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FINAL REPORT

HIGH GAIN ANTENNA CONCEPT DEFINITION AND TECHNOLOGY DEVELOPMENT

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FINAL REPORT
FOR

SPACE STATION HIGH GAIN ANTENNA CONCEPT DEFINITION AND TECHNOLOGY DEVELOPMENT

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FOREWORD

This study was performed by Lockheed Missiles and Space Company, Inc., under contract to the National Aeronautics and Space Administration, Manned Spacecraft Center, R & T Space Station Procurement Section, Houston, Texas. The period for this contract, NAS9-11874, was one year commencing June 16, 1971.

The objective of this study was to perform the first step in the laying of a technology base from which a mechanically gimballed, directional antenna can be developed to support a manned space station proposed for the late 1970's. The effort included the concept definition for the antenna assembly, an evaluation of available technology, the design of critical subassemblies and the design of critical subassembly tests.

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CONTENTS

Section

Section

ILLUSTRATIONS

TABLES

Section 1
INTRODUCTION

Requirements for a wideband data transmission via relay satellite imposes a need for a high gain beam steering antenna in the Space Station communications subsystem. Mechanically gimballed antennas have been utilized effectively on earth where maintenance is readily available. Success also has been realized for several days in the Apollo program and for a few years in unmanned satellites. However, the ten year mission of the Space Station, where maintenance is unavailable, presents considerably greater demands on design techniques and development test techniques.

This report documents the results of the work performed on the Space Station High Gain Antenna Concept Definition and Technology Development Program. This program was conducted under NASA Contract NAS9-11874 with the Manned Spacecraft Center (MSC), Houston, Texas. The purpose of the program was to determine whether currently available technology and analytical techniques are adequate to support the design and development of a high gain antenna tracking gimbal system, capable of surviving 10 years in space. The program goals were:

1. Establish design requirements typical of Space Station missions.
2. Define components which make up a tracking antenna.
3. Perform evaluation of current technology in light of the requirements.
4. Identify areas of inadequate technology.
5. Select and design the components which are considered critical in light of the 10 year life requirement.
6. Prepare accelerated test designs and plans to evaluate the life capability of critical components.
7. Identify specific areas where technology advancement is needed.

Section 2
SUMMARY OF RESULTS

Evaluation of the data compiled on this program indicates that most of the technology required to design and develop the antenna gimbal system is already available. There still remains a question as to the deterioration rates of the active components relative to the time deteriorated capability of the motor.

Tests have been designed to evaluate these effects and, on an accelerated basis, demonstrate the 10 year life capability of each of the components.

In addition, test specimens will be instrumented in order to gain an indication of their relative health and, hopefully, bring about the ability to detect incipient failure long before it actually occurs.

2.1 COMPONENT TECHNOLOGY SUMMARY

The components that make up a tracking antenna are as follows:

(1) Bearings
(2) Gears
(3) Lubricants
(4) Motors
(5) Position Sensors
(6) Power Transfer Joints
(7) R.F. Rotary Joints
(8) Limit Switches
(9) Structure (basic support booms)
(10) Reflector
(11) Feed
(12) Cabling/connectors
(13) R.F. Transmission Lines

Each of these components was investigated on the basis of failure modes, specifically related to the 10 year life requirement. At the end of this phase it was obvious that Items 7 through 13 could be eliminated from further consideration. Each of them is a mechanically passive device analyzable with current techniques and possesses no limitation on the survivability in light of the 10 year life requirement.

Items 1 through 6, however, required a detailed investigation into the causes of the failure modes and these causes will be discussed in great detail in Section 4.

2.1.1 Bearings

Roller, ball, and journal bearings were traded off on the basis of potential failure modes, ball and journal bearings coming out with almost identical scores. A closer investigation into the failure modes of these two types indicated that they could and probably should be used to complement each other in parallel redundancy. This concept is described and illustrated in Section 4.3.1.

2.1.2 Gears

The investigation of gear design and potential failure modes raised a question as to the advisability of their use. Lubrication seemed to be a problem over long periods of time. Wear particles could possibly cause catastrophic failure and there was no known way of preventing this with complete assurance. As a result of this investigation and the study of motor technology, it was determined that they would add no additional capability to the system and their use would only reduce the overall system reliability and confidence level.

2.1.3 Lubrication

At first, lubrication over the 10 year life seemed to be the major problem area in the overall system. An investigation into the generic type of lubricant to be used indicated solid films were the best choice. Failure mode studies indicate that sputtered MoS_2 on the balls and races, along with antimony oxide/MoS_2 combination on the steel ribbon retainers, provide the most optimum lubrication system.

2.1.4 Motors/Position Sensors

The results of the motor and position sensor study indicate that direct drive DC torque motors will provide the highest reliability over the design life. Synchro transmitters were found to be the best choice for position sensing and the commutation signal. This combination necessitates providing a servo followup system within the spacecraft for commutation. The actual commutation will be effected through the use of optical encoders with light emitting diodes (LED's) as the light source. There is some concern about the life expectancy of the LED's. However, placing them within the space station where they can be replaced if necessary eliminates this as a problem. The electronics required with this system are considerably more complex than that required for other type devices; i.e., stepper motors. But, again, by placing these electronics in the shirt sleeve environment, the overall reliability is maintained.

2.1.5 Power Transfer Joint

Motor power and synchro signals will be conducted across the moving interfaces of the gimbal through flat, flexible conductors. These conductors are actually copper strips that have been etched, printed circuit fashion, into Kapton HF* insulation (polyimid-FEP). EMI shielding will be provided through one of two means, silver impregnated epoxy or copper plating.

2.1.6 R.F. Rotary Joints

There is a sufficient number of non-contacting, therefore frictionless r.f. rotary joint configurations, fully developed and qualified for use with either waveguide or coaxial transmission lines. With proper design practices there is no reason to believe that the 10 year life requirement will impose any unusual problems.

2.1.7 Limit Switches

Limit switch life expectancy is based on the number of mechanical cycles. In this design they will be provided as a backup to protect against the possibility that the mechanical limit of the gimbal might inadvertently be reached. The desired number of cycles on these switches is zero. The actual number that might be imposed is unknown, but we can safely assume a very limited total.

2.1.8 Structure

The three environmental factors that are unique to the ten year life are; thermal cycling, outgassing, and ultra violet radiation effects on non-metallic elements. These factors have been experienced on flights of shorter duration and it is only the number of cycles and time duration that make them unique in this case. Sufficient

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*DuPont Trademark

technology is available to predict the effect of these factors with a high degree of confidence.

2.1.9 Reflector

As with the basic structure, thermal cycling, outgassing, and U.V. radiation are the only new concerns. Technology is available to take these areas into account during the design phase.

2.1.10 Feed

The design of the feed should be conducted prudently. Dielectric constants change with outgassing. U.V. radiation can also change dielectric properties. These factors can and should be taken into account during the design phase. Once so accounted for, there should be no concern for the life expectancy of the feed.

One word of caution; solid state devices, i.e., transistors and diodes, should not be placed on the feed or, for that matter, anyplace that cannot be maintained.

2.1.11 Cabling and Connectors

Cabling and connectors, as used here, include both motor/signal cables and r.f. cables.

R.F. cabling wherever possible should be rigid or semi-rigid if coax is used. The insulation (dielectric) should be ceramic beads as this presents the highest resistance to outgassing and temperature effects. In situations where this combination cannot be used, Kapton insulation provides the next most stable material properties.

Motor/signal cables should be insulated with Kapton. They should be shielded from U.V. radiation. Flex points should be limited to places where extreme temperatures will not be present.

With the above considerations and precautions, normal design and analysis practices are sufficient to ensure high reliability over the design life.

2.2 TEST DESIGN

The results of the technology evaluation and trade studies indicated a need to perform life tests on three components; motors, flex cabling, and bearings. Tests for these components have been designed. These tests are expected to show not only <u>if</u> the components will survive but <u>how well</u> they will perform as a function of time (cycles).

In the bearing and flex cable tests, the frictional torque will be measured and a torque vs time relationship will be derived. For the motor torque test, the torque capability of the motor will be monitored and its time dependent relationship will be determined.

The information from the three tests will then be superimposed on the inertial torque requirement to determine the ability of this system to meet the design goals.

In addition to these three primary tests, other combinations of components and testing rates have been derived for further evaluation. These tests and their parameters along with the three primary tests are discussed in Sections 5 and 6.4.

Section 3
STUDY METHODS

Section 7 is a complete bibliography of the sources of published information. Most of these data were obtained through the use of LMSC's Technical Information Center (TIC). The TIC is staffed by a group of fulltime professional literature searchers. Through computer augmentation this group is able to pick out current literature using a simple descriptive term such as "power transfer joint", "magnetic bearings", "life testing", etc. They can also limit the search to a chronological period. In this way, only the latest information is obtained.

The output of the computer searches was a bibliographical synopsis or abstract of the paper under consideration. The technical staff then reviewed these abstracts to determine if the full report would be of value. Literally thousands of abstracts were reviewed and hundreds of the reports were ordered.

The TIC also performed manual searches for specific documents not contained in the computer storage.

Technical publications such as magazines, text books, and conference reports, e.g., ASME, IEEE, etc., were also reviewed on a regular basis.

Specific vendors were contacted by mail and telecon, especially in the area of motor types and potential failure modes of those motors. We met with the vendors that responded to the requests for information and recorded their recommendations and comments.

Lockheed's own in-house specialists were contacted at each step to ensure that we had drawn the correct conclusions from the outside sources.

Section 4

EVALUATION OF AVAILABLE TECHNOLOGY

This section discusses in detail the results of the literature search, trade studies, and final decision and justification for the proposed gimbal design configuration. The techniques used and much of the information contained here is applicable to any mechanism design problem.

4.1 EVALUATION METHODS

It was obvious at the beginning of this study that some method would be necessary to arrive at a standard of comparison. The standard chosen was potential failure modes. Each generic type of component, i.e., motors, power transfer joints, bearings, etc., was considered on an individual basis. Within that generic classification, for instance bearings, one more level of detail was considered, i.e., roller, ball, and journal. This level was then used for the basic study.

Failure modes then were investigated and categorized as to the source of the failure, and possible solutions or preventative measures were devised. Where there seemed to be no real solution to the problem, it was so recorded. When the solution was possible but difficult to control this was also documented.

The next step was to devise a matrix for the failure modes. These matrices then became a tool that was used in arriving at a numerical trade study.

The method used in the trade study is known as the KTA* decision making technique. An example of how KTA is used is illustrated in the Appendix, and it is recommended that the reader study this explanation before attempting to evaluate the trade study results.

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*The Rationale Manager, Kepner-Tregoe Associates, McGraw-Hill, 1965.

4.2 BASELINE DESIGN REQUIREMENTS/GUIDELINES/PHILOSOPHY

The requirements adhered to in developing the gimbal concept are of necessity quite general. At this point in time, the actual space station concept is still in its infancy. Inferences contained in other space station program reports have been taken into consideration in the concept definition of the gimbal.

(1) Aperture - one degree half power beamwidth or maximum diameter of 13 feet. (The current indication from the space station contractors is a trend towards 5-7 foot range).

(2) Frequency - 10% bandwidth between 2 and 15 GHz. (Indications are K band, 13.5-14.5 GHz will be used).

(3) Scan Volume - full 360 degree spherical coverage with two or more antennas. (Two antennas will probably be adequate).

(4) Maximum Tracking Rate - 0.30 degrees per second.

(5) Nominal Slew Rate - 10 degrees per second.

(6) Orbital Altitude - 200-300 nautical miles (indications are 250 nm at 55 degree inclination).

(7) Design Life - 10 years (no maintenance)

(8) Reaction Moments - minimized

A philosophy was developed to ensure the design goals were met:

- Component selection based on a historical background of success
- Where historical success or usage is not available for a particular component, select devices which have fewer insoluble failure modes
- Provide for standby (passive) redundancy where possible
- Provide for active redundancy where passive redundancy cannot be employed.

Using these requirements and guidelines, the following failure mode and trade studies (section 4.3) were performed on the components common to all gimbal systems.

4.3 FAILURE MODES & TRADE STUDIES

4.3.1 Bearings

4.3.1.1 Ball Bearings. The failure mode study has revealed approximately 114 failure sources (not counting random failures). A matrix of failure mode vs source is provided in Table 4-1. A list of the failure modes, their effects and possible solutions follows.

1. CAUSE: Clearance Between Balls and Races Not Correct

   Effect: Excessive eccentricity and vibration.

   Solution: Bearing manufacturers, as a matter of routine, selectively pick races and balls to maintain the correct amount of clearance. Each bearing shall be inspected upon receipt, however, as a matter of routine and as a double check.

2. CAUSE: Balls and/or Races Not of Design Material

   Effect: Corrosion or overstressing of parts.

   Solution: This problem is highly unlikely to occur; however, bearing manufacturer(s) should be required to provide proof (metallurgical report) that proper material has been used. Complete traceability will be mandatory for all bearings and their constituent materials.

MODE NUMBER	BALL BEARING FAILURE MODE/SOURCE MATRIX — MODE \ SOURCE	DAMAGE/ANOMALY RESULTS/INDUCED DURING: DESIGN/ANALYSIS	SUPPLIER MFG. & ASSY. OPERATIONS	SHIPPING	HANDLING	INSTALLATION OPERATIONS	HANDLING OF ASSEMBLED UNIT	TESTING	STORAGE	SHIPPING OF ASSEMBLED UNIT	LAUNCH ENVIRONMENT	EXPOSURE TO SPACE ENVIRON.: VACUUM	THERMAL	RADIATION	CYCLING	FAILURE ATTRIBUTABLE TO: INADEQUATE DESIGN	INADEQUATE WORKMANSHIP	MFG. ANOMALY	INADEQUATE INSPECTION	MODE NUMBER
1	BALLS/RACES SCORED/SCRATCHED		●	●	●	●	●	●	●	●	●						●	●	●	1
2	BALLS/RACES CORRODED	●	●	●	●	●	●	●	●	●	●					●	●			2
3	INCORRECT MATERIAL	●	●													●	●		●	3
4	BALLS/RACES OUT-OF-ROUND	●	●	●	●	●	●	●		●	●				●	●		●	●	4
5	IMPROPER BALL TO RACE CLEARANCE	●	●			●							●			●	●		●	5
6	BALLS NOT OF UNIFORM SIZE	●	●													●	●	●	●	6
7	RETAINER OVERLOAD	●	●			●		●				●	●	●	●	●	●	●	●	7
8	EXCESSIVE AXIAL PRELOAD	●	●			●										●	●	●	●	8
9	LUBRICANT FAILURE — WET								●	●		●	●	●	●	●	●		●	9
9	LUBRICANT FAILURE — SOLID	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●		●	9
10	BEARNING OVERLOAD	●	●			●	●	●		●	●		●		●	●				10
11	BRG. PAIRS NOT IN LINE	●	●			●										●		●	●	11
12	RACEWAY TO O.D./I.D. ECCENTRICITY		●															●	●	12
13	BEARING SURFACE FINISH/ HARDNESS INCORRECT	●	●													●			●	13
14	ERROR IN FIT ON RACE	●	●													●		●	●	14
15	MATERIAL NOT HOMOGENEOUS		●																●	15
16	BEARING CONTAMINATION	●	●	●	●	●	●	●	●	●		●		●	●	●	●	●	●	16
17	LUBRICANT CONTAMINATION	●	●	●	●	●	●	●	●	●		●		●	●	●	●	●	●	17
18	INSUFFICIENT LUBRICANT LUBRICANT NOT INSTALLED	●	●			●										●	●		●	18

Table 4-1

3. CAUSE: Balls, Races Out Of Round

Effect: Binding, Excessive Load, Loss of Accuracy, Decreased Life

Solution: Mechanical signature analysis will identify this type of defect.

Operating the bearings during a "burn-in" period under moderate loading and high speeds should result in higher friction with subsequent localized heathing of defective ball(s) or in region of race causing a located restriction to passage of balls. The use of infrared photography or inspection/screening techniques should clearly identify the defective bearings.

Tight tolerance bearings should be selected, ABEC No. 7 or better.

4. CAUSE: Balls and/or Races Not Designed From Proper Material

Effect: Corrosion, Wear, Fatigue, Overstress of Parts.

Solution: There are already available a number of viable material candidates. Material selection at this point is not considered to be a major problem area.

5. CAUSE: Corrosion on Balls or Races

<u>Effect</u>: Premature failure of bearing

<u>Solution</u>: Bearings should be (1) designed from materials which do not rust (Example 440c), or (2) bearing materials should be protected using extensive precautionary methods such as:

- Assembly of bearing should be accomplished in a controlled humidity atmosphere.
- Persons assembling bearing should use surgical gloves.
- Bearings should be packed and sealed in inert atmosphere until ready for installation.
- Installation should be accomplished in controlled humidity atmosphere by persons using surgical gloves.
- Assembled unit should be stored in controlled atmosphere at all times and removed just prior to testing or flight.
- If oils or greases are used in the bearing, oxidation inhibitors should be used.

6. CAUSE: Balls, Races Marked, Scratched or Other Minute Imperfections

<u>Effect</u>: Unit loading increased at scar - creates asperities of greater magnitude - metal to metal contact subject to adhesion in vacuum environment.

<u>Solution</u>: (1) Vendors would be alerted to the potential criticality in the particular bearing and special procedures set up for

the vendor to follow to ensure against this possibility.

(2) There still remains the possibility that a scratch could result during assembly. The bearing would be inspected upon receipt using the Mechanical Signature Analysis (MSA) technique. Once the bearing has been accepted, extreme measures would be taken to ensure the balls or races were not damaged through handling. In the event some potentially damaging event occurs during handling or installation, it would be re-inspected using the MSA technique before being judged flightworthy and installation completed.

After installation and lubricant run-in has been accomplished, the assembly would again be inspected using MSA. After each phase of environmental testing, inspection (MSA) would be required and finally, just prior to flight, the bearings would undergo a final inspection to ensure that damage has not occurred during shipping, storage, or handling.

7. CAUSE: Retainer Wear-Out

<u>Effect</u>: Contamination of Bearing and Ultimate Failure

<u>Solution</u>: Retainer or separator wear is an extremely difficult problem to attack. This wear is due to sliding contact of the balls on the retainer and, as such, has the same inherent disadvantages as a journal or sleeve bearing.

In a personal interview with Dr. F. J. Clauss (a renowned expert in bearing analysis) he indicated that in his opinion the large majority of bearing failures were caused because of retainer failure. His recommendation was to use reinforced teflon as the retainer material. This is a relatively new technique and not much life history is available. INTELSAT IV used this concept with excellent results through flight qualification. Metal retainers can be coated with a polyimide/molybdenum disulfide coating where loads are high.

8. CAUSE: Improper Preload (Axial)

Effect: Too high a preload would cause unnecessarily high stresses on the balls and races and, therefore, a shortened life.

Too low a preload would allow misalignment of parts, and thus a loss of accuracy.

Solution: One obvious solution to this problem is to use bearings that do not require preloading. This is accomplished by sizing the balls so that there is actually a slight interference fit between balls and races. The obvious effect is that contact stresses, friction, and thus torque, are increased. The bearings will exhibit a shorter expected life. This solution is not being seriously considered for obvious reasons.

The net effect of any preload is a reduction in life. Since this is the case, it seems more advantageous at this point to use no preload if accuracy permits, and take the radial play into account when the tolerance study is conducted. It is expected some preload will be required but every attempt should be made to keep it to a minimum.

Preload can be reduced or eliminated when bearing is blocked and inoperable until vehicle or module is in orbit (no structural load other than centrifugal).

9. CAUSE: Design Lubricant Not Installed

Effect: Possible premature failure of bearings

Solution: While this problem is potentially of a serious nature, conventional Q.A. methods are considered adequate to give good confidence it will not occur.

Proof of design lubricant installation and traceability of material lot employed should be recorded.

10. CAUSE: Loss of Lubricant - Wet

Effect: Premature Failure of Bearings

Solution: One technique which has proven successful on actual flight hardware uses the inherent outgassing of these materials to their advantage. The bearings are lubricated prior to flight with oils of a singular molecular species. A reservoir is provided within the housing to replenish that which leaves the bearings. The reason for using the singular molecular species type of oil is that as the oil evaporates the residual oils left behind have the same characteristics as the original lubricant. The reservoir is composed of an oil filled

nylon fiber matrix. It is sized using conventional kinetic gas theory at equilibrium. ATS III presently has 42 months of mechanically successful operation using this technique. In that particular design, the "overkill" or over design criteria was used. The reservoir was sized to provide 240 years of lubrication (24 gm at $120^{o}F$) even though the design life was 5 years. It is assumed the reason for this margin was the lack of confidence in the analysis techniques. This, however, does not present any serious requirements on the design for 10 year life.
The same concept has been used in the design of NIMBUS E, NIMBUS 1, TIROS and OGO.

11. CAUSE: Loss of Lubricant - Solid Film

Effect: Premature Failure of Bearings

Solution: Based on information received to date, this is an extremely difficult problem to analyze. It is known from tests performed in the past that certain types of solid film lubes do give excellent lubrication for short durations at high and relatively high speeds. The lubricant eventually fails by spalling, causing flakes. The flakes are eventually moved out of the race way and provide no further lubrication.
The possibility also exists that these particles (under zero-g) could migrate to other components and cause contamination or "shorting".
Extrapolation of the available test data to present requirements is a risky proposition at best. Certain lubrication vendors have indicated that they would have no hesitancy

in using their particular processes for the 10 year life design. Individual designers and lubrication experts in the industry are not as optimistic.

At this time a solution to this problem is not in evidence. Tests could be conducted in an attempt to stimulate this particular environment but the results of such a test would not,in all probability, reveal information not already available.

The relative merits of different lubricants will be covered in detail under a separate heading.

12. CAUSE: Overload on Balls & Races Due to Thermal Environment

<u>Effect</u>: Shortened life due to Brinelling

<u>Solution</u>: Conventional analysis techniques are available to predict the thermal contraction of the individual parts. The additive effect of friction-generated heat in combination with environmental heating must be considered. This is also prevented by selection of materials and hardness. High strength alloys are preferred if they will see an extreme thermal environment.

13. CAUSE: Bearing Pairs Not In Line

<u>Effect</u>: Eccentric loading on bearings, higher stresses than expected

<u>Solution</u>: Conventional tolerance analysis and measurement techniques are adequate.

Select tight tolerance bearings at outset.

