INTERMEDIATE ACCURACY INTEGRATING GYROSCOPES DESIGN CRITERIA MONOGRAPH

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June 1970

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FOREWORD

This work, which deals with design and manufacturing consideration for intermediate-accuracy integrating gyroscopes, is one of the series on Guidance and Control; additional monographs will be issued on related topics and are listed in paragraph 1.3 of the Introduction.

The Design Criteria Series should be regarded as guides to design and not as NASA requirements. It is expected, however, that the Criteria and Recommended Practices sections contained therein, revised as experience may dictate, will eventually become uniform design practices for NASA space vehicles.

This monograph was prepared by Arthur D. Little, Inc., for the NASA Electronics Research Center, under Contract NAS 12-2077. Mr. Robert G. Haagens of ADL, the author, served as program manager and principal investigator. The effort was guided by an advisory panel, both in the evolution of its content and in the assessment of technical validity. With the assistance of the panel, the chairman prepared a subject agenda for the initial discussion meeting, following which substantive written contributions were received from the panel members. The individuals who constituted the advisory panel are listed below.

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Comments concerning the technical content of these monographs will be welcomed by the National Aeronautics and Space Administration, Office of Advanced Research and Technology (Code RVA), Washington, D.C. 20546.

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1. INTRODUCTION

1.1 SCOPE

Government agencies continue to encounter problems of gyro failure. The failure modes are related both to the basic design of the instrument and to the control of assembly processes. In this design criteria monograph, we will deal with the state of the art of single-degree-of-freedom floated gyroscopes, with emphasis on failures and performance degradation in the classical areas:

1) The gyro wheel package, ball bearing as well as gas bearing.

2) Case to float considerations, including hermetic seals and flotation/damping fluids.

3) Restraint drift torques.

1.2 EFFECTS OF THE PROBLEM

Gyroscopes require precision manufacturing of component parts, high assembly skills, innovative test and diagnostic analysis techniques, and selective component and trim assembly to obtain a properly working instrument. Their complexity requires design foresight based both on the operational theory and on practical assembly experience. Assembly errors, even on a well designed instrument, will seldom be self-correcting or forgiving; instead, they will cause premature failure. The impact of failures and poor performance of gyros on NASA space vehicles is costly in time and money. State of the art technology, however, possesses controls and solutions to problem areas and some of these are presented herein.

1.3 MONOGRAPHS RELATED TO THIS DOCUMENT

This monograph covers only a part of the total subject of single-degree-of-freedom floated integrating gyroscopes. The following subjects will be treated in future monographs:

a) The Transducing Elements

b) Output Axis Suspension Techniques

c) Temperature Control Techniques

d) Aniso Compliance of the Wheel and Float Assembly, Aniso Inertia Considerations of the Float Assembly.

2. STATE OF THE ART

2.1 EVOLUTION[1]

Floated single-degree-of-freedom integrating gyroscopes evolved from early (1935-45) SDF spring-restrained *rate* instruments built for fire control (gunnery) purposes. The accuracy of these instruments was limited by the axis output bearing friction torques; both coulomb friction and output axis bearing hangup were major error sources. From a dynamic standpoint, little attempt was made to obtain useful response characteristics from these early instruments.

The first design improvement, therefore, concentrated on the addition of dashpot dampers. Later improvements enclosed the gyro wheel in a can (float or gimbal) so that Newtonian shear damping could be provided by the addition of viscous fluids. In the search for fluids that would provide the desired damping torque, even tar was tried. Changes in viscosity with temperature soon made the need for temperature control evident.

Simultaneously, it was realized that output axis suspension friction could be reduced if the wheel could displace its own weight in the damping or flotation fluid. At that time, however, the available organic fluids with Newtonian properties characteristically had densities less than that of water, so an excessively large gimbal can was needed to float the wheel. To reduce the size of gyros, designers sought fluids of higher density.

The amount of flotation fluids used in gyros is so small that their development is of little profit to the chemical industry. Thus, progress in gyro hardware might have been seriously hampered if large quantities of high-density fluids had not also been needed by the Atomic Energy Commission. Halogenated hydrocarbon fluids, approximately twice as dense as water, began to appear in 1948, and these fluids also possessed the desired Newtonian and viscosity characteristics. SDF floated gyroscopes containing these fluids performed orders of magnitude better (in terms of friction characteristics) than the previous unfloated designs. The new flotation fluids, which provided neutral buoyancy within reasonable instrument size, also permitted the use of pivot/jewel suspension elements instead of ball bearings, which had exhibited retainer caging torques.

Other sources of friction and restraint then became evident. Contamination was the worse offender, but other uncertainty torques that resulted from hysteretic behavior of flex leads,. mass balance changes, convection in the flotation fluid, electromagnetic forces, and suspension friction, were uncovered as the development of SDF floated gyroscopes progressed. Some of these problems stubbornly resist solution and continue to plague current designs of varying degrees of sophistication.

2.2 IMPORTANCE OF EMPIRICAL KNOWLEDGE

The scientific aspects of SDF floated integrated gyros have been extensively explored and recorded in various textbooks* and papers. A recent NASA-ERC monograph, "Inertial Gyroscopes Systems Applications Considerations" illustrates today's thorough understanding of the academic aspects of this subject with a detailed mathematical and error analysis model.

However, a knowledge of the theoretical concepts is not sufficient to assure success. In any given case, there are usually several ways to achieve reliability but only one that best serves the particular application and the original design intent. Furthermore, prior experience will suggest certain design approaches that provide greater ease of assembly and a corresponding increase in reliability. These aspects of design and manufacturing know-how, which perhaps are more art than science, are the subject of this monograph.

2.3 GYRO WHEEL CONSIDERATIONS

The heart of the SDF gyroscope is the gyro wheel. For highest gyro accuracy, the gyro wheel should have the largest angular momentum obtainable within the volume and flotation capabilities of the instrument. In current designs, the wheel is supported on either ball bearings or self-levitating gas bearings; the choice between these is a trade-off that is primarily based on performance level versus cost. Gas bearing gyros generally have greater accuracy and, when properly built, provide improved reliability with increased life.

Design considerations that weigh the trade-offs available in gyro wheel and motor design are of direct interest to the gyro manufacturer and the gyro user. These trade-offs have a correlation to instrument quality, reliability and cost.

Trade-Offs of Wheel Geometrics

Wheel designs are of three basic types: spherical, I-shaped and umbrella-shaped. Spherical wheels (Figures 1 and 2) have the maximum angular momentum for a given speed and have the further advantage of symmetry, which helps to simplify the design. The I-shaped wheel (Figure 3) is also symmetrical, but its angular momentum is at least 20% less. The umbrella-shaped wheel (Figures 4 and 5) has about the same angular momentum as the I-shaped and is relatively easy to make, simple to assemble, and very reliable; unlike the other two, however, it is not symmetrical about a plane normal to its axis and special compensations are needed to obtain a properly located center of gravity.

Trade-Offs Against Ease of Wiring

A different ranking applies with respect to ease of wiring. When the motor stator is inside the wheel (as it must be for maximum angular momentum with a given gyro diameter), the stator wires for a spherical wheel must be brought in through the wheel shaft, which is difficult and conducive to electrical failure, and

* See, for example, References 2 and 3.

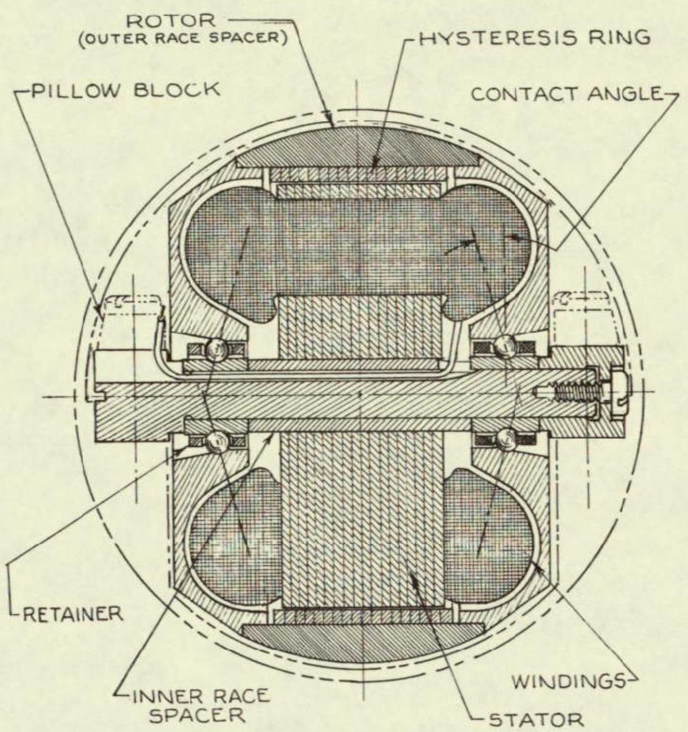

FIGURE 1 SPHERICAL WHEEL – DB PRELOAD WITH SPACERS

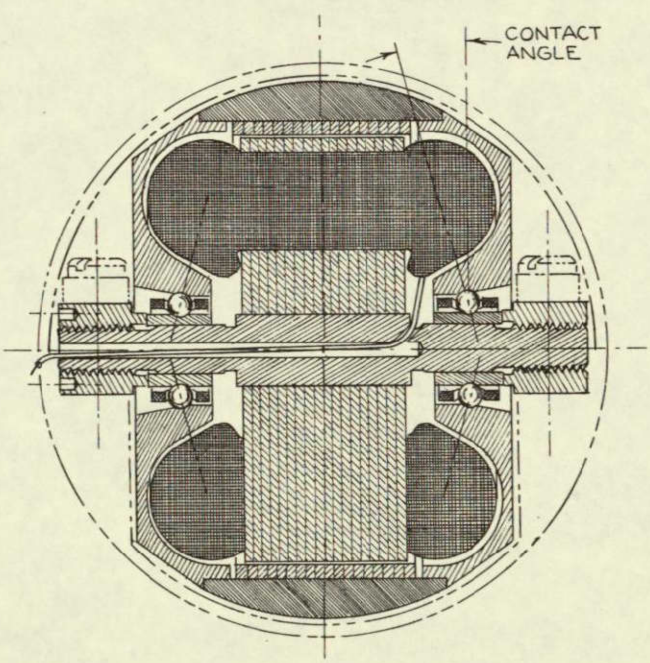

FIGURE 2 SPHERICAL WHEEL – DB PRELOAD SCREW ADJUSTABLE – NO SPACERS

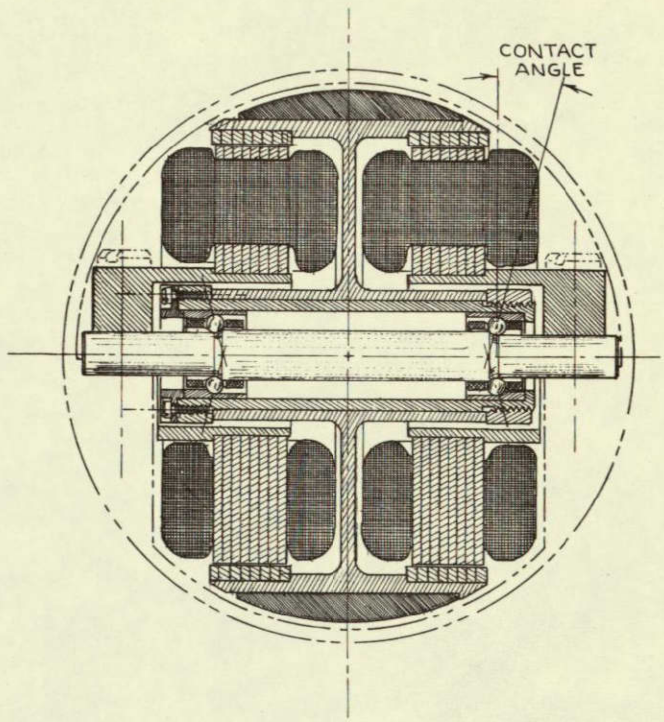

FIGURE 3 DF PRELOAD – INTEGRAL SHAFT
"I" SHAPED WHEEL
2 STATORS – COMPLETE SYMMETRY

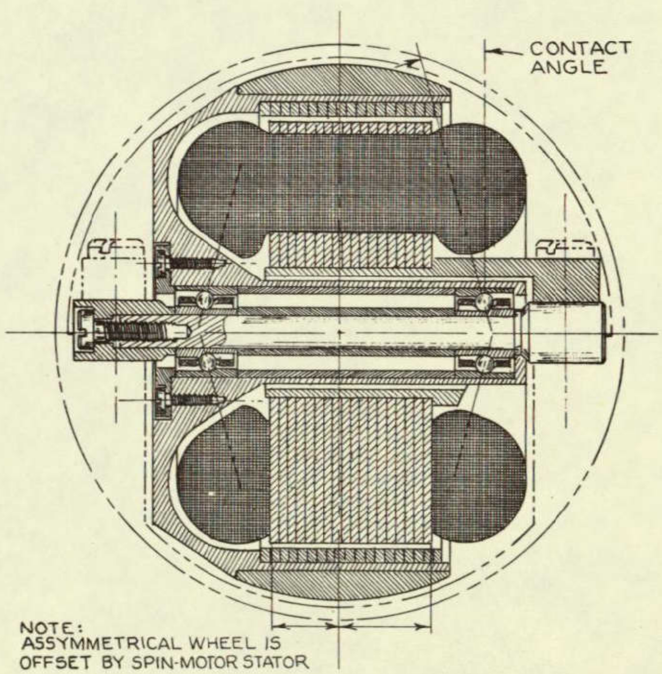

FIGURE 4 DB PRELOAD UMBRELLA SHAPED WHEEL

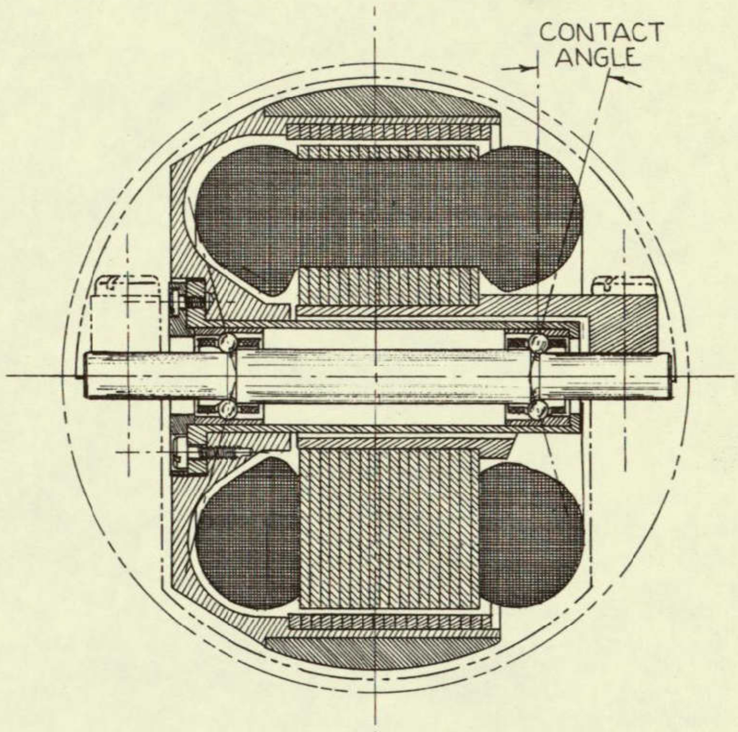

FIGURE 5 DF PRELOAD – INTEGRAL SHAFT UMBRELLA SHAPED WHEEL

impossible for gas bearing designs. I-shaped wheels have two stators, one on each side of the center section, and these are open and accessible for wiring. The single stator of the umbrella wheel is as equally accessible on its exposed side as the I-shaped wheel construction.

Trade-Offs of External Stator Construction

Although it entails additional sacrifice of angular momentum, external construction of the drive motor has certain advantages: the use of a float-mounted stator outside the wheel, for example, not only facilitates wiring connections but also improves the mass stability of the stator body and winding in the float. It also allows improved conduction of heat from the motor to the gimbal structure.

External construction of the drive motor inevitably reduces the radius of gyration of the wheel if the gyro outside case diameter remains constant. The sacrifice of angular momentum is appreciable, since momentum varies as the square of the radius of gyration:

$$H = I\omega$$

$$I = mk^2$$

where,

H = angular momentum in gm-cm^2/rad/sec or dyne-cm-sec/rad

I = SRA rotor inertia in gm-cm^2

ω = wheel speed angular velocity in rad/sec

k = radius of gyration in cm

m = mass in gm

Trade-Offs of Wheel Speed

To obtain the highest angular momentum within a particular volume outline without excessive sacrifice of operating life, the operational speed of ball bearing wheels is usually limited to 24,000 rpm. When exceptional life characteristics are desired, speeds do not generally exceed 12,000 rpm. A few recent designs, however, are operating at 48,000 rpm, but reliability data for these speeds are insufficient to indicate life and instrument performance. On the other hand, considerable life data are available for 12,000 to 24,000 rpm; the lifetimes associated with these speeds range from 3,000 to 25,000 hours with the majority concentrating at 3,000 to 8,000 hours for 24,000 rpm and 7,000 to 12,000 for 12,000 rpm.

2.3.1 Ball Bearing Wheels

2.3.1.1 Positional Stability Through Preload

To obtain the highest possible mass stability along the spin axis, the wheel bearings are preloaded to the maximum level compatible with run-up time requirements and reasonable life at the speed of operation.

The same methods of preloading can be used for all three forms of wheel design mentioned. The methods are designated DB (back to back), DF (front to front) and integral-shaft DF, and the applicable load line directions and construction are shown in Figures 1 through 5.

The DB preload method is often chosen to provide best overturning moment rigidity and to compensate for increase of preload due to thermal differentials between inner and outer rings when inners turn hotter than outers, as is usually the case.

The trade-off between DB and DF is clearly related to the ultimate environment the gyro is subjected to. DB provides more ruggedness and thermal control while DF provides smoother operation (better parallelism) and instrument life.

One advantage of integral-shaft DF construction is that the inner raceways are ground on one element (the shaft), which reduces the possibility of non-parallelism*. If a DF preload is chosen for a gyro application it should always be integral-shaft for this reason.

2.3.1.2 Ball Bearing Stress Levels for Static & Dynamic Conditions

The maximum permissible stress levels for ball bearing materials can be calculated in several ways. The most conservative theory follows the basic Hertzian method, while more recent approaches consider stress over a broader surface area of the bearing race.[4]

The stresses on a bearing set are likely to change under dynamic operation. The level of preload under static conditions can increase considerably while rotating because centrifugal forces act on the wheel. By the use of interferometer laser techniques, this action has been studied on Saturn gyro wheels at the Marshall Space Flight Center; it was found that preloads in a particular design doubled under dynamic operation.

If the designer is unaware of this phenomenon he may initiate a safe static design condition where preload and superimposed environmental stresses are kept below a maximum of 200,000 psi (for materials of R_c 60 hardness) but where dynamic operation, including environmental inputs, may cause permanent plastic flow to occur. The latter, in turn, would result in poor wheel operation, mass shifts and a considerable reduction of life.

2.3.1.3 Lubricant – Stress Transmission Through Lubricant Film

Gyro designers have not always agreed regarding the extent to which stress levels in the bearing are compensated for or transmitted by the hydrodynamic lubricant film. Recent experimental work in this area[5] indicates that the stresses are largely transmitted through the lubricant film into the basic structure of the bearing. Therefore, it is of continued importance to design gyro wheel bearing structures to safely withstand these stress levels to avoid bearing fatigue failures (see paragraph 4.2.1.3).

Wetting – The Importance of Adequate Lubrication

Studies have shown that acceptable life characteristics are dependent on proper wetting of the balls and raceways by the lubricant; a fluid film must be formed over the entire ball path of the bearing races. Bearings with metal finishes that are "poisoned" (i.e., difficult to wet with lubricant) appear to have a short life. Figures 6, 7 and 8, which are photomicrographs of bearing race surfaces, show the finishes that can be obtained by various methods of lapping and honing. Despite their appearance, all of these finishes

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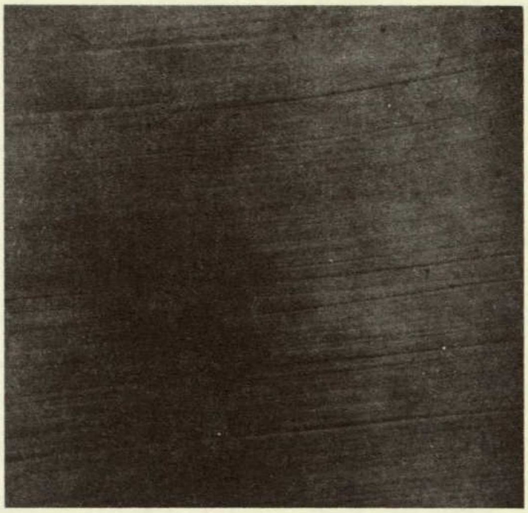
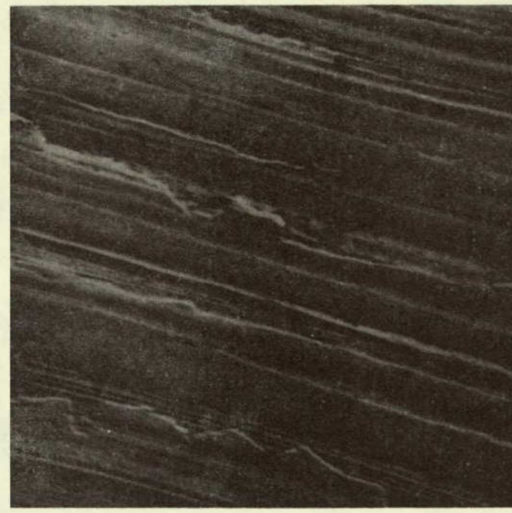
* This problem is further discussed in paragraph 2.3.1.5.



Source: NASA RASPO Office, AC Electronics, Milwaukee (Plate No. 68—122C)

Magnification: 7500X
Specimen: A-Shaft, String Lapped X26

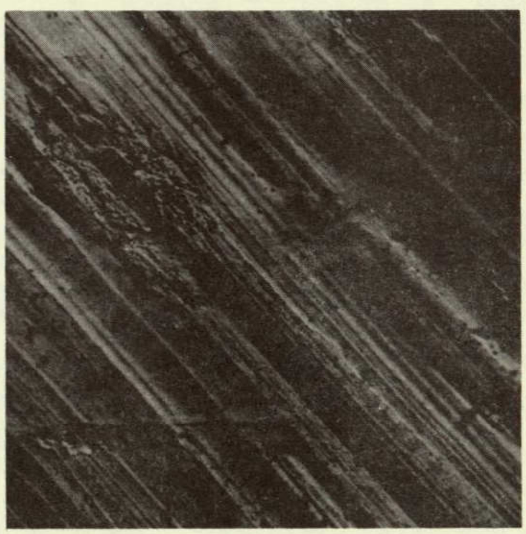
FIGURE 6 PHOTOMICROGRAPH OF STRING-LAPPED BEARING RACE (REPLICA TECHNIQUE)



Source: NASA RASPO Office, AC Electronics, Milwaukee (Plate No. 68—126A)

Magnification:7500X
Specimen: C-Shaft, Double Honed X32

FIGURE 7 PHOTOMICROGRAPH OF DOUBLE-HONED BEARING RACE (REPLICA TECHNIQUE)



Source: NASA RASPO Office, AC Electronics, Milwaukee (Plate No. 68—170C)

Magnification: 10,000X
Specimen: X-44/DB-538, Honed

Note: Intermediate finished area shows contamination on surface, which may be replica lap material. Observation is based on previous examination of bearing replicas lapped with soft, low-melting alloys.

FIGURE 8 PHOTOMICROGRAPH OF HONED BEARING RACE (REPLICA TECHNIQUE)

are compatible with reasonable bearing life if they can be wetted so as to support a hydrodynamic oil film. Of course, the likelihood of improved life and gyro wheel mass stability is further enhanced if that surface also has a minimum number of irregularities.

Kendall KG80 is the most commonly used lubricant for gyro bearings. (The older Teresso V78 oil is no longer available.) Before lubrication, the bearings are generally aged in tricresyl phosphate[6] (TCP), which improves the wettability of the surfaces and provides further protection during low-speed operation of the bearings (run-up and run-down), when the hydrodynamic oil film is more difficult to support. (See paragraph 4.2.1.3.)

2.3.1.4 Retainers

The two chief requirements of gyro ball bearings are low rotational torque and long life. These characteristics are strongly affected by the kind of ball retainer cage used. The retainer minimizes running torque by separating the balls and maintaining bearing set symmetry. Ideally, the nonmetallic retainer in a gyro ball bearing would be ball-riding to avoid rubbing contact on either race. However, the type of retainers used (cylindrical sleeves with through-holes for ball pockets) require location stabilization through close proximity to one of the races to minimize radial translation. Therefore, they are designed to be either inner- or outer-race-riding.

