thermal conductivity. The unit was accelerated to operating speed (9100 RPM) at room temperature ($\approx 70^\circ\text{F}$) and hard vacuum ($\approx 10^{-7}$ torr).

A baseline set of data consisting of time, speed, temperature, vibration and bearing torque was taken. This resulted in an initial bearing torque of 1.62 oz-in. The chamber temperature was decreased to $0^\circ\text{F}$ and the unit was allowed to stabilize overnight. After stabilization, the bearing temperatures were $10^\circ\text{F}$ and $12^\circ\text{F}$ on sides 1 and 2 respectively; however, the unit was no longer operating (was at zero speed). Without removing power from the BLTF the chamber was gradually increased in temperature until wheel rotation was detected. Once rotating, acceleration to 9100 RPM occurred in the normal interval of time and a torque test indicated the torque was slightly lower than the baseline torque (1.47 oz-in as opposed to 1.62). The unit temperature was then decreased slowly from $65^\circ\text{F}$ in approximately $10^\circ\text{F}$ increments. Torque tests were taken at each plateau. At $24^\circ\text{F}$ the bearing drag torque exceeded the motor torque and the unit stalled.

The temperature was increased in $10^\circ\text{F}$ steps to $155^\circ\text{F}$ with torque tests taken after each stabilization. The final room temperature torque test resulted in a bearing torque of 1.3 oz-in.
Holding the unit at $10^{-7}$ torr, two 24 hour temperature cycles were performed. The cycles started at 72°F, was increased to 90°F, decreased to 50°F, and returned to 72°F. Torque tests were taken at the start and finish of both cycles showing ~1.30 oz-in of torque on all occasions.

These temperature cycles were performed in an attempt to produce any of the anomalies noted in CMG #2 onboard Skylab.

CONCLUSIONS

Examining the data and the enclosed graph, we see that the viscous drag increases with decreasing temperature caused by the change in viscosity of the lubricant. The exponential increase in torque below 30°F is due, not only to viscosity change but primarily to the preload change in the BLTF. This change in preload is a result of the fact that the life test fixture configuration does not allow the thru strut to control the bearing preload spring clearance as it does in an actual unit assembly. In the BLTF the aluminum housing with its greater coefficient of expansion and its inherently stiff cylindrical configuration closes down the preload spring clearance at colder temperatures.

From the attached graph we see that the torque decreased consistently from 2.63 oz-in at 30°F to .84 oz-in at 155°F. The variations in torques above 30°F are caused
solely by the viscosity change in the oil. The two temperature cycles between 90°F and 50°F showed no variations of current, torque or speed.
ATM CMG BEARING

DRAG TORQUE VS. TEMPERATURE

NOTE: 20°F - POUR POINT OF KG-80, OIL
<table>
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<tr>
<th>HOURS</th>
<th>1/2 SPEED RPM</th>
<th>TORQUE OZ-IN</th>
<th>BEARING TEMP.</th>
<th>VIBRATION</th>
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</table>
To evaluate the effect of hard vacuum and temperature on the CMG Spin Bearings - Drag Torque Test.

The specimen was positioned on the thermal platen inside the chamber utilizing an application of thermal silicon grease. The specimen was interfaced to its test stand for functional data generated during testing. A thermocouple junction was located on the platen and was used to control the temperature of the platen. Additional thermocouples were attached to the platen in such a manner as to monitor temperature distribution along the platen surface (RE: Fig. 1).

Returned to the Engineering Project Group.
TESTING PERFORMED AND RESULTS:

The specimen was subjected to reduced pressures in the $5 \times 10^{-7}$ torr range. During this condition, various temperature test points were accomplished between $0^\circ F$ and $160^\circ F$. During these parameters, the specimen was energized and functional data generated is in the possession of the Engineering Project Group.

At the completion of this portion of testing, the specimen was subjected to two 24 hour pressure/temperature cycles as illustrated in Table I. Data generated during these tests was retained by the Engineering Project Group.

At the completion of all testing, the specimen was visually examined and no deterioration was observed. No malfunctions were reported by the Project Group.

Thermal Vacuum Log Sheets are on file in the Thermal Vacuum Laboratory.
TESTING PERFORMED AND RESULTS:

TEST SETUP

CHAMBER

SPECIMEN

THERMAL PLATEN

TC 1
(ALSO O2 CONTROLLER)

TC 2

TC 3

TC 4 (FAR SIDE)

SILICON INSUL-GREASE

FIGURE 1
SIMULATION OF SKYLAB CMG 2 DATA

MINIMAL BEARING LUBRICATION

Prepared by: A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
1.0 PURPOSE
The purpose of this report was to document the results of tests using minimally lubricated bearings in an attempt to simulate current and speed anomalies similar to those that occurred to the Skylab CMG #1 (S/N 5) and CMG #2 (S/N 6) during the Skylab mission.

2.0 BACKGROUND
The primary mechanisms used to orient and maneuver the Skylab space laboratory were three Control Moment Gyros manufactured at the Bendix Guidance Systems Division. These CMG's used two 107 H size angular contact ball bearings, each had a wheel weight of 150 pounds, ran at a speed of 9100 RPM and developed an angular momentum of 2300 ft-lb-sec. The CMG's were tested both at Bendix and at NASA Huntsville for approximately 1500 hours prior to their launch in the Skylab space laboratory on May 14, 1973.

On November 23, 1973, after 195 days of continuous operation in space, ATM CMG S/N 5 (Skylab #1) experience what appeared to be a bearing failure and ceased to function. During the remaining 75 days of the mission, close monitoring of the speed, current and temperature data from the remaining two functioning ATM-CMG's revealed periods of apparent unstable behavior in ATM-CMG S/N 6 (Skylab #2). These anomalies were in the form of slight speed, current, and bearing temperature changes from normal,
but did not affect the functioning of the gyro. However, the trend of the data indicated a possible deterioration in the performance of one of the two bearings paired in this gyro. None of these anomalies were noted in the data obtained from ATM CMG S/N 7, (Skylab #3). Both of these remaining CMG's completed the mission of approximately 6500 hours in space and were run down after the egress of the Skylab III astronauts on February 8, 1974.

The speed, current and temperature changes noted were known as "gliches" and lasted anywhere from several minutes to several hours. "Gliches" which remained constant in both current and speed for most of its duration were referred to as "plateaus". Some plateaus were of several hours duration. At no period, however, did the wheel speed decrease more than 120 RPM below the nominal speed (not necessarily 9100 RPM). This was possibly due to telemetering and instrumentation variations as well as data collection location differences. There were also other periods of "un-gliches" and "un-plateaus" where the speed appeared slightly higher than nominal and the current was correspondingly lower.

Other factors such as temperature cycling, gravity gradient desaturations, and maneuvers which may have directly or indirectly affected the initiation or termination of an anomaly period. A change in the bearing cartridge heater temperature cycle during the last seven weeks of the mission led to a prolonged
period, (four and one-half days) during which no anomalies were observed. During the first 32 weeks of the mission the gyro temperature of CMG S/N 6 (Skylab #2) cycled between +60°F and +80°F. During the last seven weeks the cycle was +72°F to +78°F, thereby increasing the lube nut (oil replacement system) flow rate, which increased the quantity of oil to each bearing and lowered the viscosity of the oil. The effect of this change was to decrease the frequency of occurrence of the "gliches" at least temporarily. It was during this later period of increased average temperature that several periods of "ungliches" occurred.

A review and investigation of this data by a committee of personnel from various NASA centers, Bendix engineering, as well as independent consultants led to the conclusion that primary factor for the occurrence of these anomalies was most likely lack of adequate lubrication in the #2 bearing on CMG #6. (Ref. 40M23157 NASA-MSFC). Therefore the tests in this study were conducted utilizing bearings with known degrees of marginal lubrication in an attempt to simulate the temperature, speed and current anomalies experienced by CMG S/N 6 (Skylab #2) and CMG S/N 5 (Skylab #1).
3.0 DISCUSSION

Three tests were conducted under the following conditions of bearing marginal lubrication.

a) Test #1

   Dry retainer
   Dry bearing metal parts

b) Test #2

   Dry retainer
   500-1 solution of freon to KG-80 oil on the bearing metal parts

c) Test #3

   Dry retainer
   50-1 solution of freon to KG-80 oil on the bearing metal parts.