14. CAUSE: Raceway to O.D. or I.D. Eccentric

Effect: Vibration and higher loads than anticipated with resulting accelerated wear/shortened life.

Solution: Conventional techniques are adequate to protect for this condition.

15. CAUSE: Bearing Surface Finish and Surface Hardness Not Correct

Effect: High friction and wear, surface fatigue.

Solution: Conventional techniques are adequate to protect for this condition.

16. CAUSE: Press Fit on Race Too High

Effect: Possible damage to bearing during installation, excessive stresses on balls and races, loss of lubrication from between balls and races.

Solution: This problem is unlikely to occur; however, from time to time out-of-tolerance parts are manufactured and inspected without discovering the out-of-tolerance condition. Therefore, parts will be re-inspected just prior to installation to "double-check" dimensional accuracy.

Occasionally, the designer will call for a press fit which is not correct. This problem is more difficult to detect. Specific limits will be established for bearing fits based on housing and shaft thermal expansion coefficients and on basic (nominal) housing and shaft sizes. A deviation from these limits will require concurrence of the Program Manager, Thermal Analysts, and Stress Analysts.

Bearing manufacturers recommended that press fits not be used because of the potentially damaging effect. Suggest method other than press fit be used.

17\. CAUSE: Non-Homogeneous Material for Balls and/or Races

<u>Effect</u>: If inclusions are located just below surface of balls or races, surface fatigue would occur prematurely and pitting would result. If the inclusions are deeper into the materials, cracks could develop and propagate to the surface causing spalling and chipping.

<u>Solution</u>: Radiographic inspection of balls and races should eliminate this as a potential problem.
Vacuum melting of bearing materials should assure purity of bearing materials with fewer inclusions.

18\. CAUSE: Bearings/Lubricant Contaminated During Assembly, Shipping, Storage, Handling, Installation, Test, Ascent

<u>Effect</u>: Vibration, binding, surface wear, decreased life.

<u>Solution</u>: This is probably the most difficult failure mode to control or detect. Extreme measures must be taken to protect against contamination and determine if the bearing is contaminated. Surgical glove handling, controlled or inert environment storage and assembly areas, and Mechanical Signature Analysis will be necessary.
While lubricants will be discussed under a separate heading, some discussion is warranted here.

Some solid film lubricants are hygroscopic in nature. Therefore, preventative measures are necessary to guard against moisture contamination which might cause corrosion or lubricant breakdown.

The conventional wet lubricants typically oxidize in air over a period of time and therefore lose their lubricity.

The extent of protection from these problems depends upon a time factor; that is, how long the lubricant is exposed to the environment.

19. CAUSE: Lubricant Not Installed

<u>Effect</u>: Premature failure of bearing.

<u>Solution</u>: Conventional Q.A. techniques considered adequate. Accurate measurement and recording of the precise weight of each bearing should give a possible means of identifying a defective (i.e. no lubricant installed) bearing.

20. CAUSE: Bearing Overload Due to Handling

<u>Effect</u>: Race brinelling or cracking and, thus, reduced life.

<u>Solution</u>: This is perhaps the most difficult failure mode to take into account in the design. One common technique used is over design. A load factor of 3 or 4 can be used to help protect the unit from handling damage. This basically has two effects; increased weight and, thus, inefficiency. It does, however, give a higher confidence in the design life. Another method for protecting against handling damage is extensive handling precautions in the form of procedures,

personnel training, and Q.A. surveillance during every operation. Present Q.A. techniques call for inspection only after a particular operation has been completed. In the final analysis, the solution might be a combination of the two methods.

One extreme method is to package the bearings in a very fragile container and place the packaged bearing and its frangible container in a second, properly cushioned and insulated container, the purpose being to verify that the bearing has not been dropped or otherwise subjected to any impact that could cause ball or race damage. If the fragile/frangible inner container is intact/unbroken, the bearing has not been damaged. Any broken inner container would be cause for rejection at any point; i.e., on receipt, during storage, or at any point until the bearing is mounted/installed.

21. CAUSE: Bearing Overload - Due to Design

<u>Effect:</u> Race brinelling and shortened life.

<u>Solution:</u> Current analysis techniques are adequate to evaluate the bearing design.

22. CAUSE: Bearing Overload - Due to Improper Functional Loads Analysis

Effect: Race brinelling and shortened life.

Solution: Functional loads on the bearing can be analyzed with great accuracy with one possible exception; i.e., vibration. The effect of a local resonance within the structure could input higher loads to the bearing than anticipated. This can be tested for in most cases, but minute brinelling of a race is difficult to detect. A technique such as MSA would reveal the discrepancy. This would mean the bearings would undergo MSA after each vibration test to ensure no damage had occurred.

Another solution which would alleviate this problem would be a bearing lock-out device. This device would remove the load from the bearing during vibration/launch/ascent shock, thus removing the concern.

23. CAUSE: Random Failure

Effect: Premature failure of a bearing.

Discussion: Random failures are those failures which are truly random; i.e., unpredictable. Precautions will be taken to preclude every known source and/or cause of bearing failure. Bearings will be designed to rigid criteria as to material composition, finish, and dimensional tolerances. Rigid quality assurance and inspection criteria will be imposed. Identical bearings will be subjected to accelerated testing to parameters in excess of the operational loading and environmental conditions of the flight articles. The bearings will be subjected to rigid screening tests, including limited operating exposure for Mechanical Signature Analysis purposes and a short term burn-in test that assures against a manufacturing, inspection, assembly, or handling anomaly. At this point we have a bearing that has an inherent operating life substantially in excess of ten years in orbit, based on the bearing having met the above criteria, and assuming that the bearing is not exposed to impact, shock, or other environmental extremes in excess of those designed and tested for. Random failures are premature failures in that they occur before the expected wear out failures. They can occur at any time during the expected normal/intended operating life and are unpredictable, although unavoidable. Random failures are caused by the cumulative combinatorial effects of undetected flaws, imperfections, anomalies, and minor defects in fabrication, assembly, materials, and workmanship, and by mechanical or environmental stresses that have exceeded those expected in severity or at point-in-time and/or location of their application. Since the inherent reliability of a bearing is at best only approached by the achieved reliability, and since random failures can not be precluded, the only way to protect against a premature failure in addition to those already enumerated is to design for failure by providing an alternate path to mission success. With the constraint against preventive maintenance or the replacement of a failed bearing by extra vehicular activity (EVA), the obvious choice is to provide limited redundancy of critical elements. This was considered in the design and trade-off study phase to attain the optimum reliability attainable within the constraints of weight, space, and system design considerations.

4.3.1.2 Roller Bearings. Roller bearings suffer from virtually the same failure modes as do ball bearings with one additional problem:

CAUSE: Rollers skewed

Effect: Excessive noise, wear, and possible seizure.

Solution: There is no known solution for this mode of failure. It can be caused by

- worn retainer
- excessive clearance
- rollers not of consistent diameter
- one end of roller carrying higher load than the other due to misalignment or tapered rollers.

All roller bearings, tapered, spherical, needle, etc., exhibit this type of failure mode.

4.3.1.3 Sleeve Bearings. Sleeve bearings do not suffer from as many potential problems as do rolling element bearings. They do, however, have one disadvantage that makes ball bearings look like the better choice. When a sleeve bearing fails by other than normal wear, it is a sudden and catastrophic seizure with no warning. Ball bearings, on the other hand, do give advance notice in the form of torque increase or noise (vibration).

Sleeve bearing failure rate, however, has been shown to be lower than that of ball bearings and for this reason they can not be ignored as a possible candidate. A list of sleeve bearing failure modes, effects, and possible solutions follows.

1. CAUSE: Shaft Bearing Surfaces Not Concentric

   Effect: Eccentric shaft motion, excessive loads and wear.

   Solution: Conventional methods adequate.

2. CAUSE: Overload - Due to Design/Analysis

   Effect: Accelerated wear.

   Solution: Conventional analysis techniques adequate.

3. CAUSE: Overload Due to Handling of Assembled Unit

   Effect: Eccentric shaft motion, excessive wear.

   Solution: It is extremely difficult to protect against or detect this discrepancy. One method is to have 100% Q.A. or supervision surveillance whenever the assembly is handled. Another possible method is the measurement of shaft eccentricity just after assembly is completed and a recheck after each handling operation is completed. These precautions would help the situation but would not eliminate the possibility of it occurring.

4. CAUSE: Cold Flow of Bushing

   Effect: Eccentricity of shaft with reduced life.

   Solution:
   (1) The use of metallic bushings would eliminate this as a consideration.
   (2) The use of non-metallics which exhibit low cold flow characteristics. This would not eliminate the problem, but would reduce it to acceptable levels.

5. CAUSE: Contamination of Sleeve/Shaft Interface

Effect: Abrasive wear

Solution: Contamination can come from numerous sources; in fact, contamination control during flight is a complete study program on its own and much work is needed to determine methods of stopping contaminant migration in a vacuum at "zero g". Contamination before flight can be controlled to a high degree but it cannot be eliminated. There is no known solution at this time.

6. CAUSE: Incorrect Surface Finish on Bushing and/or Shaft

Effect: Accelerated wear of faying surfaces.

Solution: Current technology is adequate to determine the optimum surface finish, obtain in manufacturing, and inspect that surface.

7. CAUSE: Incorrect Surface Hardness on Bushing and/or Shaft

Effect: Galling of shaft, accelerated wear of bushing, eventual seizure.

Solution: Current heat treat processes are adequate.

8. CAUSE: Thermal Expansion/Contraction of Shaft or Bushing

Effect: Decreased accuracy or seizing of shaft.

Solution: Conventional thermal analysis is adequate.

9. CAUSE: Shaft Scratched

Effect: Gradual milling of bushing (excessive wear). Shortened life.

Solution: Conventional Q.A. techniques are adequate.

10. CAUSE: Shaft Out of Round

Effect: Loss of accuracy of the system, excessive local stresses, possible binding, increased wear rate.

Solution: Conventional Q.A. techniques are adequate.

11. CAUSE: Shaft Bearing Surfaces Tapered

Effect: Excessive local stresses, increased wear rate.

Solution: Conventional Q.A. techniques are adequate.

12. CAUSE: Sleeve ID to OD Eccentricity

Effect: Shaft binding between bushing pairs.

Solution: Conventional Q.A. techniques are adequate.

13. CAUSE: Sleeve Seats Not in Line

Effect: Excessive frictional drag and premature failure.

Solution: Conventional tolerancing methods and Q.A. techniques are adequate.

14. CAUSE: Sleeve I.D. Scratched

Effect: Excessive local wear, probably not serious in result unless wear debris causes binding.

Solution: Reasonable care and Q.A. surveillance should be adequate. The scratch, if small, would be worn smooth fairly early with little or no ultimate effect.
A large scratch (gouge) would probably prohibit installation of the shaft and would require bushing replacement.

15. CAUSE: Sleeve Out of Tolerance

<u>Effect</u>: If the O.D. of a bushing is too large the resultant interference fit would be excessive causing the I.D. to shrink, the housing or the bushing to fail.

If the O.D. is too small or the I.D. is too large, an out of alignment condition would result and thus a loss in accuracy.

If the I.D. is too small, increased frictional drag would result and possible binding under thermal contraction.

<u>Solution</u>: Conventional engineering techniques for design and analysis and current Q.A. surveillance techniques are considered adequate in this case.

16. CAUSE: Sleeve Material Not Compatible With Shaft

<u>Effect</u>: (1) Possible galvanic corrosion prior to flight causing oxides to form at the interface and abrasive wear resulting.

(2) Galling, causing eventual seizing of shaft.

<u>Solution</u>: (1) There are a number of new non-metallic bushing materials available. Their composition, micro-structure, wear characteristics, lubrication characteristics, and structural properties vary as widely as the applications in which they might be used.

(2) It is virtually impossible to find a metallic bushing material which is completely compatible with a given shaft material. Some precautions can be taken to aid the galvanic corrosion problem, such as hard anodize on aluminum. Desiccant materials can be used to help reduce the exposure to moisture. This, however, does not eliminate the possibility of corrosion. At present there is no known solution, short of

complete inert environmental handling, storage, and testing. Solid film lubricants also help eliminate the seizing problem.

17. CAUSE: Sleeve Machined With Tapered Bore

Effect: Increased unit loading and excessive wear.

Solution: Conventional Q.A. inspection techniques can detect this problem if the manufacturing drawings call attention to the requirement of "zero" taper.

18. CAUSE: Sleeve Out Of Round - Installation

Effect: Excessive load and uneven and excessive wear premature failure.

Solution: Conventional Q.A. inspection techniques are adequate to protect for this failure mode.

19. CAUSE: Sleeve Out Of Round - Installation

Effect: Excessive load, uneven and excessive wear, premature failure.

Solution: Special precautions would be necessary both in the design handling and in the assembly techniques to ensure this did not occur. Lead-in chamfers on the bushing and possibly the housing would help in the installation process.

Procedures for pressing the sleeve into the housing would be established so as to ensure an even application of pressure around the bushing was effected during installation. The press would be set up in such a way that the bushing could not get cocked to one side as it was being installed. Roundness would be checked after installation to ensure against the out of tolerance condition.

Another procedure could be used, but presents some extra problems; boring the bushing to size after installation. The problem associated with this procedure comes into play because the actual amount of press (strain) in the bushing is reduced because of the reduction in hoop strength. There is also the possibility that the bushing would "spin" or slip in the housing, thus causing misalignment. Also, any mistakes/errors in the machining job could result in scrappage of a higher value assembly.

4.3.1.4 KTA Decision Analysis of Bearing Types. The KTA (Kepner-Tregoe) method of decision analysis, as described in the Appendix, was used in determining the best type of bearing for use in the Antenna Gimbal Assembly. Copies of the Decision Analysis Worksheet and Adverse Consequences Worksheet are shown as Tables 4-2 and 4-3.

Three different types of bearings were considered in this analysis, namely:

1. Ball Bearing
2. Cylindrical Journal Bearing
3. Roller Bearing

It is significant that bearings (1) and (2), the ball and journal bearings, fared better than the third.

The three different bearings were first considered on the basis of "Musts", with the following items listed as specific MUSTS:

1. Ten year life capability
2. Low speed reversing operation capability (.25 RPM)
3. Vacuum operation capability (1×10^{-13} TORR)
4. Operation over expected temperature range (+200°F to -200°F)

Decision Analysis Worksheet

Decision Statement: SELECT A TYPE OF ROTATING BEARING TO OPERATE IN SPACE FOR TEN YEARS

- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

ALTERNATIVES: A BALL BEARING; B JOURNAL BEARING; C ROLLER BEARING; D

MUST (– Is it absolutely essential? – Is it measurable?)

OBJECTIVES (MUST)	A BALL BEARING INFO	GO/NO	B JOURNAL BEARING INFO	GO/NO	C ROLLER BEARING INFO	GO/NO	D INFO	GO/NO
Ten Year Life Capability	Better	GO	Best - Based on Failure Rates	GO	GOOD	GO		
Low Speed Reversing Operation Capability (.25 RPM)	Best - Based on less friction low failure	GO	Better	GO	GOOD	GO		
Vacuum Operation Capability (1×10^{-13} TORR)	Good	GO	Best - Based on Fewer Parts to Outgas	GO	BETTER	GO		
Operation Over Expected Temperature Range ($+200^{\circ}F$ to $-200^{\circ}F$)	Good	GO	Best - Only Two Mating Parts Both Metallic	GO	GOOD	GO		

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)

OBJECTIVES (WANT)	WT	A INFO	A SC	A WT SC	B INFO	B SC	B WT SC	C INFO	C SC	C WT SC	D INFO	D SC	D WT SC
Low Friction Coefficient	10		9	90		5	50		10	100			
Thrust Load Carrying Capability	8		10	80		8	64		8	64			
Lightweight	5		8	40		10	50		6	30			
Low Corrosion Susceptibility	6		8	48		10	60		8	48			
Simple Lubrication System	4		9	36		10	40		8	32			
High Load Carrying Capability	8		8	64		10	80		9	72			
Low Cost	2		10	20		8	16		9	18			
Simple Assembly	3		9	27		10	30		9	27			
Inherent Failure Detection	5		10	50		1	5		3	15			
Simple Replacement	5		10	50		10	50		10	50			
Simple Process Control	8		1	8		10	80		1	8			
Low Launch Shock & Vibration Sensitivity	2		7	14		10	20		8	16			
High Temperature Resistance	3		5	15		10	30		5	15			
Pointing Accuracy	5		10	50		5	25		10	50			
Simple Design Analysis	4		10	40		7	28		10	40			
Low Hygroscopic Property	5		8	40		10	50		8	40			
Low Bulk Space	2		8	16		10	20		7	14			
TOTAL WT SC				688			698			639			

WEIGHTING
Which of our "Want" objectives is the most important? (10)
What is the relative importance of all other "Want" objectives?
(Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
How well do all alternatives under consideration satisfy each specific objective?
Which alternative provides the greatest satisfaction? (10)
What is the relative satisfaction provided by all other alternatives?

Table 4-2

Objective Setting Worksheet

• PURPOSE OF THIS DECISION

List the things important to you in making this decision in terms of RESOURCES desired and RESOURCES to be used and limits to be observed. — What RESULTS do I expect from this decision (now and future)? — What RESOURCES do I have available in making this decision (now and future)? — What is the ENVIRONMENT in which the selected alternative must exist?

FORM LMSC [illegible] (BACK)

Adverse Consequences

• Do any high-scored alternatives provide low satisfaction of important "Want" objectives?

• Are any alternatives close to the minimum standards of acceptability set?

• Is any information regarding an alternative's satisfaction of an objective missing or based upon assumption?

• What are the likely differences between the developmental and operational enviroments?

• What factors inherent in the alternative are likely to provide threat or dissatisfaction?

• What is the probability of this specific adverse consequence occurring? (1-10)

• How serious or damaging will it be if it does occur? (1-10)

• Does any alternative involve an unavoidable disaster? (10 x 10)

ALTERNATIVE: Ball Bearing	P	S	PXS
Ball or Race Brinelled During Handling	1	8	8
Ball or Race Corroded in Storage	1	8	8
Bearings Misaligned	3	10	30
Bearing Seizure in Operation	1	10	10
Balls Fatigued in Operation	1	10	10
Ball and Race Wear in Operation	5	3	15
Process Control Difficult	10	10	100
Contaminant in Bearing or Lubricant	3	10	30
Ball or Race Scratched	2	10	20
			231

ALTERNATIVE: Journal Bearing	P	S	PXS
Shaft or Sleeve Brinelled During Handling	1	8	8
Shaft or Sleeve Corroded in Storage	1	8	8
Bearings Misaligned	3	10	30
Bearing Seizure During Operation	3	10	30
Bearing Fatigued in Operation	1	10	10
Shaft & Sleeve Wear in Operation	5	10	50
No Inherent Failure Detection	10	5	50
Contaminant in Bearing or Lubricant	3	10	30
Bearing Sliding Surface Scratched	2	10	20
			236

Table 4-3

All three types received a "GO" with varying degrees of excellence. For example, the ball bearing can operate in low speed reversing applications with less friction torque than the journal or roller bearings. It also is less susceptible to running seizure than the journal bearing; i.e., it has rolling contact whereas the journal bearing has sliding contact.

The journal bearing has less load per unit contact area with a given load and volume constraint, and therefore less likelihood of seizure with instantaneous acceleration loads at start or stop.

It is believed that while low friction is not a significant advantage in terms of motor size and power required, the rolling friction that produces the low friction is significant in that it makes the bearing less susceptible to wear and/or seizure while running. The journal bearing is less susceptible to seizure while at rest. However, it is believed that the possibility of seizure due to surface asperities sliding against each other is more likely than seizure at rest. Therefore, the ball bearing is given the best score of the three.

A list of "Wants" was then prepared with the most desirable WANT given a weight of ten and the others a weight of less than ten relative to the most desirable.

The alternatives were then weighed against each individual WANT, with the best alternative given a score of ten and the other two a relative score based on the best.

An example of the method and arguments used to arrive at the relative weighting of the WANTS is as follows:

> Low Friction Coefficient: Among the listed wants this is chosen as the most important because of the low wear and low frictional heat generated, which could cause seizure or loss of pointing accuracy. Both of these are directly detrimental to the function of the equipment.

<u>High Temperature Resistance:</u> This WANT compared to low friction coefficient is rated as a three because proper design and fabrication can eliminate the possibility of a serious failure caused by seizure, and material temperature limits are within the extreme temperatures expected. Insulation is also possible to narrow the temperature extremes.

The relative scoring of the alternatives with respect to a WANT is given below:

<u>Roller Bearing (10):</u> Of the three types of bearings, this has the best published friction coefficient of .0011.*

<u>Ball Bearing (9):</u> The ball bearing was given a score of 9 because of the low friction coefficient inherent in rolling contact. Published data show that ball bearings have a running friction coefficient of .0015*, which is close to the best for roller bearings.

<u>Journal Bearing (5):</u> The score assigned to this type was based on the published running friction coefficient of .15, ** which is considerably higher than either of the other two.

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*Bearing Technical Journal, FMC Corporation, Link-Belt Bearing Division, Indianapolis, Indiana, 1970.

** Fuller, Dudley D., Theory and Practice of Lubrication for Engineers, John Wiley & Sons, New York, 1956.

The weights and scores are subject to the judgement of the individual making the comparison and therefore can vary from person to person. In an attempt to eliminate this weakness, two comparisons were made by two different individuals. Both of them came to the same conclusion but with differing individual scores.

The Assumptions made in these comparisons were the following:

1. The launch shock and vibration loads will be taken by a device that will "short" out the bearings.
2. The contact area of a roller or journal bearing is larger than the ball bearing for a given rated load.
3. The materials used in the bearings are all metallic except that the ball and roller bearing retainers are non-metallic and therefore not as temperature resistant.
4. All bearing types have thrust load carrying capability. Journal bearings are designed and fabricated with thrust load carrying capability.

These Assumptions were necessary in making a determination of the relative merits of each bearing.

A comparison of adverse consequences was conducted and is presented in Table 4-3. The results of such a comparison were that the ball bearing appeared better. In reviewing each adverse consequence, it becomes evident that two items stand out, namely:

Alternate	Adverse Consequence
Ball Bearing	Difficult Process Control
Journal Bearing	No Inherent Failure Detection

Since the lack of failure detection is inherent in the journal bearing, a solution to eliminate it is not readily apparent and therefore it is not considered a good choice for the primary mode of operation.