Bearings could, of course, be supplied with a full complement of balls in place of the retainer; with this arrangement, every ball could be in running operation, or every alternate ball could be in running operation with the intermediate balls serving as spacers. This might provide a sufficiently low torque drag on the bearing, but it is likely to result in poor life as a result of inadequate lubricant circulation.

Precision bearings for gyroscopes must have a circulating lubricant in order to have long life. The quantity of this lubricant must be minimized, and its center of mass must be stationary to preserve mass balance of the float and avoid drift changes. These objectives are met by the use of a porous retainer and hydrodynamic circulation of the lubricant in and out of the retainer.

The retainer is customarily outer race riding, allowing the lubricant to be absorbed by the retainer from the outer race where centrifugal action has moved the lubricant.

The porous retainer circulates the lubricant from the outer race to the ball pocket areas where ball rotation causes the lubricant to bleed out from the retainer pocket walls onto the balls.

Ball contact with outer race combined with centrifugal action recirculates the lubricant onto the outer race, allowing the cycle to be continuous.

This lubricant cycle, however, requires the selection of retainer materials, the machining of retainers and the process treatment of retainers to be complex and subject to the need of precise control to obtain desired results.

Synthane (paper base phenolic) and Nylasint (sintered Nylon) are the most popular materials used for gyro wheel ball bearing retainers and both are subject to composition, machining and porosity problems. From a technical viewpoint, the Nylasint is superior to the Synthane material, though the choice relates to the trade-off between performance and cost.

2.3.1.4.1 Synthane Retainers

Paper-base (XX Grade) Synthane® was the first long-life retainer material for gyro spinmotor bearings. The appearance of more controlled porous retainer materials (Nylasint®, see paragraph 2.3.1.4.2) over the last decade, however, would cause Synthane to take second place if the commercial market for Nylasint were to increase sufficiently to allow manufacturing to be repetitive and to controlled standards. Synthane, on the other hand, is a commonly used electrical insulation material. Its volume of production is such that a sufficient amount of the quality needed for bearing retainers can be found by suitable testing. Therefore, *given proper controls* in manufacture, treatment for use and inspection, Synthane remains nearly equivalent to more recent sintered nylon materials.

Gyro elements such as ball bearings, lubricants and materials such as beryllium, copper and alumina provide an almost negligible fraction of the total market for these products. It is not economically feasible, therefore, for vendors to develop specialized versions of these products to meet the requirements of gyro manufacturers. It is usually necessary in such cases for the gyro manufacturer to purchase fairly standard materials and components and, by imposing special processes and/or quality controls, to select or create from the total quantity purchased a frequently small fraction which can provide satisfactory performance. Synthane is this kind of material.

A mass of life test and field use data that indicates good life and performance characteristics supports this statement. One manufacturer, who uses Synthane in his inertial-grade gyroscopes, quotes a spinmotor bearing useful life of 4000 hours (see Figure 9), during which bearing reliability is 100,000 hours mtbf per motor set. These figures are based on over 2 million hours of field and laboratory tests, including stop-start cycles typical of field use. Applications range in severity from an R1-5 size bearing supporting a 24-gram rotor and preloaded to 1.75 to 2.5 pounds in a rather benign platform environment, to another R1-5 size bearing supporting a 7.5 gram rotor and preloaded to 1.75 pounds for use in a gyro which has successfully passed 600 g shock tests. Other gyro spinmotor bearings with Synthane retainers range up to R4 sizes. Many of these must operate in gyros which are not temperature controlled and are used in ambients covering the usual military range from $-65°F$ to $+160°F$.

In order to obtain superior life data with Synthane retainers under all conditions of stress, it is necessary to exercise controls on the procurement and processing of Synthane as described in paragraph 4.2.1.4.1 under "Recommended Practices."

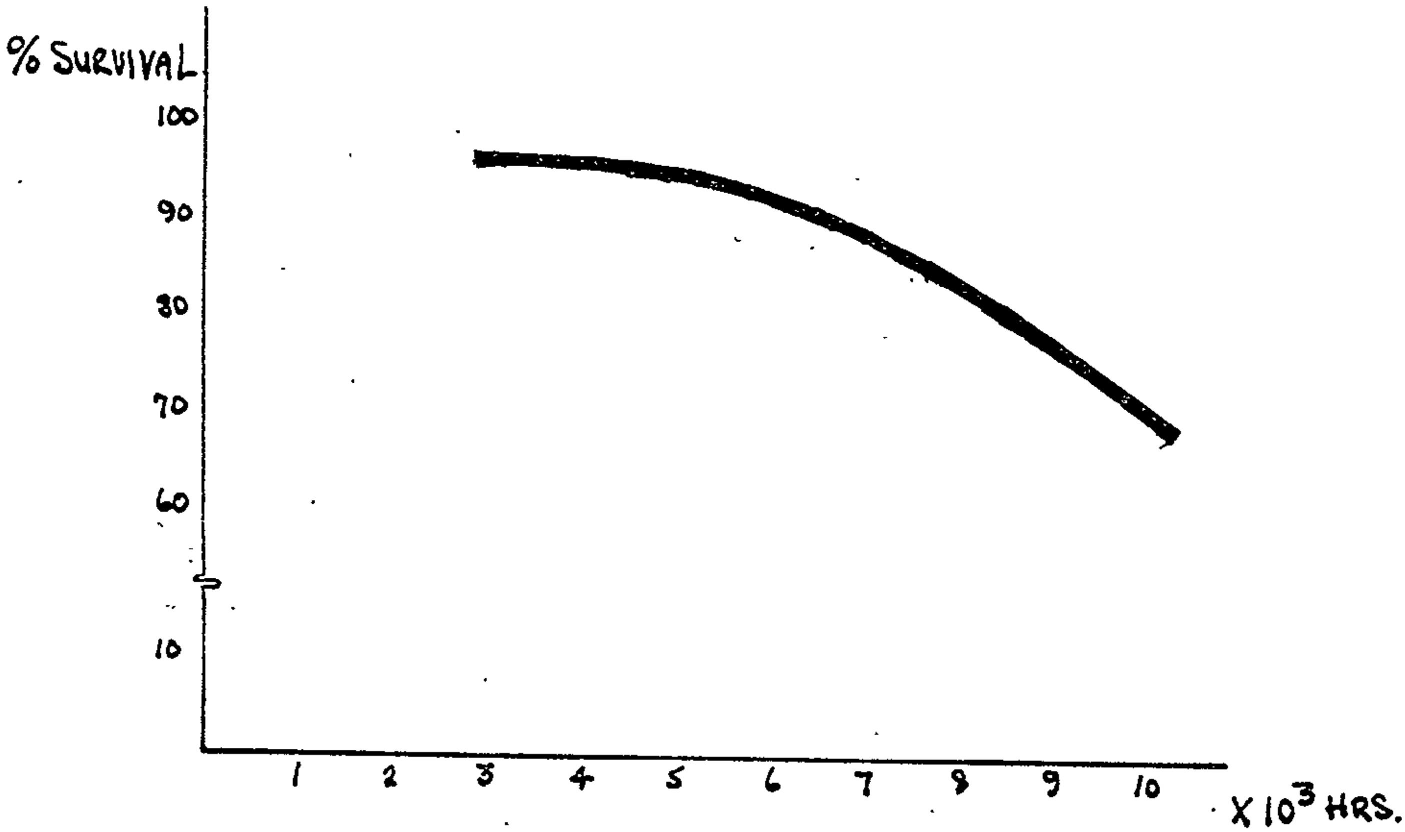


FIGURE 9. SPIN BEARINGS SURVIVAL CURVE

(Honeywell Data – Synthane Retainers)

2.3.1.4.2 Nylasint Retainers

The advantages of Nylasint over Synthane in gyro bearing retainers have been adequately presented in several MIT and other source reports. They have been particularly well described by C. H. Hannan of Miniature Precision Bearings, Incorporated, whose statements are summarized below.[7]

(1) Nylasint is precisely formulated to prescribed density, tensile strength, porosity and pore distribution. Once these properties are achieved in a given sintering lot of material, the properties of all bearing retainers produced from that lot are uniform. Synthane, on the other hand, is affected by many processing variations, and satisfactory lots are sometimes difficult to obtain.

(2) Oil can circulate through Nylasint retainers, as proved by experiments at MIT/IL[8] The non-uniform porosity of Synthane inhibits oil circulation within the bearing, which leads to lubrication starvation and early bearing failure.

(3) Other conditions in the bearing being equal (i.e., metal hardness, race finish, geometry, surface cleanliness and wettability), the Nylasint bearing can be expected to provide greater operational life than the Synthane bearing as a result of improved lubricant circulation. This is particularly true in high-performance gyro

applications, where the amount of lubricant is deliberately minimized to avoid oil jogs. On the Titan gyro, for example, AC Electronics' (Milwaukee) laboratory life tests and field experience confirmed a dramatic improvement in bearing life following the introduction of the Nylasint retainer. (Data Classified)

Nylasint, however, is not the panacea for every spin axis bearing problem. Its relatively high cost precludes its use in many applications, particularly where satisfactory life is already being obtained with Synthane. Furthermore, the Nylasint separator may exhibit unstable dynamic characteristics as a result of radial translation[9] of the retainer from the ball rotation contact in the retainer ball pocket (see also paragraph 2.3.1.6). To circumvent this, the ball pocket surface porosity is externally modified through salt-blasting until the surface porosity, as measured by the time required for a metered oil drop to be absorbed, has been reduced by a factor of 2 to 4 over the untreated areas. This process adds to the cost of the separator and introduces process control problems of its own.

Recent efforts to use a Nylasint separator in a miniature bearing have demonstrated that the requirements for processing such separators must be developed for each application at the expense of considerable engineering effort and experimentation. This effort may be prohibitive from the standpoint of both time and money, unless improvement in bearing life is of critical importance. The special processing includes:

(a) The degree of salt blasting required to achieve ball pocket surfaces that provide a dynamically stable separator,

(b) The extent of centrifuging and therefore the amount of oil retained in the separator, and

(c) The matching of separators to bearings to optimize ball-to-pocket and separator-to-bearing race clearances.

In summary, there are sound technical reasons for use of Nylasint as a bearing separator material, but the decision to use it must include consideration of the additional cost of material and the necessity to evaluate its stability under the particular conditions of the application. The trade-offs will probably favor Nylasint only where Synthane separators provide unacceptable life and/or performance.

2.3.1.5 Unwanted Torques from Raceways and Balls

Improvements in recent years have permitted closer control of surface finish of ball-bearing raceways and balls. Control of size and contour has been improved for precision bearings. Better design and greater care in manufacturing have helped to alleviate phenomenon such as ball wedging, which is caused by non-parallel paths in the opposing bearings of a set. Nevertheless, various kinds of obstructions can cause noisy performance and a correlation of unwanted torques that cause gyro drift.

Review of Noise Sources

The noise sources, and more particularly the variations of noise emanating from a rotating ball bearing gyro wheel, are the result of frictional rotational perturbations, which cause anti-rotational torques to act upon the gyro wheel. These torques, in turn, can have components that can couple into the precession axes of the gyro as a result of imperfect bearing metrology and axes misalignment.

Primary Noise Frequency

The spinmotor rotor, while it is in motion and locked into synchronism, will respond to mechanical perturbations as though it is a mass on a spring and will vibrate at a mass spring system resonance. This resonance point is at a low frequency (3-5 Hz) due to the large inertia of the rotor in comparison to the weak spring effect of the driving flux field. Therefore, any friction occurrences will cause a braking torque to be exerted around the SRA axis, and cause the rotor to lag and then lead the magnetic drive field angular velocity at the aforementioned natural frequency.

To appreciate the mechanisms of these perturbations (noise/torque sources), the following is pertinent:

Specific Torque Sources

The perturbations in rotating bearings can be caused by many sources.

(a) *Ball Wedging* – Balls are unequal in size and one is predominantly larger than others, causing a wedging action to occur at least once per revolution.

(b) *Lack of Parallelism* – During the preload cycle the curvature surfaces of inner and outer raceways are not necessarily perfectly parallel to their opposite set, due to lack of parallelness of spacers, a captured particle between preload faces, or imperfect bearing metrology. Ball wedging then results at least once per revolution.

(c) *Retainers* – Retainer translation, causing retainer whirl and squeal, which results in rubbing torques between retainer, balls and race. Retainer translation results from ball wedging in a retainer pocket.

(d) *Lubricant Channeling* – A temporary problem usually fixed by running-in time. However, it may repeat due to lubricant slump with temperature extremes and wheel in "off" condition. Some lubricant viscous torque variations will always be present and this results from lubricant circulation.

(e) *Contamination* – Particles that impede ball race rotation; they often are retainer debris.

(f) *Surface Finish & Lack of Lubricant Wetting* – So called "poisoned" race surface finishes do not support a hydrodynamic lubricant film. Poorly finished raceways are more likely not to support such a film and cause running torques.

Ball wedging and/or lack of parallelism or any of the other sources identified causes a torque in the ball race and the outer race (for outer race rotation), that may have a component of rotation torque on the gyro rotor angular momentum.

As such, then, there can be torque component coupling into either or both the IA and OA axes.

The mechanism of this residual torque is explained in its simplest form as follows:

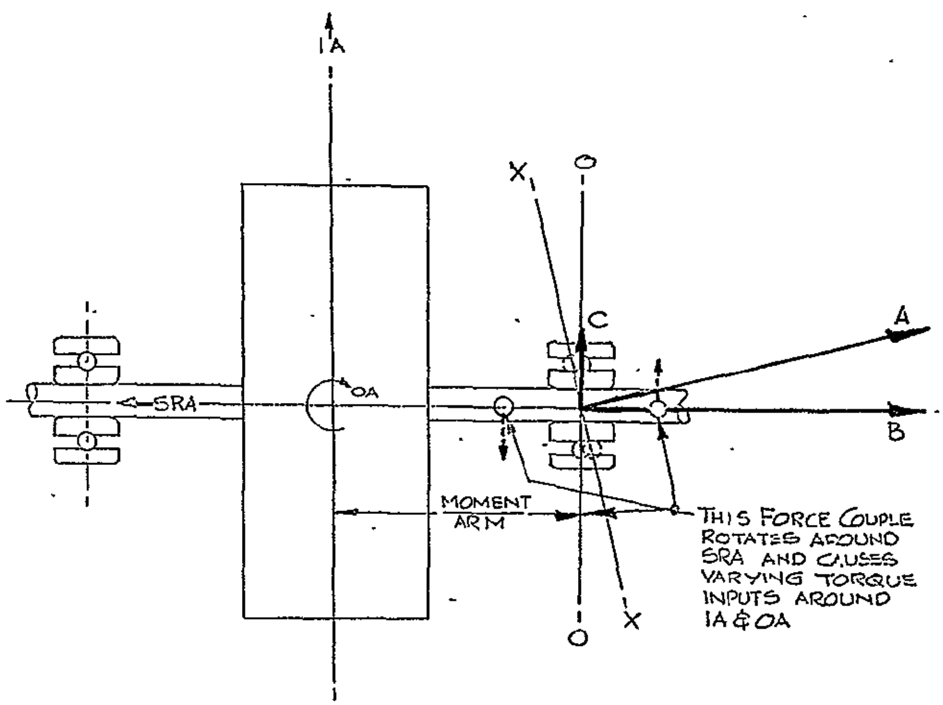


FIGURE 10. ROTATING TORQUE VECTORS

The ball races are not perfectly parallel to one another so that a plane X-X drawn through the contact points of the balls and the outer race is not perpendicular to SRA.

A torque vector "A" then lies ⊥ to plane "X-X," and component torque vectors can be projected on axes "SRA" and "OO," resulting in torque vectors "B" and "C." Torque vector B lies within SRA and has no effect on gyroscopic action or gimbal rotation. However, torque vector C is at right angles to SRA and is made up of two couple forces working around axis "O-O" and rotating around SRA.

These forces then apply a new torque around the input or output axis of the gyro wheel that is equal to the moment arm distance of the bearing location to the center (OA) axis times twice the single force of the torque component couple.

This torque input to the gyro wheel results in wheel precession and it occurs at various frequencies.

Torque Frequencies

The basic frequency is the resonant frequency of the wheel inertia acting with the spring effect of the rotating magnetic flux field.

However, since a ball wedging effect occurs at least once per revolution, another coupling frequency occurs at the wheel rotational speed. In some instances of gyro excitation, the mechanical rotation frequency of the wheel is the same as the primary excitation frequency of the signal generator (e.g., a wheel rotation of 24,000 rpm and a microsyn excitation of 400 Hz). This is an undesirable situation since the gimbal dithers approximately at the mechanical wheel frequency around the output axis and produces another 400 Hz signal.

Because these two frequencies are not in phase, they produce beats. The beat frequency is related to the mechanical accuracy of the synchronization speed, and the resulting output noise tends to appear in the 0-5 cycle region. Clearly, it is wise to separate the microsyn excitation frequency so that it is not coincident with the wheel rotational frequency or any of its harmonics.

Other frequency sources are the ball race rotation speed and one that equals the number of balls times the rotational speed of the ball race. Dynamic unbalance of the rotating wheel is still another source of gimbal motions at the second harmonic of wheel speed.

An explanation of some relevant terms might be useful at this point. A ball bearing consists of an outer race and an inner race. Both are rings with ground raceways that are finished to a high degree. A third element is the ball retainer which has pockets that retain and space the balls. The assembly of the balls in the retainer is referred to as the "ball race." Its speed or rotation is intermediate between those of inner and outer races. The formula to ascertain the ball race speed, and thereby the noise frequencies emanating from the ball race, is as follows:

$$\omega_b = \frac{\omega_h}{2}\left(1 \pm \frac{D_b - \cos \xi}{D_{pd}}\right)$$

where,

ω_b = ball race speed in rpm

ω_h = rotor speed in rpm

D_b = ball diameter

D_{pd} = ball race pitch diameter

ξ = contact angle

The ± sign stands for inner or outer race rotation design. The + sign applies to outer race rotation, which is the usual arrangement.

The higher frequencies that emanate from the ball bearing set (i.e., those equal to the product of the number of balls and the ball race rotation speed), can generally be ignored as they seldom cause performance problems and can readily be filtered from the output signal.

Coupling Axes

When the force vector of the torque is parallel to the OA, the residual torque input is around the input axis (IA) and causes normal rotational precession around the OA. Conversely, when the force vector is parallel to the IA, the torque is applied around the OA, and precession around the IA occurs. In between, the torque is coupled with both axes but attenuated by the appropriate cosine function.

Thus, the residual input torque can produce rotational precession via the gyro angular momentum around the OA, and, simultaneously, precession around the IA. The first form of precession causes normal rotational motions of the signal generator rotor around the OA axis. The second form of precession causes radial motions. Both of these rotor excursions result in a signal from the microsyn stator at the wheel rotational frequency. The variations of this signal are not necessarily symmetrical around the signal generator electrical null, so that torque inputs to maintain the null position can rectify causing drift summation errors.

Summary

The gyro wheel bearings are subject to unavoidable rotation friction that sources in:

(a) lubrication viscous friction

(b) ball wedging friction resulting from unequal ball sizes and/or lack of precise perpendicularity of the ball race to the SRA axis (lack of perfect parallelism between the opposing bearings)

(c) contamination

These friction torques cause wheel precession due to lack of precise metrology of the component parts as well as the assembly.

The wheel precession causes output noise at various frequencies of which the predominant (under 5 Hz) is difficult to filter out.

All the other frequencies are considerably higher and are normally avoided in the gyro output signal through appropriate electrical filtering techniques. Beat frequencies that result from wheel rotational frequency, ball race rotational frequency or ball race rotational frequency times ball quantity, versus pick-off excitation frequency may be avoided by appropriate divergent selection of pickoff excitation frequency.

The need for very precise pointing of precision optical systems has resulted in the use of very high gain servo loops; therefore, ball bearing noise up to several hundred Hz is now of concern in these applications.

2.3.1.6 Other Random Torques from Bearing Pair

Random torques can also result from irregular retainer motion and cause similar noise translations into the pickoff. The rotation of the balls within the retainer pockets can cause the retainers to be radially translated so that unconcentric motion (coning) can occur between the retainer and the ball sets.[9] At such times, a torque is felt in the whole rotation system which may have a rectified torque effect around the OA or IA axis combined with a power variation. The coning of the retainer causes a whirling effect and a high frequency rubbing contact between retainer and raceways which is an audible phenomenon (retainer squeal) that ultimately can cause retainer disintegration.

2.3.1.7 Raceway Finish

During the initial grinding of the bearing races it is possible to align or smear the metal fibers of the race surface as a result of the grinding speed, pressure, heating and feed. Furthermore, it is also possible to imbed microscopic carborundum particles into the race surface that ultimately affect bearing performance.

Customarily, a finishing process such as lapping or honing is used to obtain improved surface finish without raised metal or other disparities which are capable of puncturing the elasto-hydrodynamic oil film required for bearing operation. While a well finished bearing surface may also demonstrate improved wettability, this is as much a function of cleaning and surface treatment techniques as it is the basic metal finish.

The smearing of the initially ground surface or the possible imbedding of carborundum particles, however, is not necessarily alleviated by the subsequent finishing processes. Most bearing specifications unfortunately do not call for sufficient inspection of bearing parts under high enough magnification to observe the race finish (see Figures 6, 7 and 8). Inspection at sufficiently high magnification is likely to be considered too costly to be a standard production process.

Finishing techniques that are favored by bearing manufacturers include string lapping, stick or felt lapping, honing and ball lapping. The lapping technique uses an abrasive powder that has been absorbed in the medium which is held against the rotating raceway. Geometry control is difficult to maintain with these lapping procedures. Honing uses standard machine practices with shaped and abrasively charged materials. As is indicated by Figures 6, 7 and 8, the surface finish and the folded over metal surface structure is the result of machine honed race surfaces. Even though the geometry of the surface can be more accurately attained through such honing techniques the stick or hard felt lapping techniques are preferred to obtain best finish.

In rotary ball lapping a set of balls is rolled around the raceway by a cone-shaped spindle and the balls are of the proper size to lap the raceway to the precise contour desired. Other techniques of raceway finishing include etching of the raceway by electropolishing or other proprietary techniques. Aging in TCP also improves the surface conditions (see Paragraph 2.3.1.3).

2.3.1.8 Apollo Bearing Problems

A serious problem of premature ball bearing failure arose during the 1968-69 Apollo program. These failures occurred on the wheel bearings of the Apollo IRIG Block II spinmotor gyroscope, whose wheel package had a previous history of successful use on earlier Apollo and Polaris guidance systems. Although wheels of this design have a normal life of over 3000 hours, those on the Block II gyro failed at less than 100 hours.

The Block I IRIG gyro bearing procurement for the Polaris application did not include extended bearing life requirements. The Polaris gyros were neither tested as extensively prior to shipment nor were the life requirements large after shipment as compared to Apollo requirements before and after shipment.

Though these varying life requirements cannot be blamed for the very short life failures of the Apollo Block II disastrous occurrences it was realized that the wheel package design of the Block I gyros, in transferring the design to Block II, had not been fully evaluated from a life standpoint. Interestingly enough, however, it was found that Polaris Block I bearings, still available from old inventory, provided similar good performance as originally experienced on both programs.

Studies were undertaken by various organizations and laboratories to determine the cause of this large change in performance. These studies concentrated on the critical surface metallurgy of the ball bearing parts, the choice and use of the lubricants, the circulation of the lubricant in the retainer, the translational modes of the retainer, and the addition of lubricant-retaining lands to the raceway. (Figures 6, 7 and 8 show some typical raceway surfaces as viewed by replica electron microscopy.) Preload stress as well as the dynamic changes in this preload during wheel operation were also studied. Typically such studies are in direct correlation with a large part of the highlighted subject matter of this monograph.

Ultimately, wheel assembly processes were changed (mid-1969) and, as a result, the problems have disappeared without providing a finite single cause for the failure problem.

2.3.2 Gas Bearing Wheels

The gyro designer and user must initially consider the angular momentum requirements that relate to the desired accuracy. The largest possible angular momentum will provide the most accurate gyro for the available size. Similarly, the design shape considerations discussed in Section 2.3 are of equal importance for the gas bearing and must be evaluated with regard to gyro stability (accuracy), ease of manufacturing and assembly, and related costs.

The gas bearing offers some interesting alternatives to ball bearing gyro wheel designs. They include a sharp reduction of wheel perturbations typical of ball bearings (paragraph 2.3.1.5), a distinct improvement in drift errors, and increased life.

If materials are chosen to promote long life and reliability, the wheel can be run at higher speeds than is customary with ball bearing designs (12,000 to 24,000 rpm), because the lack of rolling contact in the gas bearing allows operation without wear. Some gas bearings, in fact, operate at speeds of 72,000 and 96,000 rpm. Speed limitations are imposed primarily by the centrifugal hoop stress capabilities of the materials used; the maximum stress is kept within the centrifugal explosive level of the materials.