All of these tests were performed in the spin axis vertical position with the test bearing at the bottom. The tests were conducted in this attitude to simulate a worst case condition where the oil from the lube nut must migrate or be pumped against 1 "g" to relubricate the minimally lubricated test bearings. It is assumed that in a zero "g" space environment, the dispersion and migration of the oil will be more random than in a 1 "g" field. Assuming similar temperature gradients,
the space environment would allow quantities in excess of these tests to enter the bearing race contact area. Therefore, if sufficient oil migrates against 1 "g" to replenish the minimally lubricated test bearings in these tests, then there is greater probability that the Skylab bearings in question were adequately relubricated. If the test bearings do not recover sufficient oil, there is more reason to doubt the sufficiency of the rate of oil migration into the bearing race of the Skylab bearings.

The bearing used was Barden S/N BB18, a bearing that had been previously run on life tests for approximately 1000 hours, and which exhibited the normal metal tracking associated with this amount of run-in time.

In each test the bearing metal parts were cleaned per MT-13,923 after removal of whatever residues and debris were left from previous tests.

The bearing retainers used in each test were completely devoid of oil to prevent supplementary lubrication of the bearings. This was accomplished by extraction in freon for approximately 16 hours and vacuum baked for twelve hours prior to assembly into the test bearing. The test bearing was then assembled along with a normally lubricated bearing into a life test fixture and preloaded to the normal 40# thrust load as in the Skylab ATM-CMG's. The life test fixture applies
no radial load other than the weight of the shaft and so simulates to a certain extent the weightlessness of the ATM-CMG wheel in space.

The life test fixture is driven by one standard ATM-CMG motor developing approximately nine oz-in. of torque at 455 Hz and 130 volts. Synchronous speed is approximately 9100 RPM. The normal torque of bearing paired in a life test fixture is 1.5–2.0 oz-in. Since there is very little inertia for the motor to contend with in the life test fixture, small torque changes will be reflected in pronounced speed and current changes. The inertia of the life test fixture shaft is .045 in. lb. sec\(^2\) compared with 29.0 in. lb. sec\(^2\) for an ATM-CMG.

Various test equipment was set up to monitor these changes. (See Photo 1). Visual readout of speed, motor current, vibration levels of both bearings and the temperatures of the test bearing was provided. In addition, a six channel recorder was continuously monitoring AΦ motor current, wheel speed (actually 1/2 speed), test bearing temperature, and the vibration levels of both bearings. However no meaningful data was recorded during test number 1 since the test bearing "failed" within three minutes. The speed traces recorded during tests 2 and 3 utilized an automatic reset device. During test number 2 the pen reset every ten counts, or 20 RPM, which made for very accurate recording of small speed changes. However, larger speed changes were very
difficult to interpret without visually reading the counter. During test number 3 the setting was changed to reset at 100 counts or 200 RPM intervals. Sudden speed changes in excess of 200 counts were difficult to interpret on the tape and many speed readings taken during "failure" and "recovery" periods were visual readings from the speed counter.

4.0 RESULTS

Test #1 - The completely dry bearing of Test #1 ran for approximately three minutes, including a run up time of approximately 45 seconds. The bearing torque then exceeded the motor torque and the speed dropped to zero. The period of run was too short to record any meaningful data.

Visual examination of the bearing metal parts and retainer revealed that the increase in torque was caused by the deposition of residues and debris from the bearing retainer pockets onto the bearing races and balls. This debris and resinous residues deposited on the races and balls was generated by the friction and localized overheating caused by the rub of the balls in the lead and lag segments of the unlubricated ball pocket surfaces.

The bearing metal parts showed no wear and no evidence of spalling or oxidation after removal of the debris and residues. The retainer was ultrasonically recleaned in
freon and except for the burnishing in the ball pockets which occurred during the test, there was no other physical damage. The retainer was then vacuum baked for 16 hours at 180°F.

Test #2 was conducted using the same bearing and retainer as in Test #1. In this test the metal parts were dipped into a 500-1 mixture of freon to KG-80 oil, leaving a very thin film of oil on the metal parts after evaporation of the freon. The bearing was then assembled with the cleaned and dried retainer and replaced into the life test fixture.

Test #2 ran for approximately eleven minutes before anomalies in the traces were noted. At this point in the test, calibration of the test equipment was still in progress, and direct correlations could not be made accurately. However, for the next several minutes there were periods of audible retainer squeal, accompanied by speed and current changes.

The remainder of the 3-1/2 hour run illustrated a variety of anomalies (see Figure 1). There were cases of sharp changes in vibration level both with and without changes in speed or current. There were periods of change in speed and current both with and without changes in vibration level. The period of audible bearing, or retainer noise, always corresponded to changes in vibration level but not always to speed and current changes.
The point marked A on the trace (Figure 1) was an example of audible retainer noise with a corresponding significant current increase and speed decrease. Point B was another instance of audible retainer noise but without a corresponding current increase or speed decrease. The area of the trace between the letters C contained points where the vibration levels increased and decreased sharply accompanied by drastic speed and current changes. Speed changes up to 1000 RPM were noted and current changes of over 600 ma were recorded. Area D indicated a more moderate increase in vibration level with corresponding 200-300 ma. increases in current and 150-700 RPM speed decreases.

After the second area D there was a general recovery of the bearing after which there was a gradual deterioration indicated by a steady increase in overall vibration level, motor current, and a decrease in speed. There was also a gradual, then accelerated increase in bearing temperature to +84°F from +73°F. During this one hour segment there were periods of deterioration and recovery involving changes of 100-300 ma. and several hundred RPM.

At approximately 3:20 PM the bearing stopped momentarily then ran at approximately 4450 RPM (2225 counts x 2) for several minutes. The speed then increased to approximately 9050 RPM for a minute, dropped to 2500 RPM, recovered, then stopped and the test was terminated.
The first sustained "plateau" occurs at the first "D" area. The current rose from 510 ma to 610 ma and then to 800 ma. The current then dropped to about 650 ma occurring with a corresponding decrease in the vibration level. The plateau then remained at approximately 700 ma for the balance of this period. No audible retainer noise was noted during this period. The probable cause for the vibration level and torque increase was deposition of resinous residues from the retainer ball pockets into the ball path. This plateau is several times the magnitudes noted with Skylab CMG #2 (S/N 6). This plateau may be compared with the plateau of DOY 323:16 - 323:10, Figure 2. In both cases there is an increase in current and a corresponding increase in temperature in both the test bearing and bearing #2 in the CMG. However, the decrease in speed of 660 RPM during this test period represents a torque increase of 7-8 oz-in., while the apparent change in the speed of ATM CMG #2 was in the neighborhood of 30-40 RPM or about 1 oz-in. It may also be noted that the temperature modes are dissimilar. The test bearing was approaching "failure" as the temperature was still ramping higher after the end of the "plateau". The temperature of CMG S/N 2, bearing #2 returned gradually to its former level (measured as a $\Delta$ from bearing #1).

After termination of test #2 the test bearing was removed from the life test fixture. The bearing was disassembled and the bearing metal parts and retainer
visually examined. Examination indicated that the primary cause for the torque increase was the build-up of debris and residues on the bearing balls and races. There was more defined wear in the bearing retainer ball pockets which also bore evidence of retainer instability.

The retainer flange onto which oil is centrifuged from the lube nut contained no visible evidence of oil. Whatever oil had been centrifuged onto the flange had been in turn centrifuged thru the feed holes and onto the bearing outer race land.

There was also evidence of wear on the retainer scallop pads on the pressure side of the inner race. There was evidence of burnishing on the pressure side of the inner race land (see Figure 3) indicating that the 500-1 solution (freon to oil) is insufficient to adequately lubricate the retainer pads and the area where the retainer pads contact the inner race land.

The ball pockets had considerable wear in both the lead and lag portions, and there was evidence of burnishing on the sides of the pockets, perpendicular to the direction of motion, indicating periods of retainer instability.

The bearing races contained loose retainer debris, ground in resinous residue, and minute amounts of metal wear products.
The bearing race contact areas when cleaned showed no evidence of metal distress; i.e., spalling, oxidation. The contact wear areas appeared slightly wider than prior to the test run. The balls exhibited an additional set of fairly wide wear bands caused by running in a preferential orientation for a period of time during the test. However there was no apparent metal deterioration.

The area of the outer race which was covered by fine loose debris from the retainer pockets extends from a line on the outer race land just inboard from the edge of the retainer, upward against gravity, through the outer race groove to a line at approximately a 45° angle from the opposite edge of the retainer (see Figure 3). There also is evidence of a fine film of oil on the bearing outer race below the contact zone as well as a thicker film on the lower outer race land. It is apparent that oil and some debris was pumped against gravity in a direction from the lube nut side of the bearing through the bearing outer race and out to the shallow land of the outer race (see Figure 3).