The ball bearing, on the other hand, has difficult process control as its adverse consequence, which appears to be solvable with proper controls and close monitoring of the vendor's process. Close cooperation between the vendor and prime contractor is necessary to successfully control the manufacture of bearings. Inspection procedures and check points need to be worked out and mutually agreed to by both parties.

The analysis as described and as indicated on the Decision Analysis Worksheet is not conclusive as to the best bearing type but instead, shows that the ball or journal bearings are comparable in weighted score (688 & 698) with the ball bearing slightly ahead because of solvable process control as an adverse consequence.

In considering the design philosophy as described in Section 4.2, it was concluded that these two types of bearings could be used to complement each other to provide passive parallel redundancy. With this configuration (see Figure 4-1) the ball bearings will be in usage until the torque required to cause them to rotate, increased to a point that it is equal to or greater than the torque required to cause the sleeve bearings to operate. The sleeve bearings would therefore be passive devices until needed.

An alternative to this would be use a parallel ball bearing arrangement also shown in Figure 4-1. In this case, the outer set of balls would be subjected to static loading which has been shown in past applications to be an extremely severe environment for ball bearings while it is non-detrimental to sleeve bearings.

4.3.2 Gears

Gear tooth wear and failure can be grouped into four general classes.

1. Wear
2. Surface fatigue
3. Plastic flow
4. Breakage

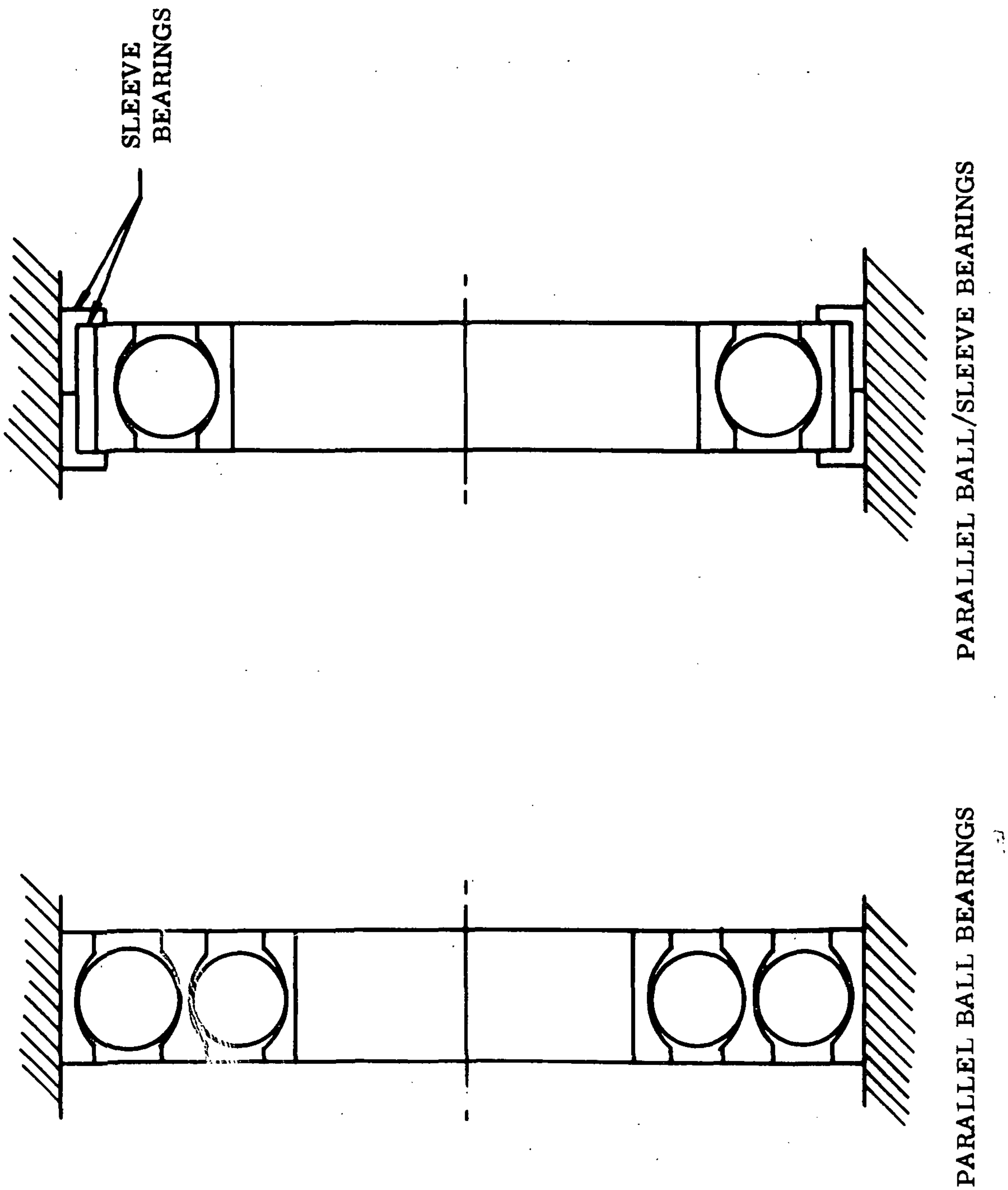

SLEEVE
BEARINGS
PARALLEL BALL/SLEEVE BEARINGS
PARALLEL BALL BEARINGS

Figure 4-1

Two or more of these types may occur simultaneously, or one may be the result of the continued or progressive nature of another. This section will discuss the four classes of failures and their causes. A matrix of gear tooth failure (Table 4-4) is included at the end of this section. The matrix summarizes the gear failure discussion.

Wear. Wear is a general term describing loss of material from the contacting surfaces of gear teeth. Normal wear is the slow loss of metal from the contacting surfaces at a rate that will not affect satisfactory performance within the expected life of the gears.

Destructive Wear. Destructive wear is surface injury deterioration, or change in the tooth shape caused by wear to such an extent that the life is appreciably shortened or the smoothness of action is impaired. It may be any, or a combination of any, of the types of wear described in the following paragraphs.

Abrasive Wear. Abrasive wear is surface injury caused by fine particles passing through the gear mesh which may be carried by the lubricant or embedded in the tooth surface. These particles may be dirt not completely removed, sand or scale from castings, impurities in the oil or from the surrounding atmosphere,or metal detached from the tooth surfaces or bearings.

Scratching. Scratching is a severe form of abrasive wear, characterized by short scratch-like lines or marks on the contacting surfaces in the direction of sliding. It may be caused by burrs, projections of the tooth surface, or material imbedded in the tooth surface, or hard foreign pieces passing through the gear mesh. Scratching damage is usually light and does not necessarily result in progressive destruction provided the cause is removed.

Scoring. Scoring is the rapid removal of metal from the tooth surfaces caused by the tearing-out of small contacting particles that have welded together as a result of metal to metal contact. Scoring is usually caused by a rupture of the lubricating film resulting from load concentration at localized contact areas. Excessive unit loading or an unsuitable lubricant has the same effect.

Interference Wear. Interference wear is a particular kind of scoring that occurs when improper or premature contact concentrates the entire load at the point of engagement of the driving flank with the mating tip or at disengagement on the driven flank and the mating tip.

Corrosive Wear. Corrosive wear is surface deterioration from the chemical action of acid, moisture, or contamination of the lubricant.

Burning. Burning is the result of excessive temperature, either from external sources or the excessive friction from overload, overspeed, or inadequate lubrication. Burning can result in severe wear and surface deterioration of the previously described types due to loss of hardness from high temperatures.

Surface Fatigue. Surface fatigue is the failure of the material as a result of repeated surface or subsurface stresses that are beyond the endurance limit of the material. It is characterized by the removal of metal and the formation of cavities or pits. Surface fatigue pitting can be characterized by three different types; corrective pitting, destructive pitting, or spalling. Pitting is fatigue failure; it is almost impossible to make a gear pit without about 10,000 cycles of contact or more. If even more load is applied than the load required to cause pitting, in the range of 10,000 to 20,000 cycles, the usual result is that the surface is rolled or peened.

Corrective Pitting. Corrective pitting occurs in localized overstressed areas and tends to distribute the load by progressively removing high contact spots. When the load is more evenly distributed, the pitting action is reduced and eventually polishes over. These pits are quite small, generally in the range of 1/32 to 1/16 inch in diameter.

Destructive Pitting. Destructive pitting is more severe and the pits are larger in size than corrective pitting. Pitting of this form progressively increases the size and number of pits until smoothness of operation is impaired. Large pits formed by the joining of smaller adjacent pits are due to a failure of the material between them and constitute a form of spalling.

Spalling. Spalling is a sporadic fatigue failure occurring usually in case hardened steel; originating with a surface or subsurface defect or from excessive internal stresses due to heat treatment.

Plastic Flow. Plastic flow is the surface deformation resulting from the yielding of the surface metal under heavy loads. It is usually associated with the softer metals, but may occur in through-hardened and case-hardened steels. Plastic flow can be characterized by three different types; rolling and peening, rippling, and ridging.

Rolling and Peening. Rolling and peening almost always occur together as the result of the sliding action under excessive loads and the impact loading from improper tooth action.

Rippling. Rippling is a wave-like formation on the surface at right angles to the direction of sliding. It may be caused by surface yielding due to "slipstick" friction resulting from inadequate lubrication, heavy loads, or vibration.

Ridging. Ridging usually appears as diagonal lines or ridges across the tooth surface, but may have a herringbone pattern occurring in the direction of sliding. Ridging is similar in nature to rippling; however, ridging is more severe because the surface imperfections are deeper. Ridging is generally associated with excessive loads or inadequate lubrication and, usually, complete failure results unless the material has a great capacity for work hardening.

Breakage. Breakage is the fracture of an entire tooth or substantial portion of a tooth.

Fatigue Breakage. Fatigue breakage is the most common type of failure by breakage. It results from repeated bending stresses that are above the endurance limit of the material. Such stresses can result from poor design, overload, misalignment, or from inadvertent stress raisers such as notches, surface or subsurface defects, etc. It originates as a crack on the loaded side usually in the fillet at the edge of the face, and progresses to complete failure - either along the root or diagonally upward across the tooth.

Overload Breakage. Overload breakage is a rather uncommon type of breakage resulting from sudden shock overload. Misalignment which concentrates the load at one end of the face is usually the cause, but it may also be caused by wedging of the teeth due to bearing failure, bent shafts, or large pieces of foreign matter entering the mesh.

4.3.2.1 Gear Failure Analysis. This section relates to gear tooth failure and preventative measures for each failure. The main body of the section discusses all known causes of failure and their individual solutions. Twelve failure causes and solutions are considered. An abbreviated chart (Table 4-5) summarizes the failure causes and their prevention.

In order to preclude failure modes, thought should be given to the following:

1. Types of friction involved in a gear train
2. Selection of gear materials and surface preparation of materials
3. Optimum lubrication for a specified application.

Type of Friction. Combination of rolling and sliding friction. When gear tooth geometry permits, rolling friction (hydrodynamic) exists in areas that can maintain a fluid film. Where the gear teeth contact we encounter boundary friction that requires a solid or semi-solid film to prevent asperities of opposing surfaces from undergoing shear or plastic flow. The use of proper lubricant becomes important and will depend on speed and load.

Gear Materials. Selection of gear materials is very important. If hydrodynamic lubrication is optimum, the opposing gear materials can be of different hardnesses so that the softer can shear readily during the brief sliding period. If boundary or solid film lubrication is required, the opposing gear materials should be of equivalent hardness $> R_C$ 55. Surface preparatives should not be of the type that would include "work hardening" or stress corrosion, e.g., case hardened depth should be adequate but not excessive. RMS or RCR finish should be as low as possible to prevent high unit loading and frictional heat.

Lubricant Selection. Lubricant selection is based on service requirements. High speeds require a grease that contains an oil and a thickener that could serve as a boundary lubricant. Low speeds require a solid lubricant. If a grease or oil is used, the gear train should be shielded and sealed to minimize loss through volatility and designed to provide replenishment when required. Solid lubricants should be applied for adequacy to number of service cycles required. If replenishment is required, an idler gear composed of lube composite can be used to transfer solid lubricating particles to the gear tooth load bearing surfaces.

4.3.2.2 Gear Failure Causes, Effects, and Solutions. Now that we have defined the failure terminology for gears, we can begin to consider the causes of those failures and the possible solutions, as shown in the following list.

1. CAUSE: Fine particles passing through gear mesh.

   Effect: Abrasive wear. The particles may be metal detached from the gear teeth or bearings, abrasives not completely removed prior to assembly, sand or scale from castings, or other impurities in the oil or surrounding atmosphere.

   Solution: Particles of contamination can be eliminated by ultrasonic cleaning of all parts and exercising control to assure that they remain free of contamination through assembly. This will become a requirement of the engineering drawings. Prudent design and lubrication will prevent particles from spalling off gears and bearings. (See para. 4.3.3.1).

2. CAUSE: Burrs, projections, inclusions

   Effect: Scratching and accelerated wear

   Solution: Existing quality assurance inspection techniques (red-line, optical enlargers, etc.) are adequate to ensure that these conditions will not exist in the gears selected for the high gain antenna drive.

3. CAUSE: Loss of Lubricant

   Effect: Scouring, burning, pitting, spalling

   Solution: (wet lubricant): The gears are lubricated at assembly using a single molecular type of oil. A wick applied to the gear meshes (perhaps as an idler gear) with a reservoir of oil will replenish the oil as it is needed.

Solution (solid film lubricant): Solid film lubricants usually fail by spalling, creating flakes of lubricant. The flakes move out of the gear tooth mesh area and provide no further lubrication. This can be taken into account for the most part by proper design (see Para. 4.3.3.1).

4. CAUSE: Unsuitable Lubrication

Effect: Excessive wear or corrosion

Solution: Conventional techniques are adequate to protect against this condition.

5. CAUSE: Irregularities of Gear Tooth Surface

Effect: Load concentration (excessive unit loading) resulting in scouring, pitting, spalling, burning, rolling, peening, overload breakage.

Solution: Existing technology is sufficient to provide adequate load carrying capability and assurance that the tooth form is correct (no local contact areas) to preclude these failures.

6. CAUSE: Impact Loading

Effect: Pitting, spalling, rolling, peening, and overload breakage.

Solution: The impact loads imposed on the antenna drive will be well defined. Adequate design will eliminate this failure mode.

7. CAUSE: Notches, Cracks, Inclusions

   Effect: Fatigue breakage

   Solution: Existing quality assurance inspection techniques will eliminate this type of failure.

8. CAUSE: Misalignment of Mating Gears

   Effect: In most cases misalignment of mating gears is catastrophic.

9. CAUSE: Misalignment - Manufacturing

   Solution: Conventional quality assurance inspection makes this a very uncommon failure, and the addition of mechanical signature analysis should completely eliminate this type of failure.

10. CAUSE: Misalignment - Bearing Failure

    Solution: Bearing failure will be prevented as discussed in Section 4.3.1.

11. CAUSE: Misalignment - Bent Shaft

    Solution: A shaft may be bent before installation or during service. The mechanical signature analysis inspection will expose this type of defect. Proper servo drive design will preclude the possibility of a shaft being bent in service.

GEAR TOOTH FAILURE MATRIX

FAILURE MODE		WHERE FAILURE MAY ORIGINATE: DESIGN	MANUFACTURING	ASSEM. & TEST	SERVICE: LAUNCH COND.	SERVICE: SPACE COND.	CAUSE OF FAILURE: FINE PARTICLES PASS THRU GEAR MESH	BURRS OR PROJECTIONS IN GEAR TOOTH SURFACE	LUBRICATION: LOSS OF LUBRICANT	LUBRICATION: UNSUITABLE LUBRICANT	LUBRICATION: LOCAL RUPTURE OF LUBE FILM	LUBRICATION: EXCESSIVE UNIT LOADING	EXCESSIVE UNIT LOADING	IMPACT LOADING	SURFACE DEFECTS NOTCHES, CRACKS	SUBSURFACE DEFECTS CRACKS, INCLUSIONS	MISALIGNMENT: MANUFACTURING TOLERENCES	MISALIGNMENT: BEARING FAILURE	MISALIGNMENT: BENT SHAFT	LARGE PARTICLES IN GEAR MESH
WEAR	ABRASIVE		●	●		●	●													
	SCRATCHING		●	●		●		●												
	SCORING			●	●	●			●	●	●	●					●	●	●	
	INTERFERENCE	●	●														●	●	●	
	CORROSIVE		●	●	●	●				●										
	BURNING		●	●	●	●			●	●		●	●				●	●	●	
SURFACE FATIGUE	PITTING AND SPALLING		●	●	●	●			●	●	●	●		●			●	●	●	
PLASTIC FLOW	ROLLING AND PEENING		●	●	●	●							●	●			●	●	●	
BREAKAGE	FATIGUE			●	●	●									●	●				
	OVERLOAD	●		●	●	●							●	●			●	●	●	●

Table 4-4

Cause of Failure	Failure Mode	Recommended Preventative
Fine particles pass thru gear teeth	Abrasive wear	Ultrasonic clean all parts. Exercise P.A. control to assure assy. free of contamination.
Burrs or Projections in gear tooth surface	Scratching (Wear)	Exercise P.A. inspection (red-line, optical, etc.). Also Mechanical Signature Analysis of Assy.
Loss of Lubricant	Scoring Burning Spalling	Replenish lubricant from reservoir as needed.
Unsuitable Lubricant	Corrosion Scoring Burning Spalling	Adequate research and design.
Local rupture of lube film.	Scoring Spalling	P.A. inspection of tooth form to preclude local contact areas. Adequate lubrication.
Excessive unit loading ruptures lube film	Scoring Burning Spalling	Adequate design, mechanical signature analysis.
Excessive unit loading, lube film intact.	Burning Rolling & Peening Overload breakage	Adequate design, especially materials.
Impact loading	Spalling Rolling & Peening Overload breakage	Define loads, provide adequate design.
Surface defects notches, cracks etc.	Fatigue	Product assurance inspection red-line, magna flux, radio-graphic
Subsurface defects cracks, inclusions, etc.	Fatigue	Product assurance radiographic inspection.
Manufacturing toler-ances create miaslign-ment	Wear Surface fatigue Plastic flow Breakage	Product assurance conventional inspection, plus mechanical signature analysis.
Bearing failure creates misalignment	Wear Surface fatigue Plastic flow Breakage	See bearing failure section in Aug. 1971 progress report; includes design, lube, installat-ion & mech. sign. analysis.
Bent shaft creates misalignment	Wear Surface fatigue Plastic flow Breakage	Mechanical signature analysis to eliminate preservice problems. Adequate design to prevent in service problems.
Large particles in gear mesh.	Overload breakage	Protect gears by enclosing in a housing. Design to prevent random fragments from entering gear mesh.

Gear Tooth Failure and Failure Prevention

Figure 4-5

12. CAUSE: Large Particles in Gear Mesh

<u>Effect:</u> Overload breakage

<u>Solution:</u> This is a rather uncommon failure even in commercial applications. Only a catastrophic failure of gear or some other component could provide the large particles.

4.3.2.3 <u>Summary.</u> To assure reliability of the gear box, the gears and other components will be:

1. Inspected by conventional QA procedures
2. Red-line inspected
3. Radiographically inspected
4. Ultrasonically cleaned
5. Assembled in a clean atmosphere
6. Tested with mechanical signature analysis.

The precautions listed coupled with adequate design should preclude any and all gear failures except those resulting from loss of lubrication.

It is proposed that we would replenish the lubricant lost via a reserve supply of lubricant fed through an idler gear. Sufficient work has been done in this field to impart a reasonable level of confidence to the concept.

4.3.3 Lubrication

This section discusses the cause, effects, and solutions for the three basic types of lubricant; oil, grease, and solid film. Justification for the final choice of lubricant is also presented.

4.3.3.1 Failure Analyses and Solutions. The following list summarizes the two primary causes of lubricant failure, the effects, and the solutions.

1. CAUSE: Loss of lubricant

   Effect: Increased torque, eventual seizure

   Solution - oil: There is no known way to prevent outgassing of an oil under vacuum conditions. Seals to date have not been advanced to provide for "zero" leakage. The technique used in the past has been to provide enough replenishment lubricant, based on kinetic gas theory calculations, so that it would last the full design life with a sufficient margin of safety. This, however, presents contaminants to surfaces on the Space Station. Previous applications were designed around the hydrodynamic lubricant film present at higher speeds than will be seen in the antenna gimbal. This method, even though practical from the standpoint of outgassing, would probably not be justified due to the boundary lubrication required at the lower speeds.

   Solution - greases: Greases suffer from the same disadvantages as oils. In fact, the thickener is nothing more than a reservoir for the oil. The thickener can become a contaminant once it has lost a sufficient amount of oil to become hardened.

   Solution - solid film: The loss of lubrication from a solid film is exhibited by chipping or spalling of the film surface as opposed to outgassing. Calculation of the life of a solid film lubricant is based primarily on the distance the mating surfaces are required to travel over the design life. These calculations are well within the "state-of-the-art" and current processes for achieving the design configuration are already available. The loss of lubricant, in this case, is not considered to be a problem.

2. CAUSE: Lubricant Contamination

<u>Effect:</u> Loss of lubricity, surface corrosion, increased torque, possible seizure.

<u>Solution - oil:</u> Extreme process control measures would be necessary to ensure the lubricant was not contaminated during manufacture, during shipping, or by the user. Care would have to be taken to ensure material compatibility within the housing so that outgassing of other materials, e.g., insulation or other organics, would not combine chemically with the lubricant to form corrosive elements, solidify into abrasive salts, or provide a catalyst for some other secondary chemical reaction.

<u>Solution - greases:</u> The same precautions taken when considering oils would be necessary. One additional problem arises, that of thickener or hardener causing particulate contamination. This problem is inherent in the use of greases with no known solution.

<u>Solution - solid film:</u> Some currently available solid film lubricants are hygroscopic. Their use requires close controls to reduce the amount of moisture absorbed to a minimum. The component on which this lubrication system is used should be corrosion resistant on its own. Particulate contamination can occur from the solid film surface through flaking. This is the main drawback in the use of solid films. This,

however, can be reduced to a minimum by using extremely thin but adequate coatings of the film without concern for degrading the 10 year capability.