The advantages of such high-speed operation are evident: (1) for the same size gyro and wheel weight, a higher angular momentum is available, and (2) for the same angular momentum, a smaller gyro can be built. The increase of angular momentum resulting from high-speed operation will cause the uncertainty torques from suspension/flotation sources to become a smaller percentage of precession torque in case (1) above. In case (2), the uncertainty torques are less in a smaller instrument and the angular momentum is unchanged. Again, the uncertainty torques become a smaller percentage of precession torque.

A combination of these two approaches permits an infinite variety of effects. An error analysis should be performed prior to the initial hardware phase to indicate the most opportune choice.

2.3.2.1 Materials and Processes

The choice of materials for gas bearings is of great importance. Various materials are in use today:

(a) Pyroceram® rotors on anodized beryllium (beryllia) shafts
(b) Hard chrome rotors on hard chrome shafts
(c) Hard chrome rotors on electroless nickel shafts
(d) Ferrotic rotors on ferrotic shafts
(e) Ceramic rotors on tungsten carbide shafts
(f) Alumina rotors on alumina shafts
(g) Lucalox® rotors on Lucalox shafts
(h) Beryllia rotors on beryllia shafts
(i) Lucalox inserts in beryllium rotors operating on aluminum oxide plasma-sprayed beryllium shafts.

While some of this variety may reflect a search for optimum materials, several of these choices are the result of trade-offs where considerations were related to material hardness, coefficient of thermal expansion and thermal conductivity.

In order to obtain sufficient bearing stiffness, gas bearings normally employ radial gap clearances of less than 150 microinches. Contaminants in this gap cause life problems, and this represents the major failure mode. Contaminants in the gap emanate from two sources: (1) gaseous matter may be drawn out of the interior of the float, particularly from epoxies and remnants of cleaning solutions in the crevices of joints and will condense in the journal because the pressure is higher there than in the float. The resulting deposits gradually reduce the gap until the available power is insufficient to start the wheel. (2) Gas bearing journals can also be contaminated by particulate matter that has been fretted out of the journal surface due to contact from stops and starts or overslewing of the gyro. The choice of materials and initial cleaning procedures are most important. Some materials fret very easily while others will generate enough heat on contact to weld together. Still others will score badly and form so many particles that the wheel cannot be restarted. (See paragraphs 4.2.2 and 4.2.2.3.) This problem can be minimized by proper choice of the organic materials within the float, the cleaning cycles, the vacuum baking cycles, the temperatures of such cycles, and the enclosures used in vacuum baking. (See paragraph 4.2.2.1.)

At least one operational instrument (i.e., a pendulous integrating gyro accelerometer) has been designed to use a Pyroceram rotor and an anodized beryllium shaft and thrust plates. The wheel weighs approximately 5 grams, axial and radial clearances are nominally 40 microinches, and the thrust pads have spiral grooves. When originally released for production, no boundary lubricant was specified. However, later results showed that a lubricant was needed, and a stearate material is now used. The instrument has achieved a very creditable MTBF, but over 50% of the failures are in the gas bearing assembly. Failed wheel assemblies usually show streaking of Pyroceram on the anodized beryllium parts. Although many failures occur on startup, many also occur during steady running operation. The latter type of failure is not typical of good gas bearing materials. The failure mechanism probably involves buildup of self-generated particle contamination. This design experiences very high angular rate inputs, due to its pendulosity, and is known to fail catastrophically in the event of high-speed touchdown.

Boundary lubricants, whose function in gas bearings is to provide low shear strength on the surface high points, can also cause contamination and must be applied with extreme care if buildup is to be avoided. However, if the bearing materials have sufficient excellence of surface geometry, the coefficient of friction may be low enough to make boundary lubricants unnecessary. One new material now being investigated (boron carbide) appears to be promising in this regard. It has self-polishing properties as well as excellent hardness.

Slewing ability is another concern of the designer. If the wheel has increased slew capability, it will withstand higher input rates without bottoming the rotor against the journal. In platform work fairly large slew rates can be applied to the gyros during erection, re-positioning, and g-testing. Not many of the gas bearings whose rotors bottom during slewing give satisfactory life and performance thereafter. For this reason platform designs must provide protection for the gyro by inhibiting large slew rates.

2.3.2.2 Construction

The shape of the gas journal is a matter that must be decided by the designer. Spherical, conical, and straight spool are the available choices (see Figure 11). Spherical designs are generally used in instruments with two degrees of freedom. Conical and spool designs are used for single-degree-of-freedom instruments. Control of alignment and spacing tends to favor the spool design.

Thrust plates with Whipple Spiral grooves or Pockets are the means whereby the bearing is customarily pressurized and whereby axial stability is obtained (see Figure 12).

In order to obtain levitation, particularly when operating in zero gravity or with SRA vertical, it is necessary to add grooves or pockets to the shaft or otherwise to lobe the shaft or rotor. These shapes will ensure a pressure differential across the shaft, which is necessary for levitation, regardless of the gravity environment or position.

Half speed whirl due to vibration at half wheel speed frequency is also minimized by these shaft treatments.

A part of the construction considerations concern the materials of the bearing and the float enclosure, which may have different coefficients of expansion. If they do, compensation must be made so that temperature variations will not cause mass shifts, which will result in drift errors. The shaft must also be prevented from microbending when it is clamp (pillow block) mounted in the float (see paragraph 4.2.2.3).

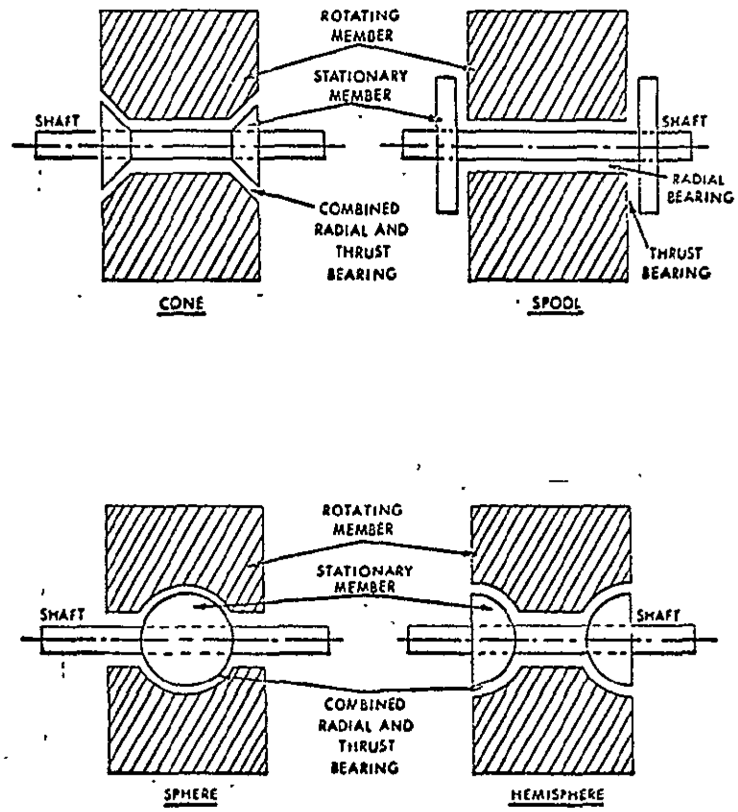


FIGURE 11. GENERAL CONFIGURATIONS OF SELF-ACTING GAS SPIN-AXIS BEARINGS

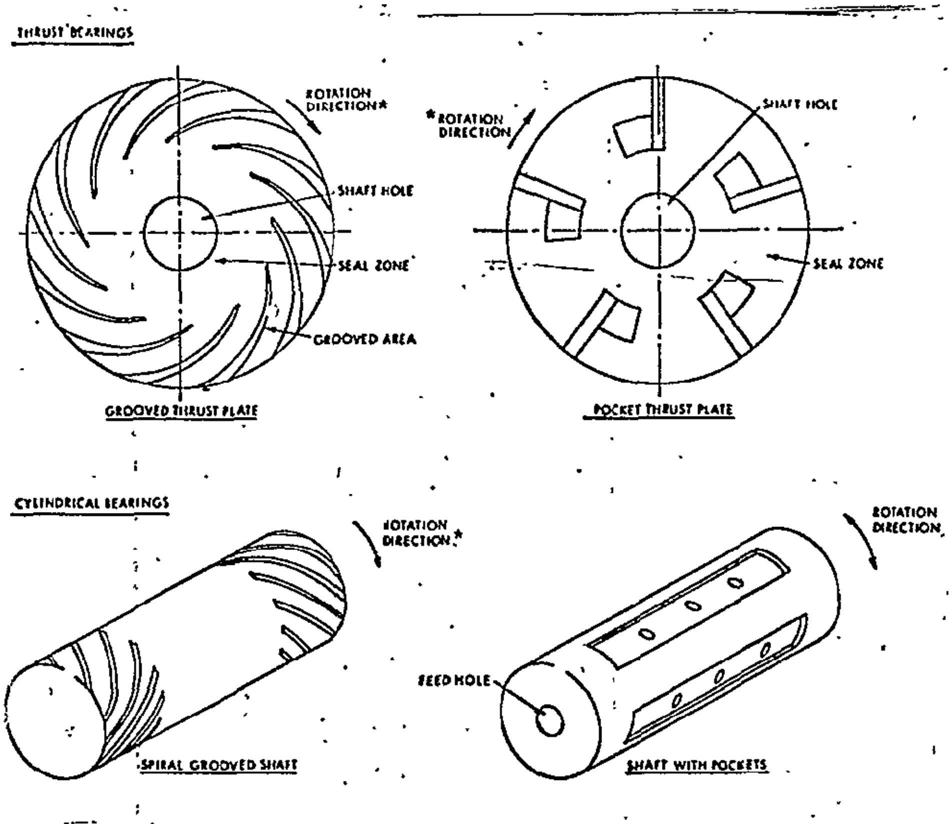


FIGURE 12. GROOVE AND POCKET CONFIGURATIONS

* Rotation directions shown are relative to the respective mating surface.

A problem that occurred on a gas bearing design for NIMBUS and related to the down-scaling of a larger design should be mentioned here. When making a miniature version of a successful design, one might assume that all components could be simply scaled down an appropriate amount. However, the available motor power is *reduced* in the process, and gaps are narrowed, and the likelihood that organics will be condensed in the journal gap and cause a motor start problem is increased. Therefore, greater cleanliness and elimination or complete enclosure (i.e., seal-in) of organic materials are necessary when a larger working design is down-scaled.

2.4 CASE-TO-FLOAT CONSIDERATIONS

2.4.1 Hermetic Seal Evolution

Hermetic sealing techniques have evolved in parallel with the overall development of SDF gyros (see paragraph 2.1). Hermetic seals on both the gimbal and the outer case are essential for the reliability of these precision instruments. Other gyroscopes, primarily aircraft instrument panel navigation aids, do not demand absolute sealing methods, not only because they are less sophisticated but also because they must be accessible for repair.

The performance of the SDF integrating gyros treated in this document would be seriously impaired if their hermetic seals were not absolute for the following reasons:

1. Fluid might leak into the float (gimbal), and change the position-sensitive drift characteristics.

2. The float ambient (an inert gas) might leak into the case-to-float fluid cavity, creating bubbles in the flotation/damping fluid. Both flotation and damping would be affected, while the bubbles would introduce a varying elastic restraint drift term.

3. Fluid might leak out of the case, or air might leak into the case. Results are the same as under item 2.

4. If the gyro uses bellows-controlled orifice damping, the damping constant would be seriously altered by all three of the above conditions.

The sealing techniques that have been used include, in chronological order, O-rings, epoxies, soldering, and welding.

O-rings (1940s and 1950s) were found to be unreliable until improved materials were available.

Epoxies were better than O-rings and were widely used, although epoxy materials and the joint design techniques left much to be desired in the 1950s.

Solder seals were a further improvement, though the lack of structural quality required developments in joint design to obtain both the hermetic seal and a secure bond. Quality and reliability problems were numerous, largely because most materials had to be plated.

Welding techniques have come into use during the 1960's, some gyro manufacturers have obtained high reliability with this type of seal. Material problems occur, as not all materials are readily welded.

During the early 1960's there was a swing back to the use of epoxies; control of thermal expansion became less of a problem when fillers made epoxy temperature coefficients approximately equal to that of aluminum. Emphasis is now more on joint designs that require a minimum amount of epoxy. Reliable hermetic seals will probably be obtainable by both welding and epoxies in the future.

With each of the above sealing techniques, designers must take many factors into consideration: joint design, expansion coefficients of adjoining materials, cleanliness of the surfaces that are being joined, performance of the seal in various environments, compatibility of the joint material with the flotation and cleaning fluids used, and the mechanical stresses caused either by unequal expansion of materials or by improper assembly techniques.

2.4.1.1 Epoxy and Their Seal Problems

The properties of epoxies are well suited to the environment of gyro applications, particularly with regard to vibration and shock. Nevertheless, epoxy joints have failed because of improper choice of gap size for the particular compound used, nearby soldering that caused localized epoxy decomposition, improper mixing and curing, or the presence of contaminants (e.g., flotation fluid) on the surfaces to be joined. In one instance the hardener for the epoxy was basically colorless, so that its addition to the epoxy did not provide a visual assurance that the hardener was added in appropriate quantity. A gyro design experienced large mass shifts as epoxied torquer coils shifted in position due to insufficient or complete lack of hardener in the "cured" compound. Repeated temperature cycling can cause joint failure when the thickness is large (several thousandths of an inch) and the expansion coefficients of adjacent materials differ considerably.

Fillers are used in epoxies to make their coefficients of thermal expansion more closely equal those of metal. Care must be taken, however, to avoid capillary leak paths formed by the connection of the minute filler particles; one technique is to use a gap of approximately 0.005 inches and a tight-fitting pilot shoulder to insure centering. Another solution is to use unfilled epoxies and to make the gap so small (0.0005 inches) that thermal expansion can be ignored.

Other than the gap size and possible plating requirements for soldered hermetic seals, joint designs for epoxied and soldered seals can be largely similar (see margin sketch on page 2-25).

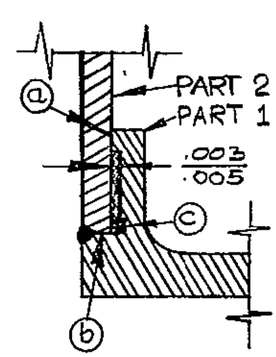
2.4.1.2 Soldering and Soldering Pitfalls

Solder hermetic seals also require special consideration of joint design. Because solder seals are not structural, stress conditions that may result from environmental inputs tend to cause a creep or cold flow of the solder seal joint. This, in turn, causes dimensional instability with associated mass drift instability when the solder joint is one of the float seals.

In order to obtain a measure of structural and dimensional stability in addition to a hermetic seal, the solder joint design must have sufficient interface and a controlled gap (0.003 inches typical). Further, a pilot diameter is needed for initial alignment and centering. In order to allow integrity assurance of the solder joint, it is wise to chamfer both edges of the mating parts to allow visual microscopic inspection of the solder bead that shows in the chamfered area and to reflow the solder if discontinuities are observed, regardless of the outcome of a leak check. Ultimately, of course, both the leak check and the appearance of the solder joint have to pass established criteria.

Problems and failure histories in solder-sealed assemblies often relate to the plating of the component part materials that cannot otherwise be soldered to one another. Entrapped acids used in the plating process will interact with materials, causing corrosion and ultimately contamination friction between float and case. The to be plated or pretinned surfaces must be very clean as lack of cleanliness will cause plating bubbles and entrapped acids.

Problems also occur with flux entrapment within the solder joint. To avoid corrosion from residual or entrapped fluxes that are not completely removed during cleaning, energized resin/alcohol fluxes are used in preference to acid flux. A solder-sealed assembly will successfully pass a helium leak test if capillary paths in the solder joint are filled with flux. However, at a later stage of assembly, or when the instrument has been field installed, the damping flotation fluids of the instrument can dissolve the entrapped flux and cause failure of the hermetic seal. Such leaks can cause a continuous drift in mass balance which is most difficult to diagnose if the fluid leakage is slow.



The sketch in the margin shows a typical float or case end seal. Part (1) has a pilot mating diameter at point (a), which centers it in Part (2) and squareness is simultaneously controlled through shoulder contact at location (b).

A flux cased solder ring is placed in groove (c), and to obtain a full outside bead, if needed, an additional solder ring is placed in the outside chamfered groove.

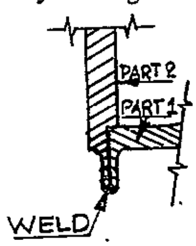
Float and case seals of this type are usually soldered by putting the entire circular joint on an induction heating machine for a few seconds. Though the resulting solder joint may be effective and reliable, other components in the float or case (epoxies, insulation materials, and wheel bearing lubricants) may experience partial decomposition and/or outgassing, as a result of this temperature shock cycle. Plating may also lift from overheated surfaces, causing contamination particles and subsequent gyro friction between float and case.

The induction soldering heating cycles require control of both time and power to minimize the overheating of nearby areas of either float and/or case, while obtaining the desired solder flow.

2.4.1.3 Welding

Welded hermetic seals have achieved a high degree of reliability and provide structural integrity. The joint is usually designed to permit two or three weld recycles in case internal repair is required. Mistakenly, however, some gyro manufacturers have avoided the use of welding for hermetic seals in the incorrect thought that repair and recycle of assemblies is seriously encumbered by welding.

Either of two welding techniques are used for hermetically sealing gyroscopes through welding; they are the fusion inert gas welding method and the E.B. (Electron Beam) method.



The fusion inert gas welding method uses a torch with a tungsten electrode through which a DC current is discharged (with a weld arc) to the piece to be welded which is at ground potential. The torch is annularly hollow so that an inert gas (Argon) may be blown by the torch tip and weld arc, displacing the oxygen of the air environment with an oxygen free gas. This keeps the weld beam from oxidizing and burning away while it is heated by the arc to its fusion temperature.

The E.B. (Electron Beam) method is accomplished inside an evacuated chamber where the to be welded parts are fixture held and rotated in front of a focused electron beam.

The fusion welding method first described is more convenient than the E.B. technique as weldments can be accomplished out in the open with the piece rotating in front of a fixed torch or with a torch that is rotated around the to be welded seam. Fixturing in either case is simpler than the fixturing required for the remote E.B. welding (inside a vacuum chamber) and the rotating torch method pretty nearly eliminates all fixturing other than the required chill members that are placed adjacent to the weld seam and which are necessary for both the fusion welding method as well as the E.B. welding method.

Though the point of fusion in any of these methods heats to a very high temperature (typically 2000°F), the chill members and the fact that only a pinpoint of material is being heated at any one time while the weld point is continuously advancing, causes the assembly to be only nominally heated within a 1/4 inch of the fusion point (typically less than 200°F).

This weld method does not introduce any additive materials (flux, plating or weld stock) and therefore provides a clean, hermetic and structural joint of great reliability. Welding of hermetic seals is, therefore, much favored by those organizations that have overcome the "permanency reluctance," i.e., the reluctance that welding implies a permanent joint, and inhibits repair recycling.

Secondary types of failure modes, i.e., failures other than the weld seal, can occur when operator skills or production tooling skills are inadequate. Improper application of the heat sink, rewelding without reapplication of the heat sink, excess current, or too slow a torch traverse speed, can cause local overheating with resultant possibilities of nearby epoxy decomposition, electrical insulation failures, and ball bearing lubricant failures (oxidation and decomposition of the lubricant).

The environmental capability of the welded seal has been excellent: temperature, vibration and shock have no effect on it. Moreover, chances of instrument contamination as a result of the sealing operation are vastly reduced in comparison to other sealing methods.

2.4.1.4 O-Rings and Their Compatibility Problems

Though O-rings were used as a sealing technique in the early designs, numerous instances of failure caused designers to be more application-conscious. O-rings tend to take a dimensional set as a result of temperature cycling and aging. This phenomenon is largely the result of a loss of plasticizer with time and temperature causing a lack of resilience of the material. Some materials, particularly Buna-N, incompatible with gyro flotation/damping fluids, can cause large-scale contamination and friction in the gyro as the damping fluid leaches the plasticizer out of the O-ring material.

Nevertheless, valid applications for O-rings remain in various gyro designs. O-rings are used for dust cap seals and often not across full ambient pressure differentials. Those that are exposed to gyro fluids are chosen to be chemically compatible. Resilient materials are now available that do not seriously compression-set with temperature (see paragraph 4.3.1).

2.4.2 Gyro Flotation/Damping Fluid Considerations

Ever since the development of SDF integrating gyros, the search has been underway for the ideal gyro fluid. This fluid would be of the highest density possible and would have Newtonian viscosity properties – i.e., its shear damping coefficient would be independent of the fluid velocity. Neither its density nor its viscosity would be affected by temperature; this would permit use of a less precise temperature control technique, or possibly none at all.

The fluid density is important because a denser fluid can support a reduced float volume of larger density. As with most aerospace components, there is a continuing emphasis to reduce the size of gyros without losing any accuracy. The advent of gas bearings allows higher wheel speeds so that angular momentum can be maintained while the float size and volume is reduced. To still obtain neutral buoyancy, fluids of increased density (greater than 2.0) offer the engineer a broader design scope. One other constraint, which becomes apparent fairly early, is that the fluid must be compatible with the other materials used in the gyro. With proper additives, present fluids can be made as dense as 2.35 gm/cc and still have desirable gyro characteristics. Such fluids, however, have to be thoroughly tested for compatibility with the gyro materials as they tend to be more corrosive (see paragraph 4.3.2).

In an integrating gyro, the total damping coefficient about the output axis must be large enough to provide the minimum input angle requirements of the system. This damping coefficient controls the gain (H/C) of the gyro. The damping constant (C) in many instances must be large enough so that large input angles can be accommodated without the gyro float reaching its limit stops. The damping coefficient is dependent upon the viscosity of the fluid and also on the gaps in which the fluid acts. If the float and case are cylindrical, the shear damping coefficient varies directly with the viscosity and inversely with the clearance between the float and the case. With these simple constraints it has been necessary in many cases to use very viscous fluids.

Fluids with the high viscosity required for simple shear dampers necessarily cause gyros to be very sluggish or have long settling times in the radial direction. Therefore, a conflict arises between high viscosity for shear damping and low viscosity for faster settling. One solution to this problem is to provide paddle or orifice damping. In these schemes the total damping is largely obtained by moving the fluid with gimbal mounted paddles through an orifice. This damping technique provides large damping torques with the use of low viscosity fluids. Together with the natural shear damping of the gyro (which is low due to the low viscosity fluid), adequate damping is obtained about the output axis (see paragraph 4.3.2).

In addition to providing short settling times in the radial direction, low viscosity fluids have other desirable characteristics. They appear to be more resistant to radiation as some are cross linked (fully bonded) without open chain connections. In view of today's radiation hardening requirements this is a desirable feature. They also make gyro filling much easier. The gyro can normally be degassed at lower pressures than when filled with a high viscosity fluid, since the danger of removing fractions of the fluid at varying pressures is greatly, if not entirely, removed when the fill time is short with a low viscosity fluid. Finally, voids (unfilled re-entrant cavities, etc.) are less likely to form with fluids of low viscosity.

Low viscosity fluids and short settling times in the radial direction cause the single-degree-of-freedom gyro, at least as a second-order effect, to behave like a two-degree-of-freedom gyro. Related problems and trade-offs are discussed in Appendix A.

Temperature controlled orifice damping is one interesting and state of the art technique to provide the necessary damping about the output axis with the use of low viscosity fluids. It has been employed on a number of gyros, some of which incorporate an orifice size control that functions over a large temperature range. One such unit permits adjustment of gyro damping by mechanical means without change of fluid. Reference 10 describes this damper, giving the basic mathematics of the model for gyro orifice damping as well as construction details.

Fluid contamination and the resultant performance degradation are common occurrences in fluid-floated gyros. This contamination can result from poor cleaning techniques, and chemical incompatibilities. As "sticky" gyros were a common failure mode in early ballistic missiles, "stiction" tests became part of the standard acceptance test for control system displacement gyros. In the early programs, failures were due to very gross contamination, such as solder balls and lint. They were performed to detect torques of the 5-10 dyne-cm level. The state of the art today is such that levels several orders of magnitude less justify rejection of units as "contaminated."

Bubbles represent a different type of contamination. They can cause gyros to exhibit variable mass unbalance, variable fixed torque, and/or large elastic restraints. Another consideration for bubble formation is the effect of low temperature on the fluid and the condition under which the fluid becomes a solid. Any lowering of the temperature below this level will enhance the formation of bubbles as well as place large stresses on gyro parts. At these cold temperatures the gyro bellows may be at the end of its stroke or could be "frozen in" by the solidified fluid. In either case, a further reduction of temperature will cause a vacuum in the gyro which, in turn, will liberate bubbles from gyro components and/or the fluid. To summarize: low-viscosity fluids facilitate filling and degassing, allow much lower temperatures (storage), and usually are more stable chemically.