Test #3 was conducted using the same bearing as in the initial two tests. The retainer was replaced with another retainer which was soxhlet extracted in freon to remove residual oil and was then vacuum baked as stated previously. The bearing metal parts were then cleaned as prior to test #2. The bearing metal parts
were dipped into a 50-1 mixture of freon and oil. The bearing metal parts and dry retainer were then assembled and replaced into the life test fixture.

Test #3 covered a period of approximately eight hours. The test was characterized by anomalies similar to that of test #2 and somewhat similar to some data from Skylab CMG #2. During the early hours of the run, while there was oil for adequate lubrication, there were no sudden changes in speed, current, or vibration level. During the middle hours of the test, variations in speed, current and vibration level occurred, but the changes in vibration level in most cases did not occur simultaneously with the speed and current changes. There were current and speed plateaus as in the data received from Skylab CMG #2, as well as ramps and recoveries. During the initial hours of the test there was no change in the bearing temperature of 73.5° F. The initial six hour period was characterized by plateaus and ripples in the current and speed traces of 10-30 ma and 60-80 RPM, although there are some instances of 50 ma current and 120 RPM speed changes.

The last two hours are characterized by more severe current changes of up to 200 ma and 800 RPM changes in speed. There were three severe speed decreases to approximately 1000 RPM with current levels reaching 700 to 1150 ma. In each case the recovery in speed was complete. The test then was stopped overnight.
The half hour run, during the following day, was similar to the last hour and a half of the previous run. However, audible retainer noise was noted during this period, and the resultant increases in vibration level were marked on the trace. The audible noise was consistent with speed and current fluctuations.

Area A occurred approximately one hour and twenty-five minutes into the test and is indicative of the initial hour and forty-five minutes of the run (see Figure 4). The current is 495 ma up 5 ma from the initial steady state reading. The speed is 9004 down from the initial 9036 RPM. The vibration level on the minimally lubricated (test bearing) bearing was steady at 60-80 mv. During this period of time, the lube end was sufficient to separate the balls and races and there were no anomalies present.

Area B (Figure 5) on the trace occurred approximately two hours and thirty minutes into the test and includes anomalies. At the beginning of this period, the bearing indicated the first "glich". The speed which was stable at 9000-9008 RPM drifted down to 8992 RPM and then dropped to 8947 RPM (point 1), while the current correspondingly increased from 495 ma to 515 ma. The speed then recovered slightly to about 8966 RPM then dropped to 8940 RPM with corresponding changes in current. There then occurred momentarily a complete recovery to 9022 RPM (point 2) and 480 ma.
Points 3 and 4 were the first accuracies of simultaneous speed, current and vibration anomalies. There was no apparent change in test bearing temperature. These anomalies both cover a ten second period. Point 4 indicated a speed drop of 63 RPM from 8979 to 8908 RPM then a recovery to 8990 RPM. The current rose from 505 ma to 530 ma then decayed to 495 ma. These anomalies were the first indication of any significant bearing torque changes.

Area C - (see Figure 6) on the trace began about fifteen minutes after area B or about 2 hours and forty minutes into the test. There was a sharp sustained increase in vibration level from an average of 70 mv to an average of 100 mv. There are vibration peaks to 260 mv. (100 mv/g.) During this period there are examples of "gliches" in both the speed and current traces. The current ranged between 470 ma to 550 ma and the speed range varied between 9058 RPM and 8768 RPM. These anomalies recurred for approximately 30 minutes.

This period began abruptly with the sharp vibration level increase. The speed dropped from 7994 RPM to 8931 and then to 8900 RPM (point 1) with a corresponding increase in current. There was also a small temperature increase of approximately 0.2° - 0.3°F over a five minute period attributed to the initial torque increase of this period. Although the magnitude of the changes are different, as well as the time periods, the series
of "gliches: in area C are somewhat similar to the "gliches" of DOY 323:16:45 to 323:18:15 (discussed later in this report).

Approximately 3-1/2 hours into the test, area D (see Figure 7) exhibited the first series of "failures" and "recoveries". The current level during the first "failure" reached 780 ma from 485 ma. The speed decreases during this period were too rapid to be recorded. The increase in vibration level was probably due to resonances in the system at lower speeds and did not precipitate the torque increases. The second "failure" was even more severe with the current level reaching 830 ma. In both cases there was a "recovery". The recovery from the first "failure" was back to 500 ma and approximately 8960 RPM. Then follows a milder speed decrease to 4425 counts or 8850 RPM from 8960 RPM, point 2. The second recovery was not as complete and was followed by another speed decrease which was not recorded. At the end of period D, the speed and current "gliches" began to smooth out and were relatively stable for approximately the next two hours and fifteen minutes.

The "failures" and "recoveries" in this area were similar to those at the end of test #2 and were typical of the failure/recovery modes of bearings on the verge of failure.
Area E (see Figure 8) occurred approximately four hours and forty minutes into the test and was indicative of the general recovery of the test bearing. This recovery period spanned approximately two and one-half hours and was punctuated by small "gliches" of similar magnitude and duration as in area E. These "gliches" were on the order of 20-55 RPM and 5-20 ma. The duration was usually 2-3 seconds. There were no temperature changes and during this recovery period the vibration level of the test bearing returned to previous levels of approximately 30-100 mv.

After approximately six hours into the test, the "gliches" became more frequent and for the next hour increased in magnitude and duration.

Area F occurred approximately seven hours into the test (see Figure 9) and followed a series of "gliches" of lesser magnitude and duration. The speed decreased from about 8950 RPM to 2148 RPM while the current increased from 520 ma to 1180 ma. During this period speed readings were impossible to determine from the trace and were read from the counter and written onto the trace. This "glich" lasted about six minutes after which the speed and current became fairly stable and the vibration level dropped to normal levels and continued so for another half hour. The test was then shut down until the following day.
Area G (see Figure 10) occurred after about twenty minutes into the next day's run and about ten minutes prior to termination of the test. Interesting points (1) and (2) on this trace were two of several instances of audible retainer noise (instability) corresponding with significant speed and current "gliches". At point (1) the speed dropped from approximately 8970 RPM to 4462 RPM. The current change was from 540 ma to 800 ma. At point (2) the speed reading was not recorded while the current peaked to 800 ma.

The bearing was then removed from the life test fixture and disassembled for examination. Visual examination of the bearing metal parts and retainer indicated greater wear than generated after test #2. There was a similar pattern of debris and oil being pumped upward in a direction from the pressure side land of the outer race thru to the shallow land of the outer race. There was a greater amount of oil on the pressure side land than in test #2, with much of the oil from the lube nut migrating downward and away from the race. However, there was oil visible on the upper area of the outer race which had been pumped into the race.

The bearing metal parts showed additional wear, more so than in test #2, with evidence of some metal to metal contact between the balls and both inner and outer races. There was some minor spalling in the contact areas of each race. The balls also showed some minor spalling.
The bearing inner race land did not indicate any polishing by the retainer pads nor do the pads show any wear. This indicated that the 50-1 solution provided a film sufficient for lubrication in this area.

There was a noticeable increase in the film of oil on the outer raceland, supplied by the lube nut. The majority of this oil migrated downward away from the bearing race. There was, however, a film of oil extending to the race edge and into the race. While no quantitative measurement of the thickness of this film or the quantity of oil could be made, it was by visual comparison at least several times the film left by the 50-1 mixture.

The retainer ball pockets were worn on the lead and lag segments to a greater extent than in test #2, and there was greater burnishing on the pocket sides. This indicated that this retainer was unstable for greater periods of time, and accounts for the correlation between audible retainer instability and the jogs in the vibration, current, and speed traces.

The inertia of the life test fixture shaft assembly is .72 oz-in-sec$^2$ or .045 in-lb-sec$^2$ (ref. MT-13,936), while the inertia of the IGRA is approximately 29.9 in-lb-sec$^2$ (ref. 2120252-9). The ratio of the inertia is 29.9/.045 = 650. Therefore, assuming similar bearing drag torques for both the test bearing and the
bearings in the Skylab ATM CMG's, the effects of a change in torque in a Skylab CMG bearing would occur approximately 650 times slower than the speed and current reactions for the same torque change in the test bearings assuming that the speed of both is in the linear portion of the torque/speed curve (8700-9100 RPM (ref. MT-15,800 and Figure 12).