4.3.3.2 Lubricant Trade Study. Attempts to perform a KTA decision analysis on the type of lubricant to be used, e.g., oil, grease, or solid film, are very revealing. One of the first steps in using this procedure is to establish the "musts" of any possible alternative. The first "must" that comes to mind when considering lubricants is that it must be capable of providing boundary lubrication for a 10 year period. As stated previously, the speeds present in the gimbal do not lend themselves to formation of a hydrodynamic lubricating film, and oils and greases will not support boundary lubrication. Therefore, there is only one alternative, generically speaking, that of solid film lube. This leaves the question of which solid film lube should be used. This choice is dependent on the particular component being lubricated and the materials from which it is fabricated.

In view of the corrosion problem involved during storage and pre-launch, it is recommended that substrates be made of CRES steel such as 440 C or 17-4 PH, or of hard anodized aluminum alloys. All surfaces should be finished to an RMS of less than 10 and where applicable, a hardness greater than Rc 55. Bearings should have an ABEC rating of more than 5, with 7 preferred. Optimum surface preparation and case depth should be considered in the design.

4.3.3.3 Selected Lubricant. The optimum lubricating solid for the system is molybdenum disulphide (MoS_2). Application of this material can be implemented in three forms:

a. A sputtered film
b. A film with resinous binder
c. As a composite using a polyimide resin as the binder.

Application should be as follows for generic mechanisms.

<u>Ball and roller bearings.</u> The MoS_2 powder should be sputtered on balls and raceways. Retainers and spacers should be metallic but coated with an 0.2 to 0.3 mil film of a polyimide/antimony oxide/MoS_2. The latter provides a transfer film of MoS_2 to the bearing surface if the sputtered film degrades.

<u>Gears.</u> The MoS_2 should be applied to gear teeth either by sputtering or as a resin bonded film depending on clearance requirements. For extended life it is recommended that an idler gear be incorporated into the system to provide solid lubricant transfer to the stressed interface. The polyimide/Sb_20_3/MoS_2 combination is worthy of investigation for both coating and idler gear composite.

<u>Plain - Bearing Surfaces.</u> The polyimide/Sb_2 03/MoS_2 composite or coating is recommended for plain bearing surfaces.

Whenever solid lubricants are used, it is good practice to "run-in" or burnish the film and blow off with dry nitrogen the particles and debris formed by the burnishing action, prior to system usage.

4.3.4 Motors/Position Sensors

Failure criteria for motors and position indicators are identified by structural, mechanical (particularly moving parts), and electrical failures. Dynamic servo components, divided here into 9 basic categories, are subject to some failure modes which are common to all of them, plus unique modes of failure peculiar to one or two categories. Interaction between mechanical failure (such as a weak brush spring) and electrical failure (poor brush contact) are described and investigated so that precautions can be taken in writing specifications, constructing components, and acceptance testing to foresee and prevent such failures.

Vendors of these components were requested to provide information on unusual failure modes that they have encountered in vacuum or temperature testing and in long-life tests. The collection of this failure data was done by mail, telecon, and personal interviews.

Step 1 of this study was the itemization of all possible failure modes for each synchro, motor, and resolver type, whether such failures have ever occurred or not.

4.3.4.1 Servo Component Failure Modes. 84 failure modes were identified; some common to all types of servo components, some particular to one or two. A matrix of failure modes vs failure causes was prepared for each component, Tables 4-6 through 4-13. These were then combined into one master matrix (Table 4-14) to simplify comparison and facilitate countermeasures to eliminate their occurrence. A list of these failure modes follows the Tables.

SERVO-COMPONENT FAILURE MODES

Table 4-6

Brush Type Synchros: / Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Slip Rings:																	
Excess wear	x	x	x	x			x					x	x		x		
Scratches			x	x			x				x						
Contamination	x		x	x				x				x	x	x			
Glazing	x	x										x	x				
Surface films	x	x								x		x	x	x			
Brushes:																	
Excess wear	x	x							x				x		x		
Arcing	x	x															
Bounce		x	x	x													
Chatter		x	x	x							x		x				
Noise		x	x	x	x		x	x			x	x	x				
Cocked in holder		x	x	x													
Weld to slip ring	x	x					x					x	x				
Non-homo material	x	x															
Excess drag	x	x	x	x			x					x	x				
Weak springs	x	x							x				x				
Fixed angle error		x	x	x		x	x	x	x		x						
Brush dust accumulation	x	x	x	x											x		
Variable null voltage		x	x	x									x				
Open windings			x	x	x	x			x		x		x				
Shorted windings	x	x	x	x							x		x	x			

SERVO-COMPONENT FAILURE MODES

Table 4-7

Permanent Magnet Motors: / Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle
Magnetic Detent Action		x	x	x											
Housing corrosion	x	x		x				x	x	x					
Shaft corrosion	x	x		x				x	x	x					
Bearing corrosion	x	x		x				x	x	x					
Brush wear	x	x	x	x			x		x						x
Brush welding	x	x					x			x			x		
Scratched Commutator			x	x	x	x	x	x	x		x				
Weak brush springs	x	x		x									x		
Shorted windings	x	x	x	x			x	x	x		x		x	x	
Open windings	x	x	x	x			x	x	x		x		x	x	
Effects of magnetic fields	x	x	x	x	x	x	x	x	x	x					
Effects of radiation	x	x												x	
Shaft bindup															

SERVO-COMPONENT FAILURE MODES

Table 4-8

Feedback Potentiometers on Torquer Motors:

Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Brush wear	x	x	x	x								x			x		
Brush weld to pot	x	x					x	x		x		x	x				
Debris buildup on pot	x	x			x	x	x					x			x		
Tap open circuit	x		x	x			x		x		x						
Tap lead flex failure	x	x	x	x	x		x		x		x						
Brush lead flex failure	x	x	x	x	x		x		x		x						
Spring pressure wrong	x	x		x									x				
Resistance mat'l redistribute	x	x	x		x		x					x			x		
Groove worn in resistor	x	x	x		x		x				x				x		
Dimples on resistor	x	x	x		x												
Rotating friction increase	x	x					x				x	x	x				
Loose coupling to motor	x	x	x	x			x		x				x				
Shaft fatigue	x	x					x							x	x		
Insulation failure	x	x	x	x			x		x		x		x	x			

SERVO-COMPONENT FAILURE MODES

Table 4-9

Torquer Motors with Brushes:

Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Shorted turns		x	x	x			x		x		x		x	x			
Brush weld to commutator	x	x			x		x		x	x		x					
Magnetic detent action	x	x	x	x													
Rotor eccentricity		x	x	x			x	x									
Lamination separation	x	x	x	x			x				x						
Lamination corrosion	x	x	x	x					x	x							
Open turns		x	x	x			x		x		x		x				
Insulation flaking off	x	x	x	x									x	x			
Lead fatigue	x	x	x	x	x	x	x	x	x		x				x		
Increasing power consumption		x	x	x					x		x	x	x	x	x		

SERVO-COMPONENT FAILURE MODES Table 4-10 Brushless Synchros: — Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Excessive phase shift		x	x						x				x				
Decreased magnetism	x		x	x	x	x	x	x	x	x	x		x	x			
Variable transformation ratio	x	x	x	x					x				x				
Overheating	x	x	x	x		x			x			x	x		x		
A rotor voltage with temp.	x	x	x	x									x				
Shorted turns	x		x	x			x		x		x		x	x			
Open turns	x		x	x			x		x		x		x				
Sensitivity to magnetic fields	x	x			x					x							
Rotor eccentricity		x	x	x			x	x									
Lead fatigue	x	x	x	x			x	x	x		x				x		
Variable null voltage		x	x	x					x								

SERVO-COMPONENT FAILURE MODES

Table 4-11

Stepper Motors with Epicyclic Gears:

Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle
Loss of registration		x	x	x									x		x
Winding resistance change	x	x												x	x
Weak welds			x	x			x		x		x				
Corrosion	x	x						x	x	x					
Overheating		x		x			x						x		x
Control logic failure		x			x	x	x		x				x	x	x
Pulse source failure		x			x	x	x		x				x	x	x
Lead fatigue	x	x	x	x	x	x	x	x	x		x				x
Shorted turns		x	x	x			x	x	x		x		x	x	
Open turns		x	x	x			x		x		x		x		
Current leakage	x	x	x	x			x		x		x		x	x	
Loss of magnetism	x	x			x	x	x	x	x	x	x		x	x	
Gear failure	x	x	x	x			x	x	x		x	x	x		

SERVO-COMPONENT FAILURE MODES

Table 4-12

Hairspring Synchros: / Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Lead fatigue	x	x	x	x	x	x	x	x	x		x				x		
Hairspring fatigue	x	x	x	x	x	x	x		x		x				x		
Hairspring corrosion	x	x	x	x				x	x	x							
Shorted turns		x	x	x			x	x	x		x		x	x			
Open turns		x	x	x			x		x		x		x				
Variable null voltage		x				x											
Weak hairspring weld	x		x	x			x				x						
Overheating	x	x	x	x								x			x		
Rotor eccentricity		x	x	x			x	x									
Variable null voltage		x	x	x					x								

SERVO-COMPONENT FAILURE MODES

Table 4-13

Failure Cause	Material Selection	Design	Mfg.	Assy.	Electrical Test	Calibration	Vib'n/Shock/Accel.	Shipping/Handling	Installation	Storage	Launch	Vacuum	Thermal	Radiation	Abnormal Duty Cycle		
Motor Rate Tach:																	
Tach Ripple over Tol.		x	x	x	x	x	x				x		x				
Load variation		x	x	x													
Insulation flaking	x											x	x	x			
Rotor eccentricity			x	x		x					x						
Cup drag on frame		x	x	x			x	x	x		x						
Damping change	x	x					x						x	x			
Weakening magnet	x	x					x	x	x	x							
Null offset variation		x	x	x	x	x	x		x			x					
2 Phase Servomotors:																	
Short within winding	x	x	x	x			x		x		x			x			
Inter-winding short	x	x	x	x	x	x	x		x		x			x			
Short to frame	x	x	x	x	x				x		x			x			
Open winding		x	x	x			x		x								
Single phase rotation		x	x	x													
Overheat		x	x	x								x	x		x		
Shaft bindup		x					x						x		x		

SERVO COMPONENT FAILURE MODE/SOURCE MATRIX

FAILURE MODES

COMPONENT	1 SHORTED WINDINGS	2 OPEN WINDINGS	3 OPEN LEAD WIRES	4 OPEN/WEAK WELDS	5 LEAD FATIGUE	6 WIRE CORROSION	7 SHORTS TO FRAME	8 INSULATION BRITTLE	9 INSULATION FLAKE-OFF	10 INSULATION LEAKAGE	11 WINDING RESISTANCE CHANGE	12 WINDING OVERHEATING	13 SHAFT/HOUSING CORROSION	14 BRUSH WEAR
1 BRUSH TYPE SYNCHROS	●	●	●	●	●	●	●	●	●	●	●	●	●	●
2 BRUSHLESS SYNCHROS	●	●	●	●	●	●	●	●	●	●	●	●	●	●
3 HAIRSPRING SYNCHROS	●	●	●	●	●	●	●	●	●	●	●	●	●	●
4 TORQUER MOTORS	●	●	●	●	●	●	●	●	●	●	●	●	●	●
5 FEEDBACK POTS			●	●	●	●	●	●	●	●			●	●
6 PERM. MAGNET MOTORS	●	●	●	●	●	●	●	●	●	●	●	●	●	
7 STEPPER MOTORS	●	●	●	●	●	●	●	●	●	●	●	●	●	
8 MOTOR RATE TACHOMETERS	●	●	●	●	●	●	●	●	●	●	●	●	●	
9 2 Φ SERVO MOTORS	●	●	●	●	●	●	●	●	●	●	●	●	●	
10 OPTICAL ENCODERS			●		●		●						●	
11 RESOLVERS	●	●	●		●	●	●	●	●	●	●		●	
12 INDUCTION POTENTIOMETER	●	●	●	●	●	●	●	●	●	●	●		●	
FAILURE SOURCE														
1 DESIGN	●			●								●		●
2 MATERIAL SELECTION	●			●	●	●	●	●	●	●	●		●	●
3 MANUFACTURING	●	●	●	●	●	●	●					●	●	●
4 ASSEMBLY	●	●	●	●	●	●	●					●	●	●
5 ELECTRICAL TEST	●						●							
6 CALIBRATION	●						●							
7 VIBRATION, SHOCK, ACCEL	●	●	●	●	●		●							
8 SHIPPING, HANDLING	●													
9 INSTALLATION	●	●	●	●			●					●	●	
10 STORAGE														
11 LAUNCH	●	●		●	●		●							
12 VACUUM/ZERO-G												●	●	●
13 THERMAL EXTREMES	●	●		●			●	●	●	●	●	●		
14 RADIATION	●						●	●	●	●				
15 ABNORMAL DUTY CYCLE												●	●	●

COMPONENT	15 BRUSH WELD TO SLIP RING	16 BRUSH ARCING	17 BRUSH BOUNCE/CHATTER	18 EXCESSIVE BRUSH DRAG	19 BRUSH NOISE	20 BRUSH SPRINGS WEAK/STRONG	21 SLIP RINGS SCRATCHED/GLAZED	22 SLIP RING FILMS/CONTAMINATED	23 LAMINATIONS SEPARATED	24 LAMINATIONS CORRODED	25 SHAFT BIND-UP WITH TEMP.	26 WEAKENED MAGNETS	27 MAGNETIC DETENT ACTION	28 LOSS OF REGISTERATION
1 BRUSH TYPE SYNCHROS	●	●	●	●	●	●	●	●	●	●	●			●
2 BRUSHLESS SYNCHROS									●	●	●			●
3 HAIRSPRING SYNCHROS									●	●	●			
4 TORQUER MOTORS	●	●	●	●	●	●	●	●	●	●	●			
5 FEEDBACK POTS		●	●	●	●	●					●			
6 PERM. MAGNET MOTORS									●	●	●	●	●	
7 STEPPER MOTORS									●	●	●			●
8 MOTOR RATE TACHOMETERS											●			
9 2 Φ SERVO MOTORS									●	●	●			
10 OPTICAL ENCODERS											●			●
11 RESOLVERS									●	●	●			●
12 INDUCTION POTENTIOMETER									●	●	●			●
FAILURE SOURCE														
1 DESIGN		●		●	●	●	●	●	●		●		●	●
2 MATERIAL SELECTION	●	●			●	●	●	●		●	●	●		●
3 MANUFACTURING			●	●	●	●	●	●	●	●	●		●	●
4 ASSEMBLY		●	●	●	●		●	●	●	●	●	●	●	●
5 ELECTRICAL TEST		●										●		
6 CALIBRATION		●										●		
7 VIBRATION, SHOCK, ACCEL	●						●		●			●		●
8 SHIPPING, HANDLING												●		
9 INSTALLATION							●		●	●		●		●
10 STORAGE								●		●		●		
11 LAUNCH							●					●		●
12 VACUUM/ZERO-G	●		●	●			●	●			●			
13 THERMAL EXTREMES	●				●	●	●		●	●	●	●		
14 RADIATION												●		
15 ABNORMAL DUTY CYCLE											●			

COMPONENT	29 LOSS OF CALIBRATION	30 DEBRIS BUILDUP	31 LOOSE COUPLING	32 NULL OFFSET VARIATION	33 EXCESSIVE TACH. RIPPLE	34 ROTOR ECCENTRICITY	35 CUP DRAG ON FRAME/COIL	36 DAMPING VARIATION	37 SINGLE PHASE ROTATION	38 CHANGING ROTOR VOLTAGE	39 HAIRSPRING FATIGUE	40 VARIABLE NULL VOLTAGE	41 RESISTANCE MAT'L REDISTRIBUTED	42 GROOVE WORN IN RESISTOR
1 BRUSH TYPE SYNCHROS	●	●	●	●		●						●		
2 BRUSHLESS SYNCHROS	●		●	●						●		●		
3 HAIRSPRING SYNCHROS				●							●	●		
4 TORQUER MOTORS														
5 FEEDBACK POTS	●	●	●										●	●
6 PERM. MAGNET MOTORS														
7 STEPPER MOTORS			●											
8 MOTOR RATE TACHOMETERS				●	●	●	●	●						
9 2 Φ SERVO MOTORS									●					
10 OPTICAL ENCODERS	●		●											
11 RESOLVERS	●		●	●								●		
12 INDUCTION POTENTIOMETER	●		●	●								●		
FAILURE SOURCE														
1 DESIGN	●	●		●	●		●	●	●	●	●	●	●	●
2 MATERIAL SELECTION		●						●		●	●	●	●	●
3 MANUFACTURING	●		●	●	●	●	●	●	●	●	●	●	●	●
4 ASSEMBLY	●		●	●	●	●	●			●				
5 ELECTRICAL TEST														
6 CALIBRATION	●													
7 VIBRATION, SHOCK, ACCEL	●		●				●							●
8 SHIPPING, HANDLING	●						●							
9 INSTALLATION			●		●	●	●		●					
10 STORAGE														
11 LAUNCH	●													●
12 VACUUM/ZERO-G		●										●		
13 THERMAL EXTREMES	●		●	●						●		●		
14 RADIATION														
15 ABNORMAL DUTY CYCLE		●									●	●		●

COMPONENT	43 DIMPLES WORN IN RESISTOR	44 EXCESSIVE ROTATING FRICTION	45 FIXED ANGLE ERROR	46 EXCESSIVE PHASE SHIFT	47 DEGRADED BY MAGNETIC FIELDS	48 CONTROL LOGIC FAILURE	49 PULSE SOURCE FAILURE	50 BEARING FAILURE	51 GEAR FAILURE	52 NON-HOMOGENIUS MATERIAL	53 RESONANT OSCILLATION	54 PLOTTING FAILURE/CRACKS	55 AMBIGUOUS READINGS	56 LIGHT SOURCE FAILURE
1 BRUSH TYPE SYNCHROS			●					●	●	●		●	●	
2 BRUSHLESS SYNCHROS			●	●	●			●	●	●		●	●	
3 HAIRSPRING SYNCHROS								●	●	●		●	●	
4 TORQUER MOTORS								●		●		●		
5 FEEDBACK POTS	●	●	●									●		
6 PERM. MAGNET MOTORS					●			●		●		●		
7 STEPPER MOTORS						●	●	●	●	●	●	●	●	
8 MOTOR RATE TACHOMETERS										●		●		
9 2 Φ SERVO MOTORS								●		●		●		
10 OPTICAL ENCODERS			●										●	●
11 RESOLVERS			●									●	●	
12 INDUCTION POTENTIOMETER			●									●		
FAILURE SOURCE														
1 DESIGN	●	●	●	●	●			●	●		●		●	
2 MATERIAL SELECTION	●	●			●			●	●	●		●		
3 MANUFACTURING	●	●		●	●			●	●					
4 ASSEMBLY				●	●			●	●			●		
5 ELECTRICAL TEST					●									
6 CALIBRATION			●	●	●								●	
7 VIBRATION, SHOCK, ACCEL	●	●		●	●	●	●	●	●			●		●
8 SHIPPING, HANDLING	●				●			●	●					
9 INSTALLATION					●	●	●	●	●			●	●	
10 STORAGE	●				●			●						
11 LAUNCH						●	●	●	●			●		●
12 VACUUM/ZERO-G		●						●	●			●		
13 THERMAL EXTREMES		●	●	●		●	●	●	●			●		●
14 RADIATION						●	●					●		●
15 ABNORMAL DUTY CYCLE						●	●	●	●					

Table 4-14

1. Brush Type Synchros

a. Slip ring wear
b. Slip ring scratches
c. Slip ring oxidation
d. Slip ring glazing
e. Slip ring surface films
f. Brush wear
g. Brush arcing
h. Weak brush springs
i. Brush dust accumulation
j. Brush bounce
k. Brush noise
l. Variable null voltage
m. Brush chatter
n. Brush cocked in holder
o. Brush weld to slip ring
p. Non-homogeneous brush material
q. Excess brush drag

2. Brushless Synchros

a. Excessive phase shift
b. Decreasing magnetic efficiency
c. Variable transformation ratio
d. High heat dissipation
e. Temperature dependence of rotor voltage
f. Shorted turns
g. Sensitive to magnetic fields
h. Rotor eccentricity

3. Hairspring Synchros, $\pm 165^{o}$ Rotation, Mech. Stop

a. Lead fatigue
b. Hairspring fatigue
c. Hairspring corrosion
d. 1 - turn shorts
e. Variable null voltage
f. Weak hairspring weld

4. Torquer Motors W/O Feedback Pot

a. Single turn short
b. Brush weld to commutator
c. Magnetic detent action
d. Rotor eccentricity
e. Bearing eccentricity
f. Lamination separation
g. Lamination corrosion
h. Brittle windings
i. Insulation flaking off

5. Feedback Pot (On Torquer Motor)

(Typ: Linearity: .825%, Travel: $\pm 50^{o}$, N/F $\pm$ 130 mv p-p, End taps at $\pm 55^{o}$, R=10K)

a. Brush wear
b. Brush weld to pot
c. Carbon buildup on pot
d. Tap open-circuit
e. Tap lead flex failure
f. Brush lead flex failure
g. Spring pressure too tight
h. Resistor material redistribute
i. Groove worn in resistor
j. Non-conducting trash on resistor
k. Dimples on resistor surface
l. Rotating friction increase
m. Loose coupling to motor
o. Shaft fatigue

6. Permanent Magnet Motors

a. Magnetic detent action
b. Housing corrosion
c. Shaft corrosion
d. Bearing corrosion
e. Brush wear
f. Brush welding
g. Scratched commutator
h. Weak brush springs
i. 1-Turn short
j. Magnet degradation
k. Effects of external magnetic fields
l. Effects of radiation

7. Stepper Motors

1. Permanent magnet type
2. Variable Reluctance type

a. Current leakage
b. 1-Turn short
c. Lead fatigue
d. Pulse source failure
e. Control logic failure
f. Overheating
g. Corrosion of conductors
h. Weak welds
i. Loss of registration
j. Winding resistance change due to temperature

8. Motor Rate Tachometers

a. Tach ripple (beyond speed-variation tolerance)
b. Load variation
c. Insulation flaking
d. Rotor eccentricity

9. 2 Ø Servo Motors

a. Inter-winding short
b. 1-Turn short
c. Single phase rotation
d. Open winding

4.3.4.2 Failure Causes, Effects, and Solutions. Each of the failure modes shown on Table 4-14 was investigated as to its cause and effect on the performance of the gimbal. Possible solutions were devised and are listed along with the applicable failure mode.