2.4.2.1 Chemical Compatibility and Radiation Hardness

Friction hang up has often been caused by contamination resulting from chemical interaction between the gyro fluid and gyro materials. An example of this was the use of Halocarbon No. 208 fluid with polyolefin-insulated lead wires. The No. 208 fluid was chosen because it has a reasonably high density (1.84 at room temperature) and a viscosity/temperature index as low as that of silicone oil. Also, as it is chemically fully bonded (no open, unattached bonds), the internal bond strength of its molecule is more difficult to disrupt than an open-chain type of molecule, which can shed or take on free radicals.

The polyolefin wire insulation was chosen because it consists of irradiated polyethylene which can withstand higher radiation than other insulations. Furthermore, the fluid manufacturer shipped the fluid in polyethylene containers, so it was logical for the gyro manufacturer to assume that the two were compatible. Unfortunately, this assumption was incorrect. At temperatures above 150°F, the No. 208 fluid is a good solvent for polyethylene. In fact, it was later learned that the wire manufacturer had been purchasing the fluid for use as a cleaning (solvent) solution for his insulation extrusion dies! However, the action is so slow at lower temperature that no effect is noticeable while the container is on the shelf.

In the gyro, swelling of the wire insulation and freed particles of this material in the float-to-case gap caused friction hang up. As a result of this experience, the fluid manufacturer switched to glass bottles and the gyro manufacturer switched to ceramic-insulated lead wires. (Teflon-coated wire would also have been acceptable, though not from a radiation standpoint.)

Fully bonded fluids are attractive for their non-stratification properties, chemical inertness, and radiation resistance (hardness). This last property is important because radiation can break down a fluid into less inert fractions that react chemically with gyro materials. Contamination and stiction can then follow. A comparison of several representative gyro fluids with respect to radiation hardness is given in paragraph 4.3.2.4.

2.4.2.2 Viscosity, Density and Stratification

Depending on the damping requirements and whether the gyro uses only shear damping or a combination of orifice and shear damping, fluid viscosities ranging from 2 to 10,000 centistrokes (cts) are used in SDF gyros. Values exceeding 600 cts are not common, however, as fluids with very high viscosity are likely to have nonlinear (non-Newtonian) shear rates.

The viscosities of gyro fluids can be adjusted to the desired value by chemically bonding selected groups of chain radicals. However, because the same radicals are not necessarily added at all free bonds, a fluid blend of varying fraction density may be obtained. Although the average density of the mixture is usually as desired, thermal diffusion across the float-to-case gap may cause stratification of the fluid into layers of differing density. The smallness of the gap in most gyros accentuates this phenomenon. Some sections of the float may then become more buoyant than others, causing the gyro to develop unwanted flotation torques in various test orientations. Fluid stratification is a classical problem that has plagued many designs and relates only to fluid quality control and fluid selection.

It is possible to avoid the problem by very tight control of the fluid as purchased, through rigid inspection and certification (see paragraph 4.3.2).

2.4.2.3 Filling Techniques

Criteria

A single-degree-of-freedom floated gyro instrument can be carefully and successfully assembled, only to be "done in" by an improper fluid fill resulting in a gyro drift restraint problem (bubbles) or a friction/stiction drift problem (contamination). It is important to enforce control techniques during the fill cycle that ensure:

a. a good solid fill (no bubbles or voids)

b. a clean fill (void of particulate and chemical contaminants).

The following descriptive paragraphs provide basic information to method and technique:

Solid Fill

Filling a gyro with fluid requires that all spatial volumes become filled. Some of these volumes may be re-entrant cavities or have such minute access that filling may not be complete.

Particular locations of this type are the crevices in bellows assemblies, the interfaces between potting compounds and winding/laminations and the porosity of the potting compounds. During the hot evacuation cycle (prior to fill) these volumes may not have been fully degassed as a result of their labyrinth paths to the "open." Consequently, voids that come about in a gyro that has limited bellows displacement compensation produce air bubbles if these improperly degassed cavities bleed out during temperature cycles, particularly when the gyro is cold-stored beyond the bellows stroke capability which causes a vacuum in the instrument.

In order to avoid these problems, pressure is generally applied to the filled gyro. It is often applied by subjecting the filled unit, including a head of fluid above the gyro, to ambient pressure for an extended time. The pressure level is often raised above ambient through various artificial means (displacement techniques, inert gas pressure bottles, etc.). Care is to be used in the application of pressure as a rapid application causes a surge and a possible gas entry into the gyro.

Through the pressure/time technique all voids in the gyro are likely to be fluid filled and captive gas bubbles are likely to dissolve into the fluid.

The above-mentioned temperature cycling, however, can still produce gas bubbles in a working instrument if the bellows displacement has not been designed to range over a sufficient temperature range. If the gyro experiences a vacuum during such cycling, dissolved gas will be liberated from the fluid. It is therefore important to have sufficient bellows/temperature design range and further, it is important to have properly degassed fluid so that it contains little or no dissolved gas as it enters the gyro from the gyro fill stand. Bubbles will redissolve in the fluid but this may take a long time if the bubble is between float and case in a pancake shape which offers little (only edge) contact with the fluid. Restraint-type drift changes result.

Pressurization techniques include the application of fluid pressure through valving off the gyro plus a head of fluid above the gyro and exerting fluid pressure at elevated temperatures and for extended periods of time through the means of a diaphragm displacement valve and to monitor it with a pressure gauge.

Care must be taken during the pressurization cycle to avoid distortion of the gyro bellows. If the bellows outside is accessible on one end of the instrument case, it is feasible to fixture-support it.

Contamination Resulting from Fill Procedure

A gyro that is contaminated during the fill procedure cannot, generally speaking, be cleaned through flushing procedures. Contaminants are not likely to emerge from the instrument as a result of the many fluid path re-entrant cavities. It is often necessary to disassemble the gyro in its major subassemblies to allow thorough cleaning of each before a re-assembly is started. This is a costly procedure so that it is obvious that the fill procedure should be controlled to avoid the introduction of contaminants. Particular attention must be paid to fill stand design, processes, and material details (see paragraph 4.3.2.5).

Fill Methods

The most commonly used fill technique is *backfilling* for which gyros are designed with a fill port at each end. The fluid is stored in an evacuated reservoir, and the gyro is evacuated, generally to a much higher vacuum. The lower port of the gyro is connected to the reservoir by a valved line, and the upper port is connected to an overflow reservoir through a similarly valved line. The overflow fluid emerging from the gyro is often checked for contamination as this provides only indication of out-of-control fill procedures and/or a contaminated instrument.

Vacuum measuring gauges (Hastings) are installed in the system near the desired measurement location and are valved off individually so that measurements can be made when desired. This avoids contamination of the gauges with gyro fluid during the actual fill; it also keeps the gauges clean and outgassed so that they cannot cause contamination of the system, including the gyro.

Evacuation is usually done at elevated temperature (the ultimate gyro use temperature is often chosen) and for sufficient time to allow both the gyro and the filling system to be adequately degassed. Both the gyro and the fill lines are kept at this temperature during the evacuation and fill cycles. The fluid is also kept at this temperature and is stirred continuously and automatically to avoid the hydrostatic pressure effect that would keep dissolved gas from being liberated. The vacuum levels that the fluid may be evacuated at are to be carefully chosen and controlled to avoid evaporation of the fluid itself or of fractions of the fluid. The use of a condenser and a gravity return is advised to minimize this.

Gravity filling, which is the other method, basically differs from backfilling in that the gyro contains only one port for both evacuation and filling. It does not provide the same assurance of a "good" fill, as the contamination check of the backfill (through filling) technique is unavailable. (See paragraph 4.3.2.5.)

The contamination precautions (valving techniques, filters and material choices) described above for the backfill procedure are equally applicable to gravity filling. In general, gravity fill is used on the less accurate instruments because it is less complex and more cost effective. For the more sophisticated instrument, backfilling appears to be more attractive, perhaps even costwise in the long run.

Fill procedures, and the certainty of a good, gas-free fill, are enhanced by the use of low-viscosity fluids, which are easier to evacuate, transfer and filter.

2.4.3 Expansion/Contraction Bellows

The SDF gyro fluids have volumetric expansion coefficients that range around $1 \times 10^{-3} in.^3/in.^3/°F$. The average SDF gyro of miniature size (1.5″ to 2″ diameter x 2.5″ to 3″ length) contains a fluid expansion absorbing bellows that has an approximate effective area (effective piston face) of 0.750 $in.^2$.

As a result, the bellows will move $\frac{1 \times 10^{-3}\ in.^3}{0.750\ in.^2} = 1.33 \times 10^{-3}$ in./°F. If the gyro is to be storable over a 230°F range (-65°F to + 165°F), the stroke of the bellows must be capable of at least .306 motion if it is not to bottom out at the cold temperature extreme. Bottoming out would cause a vacuum void in the gyro with resulting bubble formation. As a matter of course, a stroke of 0.350″ is advisable, particularly if the operating gyro temperature is near the high end of this range.

In early gyro designs, when the specified temperature range was narrow, a gas-filled soldered diaphragm bellows of one or more convolutions was used inside the gyro. The bellows was placed in a cavity, where its movement could not result in float contact. It was shaped either like a donut, with solder seams inside and out, or a circular gland convoluted to allow gas enclosure. The compression of the sealed-in gas provided pressure return when the temperature dropped. In some designs the bellows was totally free in its cavity without attachment to the gyro case structure. This caused undesirable fluid torques in the instrument as a result of acceleration-dependent bellows motion.

A more common form of bellows today is the piston type. It is designed to form a hermetic seal at one end of the gyro case. The bellows is usually arranged to compress when the fluid expands, but in some designs the opposite occurs because the fluid is inside the bellows. From a total fluid displacement standpoint, the latter is preferred over the compression version, because it uses a shorter stroke bellows to displace the same volume of fluid, allowing a shorter gyro length.

Both types of piston bellows have been used to actuate shutters that keep the orifice damping nearly constant as fluid viscosity changes with temperature.

2.4.3.1 Construction

Regardless of the way they are applied, bellows are manufactured in several ways: hydroforming, electroforming, and fusion welding.

Hydroformed Bellows

In hydroforming, a thin-walled metal tube is hydraulically forced into a separable convoluted mold. In its crushed version, the hydroformed bellows is simply pressed into a more compact form. In either case, the metal is highly stressed when used for extended strokes, so that seal reliability is not high. Due to high stress use, this type tends to have excessive positional hysteresis and is therefore unattractive for orifice damping control. Soldering is generally used to make the hermetic end seals or case attachments of hydroformed bellows.

Diaphragm Bellows

The earlier gas-filled bellows gland was made by edge-soldering two convoluted diaphragms together. This fluid absorber was modified for extended temperature range by placing a hydroformed bellows between the end diaphragms. Fusion-welded versions of this design are more typical of the present state of the art.

Electroformed Bellows

In electroforming, a bellows with continuous convolutions is electro-deposited (plated) on a removable mandrel. Because of its thin gauge (typically .001″ to .005″), this type of bellows is subject to lower stresses than one that is hydroformed. Unless the plating is under exacting control, however, its thinness is conducive to porosity and failure of the hermetic seal occurs when in motion. Its uses, therefore, have been largely in access port seals that normally are stationary.

Fusion-Welded Bellows

The fusion-welded bellows is made from a series of annular metal rings that are welded together alternately along their inner and outer edges. It can be made from lamination stock, usually convoluted, as thin as 1.5 mils for low stress levels in particular configurations. Fusion welding provides greater hermetic seal reliability than any of the other constructions, regardless of shape or size. The bellows can be a sealed, gas-filled unit or a piston type that is welded to the outer case.

The laminations of a fusion-welded bellows are in flat contact with one another at both the ID and OD welds, and the sharp "V" formed at each junction is a likely site for failure if contaminants are present. Therefore, the initial weld assembly and subsequent cleaning and vacuum baking cycles must be done with stringent control against contamination.

The use of welded bellows in otherwise soldered or epoxied instruments raises the possibility of flux or epoxy particle contamination in these crevices when the assemblies are recycled through the assembly line. Condensation of flux deposited from nearby soldering operations (lead wires) in the bellows crevices and particularly the use of epoxy softeners for disassembly purposes has caused contamination in the gyro with stiction drift characteristics, stress corrosion, and subsequent hermetic seal failure.

The welded bellows behaves much like an ideal spring; it can, therefore, be depended upon to repeat its position at particular temperatures, making it attractive for controlling an orifice damping shutter. All-fusion-welded construction, not only of the bellows but of the entire gyro, is desirable because of its high reliability and the fact that the process produces no particles or condensing gases that cause contamination.

2.5 FLOAT HYSTERETIC AND RANDOM TORQUES

2.5.1 Background

Drift torques in SDF gyros have many different causes. The minutest uncertainty torque acting on the float will cause a drift to occur. The drift resulting from a given torque will obviously be smaller at higher values of angular momentum. Some of the more important sources of uncertainty torques are these:

1. Wheel shift along the SRA axis (ADIA)
2. Suspension torques
   a. Pivot/ jewel friction
   b. Pivot/ endstone friction
   c. Center of rotation change during gyro input motions around the output axis
   d. Center of buoyancy/gravity torque changes
3. OA reversal torque changes (ADOA)
4. Transducer restraint torque changes
5. Flexlead restraint torque changes

Each of the above sources is discussed in greater detail in subsequent sections.

Normally, when the gyro is first put on test, all the identifiable error sources are measured so that an opposing summation bias torque may be applied around the output axis to the float torquer; thus, canceling these torques and keeping the gyro at its transducer null position under zero input conditions.

If the gyro is used in rate mode, torquer currents of positive or negative polarity are added to the initial bias current to keep the float at the transducer null. The magnitude of this additional current input to the torquer is therefore an indication of rate.

If the gyro is used in platform mode, the platform torquers will drive the platform gimbals so that the gyro transducer null is maintained, providing displacement attitude information from the platform gimbal rotational position transducers.

In either case, the gyro torquer bias current is continuously applied or in some instances modified to compensate for environmental inputs, with variations programmed from earlier test data for that particular unit.

2.5.1.1 Acceleration Dependent Input Axis Drift (ADIA)

Movement of the center of gravity of the rotating wheel along the spin axis is a common source of random torque change. Since the wheel is the most massive component of the total float mass, instability of its position can cause considerable drift changes in gyro mass balance.

When the CG of the wheel mass shifts, a gyro drift change occurs resulting in a mass balance drift in the IA axis (ADIA). Any of the following conditions can cause this shift:

- A change of ball track resulting from wear or environmental inputs. This shift is usually associated with a preload change.
- Variations in oil film thickness, either in average film thickness as from run-up to run-up, or transient variations as with oil jogs.
- Built in stresses of wheel component parts relieve themselves with the passage of time and subjection to temperature and other environments, causing dimensional change of wheel parts.
- A thermal gradient in the wheel, resulting from the spinmotor stator heat flow to the gimbal shell via the bearings and shaft.
- Positional instabilities in motor stator windings, iron, and potting material with thermal gradients and time.

When gas bearings are used in the gyro wheel, we can be certain of avoiding all the above bearing-oriented problems that beset the ball bearing wheel. However, the last three of the four items listed have their effect on gas bearing ADIA also.

Thermal coefficient matching and symmetry are very important for both types of bearings. Since gas bearing wheels are often made of specialized materials, dimensional stability is a critical consideration if they have different coefficients of expansion from those of the basic float materials. Suggested methods of compensating for such differences are given in paragraph 4.2.2.3.

2.5.1.2 Suspension Torques.

2.5.1.2.1 Pivot/Jewel Friction

Gyro floats with jewel/pivot suspensions along the output axis must have exact neutral buoyancy end to end as well as across the float diameter to obtain minimal friction at the pivots. When there is a net positive (or negative) buoyancy, the pivot exerts some force against the jewel. This creates a friction force opposing float precession. This friction torque is the product of the friction force and the pivot radius. In addition, at the onset of float precession, the center of rotation appears to be at this pivot/jewel contact point rather than on the centerline of the pivot. If the pivot is also touching the endstone (which limits the travel of the float along the output axis) erratic torques can exist because the two centers of rotation of contact may not be aligned. See paragraph 2.5.1.2.2 below and Figure 13.

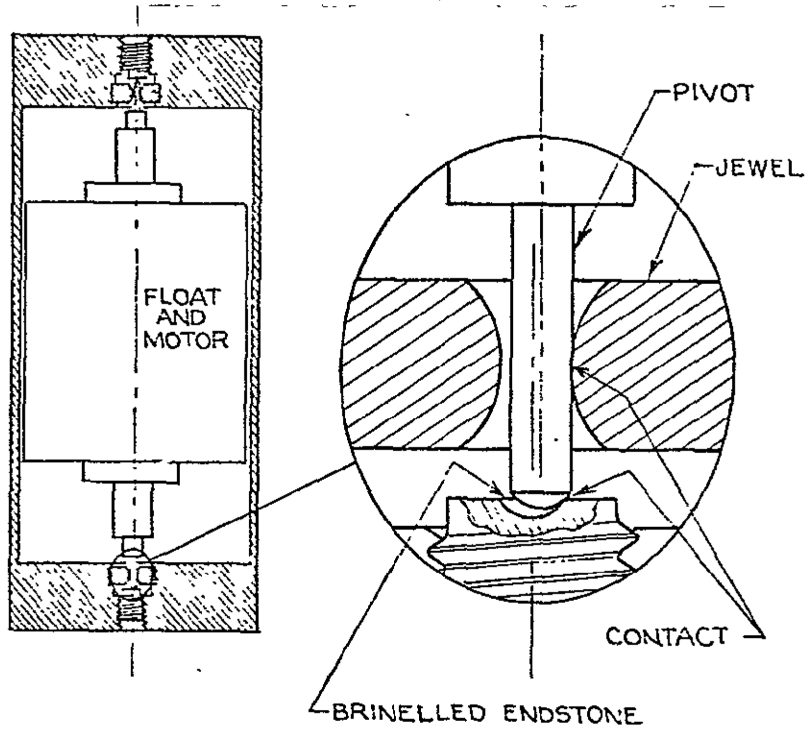


FIGURE 13 – TORQUE RESULTING FROM ENDSTONE DEFORMATION

2.5.1.2.2 Pivot Endstone Friction

Centering the float along the output axis as well as adjusting the end play along the OA, is customarily done with an unfilled gyro by inverting the unit from OA down to OA up and back. Without the buoyancy of the fluid, the full weight of the float is on the endstone; as a result, a dimpling or brinelling of the endstone may occur, and/or the pivot end may be flattened. If sapphire endstones are used, they may be cracked or fretted by the pivot.

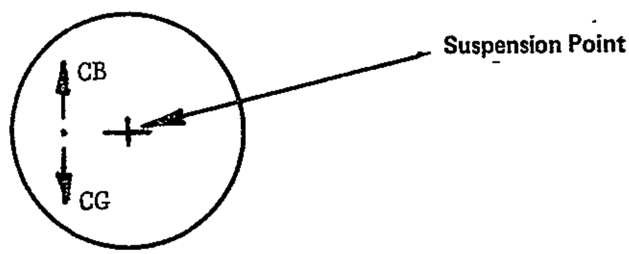
This type of damage, which occurs when the gyro is fully assembled and dry, is not observable until the unit is filled and tested. It will then manifest as erratic position sensitive torques, because precession opposing friction drift torques will be generated when the pivot contacts the jewel and the edge of the endstone deformation simultaneously (see Figure 13).

2.5.1.2.3 Center of Rotation Change for OA Inputs

Input motion that is not exactly around the IA or SRA will cause the pivots of the float to precess radially across the clearance diameter of the jewels. Since contact points will then be diagonally opposite one another, the float's initial rotation axis will be at a slight angle to the real output axis. Friction torques result similar to those described under paragraph 2.5.1.2.1 in addition to angular misalignment from the true output axis, causing cross-axis coupling changes.

2.5.1.2.4 Center of Buoyancy/Gravity Torque Changes

Three factors are involved in the flotation of a gyroscope float: The center of gravity (CG), the center of buoyancy (CB), and the suspension point. Both the CB and CG react with the suspension point to form a couple and a resultant torque. At any given temperature (hence flotation fluid density) the sum of these two torques is the net total torque.



As can be seen from the illustration, it is possible for these two forces to cancel (at one temperature) even if they do not act through the suspension point. A change in temperature, however, will cause the buoyancy force to change in magnitude, which will produce an error torque to the float. To minimize this effect the center of buoyancy should be located at the suspension point. Thermally induced changes in the magnitude of the buoyance force will then be canceled by the suspension without the generation of a couple.

2.5.1.3 OA Reversal Torque Changes

A gyro float assembly can act as a hydraulic piston when the OA is inverted with respect to thrust acceleration or gravity, if the float is not in a state of exact neutral buoyancy. If the unit is not completely symmetrical, the asymmetric fluid flow will exert hydraulic torques upon the float. The resulting gyro drift will vary as a function of settling time after inversion of the OA.

Regardless of achieving neutral buoyancy, fluid motion resulting in float motion and torques will occur if the design contains two bellows – one at each end of the case. When the instrument is inverted (OA up to OA down) the fluid/float mass sees a 2 g change in acceleration. The extent of motion is determined by the force displacement summation of bellows and axial suspension if present. The rate of the motion is determined by the axial damping characteristics.

This motion and resulting drift can be inhibited by placing a bellows of sufficient stroke at only one end of the instrument; however, a trade-off disadvantage must be realized i.e., it could result in an increasing susceptibility to axial motion resulting from asymmetrical thermal end-for-end conditions.

The inversion effect can be demonstrated by either of two methods. One is to mount the gyro on a drift table so that its SRA is along the polar axis and then cause the gyro to tumble with respect to gravity about the SRA. The result is shown in Figure 14 for two different gyros.

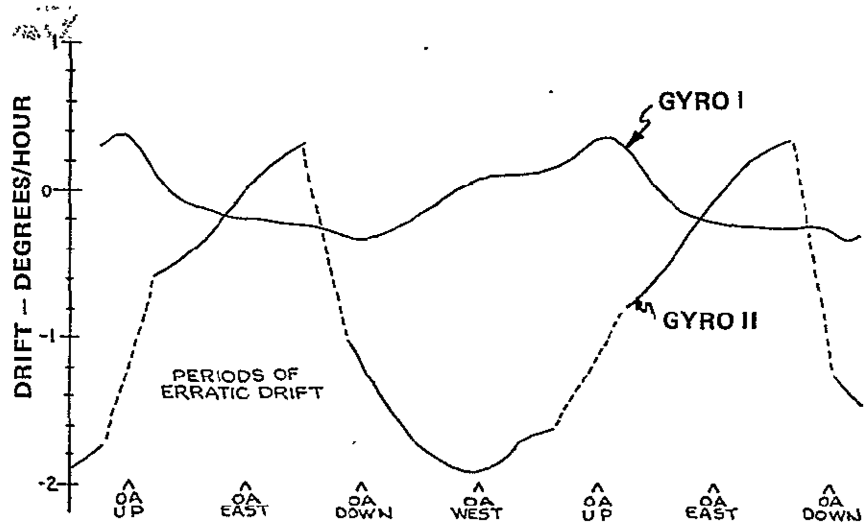


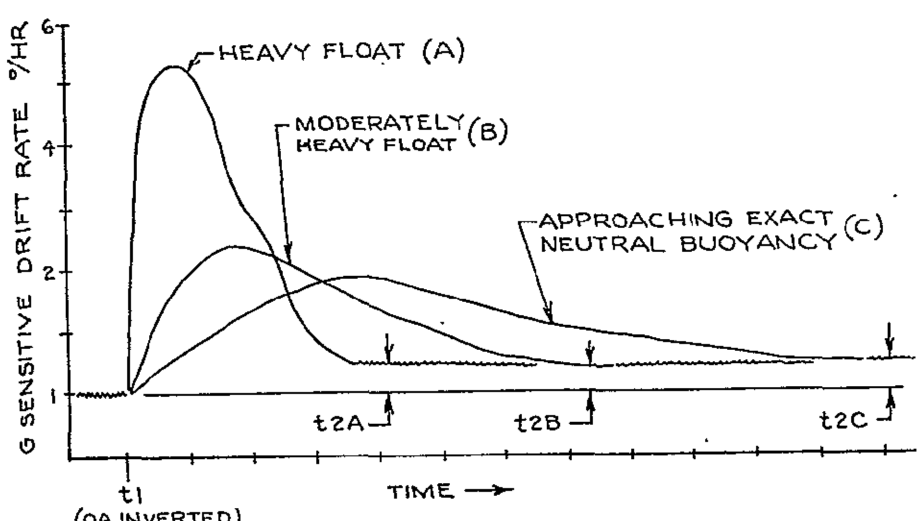
FIGURE 14. SRA TUMBLE TEST

Gyros I and II are different in construction, yet both are in the same general drift and size category (1.5 to 2 dia. x 3 to 4 lg).

The variation of drift with OA position when tumbled around SRA, which is aligned with the polar axis to minimize earth rate inputs, for Gyro I indicates repetitive torque variations due to some asymmetrics as the float translates along the OA axis.

The variation of drift for Gyro II and the erratic drift periods immediately after inversion indicates considerable lack of fluid flow symmetry, causing hydraulic torqueing during float piston action along OA.