The data from Skylab DOY 323:16 - 323:19 shows a fairly typical "glich" (see Figure 11). According to data received, the speed change during this period was from 8911 RPM to 8870 RPM, a change of 41 RPM. The torque change was approximately 1.2 oz-in from the above curve (see Figure 12). The speed change occurs over a period of approximately 20 minutes or 1200 seconds. A similar torque change in the life test fixture would occur within $1200/650 \approx 2$ seconds. The "glich" noted in area B of test #3 (Figure 4, point 4) starts with a drop in speed from 8971 to 8908, a drop of 63 RPM and a torque change of 1.1 oz-in in a period of four or five seconds. Similarly, the plateau which began at point (1) of area C in test #3 began with a speed drop from 8994 RPM to 8931 RPM (63 RPM) and occurred in six seconds.

During DOY 349:11:15 - 349:49:14:15 (see Figure 13) the speed drops from 8910 to 8850 RPM in about 31 minutes, a torque increase of approximately 1.6 oz-in. This would have corresponded to 100 RPM changes within 3 seconds for the test bearings. Instances of changes of this magnitude and rapidity occurred during the last two hours of the test. One such is illustrated (see Figure 10) in area G during the second instance of audible noise.
FIGURE 11
5.0 CONCLUSIONS

In the three tests, the primary cause for "failure" was the buildup of debris and residues from localized overheating in the unlubricated ball bearing retainer pockets due to ball rub.

The location of debris, residues and small quantities of oil in the outer race indicates that some oil is pumped or migrates against 1 g to the bearing contact area. However, with the standard retainer used in this test, the deposition of replacement oil from the lube nut was onto the outer race land and did not migrate against 1 "g" adequately to relubricate the test bearing.

The condition of the test retainers scalloped pads after each test indicates that the 50-1 mixture of freon to oil leaves the minimum oil necessary to adequately lubricate the inner race land/scallop pad interface.

In the tests, audible retainer noise did not always occur in correlation with, or cause a change in torque level (wheel speed). However, there are instances in both tests #2 and #3 where audible retainer instability is a cause for the torque level change in the test bearing. In the tests, instances of audible retainer noise did not produce "glick" plateaus of fairly constant speed and current. They generally produced current spikes and irregular speeds during the audibly unstable periods. However, in the actual CMG's with their larger inertias these changes would probably have smoothed out.
There were external factors occurring to the Skylab CMG's such as thermal cycling, gravity gradient momentum desaturations, etc. which could not be duplicated in these tests. Therefore direct correlations between the "gliches" seen in the telemetered data from Skylab ATM-CMG's 1 and 2 and the data taken during these simulation tests are difficult. There appeared to be some similarities in the data, particularly during the first half of test #3, where torque level changes were of similar magnitudes, although reactions in speed were not quite as fast as the 650/1 ratio of inertias (ATM-CMG wheel/test fixture shaft). The similarity in some of the test data to some of the telemetered data from Skylab does support to some extent the theory that minimal lubrication was the primary cause for the anomalies noted.
LIFE TEST OF NORMALLY LUBRICATED ATM-CMG BEARINGS WITHOUT LUBRICATION REPLENISHMENT

Approved by: 

R. Abramowitz

Prepared by: A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

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1.0 PURPOSE
The purpose of this report is to document the results of a bearing life test in which ATM CMG bearings were run in a life test fixture without benefit of a lubrication replacement system. The test was designed to simulate a possible condition on Skylab in space where little or no oil is replaced to lubricate the bearing.

2.0 BACKGROUND
One area of investigation into the possible causes for the failure of Skylab ATM CMG #1 (CMG S/N 5) and the anomalies in the speed current and temperature data telemetered from Skylab ATM CMG #2 (S/N 6) is inadequate relubrication of the spin axis bearings. Therefore, the purpose of the test conducted in this report is to determine the period of time bearings would operate satisfactorily without oil replenishment.

3.0 DISCUSSION
It is difficult to simulate space conditions on earth. However, some approximations to space conditions can be made. To simulate the weightlessness in space, the bearings were run in a life test fixture. The life test contains only a shaft of negligible weight compared with the 150# total weight of the CMG wheel. The bearing axial or preload was set at 40# as in the Skylab CMGs.

The life test fixture cavity was pumped down to a pressure of approximately $5 \times 10^{-3}$ Torr.

While the pressure in space will be in the area of $10^{-9}$
10^{-11} torr the labyrinth seal and other factors should keep the pressure in the ATM CMG at or below the vapor pressure of KG80 oil which is 10^{-7} at 100^\circ F and approximately 10^{-8} at 80^\circ F. However, the vapor pressure of the oil is extrapolated to be 10^{-9} at 100^\circ F after evaporative loss of 5\% by weight of the light ends of the oil and 5 \times 10^{-10} at 80^\circ F (ref 1). Labyrinth seal and molecular flow analysis estimates the loss of lubricant from each bearing to be approximately 1.6 \times 10^{-6} gm/hour if the temperature of the oil is at 100^\circ F and the surrounding enclosure is at 60^\circ F, the low point of the ATM CMG temperature range in Skylab. The escape of oil vapor thru the labyrinth seal is estimated to be less than 1.0 \times 10^{-6} gm/hr.

The ATM CMG bearings in space will contain in the order of 75 \times 10^{-3} gm of oil impregnated into the retainer and an additional 50 \times 10^{-3} gm of oil added to the races prior to assembly into the CMG. There is an additional supply of approximately 40 \times 10^{-6} gm of oil each hour the CMG is run on earth. If the worst case is considered in which the only oil available to the bearing is the 50 mg added prior to assembly and 1/3 of the oil impregnated into the retainer, there is approximately 75 mg available to lubricate each bearing. At the above evaporation rate, 5\% of this oil or 3.75 mg will take over 2000 hours to evaporate. At the point where 5\% of the available oil has evaporated, the vapor pressure of the oil has decreased by 10^{-2} torr and the evaporation rate will decrease to .016 \times 10^{-6} gm/hr.
Therefore the effect of lubricant evaporation is relatively minor with respect to bearing life.

Pumping down the life test fixture serves two purposes; it eliminates windage effects as in the ATM CMG in space, and removes oxygen from the bearing environment. This will deprive the lubricant and bearing metal interface of oxygen resupply which may have some effect on bearing life in space.

In order to simulate lubricant creep and possible loss of lubricant due to earth test in the spin axis vertical position, this test was run with the spin axis vertical such that gravity as well as thermal gradient would cause lubricant to creep from the race contact areas.

To simulate the bearing speed in space, the life test fixture was also run at 9100 rpm.

The two bearings selected for this test both have had a history of use both in ATM CMG wheels and in various life tests during the past several years. They were selected because both had accumulated several hundred hours of run with a lubrication replacement system and had the quantities and dispersion of oil within the bearing metal components and retainer similar to that of the bearings used in the Skylab ATM CMGs. Therefore, the bearing metal wear and ball pocket rub was also similar to those bearings used in the three Skylab AMT CMGs.

In late 1972 a somewhat similar test was run at Marshall
Space Flight Center utilizing a minimally lubricated bearing. This bearing contained 21.4 mg of oil in the retainer and an additional 6 mg of oil on the races. This is in comparison with the normal 50-80 mg of oil in the retainer and 50 mg in the races prior to assembly into an ATM-CMG. This test ran for 400 hours at 40# preload with the life test fixture on the spin axis horizontal position and an additional 355 hours horizontal at 190# preload to simulate the load of an ATM CMG spin axis vertical situation on earth. Although the bearing torque increased after the 355 hour period, the bearing metal parts indicated no degradation.

4.0 RESULTS
The bearings in this test ran for a total of 1688 hours. Torque and speed measurements were taken and recorded each day. The initial torque measurements on April 24, 1974 was 1.29 oz inches. The bearing then ran for 1600 hours at torque levels between 1.53 and 1.85 oz in. The speed variations during this period were 54 rpm. Due to the utilization of test equipment for other concurrent testing, continuous vibration level bearing temperature and motor current readings were not made. However, random measurements of these parameters were made during the entire period of the test with no significant variations from steady state running values.

On July 8, 1974 after running 1688 hours, the torque test indicated a bearing torque of 2.45 oz in up from 1.72 on July 5, 1974.
The speed had also decreased from 9088 rpm to 9065 rpm. This significant torque increase indicated that one or both bearings were beginning to fail and the test was terminated.

Both bearings were removed for visual examination. Bearing S/N SBB015 still contained significant amounts of oil on its metal parts and on the retainer surfaces. The bearing contact areas and ball wear pattern were visually not significantly deteriorated from when the test began. These were no significant buildup of pyrolyzed oil or wear metal residues on the ball pockets and no evidence of retainer instability.