1. FAILURE MODE: Brush springs weak or strong.

Cause: Brush springs made of wrong alloy, made to wrong dimensions, or installed in wrong position. Spring not properly tempered.

Effect: Weak springs contribute to brush noise, bounce, chatter, and arcing. Overly strong springs contribute to high brush drag, brush welding to the slip ring, and accelerated brush wear.

Solution: Quality control of spring material and manufacturing methods is adequate to assure repeatible curves of pressure vs position when properly installed. Pressure tests after installation are needed to demonstrate the pressure curve and freedom from binding. For very long brushes, "negator" springs are capable of constant pressure over the full range of brush travel.

2. FAILURE MODE: Slip rings scratched or glazed.

Cause: Contaminants adhering to the slip rings during assembly or operation. Films and oxides glazed onto rings during checkout and operation. Surface damage during handling, assembly, or tests.

Effect: Accelerated brush and ring wear, leading to failure. Increasing electrical noise, signal, and power transfer degradation. Cumulative production and entrapment of debris.

Solution: Extreme care in handling, transporting, and assembling slip ring assemblies. Constant protection from solid and vapor contaminants and fingerprints. Close inspection before and after each assembly operation to detect any minute surface damage or glazing.

3. FAILURE MODE: Slip ring contamination films.

Cause: Moisture, dust, conductive and non-conductive debris adhering to ring surface during manufacture, assembly, and operation. Vaportized lubricant from bearings depositing on ring surface in space environment. Oxide and sulfide films on improperly prepared surfaces.

Effect: Increasing electrical noise, leading to degradation of signal and power transfer. Scoring of ring surface by contaminants.

Solution: Surface preparation of rings in clean environment. Protection from contaminants during assembly and storage. Isolation of lubricant vapors from rings.

4. FAILURE MODE: Separation of laminations

Cause: Misshapen and deformed laminations. Inadequate surface preparation before bonding. Entrapped contaminants between laminations.

Effect: Increased magnetic reluctance, leading to out-of-tolerance errors and loss of accuracy. Condition is usually not catastrophic, but can compromise component capability.

Solution: Cleaning and inspection of laminations, assembly in a clean environment and protection from contaminants before and after assembly. Careful potting of assembly so as to preserve mechanical integrity.

5. FAILURE MODE: Corroded laminations.

Cause: Adherence of contaminants, such as fingerprints, oxides, and sulfides, to the surface after final cleaning.

Effect: Outgassing and rupture of protective potting. Separation of laminations, with resultant degradation of component capability.

Solution: Thorough cleansing of laminations, followed by close inspection and protection from contaminating environments, until final encapsulation.

6. FAILURE MODE: Shaft bindup with temperature.

Cause: Unequal expansion of rotor shaft and frame at elevated temperature, resulting in loss of clearance at ends of the shaft.

Effect: High friction at shaft ends produces more heat, high torque which requires more motor power, and complete stall until component is allowed to cool.

Solution: Put min/max clearance in specification, rather than max. only. Allow adequate clearance for expansion at the highest temperature of operation of the component.

7. FAILURE MODE: Weakened magnets.

<u>Cause</u>: Subjection of magnets in motors to strong shock, vibration, or inverse magnetic fields. Failure to maintain "keepers" on magnets until assembly into motors.

<u>Effect</u>: Servo motors with weakened magnets require more power to produce the required torque, may not be able to produce maximum specified torque.

<u>Solution:</u> Protect magnets from severe shock before and after assembly into motors. Maintain keepers in place before assembly. Program motor driver so that strong demagnetizing fields cannot be applied during motor operation. Do not store magnets or motors near a strong magnetic field or each other.

8. FAILURE MODE: Shorted Windings.

<u>Cause 1:</u> Turn short, due to nicked wire, insufficient enamel coverage or abrasions during assembly.

<u>Effect:</u> Component overheats, possibly causing further shorted or open conditions. Angle error on synchros increases, but may not be discovered unless component is recalibrated.

<u>Solution</u>: Strict Quality Control by wire vendor and "kid glove" treatment during winding will ensure integrity of insulation. This fault can be detected during burn-in and/or calibration.

Cause 2: Multi-turn short, due to inadequate insulation or over-voltage condition during electrical tests.

Effect: Severe heating and/or error in component. May cause burnout of associated equipment.

Solution: Extreme care in winding and testing servo component is necessary to prevent these faults.

Cause 3: Short between adjacent winding, due to over voltage during electrical tests or in use.

Effect: Overheating, severe power drain and damage to associated equipment; can cause chain reaction failures in some cases.

Solution: Place adequate insulation between windings that carry different voltage levels or out-of-phase A.C. voltages. Establish dielectric breakdown and leakage tests after manufacturing and assembly.

9. FAILURE MODE: Open windings.

Cause: Nicked or twisted wire during fabrication or assembly.

Effect: Component degraded or inoperative. May not be noticed in a multi-pole synchro or motor with light loading.

Solution: Good maintenance of coil winding equipment and training of personnel to inspect all wire and eliminate any that is not perfect. Purchase of wire from vendors with proven good quality control and 100% source inspection at the factory as this wire is being produced. Investigation of vendor winding procedures to ensure their adequacy. Information available to date indicates this to be one of the weak points in the process chain.

10. FAILURE MODE: Open lead wires.

Cause: Improper solder, wire-wrap, or splicing techniques employed.

Effect: No output, weak output, or error in output of component.

Solution: Frequent recertification of wiring personnel and equipment. 100% inspection of clips, splices and all hardware. Pull test on all joints. High frequency vibration or acoustical vibration should uncover this discrepancy.

11. FAILURE MODE: Brush Noise.

Cause: Slip ring surface imperfections. Buildup of debris under brush. Excess lubricant or non-conductive lubricant in use. Brush bounce, chatter or movement within brush holder.

Effect: Slight noise can affect system accuracy. More intense noise can also cause electromagnetic interference to other sensitive circuits.

Solution: Protect slip rings and brushes from all contaminants. Place barriers between slip rings to channel debris away from brushes. Use correct amount of correct lubricant in brushes. Design and build brush holders for a firm grip on brushes without binding. Dual brushes in parallel reduce rms noise at the expense of extra brush drag and increased hardware complexity.

12. FAILURE MODE: Excessive brush drag.

<u>Cause:</u> Excessive brush spring pressure. Inadequate brush lubrication or uneven dispersion through the brush. Brush cocked at wrong angle.

<u>Effect:</u> Inordinate servo error, brush heating and torque reflected to motor and bearings.

<u>Solution:</u> Careful attention to brush spring pressure throughout brush travel. Adequate dispersion of lubricant through brush. Close fit between brush and holder. Symmetrical spring loading on brushes.

13. FAILURE MODE: Open or weak welds.

<u>Cause:</u> Improper materials, pressure and/or current.

<u>Effect:</u> Open circuit or heating at the joint under operating conditions or environmental tests.

<u>Solution:</u> Frequent recertification of welding personnel. Close inspection of materials and dimensions of pieces to be welded. 100% visual inspection of welds under magnification with pull test, radiographic inspection, and/or dye penetrant where feasible.

14. FAILURE MODE: Lead Fatigue.

<u>Cause:</u> Improper materials, excess removal of insulating cover, excess length of wire between clamps or potting.

<u>Effect:</u> Wire shakes during vibration of component or movement of gimbals until fatigue occurs at weakest point, usually

where insulation has been stripped off, leading to a severed conductor.

Solution: Specify proper materials and inspect for conformance. Design holddown devices to avoid long loops or runs of wire. Train personnel in techniques of stripping and laying wires to avoid weak spots and resonance.

15. FAILURE MODE: Corrosion.

Cause: Atmospheric contamination, moisture, fingerprints, dissimilar metals in contact.

Effect: Open circuits, leakage between circuits, increasing errors or failure.

Solution: Clean room conditions are mandatory for manufacturing and assembly. Storage and transfer requires containers. Dissimilar metal protection specifications are available and are considered adequate.

16. FAILURE MODE: Shorts to frame.

Cause: Sharp corners, edges, and projections on frames which can abraid insulation. Wires stretched tightly around corners or forced over laminations. Wires pinched under or between metal frame members.

Effect: Heating, error and/or burnout of component. Increased electromagnetic interference to sensitive circuits.

Solution: Establish minimum radius for all corners and edges. Remove burrs and projections. Add low-friction insulation to corners where wires must pass by, to avoid abrasion. Provide adequate clearance of wires and coils around laminations. Use clamps and potting to secure wires in place and prevent relative motion between wires and edges to preclude chafing and abrading of insulation from wires.

17. FAILURE MODE: Brittle Insulation.

<u>Cause</u>: Insulation materials unable to maintain flexibility under the space environment for the required period of operation.

<u>Effect:</u> Reduction or elimination of insulation effectiveness by cracking during motion or vibration can lead to arcing, shorting, and total loss of component capability.

<u>Solution:</u> Specify and test insulating materials to withstand the space environment. Tests should demonstrate a reasonable capability of surviving twice the required period of operation without loss of insulating qualities or flexibility. Accelerated environmental tests shall be used to determine failure modes of each type of insulation.

18. FAILURE MODE: Insulation flake-off.

<u>Cause</u>: Insulating material suffers particle separation under influence of space environment.

<u>Effect:</u> Loss of voltage breakdown capability of insulation plus migration of flakes into moving parts of gimbals where they may cause friction, drag or complete stall.

<u>Solution:</u> Specify and test insulation materials to demonstrate ability to withstand the space environment for twice

the required period of operation without chipping, flaking, or cracking. No solid, liquid, or gaseous effluent shall be capable of compromising the ability of the assembly to complete its mission.

19. FAILURE MODE: Insulation leakage.

Cause: Reduction of insulation resistance by chemical, thermal, and mechanical means or by U.V. and nuclear radiation effects.

Effect: Increasing leakage, power loss, overheating and component error or total loss of capability.

Solution: Specify and test insulation materials to withstand the space environment without appreciable leakage under 4 times the maximum applied voltage in actual use. Tests to be conducted under simulated space conditions during and after exposure of insulation to accelerated environmental degradation.

20. FAILURE MODE: Winding Resistance Change.

Cause: Thermal extremes, ultra violet, and nuclear radiation in the space environment can cause a significant change in winding resistance. Copper has a coefficient of resistivity change with temperature on the order of $0.4\%/^{o}C$. Effects of radiation on resistance varies with exposure and duration.

Effect: Loss of calibration, excess current draw-in, reduction in torque capability.

Solution: Design gimbal enclosures to shield servo components from direct sun or extreme cold. Provide thermal paths so that all windings of each synchro are at approximately the same temperature. Shield as necessary again U.V. radiation. Nuclear radiation shields are not feasible for this application.

21. FAILURE MODE: Winding Overheating.

Cause: Insufficient thermal paths to conduct internally generated heat away from windings. Solar radiation impinging upon external surfaces.

Effect: Accelerated deterioration of winding insulation on servo components, leading to out-of-tolerance errors, loss of capability and complete failure.

Solution: Heat shields to protect critical components, from solar radiation. Adequate thermal paths to conduct internally generated heat to available heat sinks.

22. FAILURE MODE: Shaft/Housing Corrosion.

Cause: Atmospheric contamination, fingerprints, moisture, dissimilar metals in intimate contact.

Effect: Weakening of housing and supports. Rough areas of shaft can reduce clearance to stator, cause binding. Corrosion products may enter bearings or servo components and stall shaft. Dissimilar metals in contact in the presence of trapped moisture cause stray electrical currents which may affect servo performance and cause further corrosion.

Solution: Clean room practices and sealed storage and transfer will prevent corrosion. Sensitive surfaces can be plated or anodized. Dissimilar metal protection specifications are considered to be adequate.

23. FAILURE MODE: Excess Brush Wear.

Cause: Rough commutator or slip rings. Inadequate or improper solid lubricant included in brush material for space environment. Excess brush spring pressure.

Effect: Brushes wear down until contact is lost, component fails.

Solution: Extreme care in preparing mating surfaces. Distribute dry lube evenly throughout brush material. Adjust and test brush spring pressure; use negator springs for constant pressure.

24. FAILURE MODE: Brush Weld to slip ring.

Cause: Insufficient or improper dry lube included within brush material. Excess brush pressure.

Effect: Brush particles weld to slip ring in the vacuum of deep space. Brush disintegrates rapidly as slip ring increases surface roughness.

Solution: Distribute dry lubrication evenly throughout brush material. Set brush pressure at minimum necessary to assure good contact with slip ring over lifetime of component.

25. FAILURE MODE: Brush Bounce and Chatter

<u>Cause</u>: Improper fit of brush to brush holder. Brush cocked in holder. Brush and holder at wrong angle for bi-directional operation.

<u>Effect</u>: Momentary loss of circuit continuity. Arcing between brush and slip ring. Scarring of slip ring. R. F. interference to other apparatus. Accelerated brush wear.

<u>Solution</u>: Careful positioning of brush holders at proper angle. Close fit of holder to brush to allow repositioning of brush with wear but prohibit any motion in other directions. Proper lubrication of brush to prevent friction with holder.

26. FAILURE MODE: Brush Arcing.

<u>Cause:</u> Loss of contact with slip ring due to vibration, shock, rotor eccentricity or slip ring surface asperity.

<u>Effect:</u> Electromagnetic interference with other sensitive apparatus. Accelerated degradation of the slip ring and brush. Momentary loss of circuit continuity.

<u>Solution</u>: Adequate brush spring pressure to assure good contact with slip ring during shock and vibration. Close tolerances on rotor concentricity. Smooth surfaces on brushes and slip rings, with proper lubrication, permeating the brush material.

27. FAILURE MODE: Magnetic Detent Action

<u>Cause</u>: Salient poles or slots in a permanent magnet motor, two-phase a.c. motor or motor-generator.

<u>Effect</u>: Excess control phase voltage required to start the rotor turning. Pulsing torque produced at very slow motor speeds resulting in reflector vibration and thus possible fatigue of structural members.

<u>Solution</u>: Careful design of slots can minimize detent effect. There is no known method to eliminate this problem in this particular device.

28. FAILURE MODE: Loss of Registration.

<u>Cause:</u> Presence of magnetic fields near the component during operation or storage. Slippage of component on shaft or frame during shock and vibration. Encoder light sources or sensors knocked out of line. Worn bearings on shaft.

<u>Effect:</u> Sensor output is skewed by a fixed angle or alternates between correct and offset.

<u>Solution:</u> Careful alignment of shaft, bearings, sensors and pickups. Protect from shocks and magnetic fields. Limit applied torque to a reasonable safety factor above that required to rotate the gimbal. Protect light sources and sensors during handling and assembly.

29. FAILURE MODE: Loss of Calibration.

<u>Cause:</u> Rotation of sensor on shaft. Bent or distorted rotor or stator. Wear of gears, bearings. Electrical loading of sensor by external circuitry. Extreme temperature environment.

<u>Effect:</u> Peak error and/or root-square-sum of cyclical errors exceeds tolerance. Errors change with temperature and loading.

<u>Solution:</u> Sensor rotor and stator must be securely attached to shaft and frame, making sure the attachment methods do not cause distortion. Temperature must be controlled within design limits. External circuitry should be designed for, and calibrated with, the sensor with which it will be used.

30. FAILURE MODE: Debris Buildup.

<u>Cause:</u> Brush wear particles or non-conductive lubricant adhere to sliding surfaces or become packed under brushes. Contamination during assembly.

<u>Effect:</u> Debris building up on brushes, slip rings and close tolerance air gaps between rotor and stator will cause drag, scoring, increased wear, and may stall the rotor completely or short between rings.

<u>Solution:</u> Strict clean room procedures during manufacturing, storage, and assembly. Barriers around bearings and brushes to inhibit lubricant migration. Barriers between slip rings to inhibit short circuits. Guides to conduct

debris away from brushes, rings and bearings under zero-gravity conditions.

31. FAILURE MODE: Loose Components.

<u>Cause:</u> Lack of locking feature on threaded fasteners. Attach points too weak for maximum load. Debris captured in coupling prevents close fit.

<u>Effect:</u> Loss of registration, calibration, and/or complete failure of drive capability.

<u>Solution:</u> Clean parts and assemble in a clean room, adhering to a detailed assembly procedure. Design and test structural parts under specified vibration, shock and torque, both at piece part and assembly level.

32. FAILURE MODE: Null Offset Variation.

<u>Cause:</u> Use of a common wire for one side of rotor and stator. Unequal resistance of stator coils or connecting wires. Shaft radial play, eccentricity and diameter tolerances. Mounting tolerances of components.

<u>Effect:</u> The error angle between electrical null and mechanical angle that was obtained during calibration varies during operation. Accuracy is thus variable and unknown.

<u>Solution:</u> Avoid common wires and grounded wires. Match impedances and loading of phase windings in each system. Limit radial and end play with proper bearing design. Shaft and components must be concentric and balanced.

33. FAILURE MODE: Excessive Tachometer Ripple

Cause: Unequal winding turns, flux paths, brush-to-commutator resistance, or winding resistances on the rotor. Eccentric rotor or worn bearings.

Effect: Uneven windings produce a ripple (deviation from average voltage) of 2 x rps (revolutions per second) x number of turns. Eccentric rotor or worn bearings produce ripple equal to the rps or a multiple of it. This ripple indicates an instantaneous error in speed of the shaft to the servo electronics.

Solution: Careful winding techniques and tests are adequate to keep winding discrepancies within tolerable limits. Rotor and bearing problems can be avoided thru available technology. Added protection is obtained by filtering the tachometer output at the lowest fundamental frequency expected. Precious metal commutator surfaces are used to reduce brush/commutator resistance.

34. FAILURE MODE: Rotor Eccentricity.

Cause: Shaft eccentricity, poor mounting of rotor on shaft or bearing failure.

Effect: Null offset variation, output ripple, excessive brush wear, loss of registration and calibration.

Solution: Secure rotor to shaft per drawing. Current design, manufacturing and inspection techniques are adequate. Inspection should, however, be performed not only after but during each step in the assembly sequence.

35. FAILURE MODE: Tachometer Cup Drag on Frame or Coil.

<u>Cause</u>: Distortion or misalignment of drag cup. Inability of cup to withstand mechanical vibration and shock. Bearing failure or misalignment.

<u>Effect</u>: Excessive tachometer ripple; abrasion of coil insulation, leading to an intermittent short; additional load on shaft and motor.

<u>Solution</u>: Design drag cup strong enough to withstand shock and vibration. Allow sufficient clearance between cup and coil or frame consistent with adequate output signal at the rotational frequency of interest. Use long-life bearings, properly aligned and lubricated. Place tachometer inside Space Station where anomalies can be easily corrected.

36. FAILURE MODE: Damping Variation.

<u>Cause</u>: Variation of back-EMF in armature due to unequal windings, brush resistance variation. Non-homogeneous drag cup in tachometer. Non-linear viscous friction in bearings and viscous damper.

<u>Effect</u>: Variation of the input/speed/torque relationship for the servo motor.

<u>Solution</u>: Careful design, material selection and assembly of the viscous damped motor or conversion to rate damped (electronic) or inertial damped (revolving magnet) motors. These are more complex and more costly, but they provide smoother damping at better efficiency than the viscous damped motor.

37. FAILURE MODE: Single Phase Rotation.

<u>Cause:</u> Distortion of the fixed phase field of a 2-phase motor produces a "shaded pole" effect.

<u>Effect:</u> Uncontrolled rotation of the motor.

<u>Solution:</u> Careful design and manufacture of stator and rotor to avoid non-uniform distribution of magnetic flux in both the "air gaps" and the metallic structures. Measurement of the flux field before assembly to verify field uniformity.

38. FAILURE MODE: Changing Rotor Voltage.

<u>Cause:</u> The rotary transformer - coupled rotor winding of a brushless synchro is subject to variations in available voltage due to non-uniformity of the mutual inductance between transformer windings as rotation occurs.

<u>Effect:</u> Modulation of both rotor and stator voltages, increasing electrical error and null offset error.

<u>Solution:</u> Careful winding of transformer to assure uniformity of turns. Tests of rotor concentricity and voltage variation after final assembly of components and bearings on gimbal.

39. FAILURE MODE: Hairspring Fatigue.

<u>Cause:</u> Hairsprings which carry power to the rotor of a limited-rotation motor or synchro become fatigued from vibration and bending effect of rotation.

Effect: Catastrophic failure of component thru loss of signal or power.

Solution: Design parameters of the spring must be optimized for the specific application and design lifetime. Variable diameter spring material could decrease susceptibility to vibration fatigue at the end points of the spring. Elastomeric material at the end points would add strength and damping action.

40. FAILURE MODE: Variable Null Voltage.

Cause: The combination of quadrature fundamental null voltage and harmonic frequencies (mostly third) cause an output to be present at the null point of a synchro control system. These vary as the null point rotates.

Effect: Errors in angular measurements due to this voltage can be a significant portion of the permissible angular error.

Solution: Design and construct synchro to keep total null output to less than one-half the voltage which represents the permissible angular error. Use harmonic attenuators and quadrature discriminators at the input of synchro amplifiers to further reduce unwanted null voltages.

41. FAILURE MODE: Redistribution of Resistance Material.

Cause: Precision potentiometers, used as analog readout devices for shaft angles, experience surface wear, debris accumulation, and consequent redistribution of material.

Effect: Noise and resolution irregularity, manifested in a servo by gain variation (voltage change per unit angle) and instability.

Solution: Use solid lubricant in the potentiometer wiper and minimum contact pressure necessary to maintain electrical contact. An induction potentiometer can be substituted for the resistance potentiometer at the expense of low frequency response, weight, and the need for extra amplifiers and compensation.

42. FAILURE MODE: Groove Worn in Resistor.

Cause: Excess brush pressure during operation of a precision potentiometer, or a rough spot on the wiper, can wear a groove on the resistance element.

Effect: Material removal from the groove by the wiper results in an irregular and higher resistance element.

Solution: Inspect the wiper for a smooth surface and apply minimum pressure necessary to maintain good contact. Lubrication of sliding surfaces would reduce wear.

43. FAILURE MODE: Dimples Worn in Resistor.

Cause: Wiper of precision potentiometer exerts pressure on one spot of resistance element during long periods of storage.

Effect: Flexible mylar and soft plastic backing for film type resistance elements can be permanently deformed by light but continuous pressure on one spot during storage or inactivity, which degrades resolution and gain stability. Local wear will increase in such deformed areas during operation.

Solution: Use metal or plastic backing with good "cold flow" properties for resistance elements and minimum pressure necessary to assure contact between wiper and element.