The second method is simply to mount the gyro in a cubical fixture with an analog "torque to balance" closed loop about the gyro. After drift stability has been reached with the OA up, the OA is rotated about the SRA to the inverted position, and the drift rate is plotted as a function of time. Figure 15 illustrates the application of this method to a typical gyro (Gyro II of Figure 14). Three tests are shown: in curve A, the gyro is maintained about 1°C above neutral buoyancy, so the float is less buoyant and settles relatively quickly; curves B and C represent successively lower temperatures and closer approaches to neutral buoyancy. Points t_{2A}, t_{2B}, and t_{2C} indicate the respective settling times after inversion.



1. The drift at t_{2A}, t_{2B}, and t_{2C} is the effect of OA inversion after settling times from t_1.

2. Area under curve A≅ area under curve B≅ area under curve C.

3. The area under the curve (drift rate x time) represents a drift error of approximately 0.5 to 2.0 degrees, depending upon specific gyro.

4. This effect creates a multitude of uncertainties depending upon: The drift rate change before and after inversion; the period of time from first inversion to re-inversion; and the degree of buoyancy. Exact buoyancy is difficult to obtain, and a degree of inversion sensitivity always exists.

FIGURE 15 – OA INVERSION–SETTLING TIME

The inversion sensitivity of these gyros was improved by the following measures:

(1) Holes in a shielding disk, within the fluid path, were relocated and made more symmetrical to reduce the effects of torques produced by fluid flow during "dumping" of the float.

(2) Gimbal end play was reduced, and the temperature was adjusted to make the buoyancy closer to neutral rather than controlling H/C.

2.5.1.4 Transducer Restraint Torque Changes

Restraints are Excitation Dependent

The transducers most commonly used in SDF gyros are microsyn pickoffs, microsyn-type torquers, permanent-magnet-type torquers, and suspension elements such as ducosyns. These electromagnetic devices have a particular restraint characteristic at a specific level of excitation. The slope of this restraint torque is generally linear with angle, but the torque varies as the square power of the excitation change. Similarly, the magnetic attraction between the rotor and stator of these electromagnetic transducers varies as the square power of the change in excitation, causing a greater friction torque to be associated with pivot/jewel contact in the radial (across the gap) direction.

The primary winding of the microsyn pickoff is customarily excited with an AC current, and the voltage induced in its secondary (output) winding depends on the rotor angle. To minimize the elastic restraint characteristics of the microsyn, excitation is usually kept at the minimum useful level, and in some instances special pole shaping is used to further reduce the elastic restraint term.

The microsyn-type torquer is customarily operated with one winding excited at a specific AC current, while the other winding is excited with the control current. The resulting rotational torque is directly proportional to the control current. However, the restraint characteristics of the microsyn torquer vary with the square of the change in current of each winding, which can cause a double squaring effect if current fluctuations in each winding are in phase.

The permanent-magnet torquer is a much better device for torqueing the float. Neither its restraint characteristics nor its magnetic attraction characteristics vary with current fluctuation; it produces only polarity torque for polarity current. Nevertheless, drift can be caused by mechanical environmental inputs, undesired current fluctuations, or change of permanent magnet characteristics with temperature and time. Additionally, to avoid magnetic positional hysteresis, the flux path must be completed through a return path that is stationary with the magnet. (If the magnet is in the float, the return path must also be part of the float.)

In ducosyns, radial suspension forces are established by excitation of both windings. The earlier comments regarding the relationship of elastic restraint to the square of the excitation currents is equally applicable. Rotational torque changes will result from excitation level changes. Axial suspension is accomplished through a cone-shaped design of the ducosyn stator/rotor elements.

All of the transducers together in the gyro will produce an integrated elastic restraint term at the float null position, which is normally biased out (in combination with other torque sources) by the torquer. Therefore, gyro drift can be caused by changes in any of these elastic restraint terms of each of the transducers. For this reason, accurate excitation sources are essential to avoid current fluctuations and corresponding drift torque.

2.5.1.5 Flexlead Restraint Torque Changes

Flexlead Arrangements

The flexleads conduct electrical power from the case to the float to excite the gyro spinmotor as well as the float torquer windings, if these are a part of the float. Two basic arrangements are used:

(1) An interleaved group of crescent-shaped leads perpendicular to the output axis, or

(2) A group of coiled leads reaching from the float to the case, more or less parallel to the output axis.

With each approach, the attachment point to the float is as close as possible to the float centerline to minimize flexlead torques. There may be a residual amount of torque from the flexleads at the float null position that is biased out through the float torquer; however, since the flexleads are made of metallic materials, they could act as a spring with hysteretic properties and produce an unwanted drift torque. The residual stress conditions of the lead, particularly at its attachment point, play a role in this uncertainty.

Though flexlead torques may be biased out in reference to the float/signal generator null position, there is a finite elastic restraint term associated with the flexleads. This term adds to the drift characteristics of the gyro when the float moves appreciably from its null position. Consideration, therefore, must be given to the loop characteristics of the electronics with the possibility of programming counter torques in direct relation to float angle.

Baffles

Flotation fluid flow past the flexleads (caused by convection or float rotation) can transmit unwanted torques to the float. For this reason it is customary to envelop the flexleads with baffles. It is also good practice to shield the flexlead (i.e., keep the baffle trough at the same potential as the flexlead) to avoid electrostatic attraction between the flexlead and case ground. Such attraction causes unwanted drift torques.

Attachment

The method of flexlead attachment to the float and case terminals must be stable, because movement of the connections will change the restraint bias. The flexleads can be attached by mechanical means (a clip), soldering or tweezer welding, but all require considerable care. Failures are often caused by contamination in the form of solder balls, rosin flux or burrs scraped from the terminals through the application of the clip.

Materials

Though the flexlead is customarily made of a ribbon heat-treatable material and positioned in the gyro so that its stiff section is at right angles to its flexing motion, gravitational attraction can cause it to bend and express unwanted torques to the float. The choice of flexlead material and its density is vital in this regard. Silver copper 85-15 and beryllium copper alloys are popular flexlead materials, though their application in a high "G" environment is not recommended. In such applications, aluminum ribbon leads are used (though they are likely to cause more hysteretic behavior) because their density is much closer to the fluid density. For connections to be soldered to such leads, however, the ends must be plated; this can result in plating/lift-off contamination, positional hysteresis and drift torque problems.

Problems

Other common problems are electrical shorts (circuit to circuit or circuit to ground) and the breaking or kinking of flexleads due to fluid solidification and cracking and/or gyro precession while the fluid around the flexlead is not yet fully liquid subsequent to a cold cycle.

3. CRITERIA

3.1 INTRODUCTION

The design of a single-degree-of-freedom floated integrating gyro involves many tradeoffs which are directed at the best performance for a given application.

Even though a gyro design can yield the performance required, design and manufacturing criteria must be established to insure that each instrument made does indeed do its job.

Within these criteria lies the understanding of the application profile: Will the instrument be used for very short life spans, or will it be needed for a voyage to Venus, where the use of the on-board guidance can be repetitive, accumulating both life and on-and-off cycles for flight path corrections, perhaps including continuous operation. Some considerations may be in complete opposition to others, such as the environment that the instrument experiences during launch versus the environment it sees during space flight. It is important that simplicity and reliability be a part of every design goal to assure that design versions of the instrument be capable of meeting diverse requirements.

With these considerations in mind, the designer should review the construction of his instrument, including the planning of reliable assembly processes. The following section lists some of the design considerations that apply to instrument accuracy and reliability.

3.2 GYRO WHEEL CRITERIA

3.2.1 Angular Momentum

Within the available design volume, the highest angular momentum shall be attained to minimize drift rates from uncertainty torques. This criterion applies to both ball bearing and gas bearing wheels.

An equally important criterion applicable only to gas bearing wheels is the consideration of highest angular momentum within the constraints of levitation and slew rate capability of the gas bearing radial and thrust gap characteristics.

3.2.2 Ball Bearings

3.2.2.1 Preload

Design for maximum possible preload so that the gyro may better withstand shock and vibrations without resulting mass balance shifts. However, stay within reasonable stress levels (200,000 psi).

Analyze whether the static preload increases under dynamic operation.

3.2.2.2 TCP Treatment and Lubrication

To obtain the best wetting of bearing race and ball surfaces, which provides better hydrodynamic operation and protection during startup, the bearings shall be treated in tricresyl phosphate (TCP). Further, the lubricant selection and quantity shall be such that adequate life is assured and a measure of stability (damping) is imparted to the ball retainer through appropriate lubricant viscosity.

3.2.2.3 Retainers

Select state-of-the-art materials such as Nylasint or Synthane for gyro wheel ball bearing retainers. Choice should be based on economic factors and organization experience. Nylasint is the preferred material, if it is procured under the proper controls.

Both materials require considerable manufacturing controls and testing to ascertain oil absorption and bleedout rates. Testing is a continuing requirement on every batch of material procured.

3.2.2.4 Raceway Parallelness

Bearing and wheel dimensional accuracies, preload choice (DB and Integral DF) and the control processes to establish preload must provide excellent control of raceway parallelism of the opposing bearings of the preloaded set. Good raceway parallelism will minimize spurious torques and reduce gyro noise and drift.

3.2.3 Gas Bearings

3.2.3.1 Contaminants

Choose float and wheel materials that will minimize the condensation of gases in journal gap.

Choose journal materials that will resist fretting and galling during stops and starts.

Choose cleaning and vacuum baking procedures that will produce the least amount of condensation on the journal surfaces from residues of the cleaning processes.

3.2.3.2 Coefficient of Friction of the Gas Bearing Journal

For satisfactory startup conditions, the friction coefficient that can be tolerated has to be sufficiently low to accommodate the available motor torque. When friction coefficients of .45 to .6 are exceeded boundary lubricants are generally required to obtain satisfactory startup conditions within normal motor power regimes.

The application of a boundary lubricant to the gas bearing surfaces must be highly controlled and well proven through empirical testing for each design to establish a record of operation over many stops and starts and proof that the boundary lubricant will neither cause contamination nor accumulate in such a way as to reduce the stiffness of the bearing.

It is feasible to obtain coefficients of friction smaller than 0.45 through appropriate journal material selection and finish control. No lubricants need then to be used, which is a preferred condition.

3.2.3.3 Materials

Close attention should be given to symmetry and matching coefficients of expansion. Special design compensations shall be made if dissimilar materials (such as a ceramic gas bearing wheel in a beryllium float) are used.

3.3 CASE-TO-FLOAT CRITERIA

3.3.1 Hermetic Seals

1. Leak rates shall be less than 10^{-9} cc/sec.
2. Seals shall be capable of at least one disassembly and reassembly.
3. Hermetic seal materials must not be a source of contamination.
4. Seal joints must be both structural and hermetic.
5. Seal materials must be compatible with gyro fluids and cleaning fluids.
6. Hermetic seals must be able to withstand environmental inputs (vibration, shock, and temperature).

3.3.2 Flotation Fluids

1. Fluids shall be of high density
2. Fluid behavior shall be Newtonian regardless of whether shear or orifice damping is used.
3. Changes in density and/or viscosity with temperature should be as small as possible.
4. Fluids shall be inert chemically.

5. Fluids shall be of a narrow molecular cut.

6. Electrical insulating characteristics shall be excellent.

7. Capability under conditions of nuclear radiation shall be considered when fluids are selected.

3.3.3 Float/Gimbal Torque Uncertainties

1. Drifts due to mass shifts must be minimized, and particular attention should be given to reducing mass shifts that result from wheel positional uncertainties. Careful material selection to obtain matched coefficients of expansion and design approaches to obtain symmetry are essential.

2. All fabricated parts and assemblies must be stress-relieved to avoid dimensional and related mass-balance drift changes in the finished instrument with time.

3. Thermal design is to be optimized in terms of symmetry to minimize fluid convection and positional torque changes.

4. The center of gravity and center of buoyancy of the float should lie in a plane that contains the gravity vector and the center of rotation. In a multi-orientation gyro, they should all coincide in the center of rotation.

5. The design and excitation of transducers shall be such as to minimize electromagnetic restraint changes.

6. Flexlead containment, attachment, assembly accessibility, and material selection shall be designed to avoid contamination of the instrument and time-dependent restraint changes.

3.4 LIFE

3.4.1 Temperature Effect

The regulated temperature of the gyro should be as low as is feasible within the system application. Elevated temperatures tend to reduce the life of the instrument.

3.4.2 Operating Life

Attainment of the operating life for which the gyro is designed is strongly related to extensive testing that will eliminate all early failure occurrences.[11] A design life of 3000-5000 operational hours is a minimum attainable goal. Design lives of 7000 to 12,000 hours are considered average. A life failure is defined as the failure mode that prevents the instrument from meeting the established specification parameters.

4. RECOMMENDED PRACTICES

4.1 INTRODUCTION

When an SDF integrating gyro is needed for a new satellite application, an *experienced* gyro design and manufacturing organization will prefer to modify an existing instrument to obtain the desired characteristics. This approach permits improvements in performance and economy with minimum risk of reliability.

A considerably greater upgrading of performance is usually required when development contracts are placed for newer generation instruments, but again the organization's unique experience in design and manufacturing will have much to do with the degree of success that it achieves. This know-how is an intangible asset and is not generally put down on paper.

In the subsequent paragraphs, the basic topics of this monograph are re-examined from the standpoint of this special experience.

4.2 GYRO WHEEL DESIGN AND MANUFACTURING CONSIDERATIONS

4.2.1 Ball Bearing Wheels

4.2.1.1 Wheel Angular Momentum

Figure 16 illustrates the effect of uncertainty torques on drift rate. The abscissa scale is calibrated in both degrees per hour and milliradians per/sec and uncertainty torque in dyne-centimeters is plotted on the ordinate scale. The diagonal lines represent various values of wheel angular momentum (H) ranging from 1000 to 10^8 gram-cm^2/rad/sec. For example, an uncertainty torque of 1 dyne-cm would cause a drift rate of 2 deg/hr for a gyro with an angular momentum of 10^5 cgs units or 0.2 deg/hr if the angular momentum is 10^6 cgs units. In other words, the drift rate for a given gyro and a specific uncertainty torque will vary in inverse relationship to the angular momentum of that gyro. Therefore, the designer must consider the requirements for drift and life when choosing wheel speed and rotor inertia.

4.2.1.2 Ball Bearing Preload

Gyro spin bearings are preloaded to provide more exact radial and axial positioning, increase the rigidity of the assembly, and control the axial and radial yield rates as well as to provide increased radial load capacity.

Preload Methods

The preloading methods are either DB (back-to-back) or DF (face-to-face). Figures 1 through 5 (Section 2) show the direction of the load lines for both methods. The DB mounting is often chosen because (1) it provides the best overturning moment rigidity for gyro wheels and (2) it allows compensation for a possible increase of preload that can result if the temperature of the inner race rises under running conditions.

H = GM CM²/RAD/SEC = DYNE CM SEC/RAD   1°/HR = .004848 MR/SEC   1 MR/SEC = 206.3°/HR   1 MERU = .015°/HR

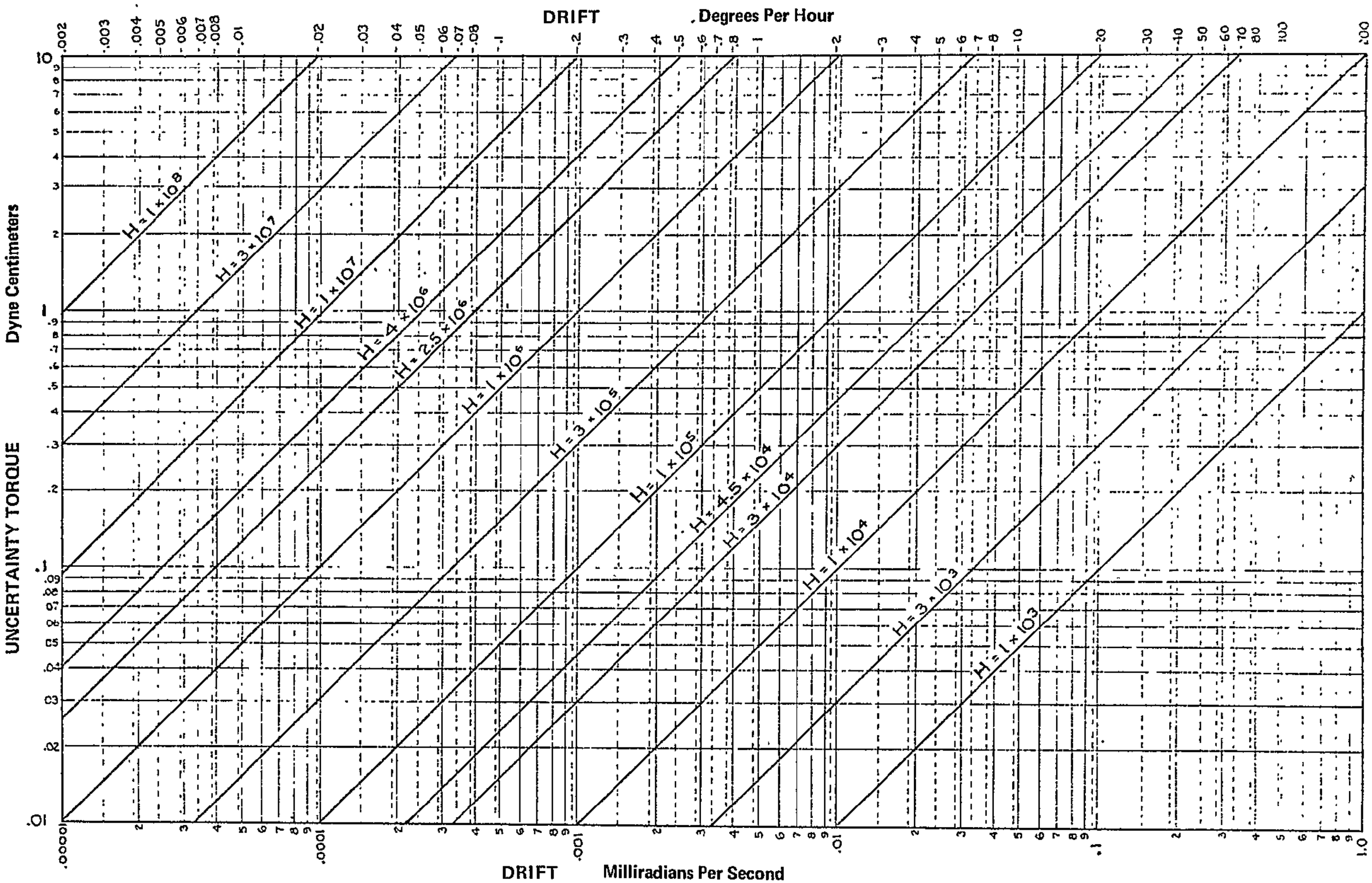


FIGURE 16. GYRO DRIFT VS. UNCERTAINTY TORQUE FOR VARIOUS VALUES OF H

For a DB matched preloaded pair, the bearing manufacturer will grind the opposing inner-race faces of the two bearings to give a specific preload offset; then when the bearing pair is mounted face to face or with equal spacers between the outer and inner races, they will be preloaded to the desired level.

Similarly, for a DF preloaded pair, the outer rings are preground by a specific preload offset, so that when the bearings are mounted face to face or are separated by equal sized spacers, the particular preload is attained. One advantage of the DF preload technique is that its sequence of assembly permits the inner races of the bearing to become an integral part of the shaft, an advantage that is not feasible with the DB preload procedure. If we assume that (1) the particular design application does not involve environmental inputs that would put large overturning moments on the spin motor assembly and (2) the heat transfer characteristics of the particular design allow adequate dissipation of heat along the spin motor shaft, the DF integral shaft design is the recommended preloading method for ball bearing wheels. It provides a tighter dimensional control and an improvement of parallelism of the raceways of the opposing bearings, thereby permitting a greater smoothness of operation with a corresponding reduction of drift. The fact that the DF method permits some misalignment of the two bearings, since it minimizes internal loading through the effects of inwardly converging contact load lines, additionally helps to obtain a smoother running assembly.

Change of Preload Under Dynamic Conditions

The preload of a bearing set is known to change in most circumstances as a result of dynamic operation from the statically established levels, and it is important to determine the amount of this change. The deflection load curves used by the designer to determine the initial static preload can also be used to determine the change of preload if a measurement is made on the rotating set at synchronous speed. This measurement is not easy; optical methods may be necessary to obtain a sufficiently accurate identification of the deflection change of the preload.

A major contributor to the change of preload (in addition to dimensional effects related to centrifugal action of the wheel component parts) is the hydrodynamic film which forms during dynamic operation and separates the balls of the bearing from the inner and outer races. The thickness of this film is no greater than a few microinches, but it is sufficient to increase the deflection and thus the total preload. The hydrodynamic film, however, is essential for the long life of the bearing system.

4.2.1.3 Hydrodynamic Films and Lubrication

Recent studies by the Instrumentation Laboratory of the Massachusetts Institute of Technology indicated that the life of ball bearings running in oil, such as Teresso V78 or Kendall KG80 was extended by orders of magnitude and friction was halved when the metal parts were exposed to various treatments of commercially available tricresyl phosphate (TCP).[12] Subsequent work showed that bearings using only the lubricants stated and without the TCP treatment would fail much earlier[12] particularly under slow speed operation. Additionally, on the Titan program at AC Electronics, it was found that bearings stored for a year in V78 oil provided a considerable increase in yield over other bearings that had only been briefly aged in oil. However, bearings treated in TCP produce yields comparable or better than "aged" bearings.

A definite recommendation can therefore be made for gyro ball bearings: aging in TCP is a necessary requirement for improved performance and life.

Teresso V78 oil (or, after it became unavailable, Kendall KG80) has been found best for lubricating spin motor bearings on the more precise gyro instruments. Minute quantities of these lubricants have provided longer life for gyro spin bearings than was possible with other oils and, in combination with appropriate retainer design, have minimized the drift rates that result from lubricant migration. The interrelated effect of lubricants, retainers, and irregular ball motions on the behavior of ball bearings has been described by E.T. Kingsbury in various Instrumentation Laboratory reports.[13-15]

Despite TCP treatment, bearing materials sometimes cannot be wetted and are considered "poisoned." It is therefore essential that every set of bearings be tested for wettability through the observation of the rate of spreading of an oil drop placed on the bearing race surface. If wetting is poor, the bearing should be replaced or at least recleaned. Additionally, "burn-in" periods ranging from 200 to 1000 hours of operation are not unusual to determine whether a gyro wheel assembly in a test enclosure, or in its final float, will operate successfully from a power consumption and running performance standpoint during and after the burn-in.

As discussed in paragraph 2.3.2, many wheel failures occurred in the Apollo IRIG Block II gyroscopes during 1968 and 1969 prior to system installation. Of this group, few wheel assemblies survived more than 50 to 100 hours of testing. The problems appeared to relate to metallurgy, race finish, retainer squeal, insufficient lubrication, and perhaps excessive preload. After considerable study by various organizations, wheel performance was improved by the following changes:

- The retainer was changed from an outer race riding retainer to an inner race riding retainer, minimizing retainer translation and squeal, while maximizing lubricant circulation to the ball/race contact area.

- The lubricant was changed from Kendall KG80 to the more viscous SRG160 (NASA Identification N-3) to damp the retainer motion; this further eliminated retainer squeal, and combined with an increase of lubricant quantity provided better film and lubrication characteristics.

- Preload was considereably reduced (3 lb to 0.9 lb) when it was realized that it would increase under dynamic operation (to approximately 1.8 lb), primarily as a result of the hydrodynamic fluid film.

- Raceway finishing techniques were changed to stick or felt lapping, limited to 2 or 3 laps to avoid geometry deterioration and providing a much improved metal surface structure and finish than the honed surface had provided on the Block II Apollo bearings.

Most, if not all, of these changes were made on an empirical basis and could be considered as art rather than science. When the performance of the gyro once again become reasonable, it was not clear just what particular item had produced the improvement.

It was concluded that there was a basic metallurgical finish problem that has not yet been fully isolated or understood. Since the preload reduction was a major factor in re-establishing acceptable performance from the bearings, it is self-evident that a contact stress reduction helped resolve the problem, while the changing method of finish lapping avoided the folder-over metal surface structure (shown in Figures 7 and 8) providing an improved finish, similar to the finish of Figure 6, while maintaining proper geometry.

No specific recommendation for the designer of a new gyro emerges from the experiences described above; their significance lies in the realization that the know-how obtained by trial and error can be applied to another design in the future, perhaps before difficulties are experienced.

4.2.1.4 Retainers

The two types of retainer materials used in ball bearings for gyro wheels are Nylasint and Synthane. It has been established that Nylasint retainers can be made of a homogeneously porous material which allows improved oil circulation within the bearing. It has also been established that Synthane has non-uniform porosity and therefore inhibits the free circulation of the lubricant. At the present, therefore, Nylasint is definitely preferable to Synthane as long as a particular batch of Nylasint can be proved to have the porosity and machinability desired. Nevertheless; many retainers are made from Synthane, not only because of its lower cost but, also because some organizations have extensive experience with it.