Bearing S/N B19 the top bearing ran dry and was the bearing which exhibited the increase in torque. Visual inspection of the bearing indicated that the bearing retainer was completely dry. The ball pockets contained buildups of pyrolyzed oil in the lead and lag portions of the pockets. There was only minor burnishing of the pockets in these areas. There was no evidence of wear or debris from contact between the retainer scallop pads and the inner race lands.

The bearing inner and outer races contained pyrolyzed oil imbedded into the contact zones and pushed out to the periphery of the contact areas. There were minor amounts of free oil visible outside of the contact areas which either migrated from the contact area or had never migrated into contact area. The bearing balls are coated with pyrolyzed oil with some balls containing areas of
pyrolyzed oil imbedded on the ball surface.

Since the test was terminated at the first instance of a significant torque change, there was no apparent metal damage, i.e. fretting or spalling, to the bearing metal parts. The torque increase was caused by the buildup of wax-like pyrolyzed oil residues in the race contact areas and on the balls.

5.0 CONCLUSIONS
Based on the results of this test and of the minimal lubrication test (ref 2), it appears that an ATM CMG bearing which is properly lubricated initially, will run without further lubrication, for at least 1500 hours in space conditions. The bearing which "failed" would have continued to perform its function on a CMG for an additional several hundred hours. However, variations in torque current, and speed would be noted.

The appearance of the "unfailed" bearing indicates that it would have run for a period in excess of 3000 hours. This 3000 hours may be closer to the average life to be expected. The minimal lubrication test (ref 2) bearing contained less than 20% of the normal quantity of oil present initially on an ATM CMG bearing. It ran for 755 hours approximately half of which was under loads five times that which would have been experienced in non-maneuver modes in space.

The failed bearing, S/N B19, had not reached a critical point where catastrophic failure was eminent. Although
much of the oil was pyrolyzed and the effectiveness of the hydrodynamic film degraded, the friction between the balls and ball pockets had not reached the paint where there is localized overheating of the phenolic in the ball pockets, and generation of resinous debris. Likewise there was no evidence of metal to metal contact.

In the ATM CMG operating temperature range of $60^\circ\text{F} - 80^\circ\text{F}$ and with a labyrinth seal, the evaporation rate of the KG-80 oil used to lubricate the bearings is minimal and is a minor factor affecting bearing life.

REFERENCES


ATM FAILURE ANALYSIS
OIL MIGRATION AGAINST IC

Prepared by: A. Carriero

Approved by: R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
1.0 PURPOSE
The purpose of this report is to document the results of a test to determine whether oil centrifuged onto an ATM-CMG bearing outer race land will flow against gravity into the race contact area. The tests were conducted with the CMG bearing spin axis vertical such that any oil migrating into the race contact area will have migrated against gravity.

2.0 BACKGROUND
The failure of CMG No. 5 (Skylab CMG 1) and the anomalies which occurred with CMG No. 6 (Skylab CMG No. 2) caused the review of all aspects of the ATM-CMG lubrication replacement system. One aspect reviewed is the migration of oil into the bearing outer race contact area from where it is deposited on the land near the race edge. MT-15,777 covers the design changes which will deposit the oil directly into the bearing race contact area.

Life tests (Ref. NCER 73-06-042) with the CMG spin axis horizontal have proved that adequate oil migrates into the bearing race contact area while running in this orientation. Life tests in excess of 30,000 hours were made in this orientation.

ATM-CMG qualification and preflight tests were also performed in the spin axis vertical position. However, the amount of test hours in this orientation was small as compared with the bulk of testing and run-in which was performed in the spin axis horizontal position.
3.0 DISCUSSION

The tests were performed under the following conditions:

1. The bearing metal parts were cleaned and then dipped into a 50-1 mixture of freon and KG-80 oil to provide for a minimal hydrodynamic film.

2. The bearing phenolic separator was completely dry except for the ball pockets which were each "wetted" with approximately 1.5 mg of KG-80 oil, the approximate amount impregnated into a flight retainer pocket. This was accomplished by applying 10 drops of the 50-1 mixture to each pocket using a #20 hyperdermic needle. The purpose of lubricating the pockets was to prevent localized hot spots in the pocket contact areas.

3. The lube nut used contained oil tinted with a red dye and had been used successfully in previous tests (Ref. NCER-73-06-042).

4. The test bearing was run in a life test fixture with the bearing spin axis vertical. The test bearing was the bottom bearing with the lube nut situated below the bearing. Therefore, the flow of any oil to the race contact area was against gravity.

Since the first test bearing failed after approximately 140 hours, but during an overnight period, a second-test bearing was also run. In both tests the bearing was removed from the life test fixture, visually examined, photographed after the initial 50 hours of run. The second bearing was also removed and photographed after 100 hours.
Both units were examined and photographed after failure at approximately 160 hours. After each examination, the bearing was reassembled without additional oil. The bearing race orientation was always such that gravity was acting against migration of oil to the race contact area.

4.0 RESULTS

Examination of the oil on the bearings surfaces after 50 hours of the first test revealed a thick ring of red oil at the bottom of the race land where it had collected after being centrifuged from the bearing retainer onto the land. See photo 1 and Figure 1. A thin film of oil was visible on the race land from this point up to the race corner. The bearing outer race contact area and balls appear "wet", but the film is too thin to detect any "redness". The film is in excess of the film left from the 50-1 freon-KG-80 dip.

The bearing retainer oil groove did not contain any excess oil. Therefore, all the oil slung off the lube nut was centrifuged thru the oil holes to the race land.

The results of visual examination after 50 hours of the second test bearing were similar to that of the first test. No visual examination was made on the first unit after 100 hours. Visual examination of the metal parts of the second bearing after 100 hours of run revealed the following: A ring of red dyed oil approximately 1/8" wide at the bottom of the outer race land (see Figure 2 and Photo 2). A thin film of oil was visible from the ring upward to the race corner. There was a thinner film of oil in the upper area of the race groove and some oil present in the race contact.
area as noted by light defraction in some areas. There is a similar very thin film on the balls and the inner race contact area. None of this film was thick enough to visibly fluoresce under ultra-violet light. Some retainer pocket wear products and pyrolyzed oil were visible at the periphery of the race contact areas. There was also pyrolyzed oil and metal wear products deposited on the lead and lag portions of the ball pockets.

After reassembly of the bearing components, the second bearing ran an additional sixty hours to failure.

Examination of the first bearing after the failure at 140 hours revealed a ring of red dyed oil approximately 3/16" wide at the bottom groove of the race land (see Figure 3). There was also a film of oil extending upward from this ring up to the race corner. In the upper part of the bearing outer race groove, there was free oil visible but covered with debris and wear products of the failure. The bearing contact area contained residues and debris, but no free oil was noted by visual means. Use of ultra-violet light to determine the presence of KG-80 oil in the vicinity of the race resulted in the following observations: the thickness of oil film among the debris in the upper area of the race was sufficient to fluoresce faintly. The film of oil in the contact area of the outer race, and the balance of the race upward from the race groove which had been exposed only to the 50-1 freon to KG-80 oil mixture, did not contain enough film thickness to fluoresce.
There was no visible fluorescence from oil present on the inner race. There was also no visible fluorescence from the retainer ball pockets or from the retainer groove overlapping the lube nut. The 3/16" wide ring of oil was readily fluorescent under ultra-violet light.

Photos 3 and 4 show the oil ring at the bottom of the outer race. Photo 4 also shows the wear products of the failure, and the discolored oil and residues in the race contact area. The brighter area above the discolored wear pattern is reflected light from the film of oil present but obscured by wear debris. Photo 5 is another of this area.

Figure 6 shows the wear and debris buildup on the ball pockets of the retainer. The elliptical contact area in the lead and lag portions of the pockets was burnished but contains no pyrolyzed oil. The debris and pyrolyzed oil was deposited or pushed to the periphery of the contact areas. The rub pattern on the ball pockets indicates minor axial retainer instability during this test.

5.0 CONCLUSIONS
The results of the two tests performed indicate that on earth some of the oil centrifuged onto the race land in the proximity of the race corner does "wet" against gravity into the bearing outer race contact area. However, most
of the oil drains downward away from the outer race groove corner and is useless for lubrication of the bearing.

The amount and rate of oil migrating into the outer race under these conditions is insufficient to sustain the hydrodynamic film in a bearing which is marginally lubricated.

The location of debris on the outer race indicates pumping action between the retainer OD and outer race land ID such that there is some flow of debris and oil against gravity.