44. FAILURE MODE: Excessive Rotating Friction.

Cause: Rotating friction in a servo system includes that from bearings, brushes on slip rings and commutators, wipers on potentiometers and encoders.

Effect: Static friction, plus slot effect, creates a "dead band" which the motor must overcome to start rotation. Running friction causes a drag on the rotor, thus a lag in response to error signals and reduction of system efficiency. Both represent basic errors in servo systems and demand increased power consumption.

Solution: Replace wipers and brushes with optical encoders and wire coils between axes. Use low friction bearings and motors that exhibit little or no slot effect (magnetic detent action).

45. FAILURE MODE: Fixed Angle Error

Cause: Failure to drive a multi-speed synchro system to near equilibrium with a 1-speed synchro before energizing the multi-speed synchro. Loss of 1 or more pulses in an open-loop stepping motor system.

<u>Effect</u>: Driven component appears to be operating properly, but is actually maintaining a fixed angle error from desired position.

<u>Solution</u>: Follow step-by-step start-up procedure to align synchros, then use the 1-speed synchros as a continuous check on alignment of the multi-speed units. Use frequent alignment checks on stepping system to indicate an out-of-step condition when in the open-loop mode.

46. FAILURE MODE: Excessive Phase Shift.

<u>Cause</u>: The rotary transformer of a brushless synchro increases the phase shift of rotor current approximately 10^{o} more than its equivalent brush-type synchro.

<u>Effect</u>: Intolerable error in the in-phase and quadrature components of synchro receiver output, with resultant pointing errors.

<u>Solution</u>: Measure and compensate for phase shift over normal operating conditions and environment or replace rotary transformer with coiled flat cables between axes for limited rotation applications.

47. FAILURE MODE: Degradation by Magnetic Fields.

<u>Cause</u>: Presence of permanent or alternating magnetic fields in proximity to synchro components during assembly, storage or operation.

Effect: Increased readout and null errors, varying with proximity, strength, and direction of alien magnetic fields and vulnerability of component to their influence.

Solution: Precautions against magnetic influence during assembly and storage. Magnetic shielding around sensitive components and/or between them and radiating components such as motors.

48. FAILURE MODE: Control Logic Failure.

Cause: Logic failure can cause the wrong windings of a stepper motor to be energized or the right windings to be energized with the wrong polarity.

Effect: Either failure can cause the stepper to stall, lose count of steps or step in the reverse direction. The rotor of a permanent magnet stepper motor will be partially demagnetized by a stator winding pulse of polarity opposed to its own.

Solution: Complex logic, with high reliability components, is necessary to drive high-efficiency steppers. A non-volatile memory would keep track of rotor position when system is shut down, to prevent rotor demagnetization during start up. A closed-loop system would provide added assurance that the motor is following logic commands.

49. FAILURE MODE: Pulse Source Failure.

Cause: Open or shorted driver transistors, logic failure or computer interface failure will degrade or prevent pulses reaching the stepper motors. Low power supply voltage will degrade high speed or high torque operation, but may seem normal at low speeds.

Effect: Stepping motor failure or restricted to low-speed or unidirectional operation.

Solution: Use transient-suppression devices to protect driver transistors from inductive surges. Use redundant drivers and amplifiers if repair is impossible. Sample pulse height and power supply voltage periodically to detect incipient failures before they become catastrophic. Employ feedback techniques to determine that each pulse has actually occurred.

50. FAILURE MODE: Bearing Failure.

Cause: Lack of lubricant. Presence of contaminants, including outgassing products from servo components.

Effect: Cumulative degradation of bearings, with increased torque necessary to overcome resultant drag, until assembly stalls.

Solution: Adequate lubrication for expected duration of mission. Provide a preferred path to space for outgassing products so they do not pass into bearings. Keep all surfaces and components clean during assembly, storage, and installation. (See Section 4.3.1).

51. FAILURE MODE: Gear Failure.

<u>Cause:</u> Lack of lubricant. Excessive drag from failing bearings. Presence of contaminants. Poor fit or alignment.

<u>Effect:</u> Welding of gear tooth particles. Cumulative degradation of teeth until gears stall or worn teeth fail to make contact.

<u>Solution:</u> Best solution is to use multi-pole torque motors and multi-speed synchro transmitters in the space environment and eliminate gears. Where gears must be used in space, extreme care in fit, alignment and lubrication will prolong life. Gears must be protected from contamination, including self-contamination by wear particles. See Section 4.3.2.

52. FAILURE MODE: Non-Homogeneous Material.

<u>Cause:</u> Voids, hairline cracks and inclusions of alien material in component frames, rotors and brushes.

<u>Effect:</u> Loss of physical strength and less reliability under shock and torque application. Distortion of magnetic paths with resultant increased reluctance and reduced accuracy and torque capability.

<u>Solution:</u> X-ray and infrared thermographic inspection of all metal structures for voids and cracks. Inclusions may be discovered in the same tests or by measuring and plotting the magnetic field at the surfaces of the rotor and frame when dissasembled and energized by a near-normal current in one winding at a time.

53. FAILURE MODE: Resonant Oscillation

<u>Cause</u>: Coincidence of the pulse frequency applied to a stepper motor at the natural resonant frequency of the motor and structure.

<u>Effect</u>: Violent oscillation of the rotor in a small arc, with no controlled advance in the desired direction. Possible fatigue of structural elements.

<u>Solution</u>: Design the servo system so that the stepping frequency is above or below the natural frequency of oscillation by a sufficient margin to avoid missed steps. If all speeds must be available, choose a different type motor.

54. FAILURE MODE: Potting Failure and Cracks.

<u>Cause</u>: Potting is out of date, wrong for this application, incorrectly mixed or applied, moved before completely cured, or subject to excessive stresses. Component to be potted is not clean and dry.

<u>Effect</u>: Voids, cracks in potting and failure to adhere or cure properly. Significant loss in strength and environmental protection of component.

<u>Solution</u>: Conventional techniques are adequate with increase of Quality Assurance surveillance.

55. FAILURE MODE: Ambiguous Readings.

<u>Cause</u>: Failure to zero a multi-speed synchro or incremental encoder. Loss of one or more light sources, photo-detectors or brush contacts in an absolute encoder.

<u>Effect</u>: The angle indicated by the synchro or encoder can be any one of several known possibilities. Speed of rotation can be determined, but not absolute position.

<u>Solution</u>: Include an accurate zero index with incremental encoders. Use redundant lights and detectors or brushes on the encoders, or redundant standby encoders to cross-check dubious readings.

56. FAILURE MODE: Light Source Failure.

<u>Cause</u>: Over- or under-voltage to incandescent lamps, open filament, blackened bulb, low light emission, poor contact to base or light out of proper position. Light emitting diodes (LED) are subject to degradation by radiation and overheating.

<u>Effect</u>: Insufficient light to operate optical encoder, loss of angular position and rate information.

<u>Solution</u>: Eliminate incandescent lamps as a light source for optical encoders. Protect LED's from radiation and voltage variation. Align LED with encoder and photo-detector and fix in place. Position electronics inside Space Station for ease of maintenance or replacement.

4.3.4.3 Servo Component Trade Study. Table 4-15 shows the results of the motor trade study. In this study, because of the ten year requirement, low rotating friction (a contributor to wear) was given the highest weighting (10), while fabrication cost was given the lowest weight (2) because of the expense involved in replacing a failed unit on orbit.

Wear phenomena were also included in consideration of both low debris production (7) and low wear of sliding contacts (5). Resistance to corrosion the lack of which would be a problem only prior to actual launch, was given a weight of 7 because of the importance of assuring there is no contamination (oxides) within the gimbal housing.

As a result of this trade study, the motor choice was reduced to two candidates, both of them being encoder commutated brushless motors, one with a wound rotor the other with a permanent magnet rotor. Of these two, the wound rotor is the prime candidate.

The choice of a position encoder was reduced to one candidate (see Table 4-16) on a basis of must criteria alone. The optical encoders are the only type capable of giving the accuracy required (0.01°). Even this encoder, unless placed in an environment where maintenance is available, would not meet the must criteria of "10 year life capability". The light source, light emitting diodes (LED), have not proven themselves capable of surviving. This concern arises because of a lack of historical information as opposed to actual failure data. LED's have been in actual usage just a few years and therefore not much is known about their absolute life capability.

4.3.4.4 Recommended Servo Component Positions. The conflicting requirements for accuracy and 10 year life in space make it extremely difficult to find a gimbal position indicator that will satisfy both. The magnetic encoder has greatest reliability and longest life, but lacks the required accuracy. The optical encoder has the highest resolution, but does not have a proven 10 year life.

Decision Analysis Worksheet

Decision Statement: Select a Type of Torque Motor for Space Antennas
- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

ALTERNATIVES: A — Encoder commutated brushless motor; B — D.C. brush motor; C — Encoder commutated Perm. magnet motor.; D — Stepper motor

OBJECTIVES — MUST (– Is it absolutely essential? – Is it measurable?)	A Encoder commutated brushless motor — INFO	A GO/NO	B D.C. brush motor — INFO	B GO/NO	C Encoder commutated Perm. magnet motor. — INFO	C GO/NO	D Stepper motor — INFO	D GO/NO
10 year life capability in 10^{-13} TORR vacuum.		Go	Brushes not proven for 10 years in space.	No		Go		Go
Low speed reversible operation capability.		Go		Go		Go		Go
Operation over $+200^{\circ}F$ to $-200^{\circ}F$ range.		Go		Go		Go		Go
Pointing accuracy within 0.02° w/o gears.		Go		Go		Go	7.50 minimum pratical step angle w/o gears	No
Capable of 30 oz-in of torque.		Go		Go		Go		Go

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)	WT	A INFO	A SC	A WT SC	B INFO	B SC	B WT SC	C INFO	C SC	C WT SC	D INFO	D SC	D WT SC
Low rotating friction	10	No friction	10	100				No friction	10	100			
Light weight	3		8	24				Heavy magnets	6	19			
Low cost	2	Expensive optical commutator	2	4				Expensive optical commutator	2	4			
Resistance to corrosion	7		8	56					8	56			
Low magnetic detent action	8	None	10	80				Slot effect	4	32			
Low debris production	7	None	10	70				None	10	70			
Simple control of torque/speed	5		6	30					6	30			
Low wear of sliding contacts	5	No contacts	10	50				No contacts	10	50			
Low bulk space	3		7	21					7	21			
Simple assembly	3		8	24					8	24			
Low outgassing	3		4	12				Only one set of windings	8	24			
Low power consumption	4		5	20				No rotor current	9	36			
TOTAL WT SC				491						465			

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2595 B (FRONT)

LOCKHEED MISSILES & SPACE COMPANY

Motor Trade-Off Study

Table 4-15

Decision Analysis Worksheet

Decision Statement: Choose Antenna Position Encoders (Pg. 1)

- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

ALTERNATIVES

OBJECTIVES — MUST (– Is it absolutely essential? – Is it measurable?)	A Optical Encoder on gimbal follower INFO	GO/NO	B Optical Encoder on Gimbal INFO	GO/NO	C Brush type Encoder or gimbal follower INFO	GO/NO	D Brush type Encoder in Gimbal INFO	GO/NO
Accurate to 0.01° (36 arc-sec) or better.		Go		Go	Insufficient resolution	No	Insufficient resolution	No
Operation over +200° to -200°F range.		Go		Go		Go		Go
10 year life capability.		Go	No proven 10 year life of light source.	No		Go		Go
Absolute position indication		Go		Go		Go		Go

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)	WT	A INFO	SC	WT SC	B INFO	SC	WT SC	C INFO	SC	WT SC	D INFO	SC	WT SC
High resolution	10	10^{19} bits possible	10	100									
Ease of alignment (phasing)	3		4	12									
Noise on signals	4		5	20									
Ambiguity of readout	7		7	49									
Low inertia	5		8	40									
Light weight	5		8	40									
Low cost	2		3	6									
Simple replacement	4		4	16									
Low corrosion susceptibility	7		8	56									
Ease of calibration	4		5	20									
Low power consumption	3		4	12									
Low rotating friction	9	None	10	90									
Low debris production	7	None	10	70									
TOTAL WT SC				531									

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2835 B (FRONT)

LOCKHEED MISSILES & SPACE COMPANY

Position Encoder Trade-Off Study

Table 4-16 (Sheet 1 of 2)

Decision Analysis Worksheet

Decision Statement: Choose Antenna Position Encoders (Pg. 2)
- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

ALTERNATIVES

OBJECTIVES — MUST (– Is it absolutely essential? – Is it measurable?)	A Magnetic Encoder on gimbal follower — INFO	GO/NO	B Magnetic Encoder on gimbal — INFO	GO/NO	C — INFO	GO/NO	D — INFO	GO/NO
Accurate to 0.01° (36 arc-sec) or better.	Insufficient resolution	No	Insufficient resolution	No				
Operation over $+200^{\circ}$ to -200°F range.		Go		Go				
10 year life capability.		Go		Go				
Absolute position indication.		Go		Go				

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)	WT	A INFO	SC	WT SC	B INFO	SC	WT SC	C INFO	SC	WT SC	D INFO	SC	WT SC
High resolution	10												
Ease of alignment (phasing)	3												
Noise on signals	4												
Ambiguity of readout	7												
Low inertia	5												
Light weight	5												
Low cost	2												
Simple replacement	4												
Low corrosion susceptibility	7												
Ease of calibration	4												
Low power consumption	3												
Low rotating friction	9												
Low debris production	7												
TOTAL WT SC													

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2695 B (FRONT)

LOCKHEED MISSILES & SPACE COMPANY

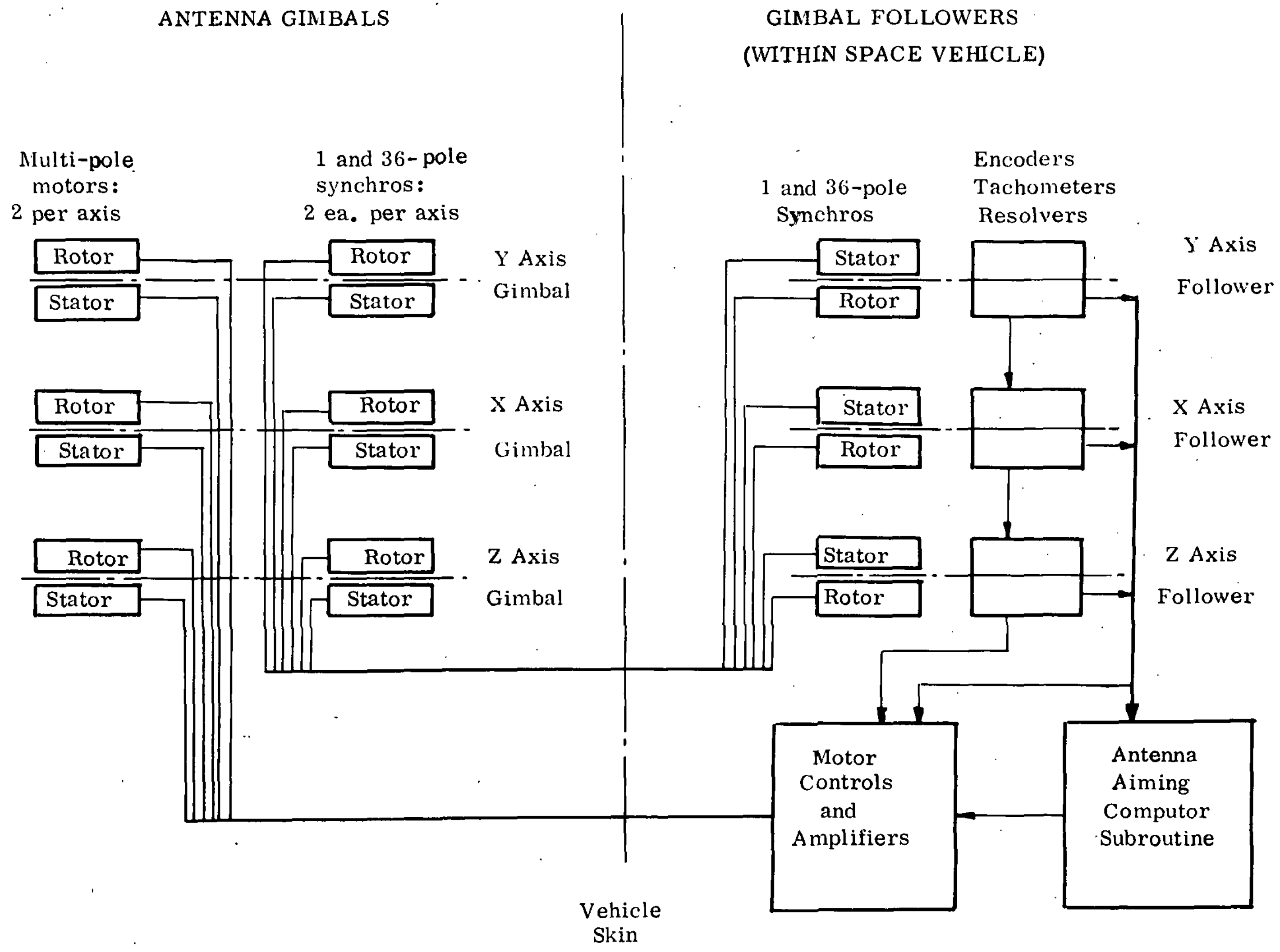
Position Encoder Trade-Off Study

Table 4-16 (Sheet 2 of 2)

A proposed method for meeting both requirements is to drive a shaft within the space vehicle in synchronism with each gimbal. A 36-speed synchro system would keep the shaft within 18 arc-sec of the antenna gimbal. All resolvers, indicators, tachometers and encoders would be mounted on these gimbal follower shafts where they can be maintained, calibrated or replaced if necessary during the mission. This would be essential for the optical encoders, since incandescent light sources do not last 10 years and light emitting diodes (LED's) have been available for only 4 years and therefore have not completed a 10 year life test. The softer environment within the spacecraft will also contribute to the long life of these sensitive instruments.

By using this method, the number of wires that must pass through the power transfer device at each gimbal will be greatly reduced. Each axis will then need only 2 motors (for redundancy), 2 dual-speed synchros and limit switches. This will increase the projected reliability with reduction of active components on the antenna structure where maintenance is difficult or impossible.

The block diagram (Figure 4-2) shows the relationship between the computator, amplifiers, motors, synchros and readout devices.



Recommended Servo Component Positions

Figure 4-2

4.3.5 Power Transfer Joints

4.3.5.1 Slip Rings. This study is concerned with three types of slip rings, namely:

a. Block Brush Types (High Current)

b. Button Brush Types (Signals and Low Current)

c. Metallic Wire Types (Signals and Low Current)

The application here involves signals and low current levels being transferred at relatively low rotational velocities and long periods of dwell such as would be encountered in the gimbal of a tracking drive. A comparison of wet versus dry slip ring lubricants for such applications would yield the following traits.

Dry Lubricants

- Higher wear rates
- Greater noise
- Higher friction

Wet Lubricants

- Requires reservoirs, seals and temperature control
- Potentially capable of catastrophic lube failure
- Unsuited to slow rotational speeds.

The types employing metallic wire brushes are usually wet lubricated where lubricant is fed to the bottom of a "Vee" groove through a wick or by vapors evaporated from a reservoir. Low velocity and dwelling obviate hydrodynamic lubrication; consequently, the lubricant serves as a contaminant to prevent galling and welding.

The selection of a particular type of slip ring will depend on the following parameters:

a. Current density

b. Peripheral surface velocities

c. Noise

A failure mode matrix (Table 4-17) is amplified by the following list of causes and effects.

1. FAILURE MODE: Excessive Noise

Failure occurs here when the noise threshold of a particular signal becomes unacceptable. The limit of acceptability is a variable ranging from the wide tolerances of high power to the low levels of some position transducers.

Cause 1: Debris from Wear

The most common cause of failure results when debris comprised of flakes of slip ring material, brush material, and lubricant forms either a coherent conductive mass or a slurry which affects the contact resistance between brush and slip ring.

Sources of Failure: Lubrication, Material, or Corrosion.

Corrective action can be taken, only, if the wear results from prolonged acceptance testing or improper storage conditions. A short "burn-in" period under simulated environmental conditions will disclose the noise level as well as its trend. Generally, most experimenters find that slip ring noise will diminish as the brushes seat and should reach acceptable limits within reasonable times.

Cause 2: Abraded Surface

The abraded surface, here, refers to a surface which initially is unacceptably rough and accelerates the generation of debris.

Source of Failure: Other than the above-mentioned causes, manufacturing, assembly and shipping.

Careless handling can scratch, abrade, or otherwise damage the slip ring surface. The corrective action may be obvious but also may require that protective containers be used during handling prior to final assembly.

<u>Cause 3</u>: Low Contact Pressure

This failure will most likely be due to brush spring pressure dropping below that needed to provide a low resistance path between the slip rings and its brushes.

<u>Source of Failure</u>: Eccentricity, out of roundness, overstress or rough handling.

Corrective action consists of specifying correct materials and manufacturing tolerances, and seeing to it that it is made and assembled to these tolerances. Other corrective action may require that vulnerable areas of the design be caged during launch and handling to prevent overstressing parts subjected to flexure, as well as avoiding reversing loads.

2. FAILURE MODE: Circuit Opens

Failure may be a continuous or intermittent open circuit. A noisy condition may precede failure, ultimately degrading to an open circuit. Several of its causes are merely extensions of the same things that cause excessive noise.

<u>Cause 1</u>: Debris from Wear

A cohesive mass of debris becomes impacted between the brushes and slip ring.

<u>Source of Failure</u>: Lubrication or Material.

Corrective Action will be similar to that outlined under <u>excessive noise.</u>

Cause 2: Damaged Surface

Source of Failure: Handling, Shipping, Manufacturing, or Assembly. This refers to surfaces that have inadvertently been damaged during handling, shipping, assembly or manufacturing. Deep scratches, gouges, and excessively rough surfaces can accelerate wear to the extent that either the brushes, slip rings, or both, fail to make adequate contact. The mechanisms for wear may be similar to that mentioned under excessive noise.

Cause 3: Worn Brushes.

Source of Failure: Lube or Material Failure, Design. Brushes or slip rings may possibly wear to a condition of an open circuit. Brush spring forces may not be great enough to unseat a brush which has worn itself a seat in the brush holder. Corrective action must begin during the design stages when proper materials and the mechanics are under consideration.