4.2.1.4.1 Synthane Retainers — Recommended Selection Procedure

The basic Synthane material is procured in tube form of a size suited to the manufacture of retainers for particular bearings. Since porosity is a basic requirement, the tubes that are the least dense out of the lot procured should be used. Selection can be based on a simple flotation process.

When Synthane is made, the process of rolling impregnated kraft paper on a mandrel results in a higher tension at the margins of the sheet, so the outer portions of the tubes are consistently more dense than the inner portions. Therefore, the tubes are cut into thirds, and only the inner third of the lighter tubes is shipped to the bearing manufacturer. The latter then fabricates retainers from this stock, lubricates them according to his own practices, assembles the bearings, and ships them back to the gyro manufacturer. If the bearings are to be used in inertial-grade instruments, they are completely disassembled at the gyro facility. The retainers are then processed through an extraction cycle in a recycling still (Soxhlet extractor) using a series of solvents and dried under a vacuum.

While hot and under a vacuum, the retainers are impregnated with a heated lubricating oil and then centrifuged at an acceleration level equal to or slightly greater than that to which they will be exposed in the bearings. If the amount of oil that remains is less than 5% of the dry weight of the retainer, that retainer is rejected. Those that satisfy this criterion are assembled into bearings and the bearings into spinmotors. The spinmotors are run for 50 hours and then torn down. All bearing components are inspected under a microscope, and any that show degradation are discarded. Bearings and spinmotors are then reassembled, and the gimbals are sealed for use in gyro assemblies.

4.2.1.4.2 Nylasint Retainers — Recommended Material Selection

In spite of the fact that Nylasint is much more amenable to the application of controls, the manufacture of the material is far from trouble-free. The Polymer Corporation, the sole manufacturer, has experienced difficulties in consistently producing a material with acceptable machining characteristics. This property is so difficult to define in specification form that sample pieces must be made and subjectively evaluated by the retainer fabricator and the bearing user. Because of the limited demand for this material, the producer does not find it economical to set up a separate facility in which process control, once established, could be maintained. Thus, when, through trial and error, a satisfactory lot of Nylasint powder is produced, it would be desirable to stockpile it in the form of pressed plugs. Since the material supplier has no assurance that the material will be consumed, the burden falls on the user, ultimately the Government, to assess long-range requirements and establish a required stockpile.

The lubricant absorption test for Synthane retainers is equally applicable to Nylasint; in fact, a test of oil absorption by weight would be more accurate than measuring the time for a metered drop of oil to be absorbed, as has been done in various facilities where the Nylasint retainer is used. Furthermore, the oil drop absorption measurement looks primarily at a surface characteristic, while the absorption-by-weight test is a volume characteristic of the entire retainer. Process controls for Nylasint retainers are further discussed in paragraph 2.3.1.4.2.

4.2.1.5 Raceway Parallelism

The parallelism of the two opposing races in a set of preloaded bearings is dependent on the dimensional accuracies not only of the bearings themselves but also of all the component parts that make up the bearing sets and gyro rotors. In addition, it is strongly influenced by the methods of establishing the preload — i.e., by the way the various pieces are assembled, by the fixtures that control alignment during this stackup, and by the way that deflection versus load is measured.

The best raceway parallelism is obtained through the use of solid spacers in the preload stackup; preloads applied by spring-tightening the bearings with a threaded shaft and nut arrangement should be avoided. Similarly, the application of preload by tightening two shaft sections together which are threaded into one another should be avoided, as the motion of screwing the pieces together will cause the races to rotate out of alignment with the central axis of the assembly and out of parallelism with each other. Such misalignment can produce ball wedging during rotation, and the resulting torques acting on the wheel can cause rectification drift errors.

The ball wedging that results from inexact raceway parallelism or ball size variation produces mechanical motion inputs to the pickoff, as discussed in paragraph 2.3.1.5. The predominant frequencies of these motions are (1) the spinmotor/drive resonant frequency, (2) the wheel speed frequency, (3) beat frequencies between wheel speed and pickoff excitation frequency, (4) the retainer (ball race) speed frequency, and (5) the product of ball race frequency and the number of balls in the bearing.

To avoid beat frequency effects (producing low-frequency noise and drift) separate pickoff excitation frequencies and wheel mechanical rotation frequencies should be chosen both at the basic frequency level and with due regard to harmonics.

After preload is established, all float assemblies should be run in for a minimum of 150 hours and a maximum of 2000 hours; during this time, variations of the operating power level should be monitored and coast-down time should be checked repetitively. The wheel assembly should be modulation-noise-tested in its gyro case as a completed gyro; the precessional output from the wheel, as expressed through the signal generator should be recorded in the frequency band of 0-50 cycles, and certain maximum limits of output voltage should be established. The signal to be monitored can be demodulated and bandpass-filtered for the frequency range of interest.

4.2.2 Gas Bearings

4.2.2.1 Contaminants

Most of the current contaminant problems in gas bearing wheels result from the condensation of organic materials in the journal gap. These materials originate primarily in the spinmotor stator, which contains epoxies and other plastics. These plastics release volatile components into the float. As the gas bearing self-levitates in the gaseous medium of the float, gas flows into the bearing journal. At the high-pressure points in the journal, the volatiles tend to condense and reduce the size of the gap. Ultimately, the available voltage will be insufficient to start or synchronize the wheel, and the gyro will fail. Other organics present in the float, such as traces of cleaning solutions, are similarly detrimental.

Select the bonding, cementing, and coating materials used on the spinmotor stator from newly available compounds (polyimides) that permit vacuum baking at elevated temperatures. The windings of the stator should be made from ceramic or equivalent insulated magnet wire, and the windings themselves can be potted by applying several coats of polyimides. These materials cure at a considerably higher temperature than any of the standard epoxies, cements, and potting compounds. Therefore, cleaning can be followed by vacuum baking at a temperature considerably higher than that at which the gyro normally operates. These temperatures can be as much as 50-200° apart, so the amount of volatiles that degas at the lower operating temperature will be negligible. High temperature vacuum baking will also eliminate the other organics (solvents) that have come from float cleaning cycles. Vacuum baking should be done in enclosures to avoid the condensation of oven contaminants on the parts upon oven cooldown.

The particle friction in the journal gap due to material fretting during stops and starts can be minimized by proper choice of journal materials and the greatest possible cleanliness. Recommended materials for the gas bearing journal are Lucalox-type alumina and boron carbide, a new material not yet widely tried. Because of its homogeneity, Lucalox can be finished to a degree which will permit it to operate as a rotor and shaft without the application of boundary lubricants. (Figure 17 compares the porosity of standard alumina with that of the dense, translucent Lucalox alumina.) Boron carbide is innately self-polishing, so stops and starts are not likely to cause the kind of fretting that leads to contamination. Both materials are very hard and can withstand bottoming during slew rates. Table 1 compares the characteristics of these and other gas bearing materials.

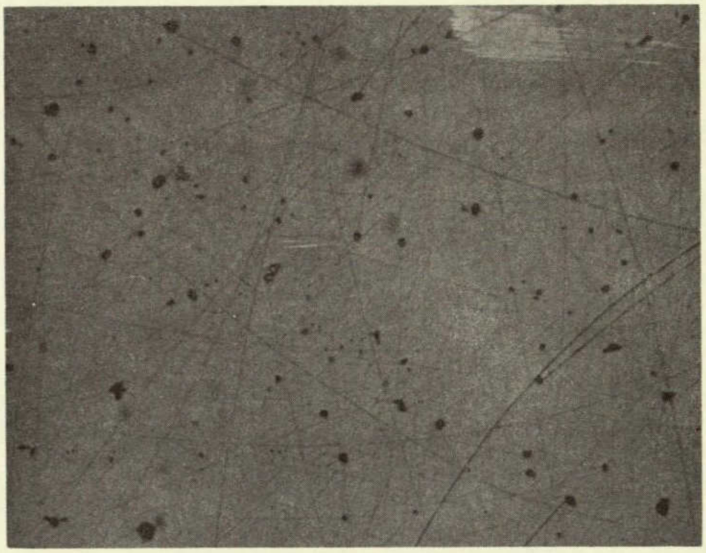
4.2.2.2 Coefficient of Friction, Boundary Lubricants

The coefficients of friction listed in Table 1 indicate that most material combinations result in dry-contact coefficients of friction that are high enough to require boundary lubricants. Only Lucalox and boron carbide appear to have sufficiently low coefficients of friction to make the addition of a boundary lubricant unnecessary. In all likelihood it is not possible to get any material perfectly chemically clean; therefore, the friction coefficients of Table 1 are a result of the material characteristics combined with cleaning fluid or moisture residues on the journal surfaces acting in part as a boundary lubricant.

Though Lucalox is being used in a gas bearing by several gyro manufacturers, boron carbide is still considered experimental because of its high cost and machining difficulties. The advantage of boron carbide lies in the extreme hardness coupled with a self-polishing tendency, which provides a low coefficient of friction.

Because Lucalox material is dense and of low porosity, it permits a far better surface finish and lower coefficient of friction than standard alumina.These qualities enhance its ability to survive touchdown, tough fretting will ultimately occur.

For nearly all other material choices, a boundary lubricant is needed to obtain effective stop and start operation. This lubricant provides a low shear strength condition between the journal shaft and rotor, thereby reducing the rubbing friction. In order not to cause contamination, the coating must adhere firmly to bearing surfaces and be of such little quantity (i.e., not more than two or three molecular layers) that it cannot accumulate in the grooves or pockets of the thrust plate or herring-bone grooves on the shaft; otherwise, it would change the bearing stiffness characteristics. In addition to fulfilling all the requirements of use over the required temperature range of start/stop operations, the boundary lubricant must permit application in a practical manner.



LUCALOX 110X

ALUMINA 110X

FIGURE 17. COMPARISON BETWEEN STANDARD ALUMINA AND LUCALOX

TABLE 1

PROPERTIES OF SELF-ACTING GAS-BEARING MATERIALS

Class of Material	Principal Ingredient	Form[1]	Young's Modulus (10^6 psi)	Density (gm/cc)	Hardness Knoop (100 gm)	Coeff. of Linear Exp. (10^{-6}/°F)	Coeff. of[4] Friction	Thermal Conductivity Btu/hr ft^2 °F/ft
			Properties of Principal Ingredient					
Alumina	Al_2O_3	b, c	55	4.0	3000	4.0	0.5-0.8	10.7
Lucalox	Al_2O_3	b, c	55	4.0	3000[3]	4.0	0.2-0.5	10.7
Chromia	Cr_2O_3	c		5.2	1200	3.0		
Tungsten Carbide	WC, Co	b, c	105	14.9	1250	4.8	0.5-0.8	
Beryllia	BeO	b[2], e	52	3.0	1300	3.8	0.6	55
Pyroceram	$MgO-Al_2O_3-SiO_2-TiO_2$	a	17	2.9	700	5.3	0.8	
Ferro-Tic	Fe, TiC	b	44	6.6	1000	4.0	0.4-0.7	35
Chromium Plating	Cr	d		7.5	1200	3.5	0.4-0.7	
Boron Carbide	B_4C	b[2]	65	2.5	2800	2.1	0.2-0.4	16

1. a. Solid from melt
   b. Solid from compacted and sintered grains
   c. Melt sprayed coating
   d. Electroplated coating
   e. Anodized coating
2. Experimental as gas-bearing material
3. 50-gram load
4. Material on itself, better than one microinch finish

Note: The approximate properties of the principal ingredients allow speculative comparisons. Very substantial departures from the above are obtained on actual gas bearings because of the influence of binders, porosity, compaction, formation or deposition processes, as well as heat treatments, departures from stoichiometric composition, and several other factors.

Materials that are applied as coatings to reduce rubbing friction for start and stop cycles have included metal soaps and surface-active compounds, such as lead stearate, N,N-dimethyl stearamide, sodium stearate, fluorolubes, and TCP.

In view of the above, the designer shall choose hard, nonfretting materials and avoid lubricants. If a boundary lubricant must be used, controls must be applied to insure that it does not cause contamination or reduce the operating capabilities of the wheel. Moreover, the lubricant should be thoroughly proven by empirical testing of each design to establish a record of operation over many stops and starts.

4.2.2.3 Mechanical Design

The design of gas bearings usually follows one of these approaches:

- Solid structures are used where the gas bearing, the shaft, and the rotor are all made of the same material.

- Solid inserts are used when the gyro wheel is made of, say, beryllium and has a core of hard bearing material.

- Coatings are used where hard bearing materials are flame- or plasma-sprayed or are plated onto a softer base material such as beryllium or steel.

While each design approach must be judged on its own merits, the solid structure is generally recommended as long as mismatched coefficients of expansion are compensated. Difficulties of porosity control arise with applied coatings, and solid inserts with thin walls are subject to hoop stress problems (fractures).

When the instrument contains dissimilar materials, such as a ceramic wheel in beryllium float enclosure, it is recommended that flexure diaphragms be used in combination with reduced-diameter shaft ends to compensate for the thermal expansion differential and for possible shaft distortion resulting from the pillow block/yoke clamping. The flexure diaphragms, which compensate for the expansion of dissimilar materials along the spinmotor shaft, must have spring constants that are matched to each other in the particular assembly, so that the center of mass will remain at its original location regardless of temperature.

4.3 CASE-TO-FLOAT CONSIDERATIONS

4.3.1 Hermetic Seals

In order of preference, the four techniques now used for instrument hermetic seals are: (1) welding, (2) epoxy, (3) solder and (4) O-rings.

Welding is recommended wherever possible; it provides a hermetic seal with structural strength and does not contaminate the assembly during the sealing process, as no material is added to the two elements to be joined. The joint design for the welded seal is quite simple: two weld lips (8-20 mils thick) nest into one another to insure concentricity, and an internal shoulder provides the customary seating of the parts. The weld lips are long enough (about three times the weld penetration depth, or 60-90 mils) to permit two or three repair operations (i.e., machining through the weld to open the enclosure, a fix and a reweld). (See paragraph 2.4.1.3.)

Some steels weld better than others, particularly the stainless steels of both the 300 and 400 series, which fusion-weld quite well, though certain alloy types (No. 317, 321, 341, 350, 410, and 416) are preferred. Stainless steels also weld well to Cupro Nickel 30, which has the advantage of being magnetically transparent and will therefore accommodate transducing requirements such as a spinmotor rotation detector (SMRD).

For greater stress/yield stability, bellows made of particular alloys may be heat treated after welding in a particular position (such as a fully nested position) allowing improved repetition (a reduction of mechanical hysteresis).

Titanium and its alloys weld to themselves beautifully but are difficult to weld to other metals.

Neither aluminum nor beryllium fusion-weld satisfactorily with an inert gas torch (TIG) but they do weld reasonably well under a vacuum with the electron beam method.

Epoxy hermetic seals are the next best choice, particularly for materials that are not readily weldable. The more liquid, unfilled epoxies are preferable to those that are filled. The pieces that are to be joined and hermetically sealed should have a pilot diameter to permit nesting; this can also be the epoxy joint if unfilled epoxies are used. The radial gap size should be in the order of 0.5 to 1.0 mil, and the joint should have a shoulder for seating purposes.

When epoxies are used, the basic resin and the hardener must be properly mixed by weight rather than by volume, and the cure cycle must be sufficiently long to allow the hardener to fully cure the epoxy. (It is helpful to have the resin and the hardener of distinctly different colors, so that one can easily verify that the hardener has been added before the cure cycle.) Before the surfaces of the joint are coated, they must be thoroughly cleaned so that the epoxy will bond firmly to all areas. Before epoxies are applied to the cement joint, they must be thoroughly degassed to eliminate bubbles in the cement. After curing, the assembly should be vacuum baked at as high a temperature as it can safely withstand, to drive off all volatiles that may come out of the cement joint. The shoulder of the joint should have a very slight chamfered lip to permit visual inspection and assurance that the bead of cement is unbroken around the whole perimeter.

Filled epoxies have been used for many years. Through the use of various fillers (such as mica), they can have coefficients of expansion ranging from 15 to 40 microinches per °F per unit length, thereby approximating the coefficients of metal gyro parts, particularly aluminum. In contrast, thin liquid epoxies, without fillers, range between 50 and 150 microinches per °F per unit length. The thin, unfilled epoxies are used in such small quantities and thicknesses that they are elastic and can move with the enclosures they bond together.

The filled epoxies are usually used with metals such as aluminum, whose expansion over the temperature range may be too great for a thin epoxy to handle. These filled epoxies require a considerably larger gap (typically 3-5 mils), because if the layer of cement were very thin, a capillary path might form from one filler particle to another and thereby permit a slow leak. Accordingly, it is necessary to have separate pilot diameter in addition to the customary shoulder for nesting the pieces together. Again, the lip should be slightly chamfered to permit visual inspection of the cement.

A disadvantage of filled epoxies is that they sometimes absorb cleaning fluids and gyro fluids, particularly if excess cement has been sanded or scraped from the joint. The fluids will penetrate the cement to some minimum depth and, since the action is time-dependent, can cause an apparent change of balance of the float.

Soldering an hermetic seal usually requires that the parts first be plated. Because plating quality is an exceeding difficult variable to control, soldering is ***not recommended.*** Solder has poor structural properties and tends to cold-flow when stressed. If soldering is chosen as a sealing/construction technique, the following is relevant: To provide some structural integrity, a section of the solder joint must be almost a snug (snap) fit. This section, which should accomodate a solder bond between the parts, then also functions as a pilot diameter. The remainder of the joint should have a gap of at least 3-5 mils so that it will contain enough solder to minimize the danger of capillary paths. Lastly, the shoulder of the joint must be distinctly chamfered on both pieces so that the solder bead can be visually inspected all the way around.

The energized rosin-alcohol fluxes that are used with solder are often captured in the joint and can seal capillary leaks so that they are not detected during assembly. At some later time, however, gyro fluid may leach the flux out of the solder and thereby cause a leak. If the joint is properly chamfered, the presence of flux on the surface of the solder can often be detected by visual inspection and the solder can be reflowed.

Another disadvantage of soldering is that a temperature of approximately 400°F is briefly reached during RF induction heating. This high heat can cause reliability problems with susceptible materials nearby (epoxies, bearing lubricants, etc.). Also, the interior side of the solder joint cannot be visually inspected, so the assembler cannot be sure that solder balls have not formed during the heating cycle. These particles can contaminate the interior of the gyro or cause mass shifts inside the float.

O-rings are often affected by gyro fluids and cleaning fluids. Many take a compression set and cause seal failure during temperature cycling. Viton A and fluorosilicone materials are least subject to these shortcomings. When O-rings are used for hermetic seals in gyros, the application should always be static. Furthermore, the O-ring cavity should be nearly 100% filled, so that the elastomer is under fairly heavy compression.

Silicone rubber and Buna-N should be avoided, as the plasticizer used in this material can be leached out by either cleaning fluids or gyro fluids. Also (unlike fluorosilicone rubber), silicone rubber tends to become porous and absorbs fluids; when this happens, it swells, softens, and disintegrates, causing contamination of the gyro fluid.

In applications that involve very low temperatures (e.g., -100°F), O-rings of ethylene propylene perform reliably, because they retain a high degree of resilience at such temperatures.

For a good O-ring seal, it is necessary for one part to be held in compression against another, usually by screw-rings or separate screws. This method of assembly is not attractive from either a space or reliability standpoint, and contamination may result from the rubbing motion of screw threads.

Although metal O-rings of various types are available and have been used on static seals, they are *certainly not recommended* for gyro applications. Because they have exceedingly little resilience, they are likely to fail under temperature cycling when their coefficients of expansion do not match the coefficient of the parts being sealed. If the application requires their use, eliminate all cross machining marks on the metal pieces; i.e., all machining marks must be circular and concentric with the O-ring.

4.3.2 Flotation Fluids

4.3.2.1 Fluid Compatibility

The denser the flotation fluid, the more efficient a gyro design can be. The densities of standard fluorochloro and Fluoro-Chem fluids range from 1.8 to 2 between 80°F and 140°F. Fluids of higher density (approximately 2.35) are available in the BTF (bromotrifluoro ethylene) polymers; however, these are not as compatible with gyro materials as the straight fluorochloro fluids. For this reason, BTF fluids should be used only after careful consideration of all compatibility problems.

Since some degree of incompatibility can be encountered with almost any type of gyro fluid, the one that is selected must be tested for reaction with the gyro materials at a temperature at least as high as that of normal operation. All of the materials should be in the fluid test bottle together and electrically connected so that galvanic action can take place. This test, which normally takes at least three months, provides data on life and contamination that are very useful during the design phase; with proper choice of materials, a gyro can last two years or more insofar as compatibility/contamination is concerned. Particular attention is to be paid to this test and its duration is to be continued during the development period of the instrument, to determine whether long-term effects of contamination will show. Five- to six-year storage lives are now part of some specifications.

4.3.2.2 Fluid Damping

The gyro fluid provides damping as well as flotation, and this damping must be constant for various torque inputs to the float. To test for this, perform velocity tests with the float assembly operating in the gyro fluid and within its outer case (actual or simulated). A dummy float without an operating spinmotor can be used.

When the requirements of damping indicate the use of a highly viscous fluid, to maintain a low gain condition (H/C), a disadvantage of increased radial settling time will occur. Suspension elements in precision gyros do not generally have large centering force gradients. Therefore, the highly viscous fluid necessary in shear damping to obtain a H/C ≈ 1 or less will cause a slow response to output axis float centering after angular inputs around OA. One solution to this problem is to use orifice damping (paddle damping). In these schemes the total damping is largely obtained through orifice metering of the paddle displaced fluid in combination with only minor shear damping. Low viscosity fluids can now be used while the gyro gain can be maintained near 1 (H/C = 1), while short settling times in the radial direction are simultaneously obtained.[10]

When the gyro uses orifice damping, fluid turbulence is likely to make the damping non-Newtonian; therefore, the L/D ratio of the float pistons as well as the orifices must be considered. To obtain laminar fluid flow over the range of velocity inputs, an as large as is possible L/D ratio should be used (5 to 10). If this is impossible for the orifice, a maze of smaller orifices, each with an L/D of 10, should be substituted. When temperature damping compensation is desired on an unheated unit, the orifice block can have a series of holes arranged to compensate for the fluid temperature/viscosity index as a shutter mechanism moves by.

Orifice damping permits the use of gyro fluids with lower absolute viscosity than those needed for shear damping. The low-viscosity fluids are often more stable chemically, and can withstand higher levels of radiation. Furthermore, they generally undergo a smaller change of viscosity with temperature, resulting in more uniform damping over a range of temperatures. Examples of such fluids are the Fluoro-Chem Series 11-14 and 4-11, Halocarbon 208, Hooker Corporation M010 and FS5, and 3M Company FC75 and FC43.

4.3.2.3 Stratification

When shear damping is used and fluids of higher viscosity must be employed, the viscosity improvers must consist of radical fractions that are nearly identical in molecular weight to the original fluid. The fluid source should certify that the range of molecular weight of the fluid constituents is narrow. Additionally, gas chromatography should be used on a sampling basis to identify and control the fluid for inclusion of different molecular weight fractions. These precautions are necessary to avoid possible thermal stratification of the fluid in the gyro, which will cause apparent mass balance ramps. In the gyro, the presence of two different fluid densities can be detected by a gravity transfer technique, in which the gyro is tested first in one axis attitude and then in another to reveal apparent transfer of mass balance. For this test to be meaningful, the fluid must be allowed to stratify, which may take many hours; in some cases, stratification can continue for weeks before it is complete.

4.3.2.4 Radiation Effects on Fluids

Fluids that are subjected to radiation are likely to liberate chlorine and, sometimes, fluorine. Other than the fact that the bubbles can cause the gyro to drift, the free gas can combine with any moisture that may be in the assembly and form an acid. Should this occur, the dielectric properties of the fluid are likely to deteriorate, fluid viscosity is likely to change and corrosive contamination will start. Table 2 presents the chemical formulas, densities, viscosities, and radiation capabilities of several typical gyro fluids. It is provided to give the designer the awareness of alternatives and tradeoffs, so that he can make the best choice for the specified application environment. Even if radiation is not specified, the designer should prefer a fluid that is radiation-resistant to one that is not (other properties being equal), because an application for that gyro with radiation hardness capability may well arise later.

4.3.2.5 Gyro Fill Procedure — Specific Recommendations

Paragraph 2.4.2.3 discussed the state of the art of gyro filling techniques without providing specific recommendations in terms of fill stand components and materials selection, such as:

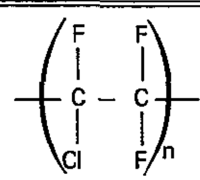
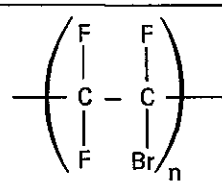
The various valves of the fill stand system must be chosen to contain inert materials for seal purposes to avoid chemical reaction with the gyro fluid.