There was insufficient quantities of red dyed oil transferred from the outer race to the inner race to be visibly detected on the inner race.
PHOTO 1
FAINT RED RING AT BOTTOM OF DISASSEMBLED OUTER RACE 50 HOURS 1ST TEST

PHOTO 2
RED RING AT BOTTOM OF ASSEMBLED OUTER RACE 100 HOURS TEST 2
PHOTO 3

OIL RING AT BOTTOM OF ASSEMBLED OUTER RACE 160 HOURS TEST 2

OIL RING AND WEAR DEBRIS DISASSEMBLED OUTER RACE
160 HOURS TEST 2
PHOTO 5
OUTER RACE GROOVE
REFLECTED LIGHT
FROM OIL IN RACE GROOVE

PHOTO 6
RETAINER BALL POCKETS 160 HOURS
HYDRODYNAMIC FILM TESTING
OF THE
ATM, CMG INNER GIMBAL
AND
ROTOR ASSEMBLY (IGRA)
UTILIZING 107-H ANGULAR CONTACT SPIN AXIS BEARINGS

Prepared by: C. Gurrisi

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608
PURPOSE

This test program was initiated to determine the wheel speed that develops a sufficient hydrodynamic film in the spin axis bearing as a function of temperature, cavity pressure, and lubricant quantity.

DISCUSSION

For this test program two ATM CMG Inner Gimbal and Rotor Assembly (IGRA) angular contact ball bearings were normally lubricated with KG-80 oil, torque tested for quality, and assembled into an IGRA Bearing Life Test Fixture (BLTF). This BLTF was modified for special capacitance measurements to determine the development of a hydrodynamic film in the spin axis bearings. The stiffening thru strut was electrically insulated from the bearing outer race (i.e. the case of the BLTF) by employment of lucite end caps. Any capacitance increase between the thru strut and case of the BLTF would indicate the presence of a hydrodynamic film within the ball bearings. The circuit used to monitor this capacitance is shown in Figure 1.

The first attempt at determining the wheel speed that develops a hydrodynamic film was performed on this BLTF at room temperature, and a cavity pressure of 5 microns. The capacitance measurement indicated a film was present before the wheel speed indicator was triggered. Therefore, the second test was performed at
HYDRODYNAMIC FILM SCHEMATIC.

Fig # 1

CHANGE IN SINE WAVE AMP.
INDICATED INCREASE IN CAPACITANCE
atmospheric pressure with the side covers removed so wheel speed could be detected by means of a strobe light. This set-up also produced unsatisfactory results because the strobe light introduced pulsating electrical noise into the capacitance measurement. The third attempt was also performed at atmospheric pressure with the side covers removed. However, the bearing lock nut was marked so that the wheel speed could be determined by counting the revolutions per unit time. The wheel speed of the BLTF was gradually increased until a lubricant film was indicated by the capacitance measuring electronics. This technique showed a hydrodynamic film was present in both bearings at speeds of 66-84 RPM.

Since speed detection at low speed presented a large problem in these initial tests, successful testing in thermal-vacuum conditions was doubtful. The BLTF was subjected to a temperature of 165°F and a vacuum of $10^{-7}$ Torr in the thermal vacuum chamber. Once thermal equilibrium was reached, the unit was slowly accelerated to operating speed. Throughout the accelerating period the capacitance electronics were saturated with noise, caused by long cabling to the thermal vacuum chamber. To obtain any satisfactory results in thermal vacuum environments a more sophisticated means of measuring speed and a special capacitance bridge will be needed. Since these items are considered special test equipment, not funded by the present contract, and the results of further testing was uncertain, it was decided in conjunction with NASA personnel to discontinue the task.
CONCLUSIONS AND RECOMMENDATIONS

This test indicated that a hydrodynamic lubricant film is developed over a speed range of 66-84 RPM and is maintained throughout the run-up period to 9100 RPM. Since this hydrodynamic film was established at a very low speed, in comparison to the operational speed (9100 RPM) of the CMG, insufficient hydrodynamic film lubrication does not appear to have been a problem in the ATM CMG.

If any further hydrodynamic film testing is to be considered, a very accurate and low speed detection system will have to be developed, along with a sophisticated capacitance bridge.
LIFE TESTS WITH
REDESIGNED ATM CMG RETAINER

Prepared by: A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
1.0 PURPOSE
The purpose of this report was to document the results of life tests utilizing ATM CMG bearings and redesigned bearing retainers. The bearing retainer was redesigned to more effectively capture and distribute oil to the bearing contact surfaces from the lubrication replacement system (lube nuts). The purpose of these tests was to evaluate the effectiveness of the design changes.

2.0 BACKGROUND
The primary mechanisms used to orient and maneuver the Skylab space laboratory were three Control Moment Gyros. These CMG's used two 107H size ball bearings with phenolic bearing retainers. The retainers were impregnated with KG-80 oil and served as a mechanism by which additional oil was centrifuged from the lube nut oil replacement system to the retainer flange and was then centrifuged through feed holes in the retainer to the I.D. of the bearing outer race.

A review of the bearing design parameters and further tests were instigated by the premature failure of CMG 1 (S/N 5) aboard Skylab, and anomalies encountered with CMG (S/N 6). These studies disclosed that in certain conditions most of the oil centrifuged from the retainer feed holes would not enter or migrate into the bearing outer race contact area (Ref. MT-15,776). Therefore, a redesign of the bearing retainer (Ref. MT-15,777) was performed such that oil from the flange would feed directly into the bearing outer race groove in all conditions. One set of holes would feed the oil into the area of the race groove above the ball
path. The second set would feed oil directly into the center of the ball path under nominal conditions. In addition the retainer flanges were redesigned to further overlap the lube nut as well as to allow for the more acute angle of the oil feed holes.

3.0 DISCUSSION

To determine the effectiveness of the retainer redesign three life tests are being performed in life test fixtures and an additional life test is being performed in an ATM CMG Inner Gimbal and Rotor Assembly (IGRA). This report is concerned only with the three life tests performed in the life test fixtures.

The life test fixtures were designed originally to simulate to a certain extent the conditions of vacuum and weightlessness (lack of radial load) the bearing will encounter in an ATM CMG in space. The weight of the fixture shaft is negligible when compared with the 150 pound weight of the actual ATM CMG wheel. The life test fixtures are powered by an ATM CMG motor developing approximately nine oz-in of torque at 130 volts, 455 Hz, and at 9100 RPM shaft speed. The bearings are preload to 40 pounds, the same preload applied to the bearings used on the Skylab CMG's.

The life test fixtures were run in a spin axis vertical orientation such that the effect of gravity on the dispersion of the oil supplied to the bearings from the lube nut could be studied, as well as the effect of the pumping action of the retainer to disperse oil against gravity.
To accomplish this, the test bearings in tests 1 and 2 were the bottom bearings in the fixtures and were minimally lubricated initially by being dipped into a 50-1 mixture of freon to KG-80 oil. The redesigned retainers in these test bearings were assembled dry except for a small quantity of oil in each ball pocket to prevent excessive friction between the balls and the pockets. Therefore, the location and amount of any oil in excess of the initial thin film on the metal parts would be directly attributed to the oil centrifuged from the lube nut to the retainer flange and dispersed by the retainer feed holes onto the outer race. Any further migration of the oil would then be the effects of migration due to gravity or pumping action of the retainer.

The test bearing in Test #1 was run with minimal lubrication and was resupplied with oil tinted with a red dye. The test bearing on Test #2 was similar to that in Test #1 except that the resupplied oil did not contain a dye. The life test bearings on Test #3 were wetted normally and contained normally lubricated bearing retainers as were the retainers on the bearings paired with the test bearings in tests 1 and 2. In each test the bearings were prepared as follows: the test bearing metal parts used in Test #1 were assembled for the first time specifically for this test and were prepared per (MT-13,923). The metal parts were then dipped into a 50-1 mixture of freon to KG-80 oil leaving a thin film of oil on the metal parts after evaporation of the freon. The redesigned retainers used in the bottom bearing (the test bearing) was soxhlet extracted in freon and baked dry at 220°F for 54 hours.
Therefore, this retainer was completely void of oil except for the ball pockets of which each were "wetted" with approximately 1.5 mg of oil. This was accomplished by applying ten drops of a 50-1 mixture of freon to oil to each dry pocket with a #20 hyperdermic needle. This was done to reduce the friction and localized over heating in the ball pockets as would occur if the pockets were left dry. Therefore, the bottom bearing when assembled into the life test fixture contained only minimal oil to support an elasto-hydrodynamic film and to prevent rusting of the metal parts. Any additional oil found would necessarily have flowed from the lube nut/oil replacement system.