3. FAILURE MODE: Cross Talk

Cross Talk is a malfunction characterized by two or more signals appearing at one channel. Such spurious signals are experienced when large spikes or pulse type signals are inductively or capacitatively cross-coupled by an inadequate dielectric, but can also occur as a resistively coupled signal by conductive debris that has accumulated between adjacent channels.

Cause 1: Debris from Wear.

Source of Failure: Lubrication or Material Failure.

Corrective Action is similar to that outlined under excessive noise.

Cause 2: Lateral Shift of Slip Ring.

This type of failure is heavily dependent upon the ability to maintain alignment in the slip rings where support is achieved outside the slip ring structure itself.

Corrective Action: Corrective action consists of making the proper installation of bearings.

4. FAILURE MODE: Increased Torque

Experimentation indicates that brushes exhibit a characteristic reduction in friction during their "burn in" time. After seating, a substantially long period follows during which little change in friction occurs. Ultimately, an increase in both noise and torque precedes failures. An examination of the failed parts can show an accumulation of debris which contributes both to noise, as well as wear. In those instances where signals are transferred, the limit of tolerable noise may be substantially below that where the slip rings are solely used for power transfer.

Cause 1: Debris from Wear.

Failure involves familiar parameters mentioned earlier in this paper.

Source of Failure: Lube or Material Failure.

Corrective Action: Burn in period as discussed earlier.

Cause 2: Abraded Surface.

Previously discussed.

Source of Failure: Lube or material failure, corrosion during storage, rough handling, or improper handling during manufacturing and assembly.

Corrective Action: Burn-in period will help in accelerating failure mode. Careful handling during manufacture, assembly, and shipping can help to prevent damage to surface and rotor bearings.

Cause 3: Lateral Shift of Slip Ring.

This type of failure results when the alignment of the slip ring with respect to its brushes is lost because of wear in bearings outside of the slip ring structure.

Corrective Action: Special care in the selection and installation of the bearings.

5. FAILURE MODE: Freeze-Up

Freeze-up here refers to torque increases which exceed the maximum available gimbal torque, and the assembly freezes up.

Cause: Cold welding.

Cold welding is a phenomenon where a spot welding process occurs at relatively low temperatures. Material is virtually plucked out along the contact area leaving a galled abrasive surface.

Source of Failure: Cold welding can occur during dwell periods on clean uncontaminated surfaces.

Corrective Action: Cold welding is usually abated by the presence of a thin oil film.

TABLE 4-17

FAILURE MODES FOR SLIP RINGS

(Block, Button, and Wire (Flat or Round) Types)

FAILURE MODE	CAUSE	LUBRICATION	MATERIALS	CORROSION (Launch Site)	FATIGUE	HANDLING	SHIPPING	*THRUST BRG WEAR	PROLONGED STOR-AGE	ACCEPTANCE TEST	ASSEMBLY	MFG.	DESIGN	OPERATIONAL VELOCITY	RECOMMENDA-TIONS AND REMARKS
1. EXCESSIVE NOISE	DEBRIS FROM WEAR	X	X	X						X			X		
	ABRADED SURFACE	X	X	X		X	X				X	X			
	LO CONTACT PRESS.		X		X	X					X		X		
2. CIRCUIT OPENS (Continuous or Intermittent)	DEBRIS FROM WEAR	X	X							X			X		Debris here is an elect. non conductor.
	DAMAGED SURFACE					X	X				X	X			
	WORN BRUSHES	X	X										X		
	NO CONTACT PRESS		X		X	X					X		X		
3. CROSS TALK	DEBRIS FROM WEAR	X	X							X			X		
	LAT. SHIFT OF SLIP RING		X					X			X				
4. INCREASED TORQUE (Causing Excessive Drain on Veh. Power)	DEBRIS FROM WEAR	X	X							X			X		*Refers to Slip Ring Support Brgs.
	ABRADED SURFACE	X	X	X		X	X				X	X			
	LUBE FAILURE	X	X						X			X	X		
	LAT. SHIFT OF SLIP RING							X						X	
5. FREEZE UP	COLD WELDING	X	X										X	X	

4.3.5.2 Rotary Transformer. The Rotary Transformer is an inductively coupled device which directly transfers AC power for components such as motors, synchros, etc. When used for the transfer of analog signals, signal conditioning and other associated electronics must also be considered as part of the power transfer system, and the additional weight, volume, and complexity of the overall system with respect to a more simple system, such as slip rings, must be evaluated.

Although phase shifts, inherent in this device, are predictable from its geometry, flexure, thermal expansion, etc., can cause phase aberations that must be overcome by resorting to carrier or subcarrier systems. Cross coupling can result from a relative shift between rotor and stator; therefore, precise axial constraints must be maintained. Additionally, each portion of the rotor/stator stack must be decoupled from the effects of the adjacent channels.

For a given number of circuits, a rotary transformer can prove to be substantially heavier and more volume-consuming than any other system. Justification for its use must be based on its sole capability of eliminating sliding brush contacts which can wear or add noise to the signal. The matrix (Table 4-18) and the following listings delineate Rotary Transformer failure modes, causes, and corrective actions.

1. FAILURE MODE: Cross Talk

Cause: Lateral Misalignment of Core

Adjacent channels can cross couple because of misregistration of corresponding rotors and stators.

Source of Failure: A relative shift between the stator and rotor can result from wear in a gimbal bearing that is not suited to this application. Bearings should be selected on the ability to maintain axial stability as wear occurs.

Corrective Action: Angular contact bearings appear to be best suited for this application. The electric static and magnetic shielding between channels must be thick enough to accommodate small axial misregisters or anticipated misalignment due to gimbal bearing wear. These increased thicknesses will be over and above the thickness needed for channel isolation.

2. FAILURE MODE: Increased Torque.

Cause 1: Rotor and Stator Rub

Source of Failure: This type of interference can result from radial wear in the gimbal bearings, or flexture of the gimbal shaft resulting from an unexpected load condition.

Corrective Action: A careful inspection for radial run-out must be made on all involved parts such as rotor and stator bearing housings and bore, etc. Gimbal bearings should be preloaded in order to maintain the concentricity tolerances.

Cause 2: Dirt Between Rotor and Stator Gap

Source of Failure: Electromagnetic dirt such as steel chips or metallic oxides are attracted by magnetic flux. Other dirt may come from the launch pad environment.

Corrective Action: A protective shroud should be provided to prevent the entry of dirt or moisture to critical areas of the gimbal when the vehicle is either in storage or on its launch pad. Special care during assembly will prevent the entry of ferromagnetic dirt particles.

Cause 3: Thermal Contraction Between Rotor and Stator

Source of Failure: Differential thermal expansion between close fitting components can cause interference. This type of expansion must be considered in space applications since the only means of cooling, ultimately, is radiative. Local heat sources such as coils and electronics apparatus can heat parts conductively while an adjacent part can be substantially cooler and losing heat to space.

Corrective Action: All designs should be analyzed for thermal gradients and dimensional changes that can result from the vehicular configuration in relation to the space environment. Gaps or clearances should be adequate to accommodate anticipated temperature gradients and the resulting thermal expansion.

3. FAILURE MODE: Freeze-Up

Cause: Rotor is Locked by Stator

Failure here resembles increased torque except that the interference between the Rotor and Stator are sufficient to cause complete lock-up.

Source of Failure: A complete lock-up of Rotor and Stator is more or less a continuation of the source of failure that brings about an increase in torque interference resulting from gimbal bearing wear, the wedging of debris between Rotor and Stator, and differential thermal expansion between close fitting moving parts all can bring about a lock-up.

Corrective Action: Consists of all the precautionary measures listed under Increased Torque.

TABLE 4-18

FAILURE MODES FOR ROTARY TRANSFORMERS

FAILURE MODE	CAUSE	PHYSICAL SOURCE: Materials	Corrosion (Launch Site)	Gimbal Brg. Wear	Radiation	Aux. Equipment	CONTRIBUTORY EVENTS: Handling	Prolonged Storage	Acceptance Test	Assembly	Vibration	Design	RECOMMENDA-TIONS AND REMARKS
1. CROSS TALK	a Lateral Mis-align. Core			X						X		X	
2. INCREASED TORQUE (CAUSING EXCESS-IVE DRAIN ON VEH. PWR.)	b Rotor & Stator Rub			X						X			
	b Dirt		X				X	X	X	X			
	c Thermal Contraction	X										X	
3. FREEZE-UP	a Rotor Lock	X	X	X				X					
4. ELECTRONIC FAILURE	a Open Circuit					X			X	X	X		
	b Non-Linear Response					X			X				

4.3.5.3 Flexible Cables. The scope of this study is confined to flat conducting cables (FCC) which are etched on a flexible kapton or polyester substrate. The conducting elements are generally of thicknesses ranging from a few mils to 20 mils, with potential flexural life spans well within the requirements of this application.

The cable presents the simplest type of power transformer system. It virtually has no mechanical or electrical interfaces other than its connections, and completely eliminates coupling problems. It does, however, require a means of limiting rotation and provisions for re-cycling at the end of its excursion. The modes of failure are presented in Table 4-19 and discussed below.

1. FAILURE MODE: Circuit Opens

Failure here is a complete loss of signal which results from a broken conductor.

Cause 1: Fatigue. Refers to a condition where the safe number of flexural cycles has been exceeded.

Source of Failure: Events other than normal operation may hasten failure as follows:

a. Acceptance testing can contribute to failure by excessive exercise before launching the vehicle.
b. Gimbal support bearing failure can create dimensional changes and positional shifts which tend to increase flexural and shear stresses, ultimately hastening fatigue failures.

Corrective Action: May consist of merely recording the total number of cycles undergone during acceptance testing and making certain that a maximum limit, as established by tests, is not exceeded; also, gimbal bearing

TABLE 4-19

FAILURE MODE FOR FLEXIBLE CABLES

FAILURE MODE	CAUSE	Physical Source: Conductor Mat'l.	Physical Source: Insulator Mat'l.	Physical Source: Laminate Adhesive	Contributory Event: Radiation	Contributory Event: Acceptance Test	Contributory Event: Launch	Contributory Event: Limit Stop Fail.	Contributory Event: Support Brg. Fail.	Contributory Event: Design	Contributory Event: Flexure	Contributory Event: Temperature	RECOMMENDATIONS AND REMARKS
1. Circuit Opens	a. Fatigue	X				X			X	X	X		
	b. Overstress	X					X	X			X		
2. Circuit Shorts	Deterioration		X	X	X					X		X	
3. Noise	a. Fatigue	X				X			X	X	X		
	b. Deterioration		X		X					X		X	
4. Cross Talk	Delamination		X	X	X						X	X	*

*Delamination here refers to failure of the adhesive bond between conductor and substrate.

installations should be reviewed for potential wear from misalignment due to deficiency in the basic design.

Cause 2: Overstress. A condition where strain on the conductors exceeds the elastic limit, ultimately causing breakage because of the repeated number of strain cycles.

Source of Failure: Overstress can result from launch loads or because of strains occurring during periods of over travel (limit stop failure).

Corrective Action: Protect the coils of flat cable during launch by monitoring them in a housing that limits coiled distortion and provides enough overtravel of the cable wrap so that it is physically impossible to apply a load to the connection.

2. FAILURE MODE: Circuit Shorts

Failure here is a complete loss of signal which results from a grounded conductor.

Cause: Deterioration. Reference to the insulating materials or laminating adhesives which can become embrittled by radiation or temperature extremes.

Source of Failure: The insulating material or adhesives that cement the laminates into a flat cable can deteriorate because of direct exposure to space radiation (nuclear and U.V.). The physical breakdown of insulation can cause adjacent conductor to make contact with supporting structure thus shorting the signal to ground.

Corrective Action: Where kapton substrates are bonded using FEP as an adhesive, good resistance to radiation can be expected. Should this prove to be inadequate, radiation shielding should be provided.

3. FAILURE MODE: Noise

Failure occurs when the signal is masked by a background of static or random noise.

Cause: Fatigue. This can be a precursor to a developing open circuit and is due to the rubbing of the broken ends of a conductor.

Source of Failure: Since failure here is a matter of degree, ultimately resulting in an open circuit, the source and corrective action are the same as for Item 1, Cause 1.

4. FAILURE MODE: Cross Talk

Failure here is characterized by interaction of signals on adjacent conductors. This is more evident on pulse type signals which are easily cross coupled. The malfunction appears as an unwanted signal in adjacent channels.

Cause: Delamination of the F.C.C. This may occur in areas where radiation can reach. Radiation acts upon the F.E.P. adhesives that hold substrate and cover layers together causing the conductors to detach. The adjacent conductors may touch ground or each other or they may move close enough to have their signals inductively, resistively, or capacitively coupled to nearby conductors.

<u>Corrective Action:</u> Materials which are radiation sensitive should be properly shielded with enclosures that are opaque to the offending radiation.

4.3.5.4 <u>Rolling Contacts.</u> This study is concerned with three types of rolling contacts; namely, conventional rollers, gears, and inside-out rolling contacts. Rotary Brushes of a conventional design are presently used in low friction electrical motors. The gear and inside-out type roller is more of a novelty than a design that can be readily adapted to a space application.

The most likely candidate for space use is a conventional roller design in which the roller is a ball or a roller bearing made of a conducting metal such as hardened Beryllium copper. Lubrication may be fluid or dry Lamellar materials, such as Niobium Di-Selenide or Molybdenum Di-Sulfide.

The lack of wiping action and resultant self cleaning, such as found in conventional slip ring designs, make the rolling contact a potentially noisy means of transferring signals. It can be expected that dirt or debris will always be present in small amounts which will tend to either modulate or intermittently interrupt the signal.

The rolling contact can be expected to have failure modes that are similar to slip rings and also additional modes characteristic to that of ball or roller bearings.

A failure mode matrix (Table 4-20) follows the discussion of failure modes.

1. FAILURE MODE: Excessive Noise

When the noise threshold of a particular signal becomes unacceptable, it is considered that failure has been reached. The limits of acceptability range from wide tolerances for high power to small tolerance for low power position transducers.

Cause 1: Debris

This cause of failure results when debris comprised of flakes of ring and roller material collects at the rollers, causing intermittent signal modulation.

Source of Failure: Lubrication, Material or Corrosion.

Liquid lubricant can combine with wear debris or corrosive products to form a coherent conductive mass or a slurry which effects the constant resistance between a roller and its ring.

Corrective Action: Can be taken, only if the wear is caused from prolonged acceptance testing, or corrosion is caused by improper storage conditions. A short burn-in period under simulated environmental conditions will disclose the wear trends, as well as the noise levels. Generally, it will be found that roller ring noise will diminish as the rollers seat and should reach acceptable limits within a reasonable time.

Cause 2: Abraded Surface.

The abraded surface, here, refers to a surface which initially is unacceptably rough and accelerates the generation of debris.

Source of Failure: Manufacturing, assembly and careless handling can be the source of scratches or abrasions on the ring or roller surfaces.

Corrective Action: May be obvious, but also may require that protective covers be used during handling prior to final assembly.

Cause 3: Low Contact Pressure.

This failure will most likely be due to spring pressure dropping below that needed to provide a low resistance path between the rings and the rollers.

Source of Failure: Eccentricity, fatigue, overstress, or rough handling.

Corrective Action: Consists of specifying correct materials and manufacturing tolerances, and seeing to it that it is made and assembled to these tolerances. Other corrective action may require that vulnerable areas of the design be caged and/or covered during launch and handling to prevent overstressing parts subjected to flexure and exposure to snagging.

2. FAILURE MODE: Circuit Opens.

Failure may be a continuous or intermittent open circuit. A noisy condition may precede failure, ultimately degrading to an open circuit. Some of its causes are merely extensions of the same things that cause excessive noise.

Cause 1: Debris

A cohesive mass of debris becomes impacted between the rollers and their ring.

Source of Failure:	Lubrication or material.
Corrective Action:	Will be similar to that outlined under excessive noise.
Cause 2:	Damaged Roller
Source of Failure:	Handling, shipping, manufacture or assembly. This refers to surfaces that have inadvertently been damaged during handling, shipping, assembly of manufacturing. Deep scratches, gouges, and excessively rough surfaces can accelerate wear to the extent that either the rollers, rings or both fail to make adequate contact. The mechanisms for wear may be similar to that mentioned under excessive noise.
Cause 3:	Worn Rollers
Source of Failure:	Lube or material failure, design. Rollers and/or their rings may possibly wear to a condition of open circuit.
Corrective Action:	Must begin during the design stages when materials, lubrication and the mechanics are under consideration. Be sure to design to the latest state-of-the-art in materials and lubrication for a roller and ring.
Cause 4:	No Contact Pressure
	This failure is caused when the spring loading of the roller is relaxed to the point that the roller is no longer in contact with the ring.

Source of Failure:	The primary sources are spring arm material with improper heat treat, roller wear and damage during assembly or handling.
Corrective Action:	Confirm the spring temper status of the spring arm material before assembly, protect the assembly with a cover during handling.

3. FAILURE MODE: Cross Talk

Cross talk is a malfunction characterized by two or more signals appearing at one channel. Such spurious signals are experienced when large spikes or pulse type signals are inductively or capacitatively cross coupled by inadequate shielding. They can also occur as a resistively coupled signal by conductive debris that has accumulated between adjacent channels.

Cause 1:	Debris
Source of Failure:	Lubrication or material failure.
Corrective Action:	Is similar to that outlined under excessive noise.

4. FAILURE MODE: Increased Torque

Experimentation indicates that rollers exhibit a characteristic reduction in friction during their "burn-in" time. After seating, a substantially long period follows during which little change in friction occurs. Ultimately an increase in both noise and torque precedes failure. An examination of the failed parts can show an accumulation of debris which contributes both to noise, as well as wear.

In those instances where signals are transferred, the limit of tolerable noise may be substantially below that for rings which are used for power transfer.

Cause 1: Debris

Failure involves parameters previously described.

Source of Failure: Lube or material failure.

Corrective Action: Burn-in period as discussed under excessive noise.

Cause 2: Abraded Surface

Previously discussed under Item 1, Cause 2.

Source of Failure: Lube or material failure, corrosion during storage, rough handling, or improper handling during manufacturing or assembly.

Corrective Action: See Item 1, Cause 1; a "run in" period will aid in detecting anomalies in torque vs time for those particular rings which are suspected of having abraded surfaces. Careful handling during manufacturing, assembly and shipping can help to prevent damage to surface and rotor bearings.

5. FAILURE MODE: Freeze Up

Freeze-up here refers to torque increases which exceed the maximum available gimbal torque, and the assembly freezes up.

Cause: Cold Welding

Cold welding is a phenomenon where a spot welding process occurs at ambient temperatures and between uncontaminated (unlubricated) surfaces. The

TABLE 4-20

FAILURE MODES FOR ROLLING ELECTRICAL CONTACTS

(Including gears, rings & rollers).

FAILURE MODE	CAUSE	SOURCE OF FAILURE: PHYSICAL SOURCE: LUBRICATION	MATERIALS	CORROSION	FATIGUE	*BEARING WEAR	ROLLER WEAR	CONTRIBUTORY EVENT: SHIPPING	HANDLING	ACCEPT. TEST	ASSEMBLY	MFG.	DESIGN	OPER. ENVIRON.	RECOMMENDATION AND REMARKS
1. Excessive Noise	Debris	X	X	X		X	X			X					
	Abraded Surface	X	X						X		X	X			
	Low Contact Press.		X		X				X		X		X		
2. Circuit Opens (Continuous or intermittent)	Debris	X	X	X		X	X			X					Debris here refers to non-conductor
	Damaged Roller							X	X		X	X			
	Worn Rollers	X	X				X						X		
	No Contact Press.		X				X		X		X				
3. Cross Talk	Debris	X	X	X		X	X			X					
4. Increased Torque (Causing excessive drain on vehicle power)	Debris	X	X	X		X	X			X					*Refers to bearing in rollers & rotor support bearings
	Abraded Surface	X	X						X		X	X			
4. Freeze Up	Cold Welding	X	X										X	X	

phenomenon is characterized by material that has been virtually plucked out along the contact area leaving a galled abrasive surface.

Source of Failure: Cold welding can occur during dwell periods on clean uncontaminated surfaces.

Corrective Action: Cold welding is usually abated by the presence of an oil film, however thin. A lubricant should be selected that will insure a lubrication film at the contact surface. For this very low rotational velocity, a dry film lubricant should be more suitable than wet lubricant as the wet lubricant can be squeezed out in a metal to metal dwell situation.

4.3.5.5 Additional Transfer Devices. In addition to the devices that have been considered previously, other signal and power transfer devices were considered. They were not included for more detailed discussion because of basic disqualifying deficiencies in the category of the "must" objectives for this particular application. Two are discussed here in order to present a record of their potential. They could become eligible for consideration in the future, should their state of development and their limiting capacities be enhanced.

Liquid Metal Slip Rings. Description - For electrical current transfer in this type of system, the interfaces of rotating and stationary Beryllium plates are covered with an ion plated gallium film. Beryllium is used because of its resistance to the highly corrosive action of gallium when at elevated temperature. The gallium provides a low shear strength wetted metallic interconnect having low electrical resistance and good lubricity. It has the further advantage of providing a large area of metal to metal contact for electrical conductance without being dependent upon the mechanical properties of brush materials and their required contact loading. Gallium has low vapor pressure for operation in vacuum and a low melting point ($30^{o}C$) for providing a liquid film at normal internal operating temperatures.

Test - A 500 hour test has been run by the Lewis Research Center on this type of Be plate - a gallium configuration, for transferring 20 amp D.C. After initial run-in, the contact noise remained very low (within 0.2 milliohms peak to peak). The contact resistance stabilized around a value of 1.2 milliohms. The basic results indicated that this configuration had a lower electrical noise level than any of the presently available electrical brush compacts containing MoS_2.

This liquid metal device is not recommended at this time for the antenna application. It is not yet developed to the degree that it could be considered reliable for 10 years in a space environment.

Flex Capsule. Description - A Flex Capsule, made by Poly-Scientific, is capable of transferring small current densities through a $\pm$ 110 degree oscillating joint. The current rating is 5 amps for 6 circuits and .1 amp for 14 circuits minimum or 26 circuits maximum. In size, a 32 circuit unit has a 2.16" diameter x .239" thick (rotating head), with a .750" dia. x .961" long body.

The basic principle is a pre-packaged flex cable, which accounts for the limited rotation.