Valves used must be of a type that can be readily disassembled for cleaning. Materials of the fill stand must be compatible with the gyro fluid; the use of plastic or rubber tubing is to be avoided, as plasticizers in these materials tend to be leached by the gyro fluid and cause contamination. Preferred materials are glass, stainless steel, Teflon, and Viton-A.

Micro filters are customarily installed in the fill stand plumbing, one at the inlet to the gyro and another at the gyro exit port. As the fluid is backfilled into and through the gyro, the first filter will catch any particles that may come out of the system and the fluid. The second filter will catch particles coming out of the gyro. A particle size and number count criterion must be established by which the gyro fill is considered acceptable or not. If a second fill (after complete flushing with solvents, followed by a vacuum bake) still produces an excessive particle count, the instrument is torn down for closer analysis, a recleaning of subassemblies, and a subsequent reassembly with substitution of parts if necessary. Refer to paragraph 2.4.2.3 for filling procedures and techniques.

TABLE 2

RADIATION CHARACTERISTICS OF REPRESENTATIVE GYRO FLUIDS

Designation	Formula	Approx. Density at R.T. (gm/cc)	Approx. Viscosity at R.T. (cts)	Radiation Rate Capability w/o Appreciable Chemical Change (rads)
Hooker M010		1.85	10	10^5
Fluoro-Chem Halocarbon 208	$C_4Cl_5F_5$	1.84	2.8	10^7 to 10^8
Fluoro-Chem BTF (Polybromo tri-fluoro ethylene)		2.2 to 2.5	7 to 10,000	10^5
3M Co. FC-43 (Perfluoro tri-n-butylamine)	$(C_4F_9)_3N$	1.88	2.8	10^6 to 10^7
Dow Corning Silicone Oil	$-\!(Si(OCH_3)_2)_n\!-$	0.9	2 to 10,000	10^3
Shell Polyphenyl Ether	$C_6H_5O-(C_6H_4O)_n-C_6H_5$	0.6 to 1.6	High	10^{13}

4.3.3 Torque Uncertainties

4.3.3.1 Mass Shifts Along SRA

To minimize drift changes that result from wheel shifts along the SRA, the spinmotor ball bearing should be of the largest possible size that can fit in the available space. The largest possible bearing allows a higher preload across the bearing system within a reasonable Hertzian ball bearing stress. The greater the preload, the less likely that a ball track shift will result from environmental inputs. Further, it is recommended that the compliance of the wheel assembly along the SRA versus that at right angles along the IA be approximately the same, so that acceleration will not cause a g-square effect. ANISO measurements at right angles to the spin axis (and also along that axis) should take into account the combined compliances of the gimbal structure and the wheel structure. However, to approach iso-elastic conditions, bearing contact angles between 23° and 33° are recommended.

The windings of the spinmotor stator should not move relative to its potted enclosure or relative to the established float center of gravity. To achieve this stability, the coefficients of expansion of the compounds used on the spinmotor stator windings should be as low as possible and approximate those of the copper iron material in the winding and lamination stack. It is further suggested that these compounds be able to withstand vacuum baking at considerably higher temperatures than the gyro operating temperature to *ensure* little or no degassing of volatiles in the completed instrument. Polyimide (kaptan) brush-on plastics are recommended.

4.3.3.2 Stress Relief Changes

All parts of the gyro wheel, whether ball bearing or a gas bearing, must be stress-relieved through hot and cold temperature cycles (not only during and after machining but during subassembly and assembly) to avoid time-related stress relief changes after the instrument is completed. Temperature cycling should take the form of shock cycling between -100°F and +200°F on finished pieces. During the machining process, metal parts should also be stress-relieved at higher temperatures, typically 600°F. These stress relief cycles are undertaken on semifinished machined pieces, just prior to the final finish cut.

4.3.3.3 Fluid Convection

To avoid unsymmetrical thermal gradients that cause temperature variations and fluid convection in the gyro, it is important that the gyro be mounted in a fully symmetrical manner in its environment; external thermal gradients will then tend to cool or heat the instrument symmetrically.

4.3.3.4 Suspension Torques

Suspension Elements

The preferred material for pivots, jewels, and endstones is tungsten carbide. Tungsten carbide will not fracture as rapidly as sapphire suspension components. For a fully floated gyro, it is important to make the pivot diameter as small as possible to minimize friction torques when the pivot contacts the jewel; a diameter of 15 mils is probably the smallest that will permit assembly of the instrument without breakage. The jewel contour should be olive-shaped if the pivot has a straight-shank. However, if the pivot is a small sphere welded to the end of a stem, the jewel should have a straight through bore. This latter method of construction allows greater misalignment between opposite suspension ends than the straight-shank pivot and olive-shaped jewel.

The endstone that faces the polished spherical end of the pivot and limits the axial excursion of the float should be positioned so that no compression contact takes place as the result of thermal gradients between the float and the case structure over the full temperature range of the application. Further, to avoid brinelling damage to the endstone or the pivot, it is customary to support the endstone on a resilient member to allow absorption of motions during the assembly process, while the gyro is dry.

Taut wire suspensions are strongly recommended for many applications where environmental levels are not severe. A fine wire taut suspension system will eliminate OA float motion hydraulic torquing that normally results from OA inversion.

Balancing

To adjust the center of gravity and the center of buoyancy so that they lie in a plane that contains the center of rotation, the assembler should first dry-balance the float, preferably in a set of ball bearings to get as close to the rotational axis as possible. This coarse gravity balance should be completed in both the SRA and IA. Rotation of the float can be used as an indication of lack of balance, although electrical readout from a pickoff would be a more sophisticated method.

Next, the float must be buoyancy-balanced. The balance tank is filled with a fluid which at room temperature will neutrally buoy the float. The float is then end-for-end balanced in the tank. The addition or subtraction of weights at the float ends must be done symmetrically around the output axis so as not to disturb the already achieved gravity balance. Similarly, weights added for gravity balance can also affect the end-for-end buoyancy balance, so it may be necessary to repeat these two balancing operations to satisfy the requirements of each. The float is considered end-for-end balanced when it floats horizontally in the flotation tank. To buoyancy-balance the float around the center of rotation, weights of the same density as the fluid, such as beryllium weights, are inserted in the float in the plane under test. The volume of these weights displaces the fluid, thereby changing the symmetry of the float and adjusting the buoyancy balance. Weights must also be inserted on the opposite side of the buoyancy weights to maintain the previously established gravity balance; these are usually of a high-density material, such as Inconel, so that their displacement is small. However, the buoyancy-balancing weight must be similarly increased to maintain both the buoyancy and gravity balances. Both the SRA and IA axis are so treated.

When coarse balancing of the float is achieved, the gyro is further assembled. Fine balancing of the float in the IA and SRA axes is done when the gyro is filled with fluid. During gyro test, bellows ports provide access to friction-held adjustable float weights, so that adjustments can be made without breaking the hermetic seal of the case. See Figure 18.

4.3.3.5 Transducer Restraint Torques

The a-c transducers, such as the microsyn pickoff, the microsyn-type torque, and the float suspension ducosyns, react to excitation fluctuation as the square power of the current change, causing change of magnetic attraction across the stator/rotor gap as well as change of elastic restraint torque. Therefore, the excitation levels of all a-c excited transducers should be controlled as closely as possible.

The permanent-magnet torquer, on the other hand, is only linearly affected by excitation levels. Therefore, since it is used to provide a bias torque to oppose the known restraints of the gyro, its excitation source must have considerable current control; any fluctuation of the bias currents supplied to the torquer will cause gyro drift.

It is further recommended that the flux return path for the permanent-magnet torquer always be mechanically fastened to the same element as the permanent magnet itself to avoid magnetic mechanical position hysteresis.

Lastly, it is recommended that the magnet assembly be passed through several cycles of magnetization with 5-15% "knockdowns" of gauss level for stabilization. A keeper must be placed around the magnet, if it is separately magnetized, and slid off the magnet during assembly in the gyro (where the return path will take the place of the keeper) without a moment's disruption of return path. It is better, however, to magnetize the magnet in place, after assembly in the gyro and with its final return path. It is also better to use an internal-pole magnet and place it outside the gyro case, so that contamination is lessened within the fluid cavity of the gyro. In this approach, the case wall is part of the magnetic gap of the torquer and must be magnetically permeable.

4.3.3.6 Flexleads

Materials

Flexleads must be very well annealed and heat treated so that no stress relief problems remain to cause gyro drift. Silver copper 85-15 and beryllium copper are excellent materials as they simultaneously provide good electrical conductivity and solderability. Aluminum is recommended only when the environment is severe enough to cause failure (distortion with subsequent gyro drift) of the high density materials. The choice is sometimes dependent on the length of the leads. Crescent or moon-shaped flexleads that are of high density, and large width-to-thickness ratios are customarily used. If the leads sag under their own weight because the span is too great, this will impose unwanted torques on the float and increase the

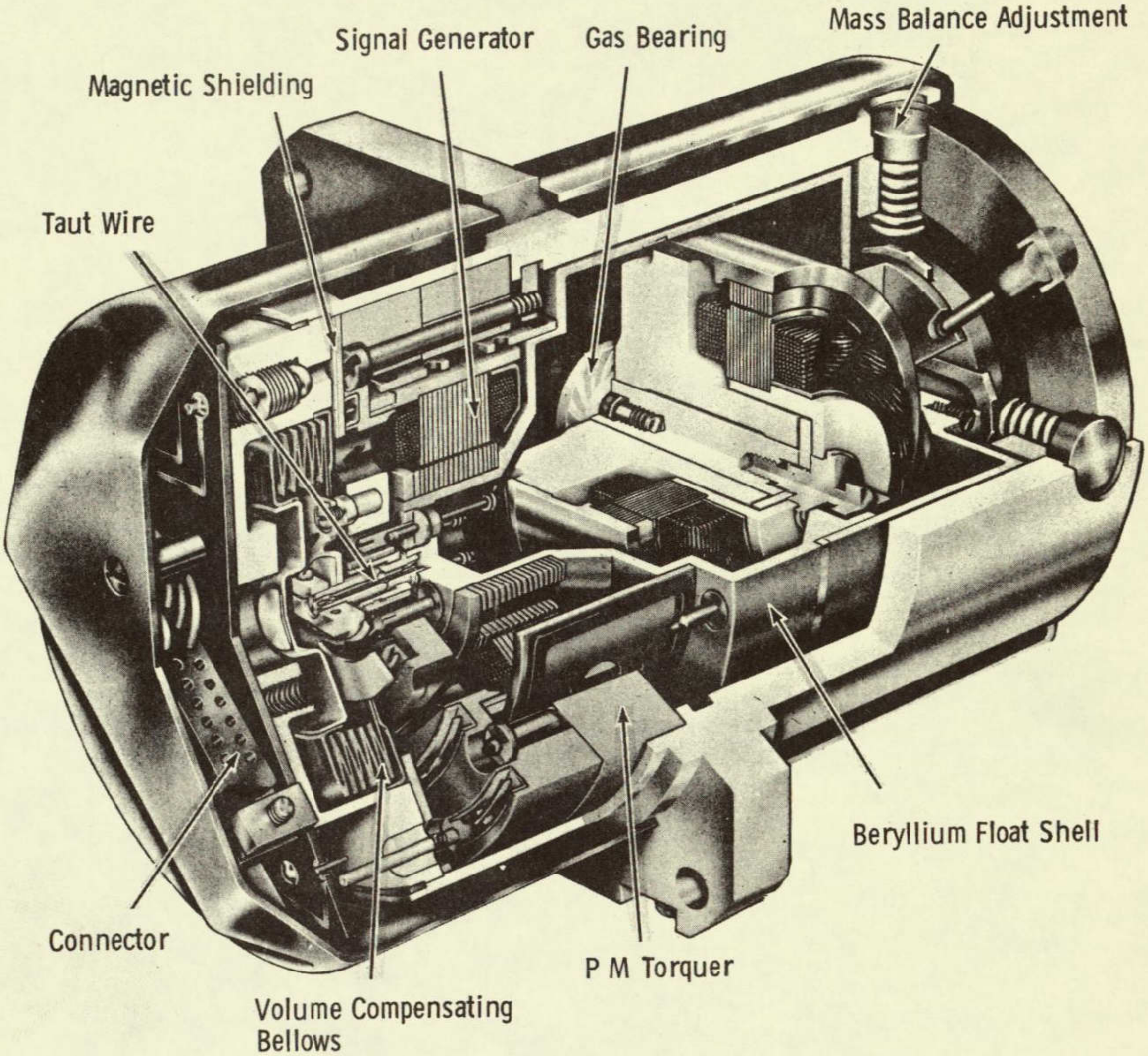


FIGURE 18. GI K7G GYRO (PRECISION PRODUCT DEPARTMENT, NORTHROP CORP.)

likelihood of electrical shorts. With a density of 2.7, aluminum is more buoyant in the fluid and therefore sags less under acceleration inputs.

As an alternate to the crescent configuration, a vertical coil (axial) type of flexlead, running almost parallel to the output axis of the gyro, is often used. Because it is suspended under slight tension, it is better able to support its own weight. The same considerations of material, residual stress, hysteresis, and attachment apply as to crescent-shaped leads.

The use of small spring clips which can be fixture-held and soldered to the flexlead prior to assembly in the gyro provides the advantage of a solder joint that can be inspected outside of the instrument and avoids soldering inside the gyro, which is a common source of contamination. The spring clip, however, can also cause contamination in the form of scraped metal burrs; during installation, the assembler should avoid scraping the terminal, and after the clip is in place it should hold onto the terminal firmly.

As an assembly aid and to avoid contamination, the installation of flexleads may be done with the instrument upside down. Any metal scrapings from the installation of spring clips onto terminals, or any particles of solder or flux, can be washed out of the area with a greater measure of confidence when the gyro is inverted and the interior is viewed with a mirror.

Since any motion of attachment will cause a torque change and an associated drift change, the flexleads must be firmly connected both mechanically and electrically, to their terminals. When the attachment is made by soldering, regular 60/40 (or 63/37) solder should be used to avoid loosening of the connecting lead wires from the opposite end of the flexlead attachment terminals; such leads should be soldered with high temperature (95/5) solder to avoid remelting. All parts should be pretinned so that only momentary heating is necessary. After assembly, any remnants of flux and solder particles should be removed with cleaning solution and a vacuum tip.

Baffles

It is recommended that the flexlead be provided with an electrostatic shield. If a crescent-shaped flexlead is used, a baffle plate with grooves that match the contour of the lead should be installed. These grooves must be coated with a conductive material that is insulated from ground. The conductive material should be at the same potential as the flex lead, so that no electrostatic attraction exists between the flexlead and the grounded gyro case. The baffle also minimizes the effect of fluid convection on the flexlead, which would be transmitted to the float and cause gyro uncertainty torques and drift.

A vertical coil flexlead can be electrostatically shielded and baffled simply by placing it inside a tube made of an insulating material with an internal metal coating that is maintained at the same potential as the lead.

The baffle also protects the flexlead from damage caused by low temperatures, causing high viscosity or freezing of the gyro fluid. Cracking of frozen fluid in the baffle grooves is not as likely to occur as in larger fluid volumes. Flexlead breakage or kinking is thereby avoided.

When gyros are removed from cold storage, their spin motors should not be energized until the gyro heaters have warmed the fluid to a liquid state. Preferably, fluids should be used that do not solidify or get slushy at the lower end of the operational temperature range.

4.4 LIFE

4.4.1 Influence of Temperature

It is often said that for every ten degrees of temperature increase the life of an instrument is halved. This is, of course, an empirical statement; nevertheless, high temperatures have a deleterious effect, because they promote chemical interactions between materials, increase the flow of plastics and solder, and accelerate the degassing of volatiles from plastics in the float. It is therefore recommended that the gyro operating temperature be kept as near as possible to room temperature and that due consideration be given temperature controls (cooling in combination with heating) to allow such operational temperatures.

4.4.2 Design Life

Every instrument should be designed to have a life of at least 5000 operational hours; the average lifetime should be 10,000 hours. To eliminate causes of failure, extensive gyro testing is recommended. Because complete missions of large space systems are often dependent on the successful operation of a single instrument, it is economically justifiable to cause an instrument to fail by extensive testing, both of the subassemblies and of the completed gyro (see reference 11).

APPENDIX A

NOTES ON THE APPLICATION OF HIGH GAIN, FLOATED, SINGLE DEGREE OF FREEDOM GYROS TO SPACECRAFT ATTITUDE CONTROL SYSTEMS

NOTES ON THE APPLICATION OF HIGH GAIN, FLOATED, SINGLE DEGREE OF FREEDOM GYROS TO SPACECRAFT ATTITUDE CONTROL SYSTEMS

Some spacecraft attitude control systems have utilized attitude reference gyros in a somewhat unconventional manner in order to minimize system power requirements. In these systems, precision gyros must be capable of operating with satisfactory performance over a wide specified temperature range, such as ± 20°F, without the requirement for heater power. The desirability of operating the attitude reference without heater power is obvious because of the power and attendant weight reductions that are possible with this design approach. Within certain limitations, this can be accomplished with floated, single degree of freedom gyros because, for most spacecraft applications, the steady state acceleration environment is either zero (free fall) or very low. Thus, the desirability of the float being neutrally buoyant for suspension and stiction reasons becomes a less dominant consideration.

In these applications, the gyros are operated in a closed loop rate mode at all times because of the effect of the wide temperature range on the characteristics of the gyro transfer function. That is, the gyro float is servoed to its pickoff null by means of electronics which sense the pickoff output signal and supply a proportional current to the gyro torquer. In this configuration, the steady state torquer current is proportional to the average input angular rate sensed by the gyro. The torquer current is usually passed through a dropping resistor placed in series with the torquer coil and hence a voltage proportional to input rate is obtained. In order to obtain an electrical signal proportional to angular position, the rate output signal is then integrated by means of (typically) a large capacitor also placed in series with the torquer coil or with an analog operational amplifier integrator. Note that the integration obtainable from the float/damping combination with the gyro operating open loop is not utilized to obtain the position signal. The reason for not using this seemingly simple choice is that the large allowable ambient temperature range associated with this application causes large variations in the gyro damping coefficient. Changes in this parameter result in a proportional variation in open loop transfer function or gain, $\frac{H}{C}$. In this representation, H is the angular momentum of the gyro wheel and C is the damping constant. In this case, gain is a dimensionless quantity and can be represented by $\frac{\text{Gimbal Pickoff Angle}}{\text{Gyro Input Angle}}$. This quantity represents a servo gain in the overall spacecraft attitude control system stabilization loop and as such there are usually constraints placed on its magnitude and allowable variation which are dictated by system design considerations.

A "rule of thumb" which is approximately true for the flotation fluids utilized in most single degree of freedom gyros is that the fluid viscosity doubles for each 20°F decrease in ambient temperature. Thus, a ± 20°F ambient variation will cause the viscosity, and hence the damping coefficient, to vary over the range from half to twice its nominal value. Since the angular momentum H, is constant, the gain thus varies over a 4 to one range. This gain variation is usually not acceptable for satisfactory attitude system performance, which is why the open loop gyro position signal is not utilized for this purpose. The damping constant variation also affects the dynamics of the rate loop, however, these effects can be made acceptably small and this approach is described below.

The gyro output axis transfer function in Laplace transform representation is:

$$\frac{\theta}{T} = \frac{1}{J S^2 + C S} \tag{1}$$

$$= \frac{1}{S(JS + C)}$$

$$= \frac{1/C}{S(\tau S + 1)} \quad \text{where} \quad \tau = \frac{J}{C}$$

or alternately,

$$= \frac{1/J}{S(S + C/J)}$$

where:

θ = gimbal pickoff angle – radians

T = torque applied to the gimbal – dyne cm.

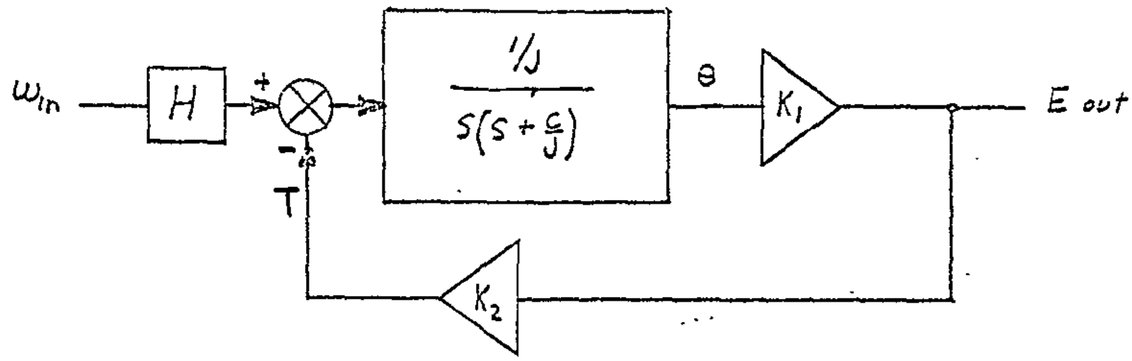
J = gimbal inertia – g-cm^2

C = Gimbal damping constant – dyne cm sec.

τ = gimbal time constant – sec

S = complex frequency rad/sec.

A simplified block diagram associated with the rate loop is shown below:



where:

ω_{in}	=	input angular rate – $\frac{\text{rad}}{\text{sec}}$
H	=	angular momentum – $\frac{\text{gcm}^2}{\text{sec}}$
T	=	torque – dyne cm
K_1	=	gain from pickoff position to output voltage – $\frac{\text{volts}}{\text{rad}}$
K_2	=	gain from output voltage to feedback torque – $\frac{\text{dyne cm}}{\text{volt}}$

All other symbols are the same as defined above.

Utilizing root locus techniques in analyzing the problem, it can be seen that the only effect of variation of gimbal damping coefficient is to change the location of the open loop pole located at $\frac{-1}{\tau}$ or $\frac{-C}{J}$. Changes in the location of the open loop pole will affect the locations of the root loci and, hence will affect the dynamic response of the loop.

Gyros utilized for spacecraft attitude reference purposes typically have nominal open loop gains $(\frac{H}{C})$ and gimbal time constants $(\frac{C}{J})$ of approximately 1 and 0.001 sec, respectively. Thus, in this example, the nominal open loop pole location is at $-\frac{1}{\tau} = -\frac{1}{0.001} = -1000$ rad/sec. Since the value of C is assumed in this case to vary over a range of double to half its nominal value, the location of this pole varies from -500 to -2000 rad/sec. If a bandpass of approximately 20 Hz is desired, the dominant closed loop pole (or poles if a complex pair is involved) will be located approximately 126 radians/second from the origin of the root locus plot on the complex frequency plane. A little reflection will indicate that the temperature caused migration of the open loop pole is likely to have major impact on the location of the dominant closed loop poles for fixed rebalance loop gain. The above argument assumed that the rebalance amplifier signal rolloff frequency (associated with the transfer functions K_1 and K_2) is very high relative to the frequencies being discussed here (typically a rolloff frequency of 1000 Hz or 6280 rad/sec), and hence have negligible effect on the root locus in the low pass frequency region. The result for the nominal and extreme temperature cases is depicted in Figure A-1. In the cold case, the system is overdamped with a rolloff frequency of 60 radians/second. At the high temperature, the system is oscillatory with an undamped natural frequency of 330 radians per second with a damping constant of 0.8. This combination results in a bandwidth of 290 radians/sec. Thus for this example, the system can exhibit an upper rolloff frequency which can change from 60 to 290 radians per second or 9.5 to 46 Hz. In addition to these effects, there are also non-trivial phase variations which occur simultaneously with the above frequency changes.

Many of these difficulties can be avoided by selecting a gyro with suitably high gain. By lowering the nominal value of the damping coefficient C which is equivalent to raising the gain, the range of motion of the open loop pole located at $-\frac{C}{J}$ becomes smaller from an absolute standpoint, and this motion occurs in a region of the S plane where its impact on the closed loop response is reduced. Specifically, reducing the damping causes the pole at $-\frac{C}{J}$ to move closer to the origin, and hence its nominal absolute value is smaller than before. Since the temperature induced damping variations are approximately proportional to the nominal value (i.e., 1/2 and twice nominal for a ± 20°F temperature change), the absolute range of motion of this pole will be less than that of the former case. With this in mind, it is possible to pick a value of C such that the location and temperature induced range of motion of this pole on the negative axis of the complex frequency plane, is small compared to the distance from the origin of the dominant complex closed looped poles which determine the loop response. By this means relative temperature insensitivity can be achieved. There is a tradeoff associated with the selection of this parameter. The choice of extremely small values of C will result in a gyro with essentially a double open loop pole at the origin (i.e., a doubly integrating gyro) and hence, a somewhat more difficult servo loop to stabilize.