In Test #1 the top bearing used to pair with the test bearing was a bearing utilized in previous tests and did not contain a redesigned retainer.

The test (bottom) bearing in Test #2 and its redesigned retainer were prepared exactly as the test bearing and retainer in Test #1. In this test the oil in the lube nut to resupply the test bearing contained no red dye. In Test #2, the top bearing used to pair with the test bearing was also assembled for the first time for this test and the metal parts were cleaned per MT-13,923. However, the metal parts were dipped into the normal 5 to 1 freon to KG-80 oil mixture, leaving a thicker film on the metal parts. The retainer used in this bearing was a redesigned retainer and was fully impregnated.

In Test #3, both bearings were assembled for the first time for this test and the metal parts prepared per MT-13,923.
The metal parts were then dipped into the 5-l freon to KG-80 oil mixture to coat the metal surfaces as is normal for bearings for use on an IGRA. Likewise, the redesigned retainers used in these bearings were fully impregnated. The lube nuts used with these bearings also did not contain dyed oil.

4.0 RESULTS

As of December 6, 1974, Test #1 has run for 3588 hours and shows no visible signs of distress to the test bearing metal parts or retainer. There is a meniscus of oil visible between the balls and outer race contact surface. There is no evidence of degraded or pyrolized oil in the visible portions of the retainer pockets, on the retainer pads or in the bearing outer race. There are visible quantities of red tinted oil which have been pumped against gravity to the upward land of the outer race. There is excess oil in the outer race groove and contact area. There are greater quantities of red tinted oil which have migrated from the outer race groove downward with gravity to the bottom of the bearing (see Figure 4).

Visual examination of the bearing metal parts and retainers after 2101 hours in Test #2 and 3331 hours in Test #3 are similar to that in Test #1 in that there is only normal light wear banding visible on the bearing balls and outer race contact areas and no visible signs of metal or retainer distress. There is a visible meniscus of oil between the outer race contact area and the balls. There is likewise no visible evidence of any lubrication degradation. There is evidence of quantities of oil, pumped against gravity,
visible on the upward land of the outer race of both bottom test bearings. There are larger quantities of oil visible in the race grooves and more which has migrated due to gravity from the race groove to the bottom of the lower land of the test bearings' outer races.

The progressive visible inspections of the lower bearing in Test #1 gives a graphic description of the flow and dispersion of the oil. At the 90 hour examination, see Figure 1, there is no visible additional oil on the outer race lands, and while there appears to be some additional oil in the outer race contact area and balls. The quantity is sufficient for a red tint to be noted. At the 328 hour examination (see figure 2), there is some evidence of additional quantities of oil on the bearing upper outer race land, pumped against gravity. There is definitely additional quantities of red tinted oil in the outer race land and balls, and a small meniscus of oil with a faint red tint between the balls and the contact area.

After the 1159 hour visual examination (see Figure 3), there is an additional quantity of red tinted oil pumped against gravity to the upward land of the outer race. The bearing outer race groove contains enough extra oil to be visibly pushed out of the ball contact area by the balls as the bearing is slowly rotated. There is a meniscus with a red tint visible between the balls and contact area. There is also additional oil from the outer race groove on the lower land and ending in a red ring at the bottom of the bearing. The examinations after the 2075, 2621 and 3588 periods (see Figure 4) show similar concentrations of red
dyed oil in the races, balls, and upper land but with increasing quantities of oil being drained from the race groove as additional quantities are centrifuged into the bearing outer race.

The results of visual examinations of the bearings in Tests 2 and 3, are similar except that they are not as graphically visible since the oil used does not have a red tint.

Chart 1 below shows the hours and date of each visual examination of the test bearings.

<table>
<thead>
<tr>
<th>Test/Visual Examination</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimal Lube 90</td>
<td>90</td>
<td>328</td>
<td>1159</td>
<td>2075</td>
<td>2621</td>
<td>3588</td>
<td></td>
</tr>
<tr>
<td>Minimal lube 480</td>
<td>480</td>
<td>978</td>
<td>1677</td>
<td>2101</td>
<td></td>
<td></td>
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<tr>
<td>(no red dye) 9/24/74</td>
<td></td>
<td>10/15/74</td>
<td>11/18/74</td>
<td>12/6/74</td>
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<td>Normal lube 484</td>
<td>484</td>
<td>1286</td>
<td>2384</td>
<td>3331</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(no red dye) 8/2/74</td>
<td></td>
<td>9/4/74</td>
<td>10/21/74</td>
<td>12/6/74</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

CHART I
5.0 CONCLUSIONS

The redesigned retainer does effectively capture all of the oil centrifuged from the lubrication replacement system and effectively distributes it to the bearing outer race groove and contact area.

The presence of oil on the upward land of the bottom test bearing in each test proves that some of the oil centrifuged into the bearing outer race is pumped upward against gravity through the race groove.

The fact that all of the oil centrifuged from the lube nut is captured by the retainer flange and is deposited into the outer race groove is proven by the absence of oil on the lower or lube nut side of the outer race land, during the first several hundred hours of each test. Only after sufficient excess quantities of oil are dispersed into the outer race does oil appear on the lower land due to drainage.
BEARING LIFE TEST WITH REDESIGNED RETAINERS IN AN INNER GIMBAL AND ROTOR ASSEMBLY (IGRA)

Prepared by: C. Gurrisi

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608
PURPOSE

This test program was initiated to validate and life test the present lubrication system with bearings containing a redesigned retainer in an inner gimbal and rotor assembly (IGRA). The main purpose was to demonstrate that the redesigned retainers would supply lubricant (KG-80 oil) directly to the ball/race contact zone while operating in a "1 G" field opposing the flow of the lubricant.

ABSTRACT

The failure of ATM-CMG No. 5 (Skylab CMG 1) and the anomalies which occurred on ATM-CMG No. 6 (Skylab CMG #2) resulted in an investigation of all aspects of the ATM CMG active lubrication system. As documented in MT-15,777, design changes to the ball bearing retainer were made to deposit lubricant directly into the ball/race contact zone. This test program was initiated to validate the design changes of the bearing retainer while operating in an IGRA with a one "G" field opposing the flow of lubricant to the race contact zone.

DISCUSSION

Inner Gimbal and Rotor Assembly S/N E-2 was assembled with two normally lubricated 107H angular contact ball bearings containing the redesigned retainers. After the unit was assembled an abbreviated functional test was performed to verify all performance parameters.
This test accumulated a total of 36 hours on the bearings. Since all performance parameters were satisfactory the unit was partly disassembled for bearing visual inspection and installation of the active lubrication systems. The lubrication systems installed at this time had flow rates of .092 Mg/Hr on Side 1 and .073 Mg/Hr on Side 2. The unit was reassembled and a final acceptance and screening test was performed. This functional test is outlined below.

<table>
<thead>
<tr>
<th>Time</th>
<th>Operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1. Normal run-up to 8000 RPM (400 Hz, 115 Volts, Pump On)</td>
</tr>
<tr>
<td>24</td>
<td>2. Stable torque test spin axis horizontal, start-spin axis down</td>
</tr>
<tr>
<td>32</td>
<td>3. Stable torque test spin axis down, start stable spin axis horizontal</td>
</tr>
<tr>
<td>89</td>
<td>4. Stable torque test spin axis horizontal, start spin axis up</td>
</tr>
<tr>
<td>97</td>
<td>5. Stable torque test spin axis up, start stable speed spin axis horizontal</td>
</tr>
<tr>
<td>114:30</td>
<td>6. Stable torque test spin axis horizontal, run-up to 9100 RPM, 455 Hz, 130 volts</td>
</tr>
<tr>
<td>137</td>
<td>7. Stable torque test spin axis horizontal, start stable speed spin axis down</td>
</tr>
<tr>
<td>145:15</td>
<td>8. Stable torque test spin axis down, start stable speed spin axis horizontal</td>
</tr>
<tr>
<td>161</td>
<td>9. Stable torque test spin axis horizontal, start stable spin axis up</td>
</tr>
</tbody>
</table>
All performance parameters of this test are summarized in Table 1.

At the conclusion of this functional test the unit was vented, the side covers, end cap and strut assembly, and lubrication systems were removed to visually inspect the ball bearings.

It should be noted that the ball bearings were not removed, and the end shake (bearing preload) was not disturbed.