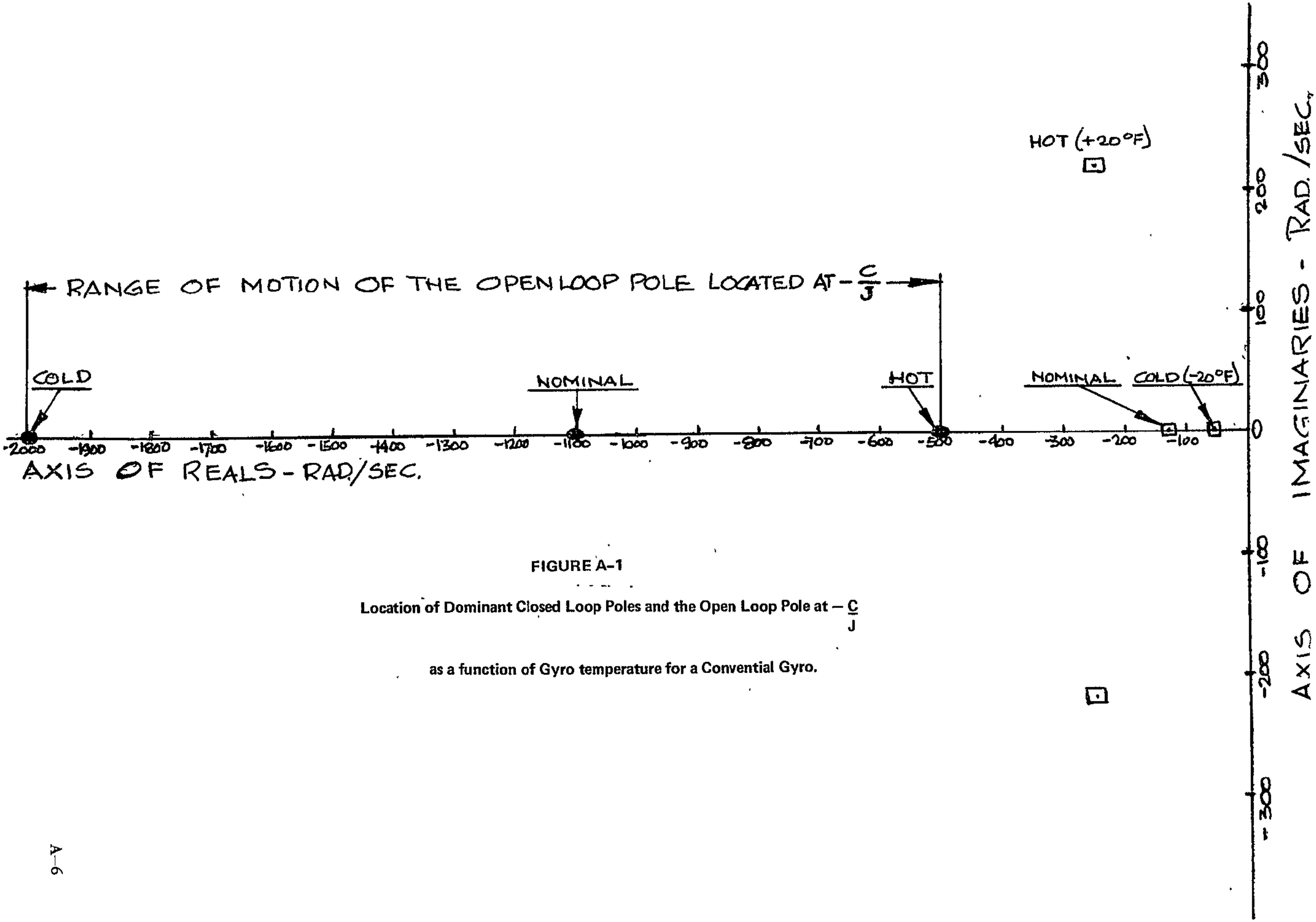


FIGURE A-1

Location of Dominant Closed Loop Poles and the Open Loop Pole at $-\frac{C}{J}$

as a function of Gyro temperature for a Convential Gyro.

A gyro that has been successfully utilized in this manner has a gain of 240 and a gimbal time constant of 0.1 seconds. In this case, the nominal open loop pole occurs at $-\frac{1}{\tau}$ or -10 rad/sec. For the same fluid viscosity temperature characteristics which actually tend to be somewhat smaller, the pole location varies from -5 to -20 radians/second. In this case, the pole is quite near the origin and in order to obtain reasonable values of bandwidth, lead compensation will be required. Given proper compensation, the movement of the open loop pole because of its relatively short travel, has an almost negligible effect on the closed loop response which was set to have an undamped natural frequency of 20 Hz or 126 radians/second. The result is shown in Figure A-2. Thus, the high gain gyro, while it suffers essentially the same percentage change in damping coefficient, yields a rate measuring system which for all practical purposes has a dynamic response independent of temperature. This argument is the dominant reason for the choice of a high gain gyro for reference applications involving relatively loose temperature control.

A further benefit is derived from the use of low viscosity fluid in addition to the temperature stable high gain characteristics since such fluids are easier to use during gyro manufacturing and gyro fluid filling cycles, and are often more stable chemically.

There are some complications associated with the use of these gyros which have occurred in actual practice and should be considered in subsequent spacecraft attitude reference designs. The paragraphs below describe these effects.

The commonly used representation of this output axis transfer function is given in equation (1) and again repeated here.

$$\frac{\theta}{T} = \frac{1}{S(JS + C)} \tag{1}$$

This equation assumes that the unit is an ideal single degree of freedom gyro and hence, the gimbal is ideally restrained so that its only allowable motion relative to the case is rotation along the output axis with zero spring restraint. This ideal is not completely met in an actual gyro design. Most single degree of freedom gyros utilize a pivot and jewel suspension which constrains the gimbal. This suspension is non-ideal since there are small but non-zero clearances between the pivots and jewels which allow motion other than rotation about the output axis.

Under certain conditions, these clearances allow the gimbal to be a two degree of freedom gyro and the resultant gyro dynamic response is quite different from that indicated in equation (1). Of interest here is the freedom of the gimbal to rotate about its input axis. This capability when combined with the low damping associated with high gain gyros can result in significant deviations from the open loop dynamic response indicated by equation (1) and the resulting closed loop response predicted in Figure A-2. The predicted and actual closed loop response actually experienced in a design utilizing high gain gyros is shown in Figure A-3.

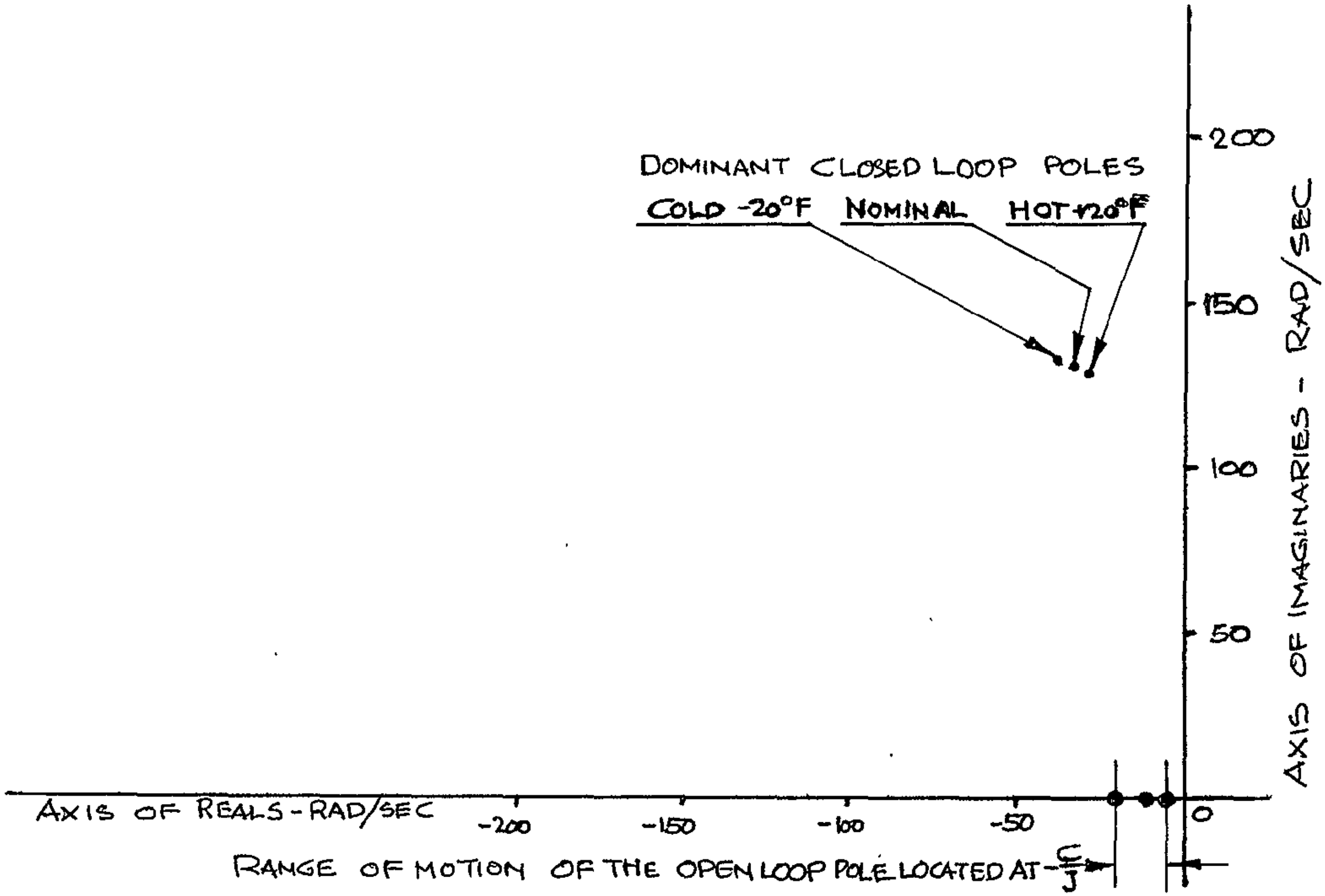


FIGURE A-2

Location of the Dominant Closed Loop Poles and the Open Loop Pole at $-\frac{C}{J}$ as a function of Gyro temperature for a High Gain Gyro. Note that only the upper left quadrant is plotted.

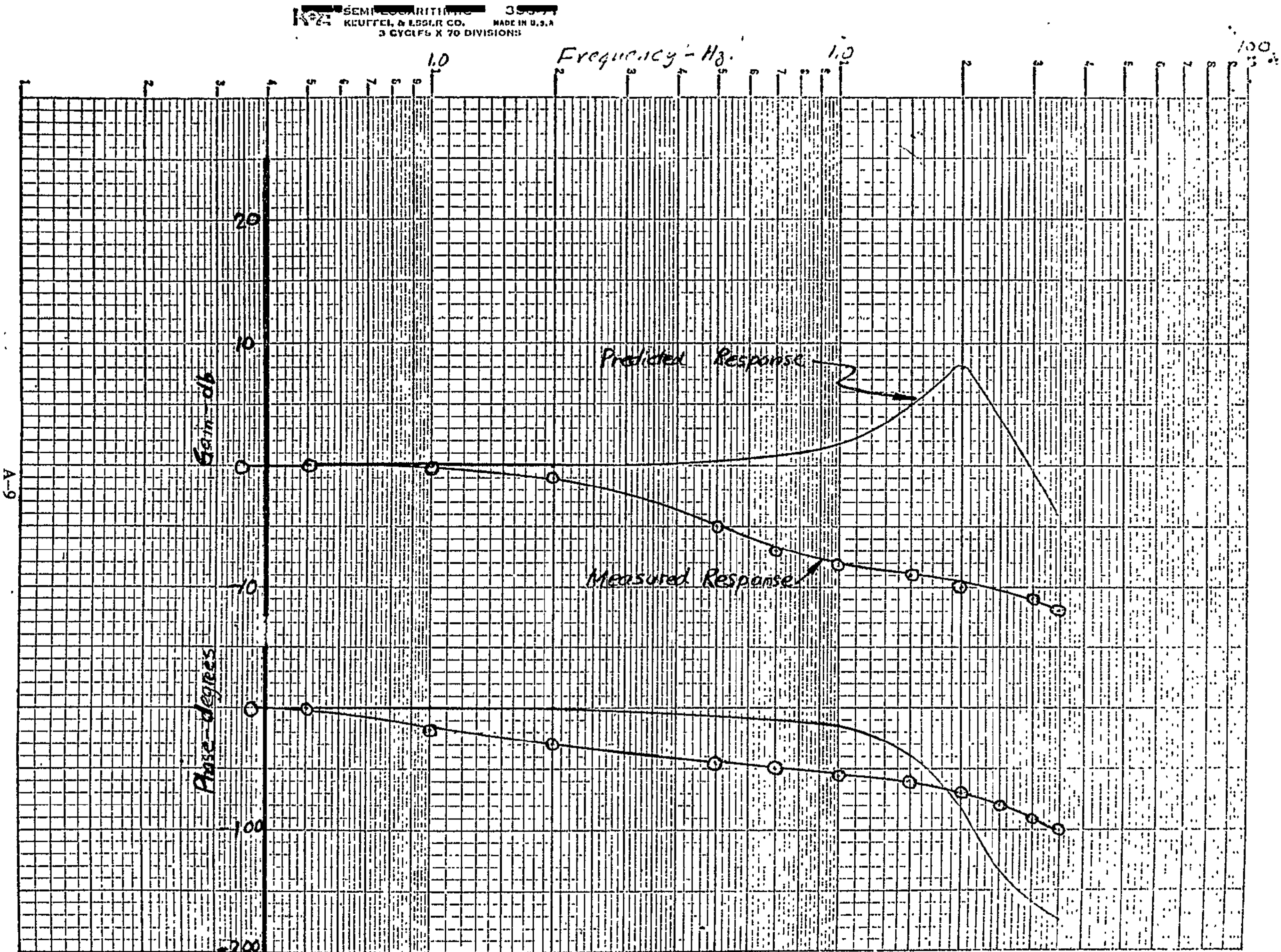


FIGURE A-3

Predicted and Measured Gain and Phase Versus Frequency for a Gyro Rate Loop

Note that the predicted undamped natural frequency corresponds to that of the previous example, being approximately 20 Hz. The actual results were obtained by tests performed on an oscillating rate table and obviously differ significantly from the prediction.

The difference has been shown to be caused by the angular freedom of the float along the input reference axis in combination with a low input axis damping constant. The input axis damping constant is a function of the flotation fluid viscosity and, other things being equal, tends to be low when the output axis damping constant is low as is the case in a high gain gyro. Float motion about the input reference axis results in a more complex dynamic situation which is shown in block diagram form in Figure A-4. This diagram is somewhat simplified from that obtained from a rigorous derivation in order to simplify comprehension. Note that the output axis transfer function has an additional feedback path between the output axis gimbal angle θ and the torque summing junction. This path involves torques about the input axis and their associated motion about that axis. The input axis transfer function is similar in form to that of the output axis except for the term K_i which is the spring restraint of the gimbal about the input axis and has the units of $\frac{\text{dyne-cm}}{\text{rad}}$. In an ideal gyro $K_i = \infty$ in which case the input axis transfer function is zero for all frequencies, and the block diagram reduces to that of an ideal gyro. In a pivot and jewel restrained single degree of freedom gyro, K_i is a non-linear function of input axis angle, θ_i as shown in Figure A-5. K_i is zero when the pivots are free of the jewels, however, when the clearance is taken up in either direction and hence $\theta_i = \pm\delta$ the spring constant increases abruptly to some very high value determined by the structural stiffnesses (which would ideally be infinite).

Note that the input axis feedback path serves to attenuate the amplitude of the ideal gyro transfer function which is normally assumed in most analyses. The result of this effect is an actual gyro transfer function which is significantly lower in amplitude and this materially affects the closed loop frequency response. In most rate feedback loops, this will tend to lower the frequency response as shown in Figure A-3. Because of the K_i nonlinearity, the response is dependent on input amplitude and frequency. For small signal sinusoidal inputs, the pivots never impinge on the jewels, in which case the gyro acts as a two degree of freedom device. On the other hand for large signals, the pivots will be in contest with the jewels most of the time, resulting in performance that is very close to ideal. However, since spacecraft motions tend to be low amplitude at low frequencies, the first case should not be overlooked in the system analysis.

Another effect which tends to attenuate the input axis feedback path is the input axis damping coefficient C_i. If C_i is large, then this coefficient tends to attenuate the feedback and hence makes its effect less apparent. This is probably why the gain attenuation has not been generally observed in conventional single degree of freedom gyros. Reversing the argument, the effect is much more pronounced when high gain gyros are utilized in the design since, other things being equal, the input axis damping coefficient in the above example is some 240 times smaller.

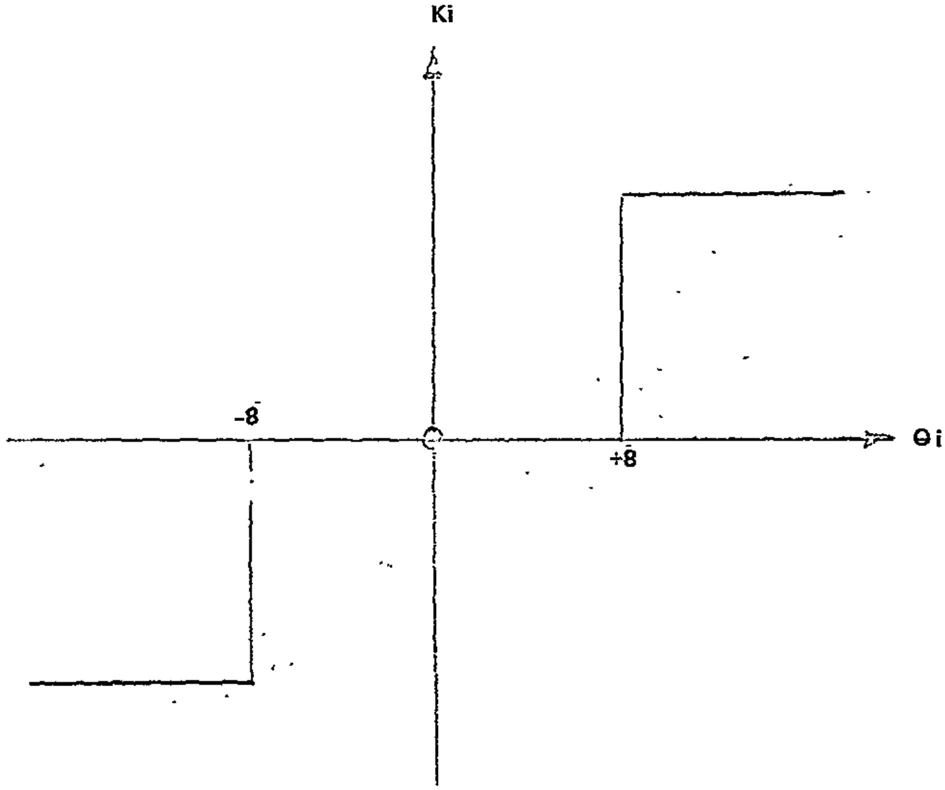


FIGURE A-5

INPUT AXIS SPRING CONSTANT VERSUS INPUT AXIS DISPLACEMENT ANGLE FOR A PIVOT AND JEWEL SUSPENDED GIMBAL

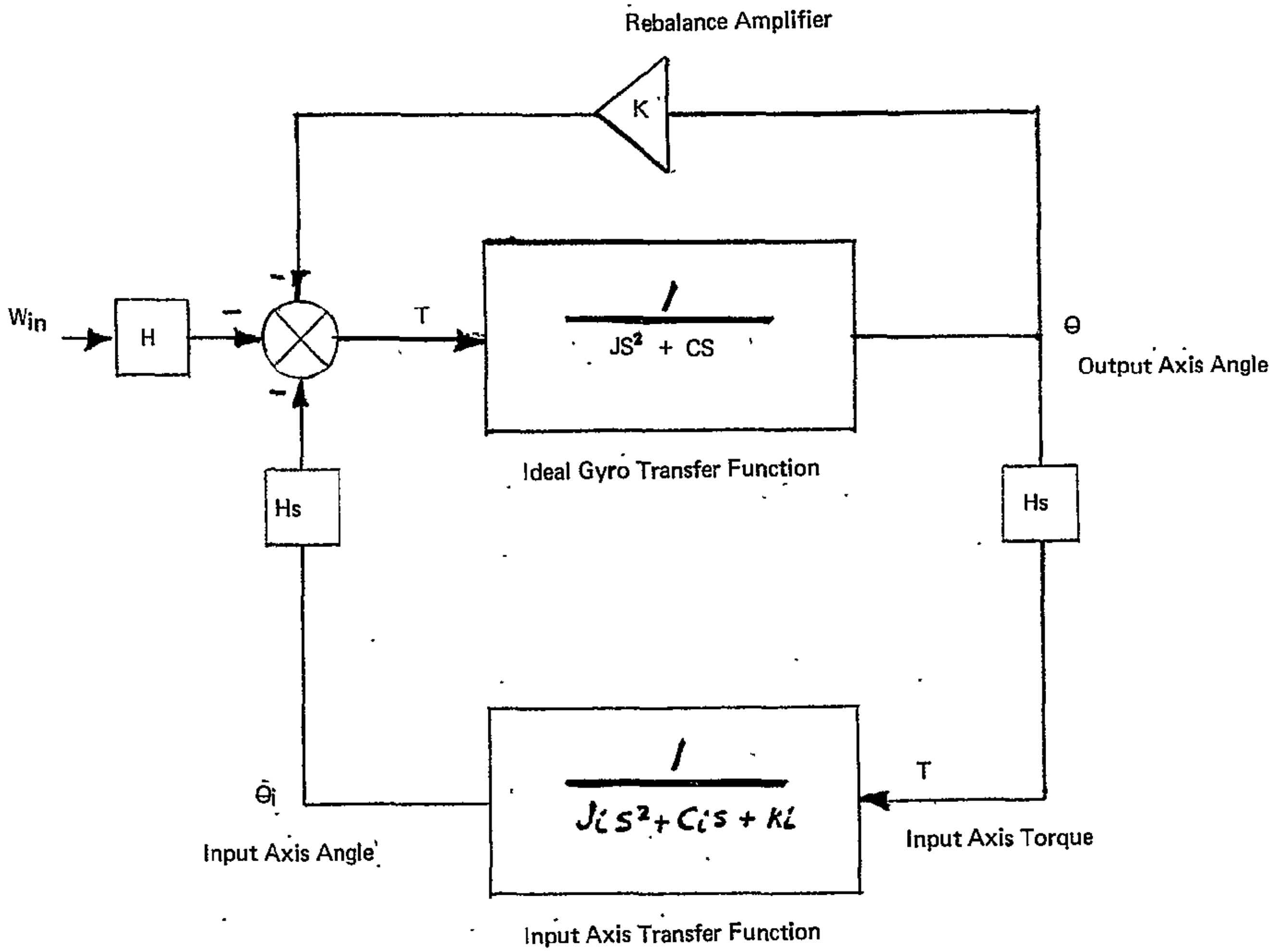


FIGURE A-4

SIMPLIFIED BLOCK DIAGRAM OF A NON-IDEAL SINGLE DEGREE OF FREEDOM GYRO

It should be noted from the block diagram that the attenuation effect is only present when the gyro wheel is running, i.e., $H > O$. When $H = O$, the series elements before and after the input axis transfer function are identically zero and hence the attenuation path is eliminated. This requirement has been verified empirically; by using equivalent electrical torque disturbances with the spinmotor off, the predicted result of Figure A-3 has been observed within measurement error.

The high gain gyro has another possible disadvantage which should be given attention during the system design. The low output axis damping can result in a potential gyro handling problem. Because of the low damping, the viscosity coupling between the float and the gyro case is very small. This means that the float is almost decoupled from the case, being acted upon essentially only by small friction torques from the suspension pivots and jewels. In an open loop mode, if the gyro case output axis experiences an angular acceleration, the decoupled float does not until the case stop impinges upon the float stop, at which time a "step input" of angular rate occurs, resulting in potentially large float angular accelerations. The force couple required for this acceleration is provided by the gimbal stop and the pivots and jewels. Since the forces can be large and the pivots and jewels are delicate, brittle items, there is danger that damage to these parts can occur. The result is that when handling either the gyros or the assembled reference package, extreme care must be exercised in order to prevent gyro damage. This characteristic can be tolerated; however, it does require special precautions and handling, and there is always the fear that damage might have occurred unbeknownst to those responsible for the unit. Conventional (lower gain) gyros are much less sensitive to this condition because the higher viscosity fluid tends to couple the float to the case, resulting in much lower impingement velocities at the gyro stops.

There is another characteristic of ball bearing gyros that are operated over relatively wide temperature ranges which should be considered. As the gyro temperature is lowered, the viscosity of the spin bearing lubricant increases. In some cases, this situation can cause a phenomenon known as "retainer squeal." When this condition exists, the bearing retainer acquires a whirling motion in addition to its normal rotation with the balls. This motion tends to increase the frictional load on the drive motor and has been known to cause wheels to drop out of synchronism with the driving excitation. The resulting loss of wheel angular momentum, H, can cause large changes in the apparent gyro torquer scale factor which is usually detrimental to system operation.

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