This inspection revealed that both bearing retainers were well lubricated. Both bearing inner raceways contained significant quantities of oil, and no evidence of polymerized, or degraded oil was present on the retainer pads. Although both retainers were wet with oil
### PERFORMANCE SUMMARY FOR F-2 FROM 10/10/74 TO 10/30/74

<table>
<thead>
<tr>
<th>SPIN AXIS (UP)</th>
<th>SPIN AXIS (DOWN)</th>
<th>9000 RPM</th>
<th>9100 RPM</th>
<th>1000 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Bearing Temp Side 1 °F</td>
<td>5.63</td>
<td>5.93</td>
<td>92</td>
<td>96</td>
</tr>
<tr>
<td>2 Bearing Temp Side 1 °F</td>
<td>85</td>
<td>5.93</td>
<td>85</td>
<td>85</td>
</tr>
<tr>
<td>3 Bearing Temp Side 2 °F</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>4 Bearing Vibration Side 1</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>5 Bearing Vibration Side 2</td>
<td>11</td>
<td>11</td>
<td>11</td>
<td>11</td>
</tr>
<tr>
<td>6 Cavity Pressure Microns</td>
<td>6.5</td>
<td>6.5</td>
<td>6.5</td>
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<tr>
<td>7 Power Watts</td>
<td>38</td>
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<td>38</td>
</tr>
<tr>
<td>8 Peak Speed RPM</td>
<td>7/5</td>
<td>7/5</td>
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</tr>
<tr>
<td>9 Hours Spin Axis Up</td>
<td>8.03</td>
<td>8.03</td>
<td>8.03</td>
<td>8.03</td>
</tr>
</tbody>
</table>

### GENERAL DATA
1. Run-up time hours (2600 rpm only) side 15 wins. 36 Power Supply
   - 9.56
2. Axial end snake (inches)
   - 0.6595
3. Speed pick-up gap (inches)
   - 4.804
4. Run time of test (hours)
   - 2040.7
5. Bearing temperature at start of test
   - Side 1
   - 70°F
   - Side 2
   - 73°F

### TABLE 1
there were no large accumulations of oil in the retainer edge undercut as found in some of the bearings life tested with the original retainer design. Therefore the design changes of the oil feedthrough holes provides more efficient oil flow directly into the ball/race contact zone.

The unit was reassembled and accelerated to 9100 RPM in the spin axis horizontal position, this unit is currently on life test in the spin axis down position at BGSD. To date this unit has accumulated 167 hours in this orientation. When 1000 hours of operation in this attitude are accumulated, a bearing visual inspection will be performed, and if the results are satisfactory, the test will continue.
To: Engineering File MT-15,777
From: A. Carriero

ATM CMG BALL BEARING

RETAILER REDESIGN

Prepared by: A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608
1.0 PURPOSE

The purpose of this report is to document the modifications made to the ATM CMG Ball Bearing Retainer. These changes are required to more effectively distribute the oil from the annular groove on the side of the retainer to the bearing outer race.

2.0 BACKGROUND

A complete design review of the ATM CMG system was required due to the failure of CMG S/N 5 on Skylab and the speed changes in CMG S/N 6 also on Skylab. During review of the ball bearing phenolic retainer it was noted that although the center of the oil feed hole was well past the edge of the outer race dam, the rear of the hole, where the oil drops actually formed, did not clear the edge. Therefore, the oil drops centrifuged from the rear of the feed hole, onto the outer race land approximately .027" from the dam edge, and migrated into the outer race. There is a question as to whether sufficient oil migrated into the outer races of the CMG's on Skylab and affected performance or propagated the problems encountered in CMG S/N's 5 and 6. The third CMG, S/N 7 ran without any problems noted, throughout the entire 6,500 hour mission in space.
3.0 DISCUSSION

The redesign of the retainer was effected to assure that oil droplets would centrifuge directly into the outer race in all cases. To accomplish this the following changes were made:

1) The original holes were drilled at a more acute angle to overlap the race dam by a minimum of .018" in the worst conditions and .0315" at nominal conditions. See Figures 1, 2 and 3.

2) An additional set of holes were added to centrifuge oil droplets into the bearing outer race at the nominal contact angle of 14.5°, see Figure 4.

3) The width of the oil groove was increased to allow for the more acute angle of the oil holes, without loss of groove dam diameter. The additional width also increases the overlap of the groove with respect to the lube nut by .036" from a minimum of .0035 to .0395. See Figures 5 and 6.

4) The retainer outer diameter was increased by .012". This decreases the minimum radial clearance between the outer race land and retainer O.D. from .026" to .020". However, it allows the nominal 19° angle hole to extend .017" further into the outer race, and the nominal 14.5° hold .023" further extension into the outer race.
The oil droplets will now centrifuge off into the bearing outer race in two areas: One area is above the contact zone from which oil will migrate into the contact zone of the outer race. The second area is in the center of the nominal contact zone. Therefore, the bearing outer race contact zone will be lubricated both directly and indirectly providing an adequate supply of oil to be transferred by the ball to the inner race and retainer pocket and ensuring an adequate hydrodynamic oil film. See Figure 7.
## Retainer Design Calculations

<table>
<thead>
<tr>
<th></th>
<th>Worst Case 29°</th>
<th>Worst Case 20°</th>
<th>Nominal Case 19°</th>
<th>Nominal Case 14.5°</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>A. Contact Angle °</strong></td>
<td>17°</td>
<td>17°</td>
<td>15°</td>
<td>15°</td>
</tr>
<tr>
<td><strong>B. Ball Size</strong></td>
<td>.3120</td>
<td>.3120</td>
<td>.3125</td>
<td>.3125</td>
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<tr>
<td><strong>C. Ball Pocket Dia.</strong></td>
<td>.329</td>
<td>.329</td>
<td>.328</td>
<td>.328</td>
</tr>
<tr>
<td><strong>D. Outer Race Curvature</strong></td>
<td>.1646</td>
<td>.1646</td>
<td>.1656</td>
<td>.1656</td>
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<tr>
<td><strong>E. Outer Race Groove Dia.</strong></td>
<td>2.2219</td>
<td>2.2219</td>
<td>2.2222</td>
<td>2.2222</td>
</tr>
<tr>
<td><strong>F. Outer Race Land Dia.</strong></td>
<td>2.1110</td>
<td>2.1110</td>
<td>2.1100</td>
<td>2.1100</td>
</tr>
<tr>
<td><strong>G. Outer Race Groove Depth</strong></td>
<td>.05545</td>
<td>.05545</td>
<td>.05610</td>
<td>.05610</td>
</tr>
<tr>
<td>(E-F)/2</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>H. Outer Race Center to Land Corner</strong></td>
<td>.1233 - [(1646)^2 - (.1646 - .05545)^2]^{1/2}</td>
<td>.1242 - [(1.1656)^2 - (.1656 - .05610)^2]^{1/2}</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>I. Ball Center to Land Corner</strong></td>
<td>.1208 - .1233 - (.1646 - .1560) sin 17°</td>
<td>.1216 - .1242 - (.1656 - .15625) sin 15°</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>J. Retainer Displacement</strong></td>
<td>.0085</td>
<td>.0085</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td><strong>K. Ball Center to Oil Hole Lip</strong></td>
<td>.0085 - .258 + .519</td>
<td>.0085 - .294 + .591</td>
<td>.295 - (.250 - .1475 sin 29°)</td>
<td>.295 - .255 = .040</td>
</tr>
<tr>
<td></td>
<td>= 1625</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>L. Overlap - Oil Hole Lip to Land Corner</strong></td>
<td>-.0417 - .1208 - .1625</td>
<td>-.0183 - .1208 - .1025</td>
<td>.0315 = .1218 - .0903</td>
<td>.0818 = .1218 - .040</td>
</tr>
<tr>
<td><strong>I. Location - Ball Center</strong></td>
<td>.5310 + .0015 = .5325</td>
<td>.2900 - .0025 = .2875 , .5325 - .2875 = .2450</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>K. Location - Ball Center</strong></td>
<td>.517 - .260 - .0085</td>
<td>.589 - .296 - .0085 =</td>
<td>.2485 =</td>
<td>.2485</td>
</tr>
<tr>
<td><strong>III. Overlap - Retainer Groove to Lube Nut</strong></td>
<td>.2485 - .2450 - .0035</td>
<td>.2945 - .2430 - .0395</td>
<td>SEE FIGURE 5</td>
<td>SEE FIGURE 6</td>
</tr>
</tbody>
</table>

**TABLE I**
WORST CASE 2.9° HOLE
ORIGINAL RETAINER

Figure 1
WORST CASE 20° HOLE
REDESIGNED RETAINER

Figure 2
NOMINAL CASE 45° HOLE

Figure 4
FIGURE 7
OIL DEPOSITION