Within the limitations of current capacity and angular rotation, this type of unit could be considered for use in a long life space application. The Flex Capsule is being used in the following projects:

a. The inertial guidance package, both Mark 3 and Mark 4, on Poseidon. (56 circuit 2 amp/circuit).
b. The Apollo Applications Study
c. Agile Missile
d. Viking - Mars Lander (46 circuit 1 amp/circuit).

4.3.5.6 Trade-Off Power Transfer Joints. A KTA trade-off was performed, the results of which are shown in Table 4-21. The best candidate, by an overwhelming margin, was flex-cables. The next best candidate was slip rings.

These results correlate well with what was discovered on the S/S Solar Array Study. They chose slip rings because they were required to rotate continuously in one direction, but they state in their second topical report, LMSC A/995719, that if this condition did not exist they would revert to the flex cable concept.

4.3.5.7 Signal and Power Transfer Device Design Concept. The most promising Transfer Device is the flexible flat cable. There are two different types of flat cables that are commercially available at this time:

1. Round Conductor Cable (RCC)
2. Flat Conductor Cable (FCC)

Decision Statement: SELECT A SIGNAL/POWER TRANSFER DEVICE CAPABLE OF LASTING 10 YEARS.

- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

Decision Analysis Worksheet

OBJECTIVES		ALTERNATIVES: A Slip Rings				B Rolling Contacts				C Rotary Transformer				D Flex Cable			
MUST (– Is it absolutely essential? – Is it measurable?)		INFO			GO/NO	INFO			GO/NO	INFO			GO/NO	INFO			GO NO
Operate to $\pm$ 180° of Rotation					Go				Go				Go				Go
Temperature Range $\pm$ 200° F					Go				Go				Go				Go
Low Speed Operation (.25 RPM)					Go				Go				Go				Go
Vacuum Conditions (1 x 10^{-13} Torr)					Go				Go				Go				Go
Noise Level 100 V/ma max.					Go				Go				Go				
WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)	WT	INFO		SC	WT SC	INFO		SC	WT SC	INFO		SC	WT SC	INFO		SC	WT SC
Provides no Contamination to other Components	10	Wear Debris		3	30	Wear Debris		7	70	None		10	100	None		10	100
Low Noise Insertion	6	Predictable		6	36	Subj. to Noise with Wear		5	30	None		10	60	None		10	60
Good Circuit Isolation (Cross Talk)	7	Axial Displacement Can Cause It		7	49	Axial Displacement and Roller Wear		6	42	Magnetic Flux Must Be Channeled		0	0	None		10	70
Redundancy Feasible	7	Not Difficult		9	63	Significant Increase in Mechanical Components		5	35	Excessive Weight and Size Penalty		0	0	Easily Provided		10	70
Min. Mech. Interfaces (Sliding)	6	One		9	54	Two		6	36	None		10	60	None		10	60
Min. Elec. Interfaces	5	One		9	45	Two		6	30	Multiple		2	10	None		10	50
State-Of-The-Art	4	Most Used		10	40	Not Prevalent		3	20	Limited Usage		4	16	Used with Non-Continuous Rotation		9	36
Adequate Lubrication	9	Adequate Lubrication Available		6	54	Difficult to Lubricate		3	27	None Required		10	90	None Required		10	90
Low Friction (Sliding, Flex, Magnetic)	8	Sliding		7	56	Sliding		9	72	Magnetic		10	80	Flexing		7	56
Gradual Failure (Early Detection of Progressive Failure)	6	Detected through Increased Noise		8	48	Detected through Increased Noise		7	42	No Warning		5	30	No Warning		6	36
Min. Aux. Equipment	3	None Required		9	27	Rollers		8	24	Excessive		2	6	None Required		10	30
Low Wear Rate (Deterioration)	10	One Interface		5	50	Two Interfaces		6	60	Numerous points of Deterioration		4	40	Insulation Only		10	100
Low Mass	2			9	18			6	12	Heavy Magnetic Coils		2	4			10	20
Low Volume	1			8	8			7	7	Many Coils Required For Multiple Channels		1	1			10	10
					578	TOTAL WT SC			507	TOTAL WT SC			497	TOTAL WT SC			788

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2695 (FRONT)

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Table 4-21

KTA Tradeoff Results

Both of these flexible cable types have their advantages and disadvantages, as stated below:

	Advantages	Disadvantages
RCC	1. Conventional manufacture	1. Bulky
	2. Easily stripped	2. Heavy
	3. Conventional connector termination	
FCC	1. Compact	1. Difficult to strip
	2. Light	2. Delamination possibility

Cicoil Corporation fabricates RCC and uses its own proprietary silicone compound as an insulation material. The cables are extremly flexible and capable of 70,000,000 flexures. They have been used by General Electric and Ball Brothers Research Corporation. However, the silicone compound poses a potential hazard in the formation of silicone oxide, which could be detrimental to the bearings. The Cicoil Corporation does not have any experience with polyimide insulated RCC.

Raychem Corporation fabricates polyimide insulated RCC; however, the samples supplied are much stiffer than the polyimide FCC samples.

Lockheed Missiles and Space Company, Incorporated (LMSC) has the capability to manufacture polyimide FCC and has tested it to MIL-C-55543, but has not gone beyond the specification limit of 1000 cycles of flexure.

LMSC has also tested RCC to 1,000,000 cycles but with a large bend radius (approximately 2 inches).

Because of the large number of flexures (60,000 cycles based on 16 cycles per day over a ten year period) and the relatively small bend radius desirable in any gimbal design, it is believed that a flexure test is necessary to substantiate a ten year life.

The type of insulation recommended in the cable is Kapton HF* which is good in terms of:

1. Radiation Resistance
2. High Temperature Aging
3. Abrasion Resistance
4. Flammability
5. High Temperature Resistance
6. Outgassing

In tests[1] performed by DuPont, a Kapton HF Insulation (polyimide-FEP) was the best in Heat Aging Performance, Cut-Through Resistance, Flammability, Radiation Resistance and Outgassing, when compared to five other types of insulation. It was rated second in Abrasion Resistance with MIL-W-81044/1 insulation first.

Shielding of the flexible cable should be considered to eliminate cross talk between signal and power wiring. Shielding is available in braided, silver impregnated epoxy, and copper plated shields. LMSC has developed the process for silver impregnated epoxy and is currently developing a copper plated shielding for future use. Flexure test information on copper or silver impregnated epoxy shielded wire is not available; however, the vendor survey is still in process and not conclusive at this time.

Samples of Kapton insulated wire, both round and flat conductor, show that the flat conductor is more flexible and therefore can be used with a smaller bend radius than the round conductor.

Considering the foregoing observations and data, it is recommended that Kapton insulated flat conductor cable be tested with a bend radius similar to the preliminary design for a minimum of 80,000 cycles. Shielding on the test cables should be both the silver impregnated epoxy and copper plated variety.

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*DuPont Trademark

1. L. L. Lewis, Kapton Polyimide Film, A Thin Wall High-Temperature Insulation for Wire and Cable, E. I. DuPont De Nemours & Co. (Inc.), Wilmington, Delaware.

To avoid scuffing of the flat cable; a design similar to that shown in Figure 4-3 is recommended. The cables, if arranged as shown, will not slide relative to one another or relative to the shaft and/or housing. Therefore, it will only experience the flexure as the deteriorating cause. Another advantage to this concept is that the entire cable is flexed while rotating and not just one portion.

In order to assure an adequate amount of flexure cycles in the Transfer Device, it is recommended that the bend radius be made as large as possible and still be compatible with the gimbal design. The amount of flexure that can be endured without damage would be a function of the bend radius and therefore the testing of flexible cables should be conducted with two or three different bend radii.

The flat cable should be arranged in a way that the load on the bearings will be minimized and a minimum restoring spring force is present. This can be accomplished by equally spacing the flexure points (assuming that more than one cable is necessary) as shown in Figure 4-4.

Fabrication of FCC terminations is difficult at this time and therefore should be taken into consideration in the design. Connector terminations can be made by soldering, brazing, or welding with the brazing or welding method recommended for this design. However, both processes require a stationary machine and therefore require that the cable be pre-fabricated before installation in the gimbal. A careful detailed cable design early in the design phase is necessary to assure that the FCC can be installed correctly.

Figure 4-5 is an illustration of the cable routing that could be applied to this particular concept.

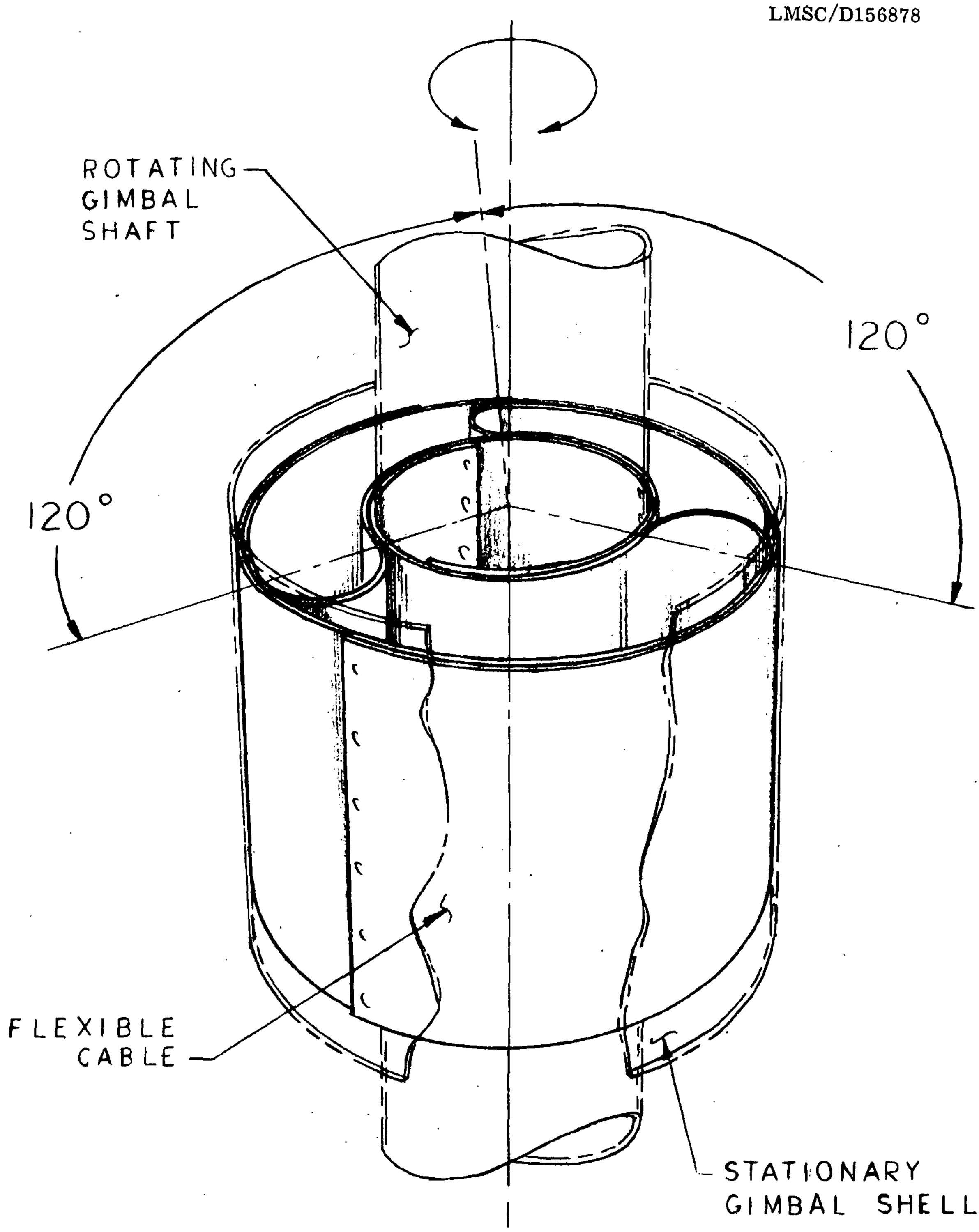
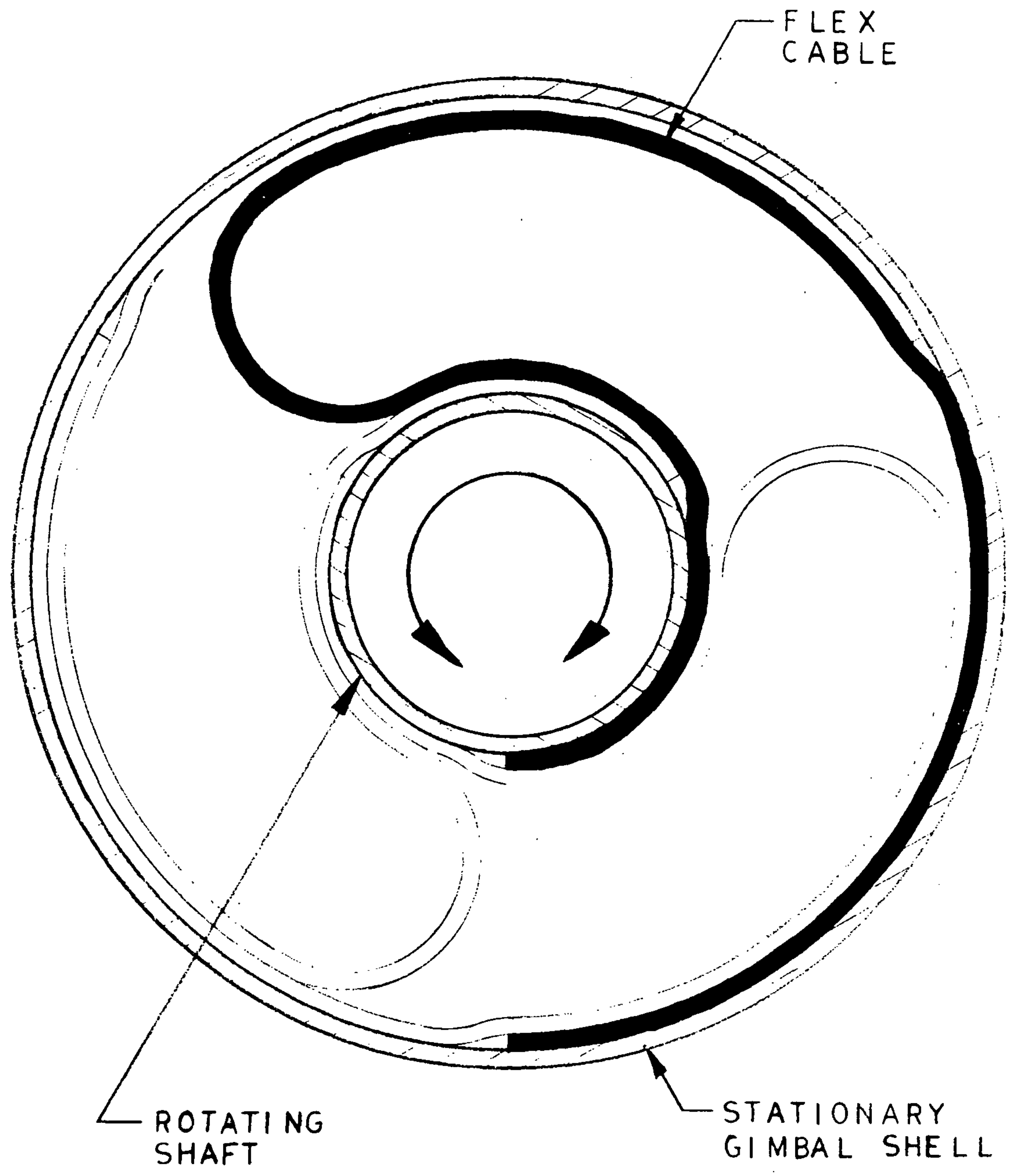


Figure 4-3

FLEXIBLE CABLE CONCEPT



PLAN VIEW
FLEXIBLE CABLE CONCEPT

Figure 4-4

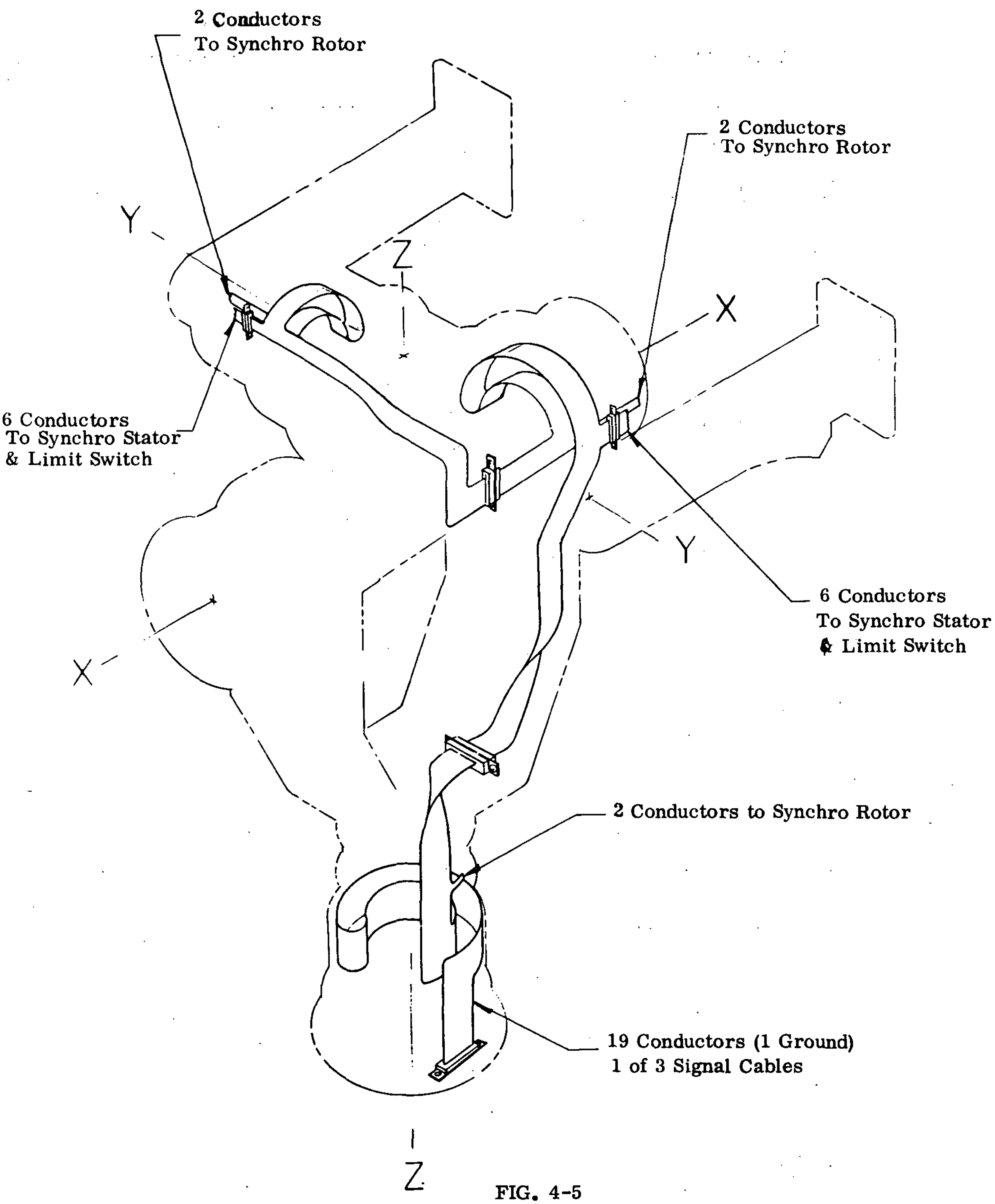


FIG. 4-5

TYPICAL FLAT CONDUCTOR
CABLE ARRANGEMENT

4.4 CONCEPT TRADE STUDY

A concept trade-off was performed on the types of gimbal that can be used to position the reflector, using the KTA method, results of which are shown on the Decision Analysis Worksheet, Table 4-22.

Five different gimbal concepts were used in this analysis, namely:

(1) 2-axis torque tube (Figure 4-6)
(2) 2-axis 360^{o} coverage (Figure 4-7)
(3) 2-axis Az-El (Figure 4-8)
(4) 3-axis coplanar (Figure 4-9)
(5) 3-axis offset (Figure 4-10)

Of all the WANTS considered, "low inertia" and "low bearing load" were considered the most desirable, and therefore given a weight of ten (10). The lowest weighted WANT was "low cost", which was given a weight of two (2).

Low Inertia (10). Among all the wants, this was given the highest weight because of the importance of inertial load on the motors and bearings within the gimbals. All of the loads on the bearings and motors in operation will be due to inertial acceleration or deceleration loads, both thrust and radial (not considering the effects of temperature or preload). Since ball bearing fatigue life is an inverse function of load to the third power[1], it becomes very important in considering a ten year life capability.

Low inertia also indicates a lower torque requirement for a given acceleration, which means that power consumption during acceleration will be minimized.

Another advantage of low inertia is that the counter weighting that may be necessary to test the gimbal can be kept to a minimum.

1. Bisson, Edmond E. & Andersen, Wm. J., Advanced Bearing Technology, NASA, Washington, D. C., 1964.

Decision Statement: SELECT A GIMBAL CONCEPT FOR USE AS AN ANTENNA POSITIONER

- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

Decision Analysis Worksheet

MUST (– Is it absolutely essential? – Is it measurable?)

OBJECTIVES	A 2-Axis Torque Tube INFO	GO/NO	B 2-Axis 360°/Axis INFO	GO/NO	C 2-Axis Az-El Axis INFO	GO/NO	D INFO	GO/NO
Hemispherical Coverage								
Ten Year Life Capability								
Vacuum Operation (1×10^{-13} Torr)								
Operation in Expected Temp. Range (-200°F to +200°F)								
Pointing Accuracy Better than .05 Degree								

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)

OBJECTIVES	WT	A INFO	SC	WT SC	B INFO	SC	WT SC	C INFO	SC	WT SC	D INFO	SC	WT SC
Low Inertia	10	Off Axis Load about Both Axes	7	70	Centered load about both axes	10	100	Off axis load about both axes	7	70			
Lightweight	5	Torque Tube Adds Wght.	8	40	Longer yoke arms add weight	8	40	Most compact design	10	50			
Design Simplicity	4	Compact but not symmet.	9	36	Symmetry makes for simpler design	10	40	Compact design but not symmetrical	9	36			
Fab and Assembly Simplicity	8	Compact Design plus torque tube makes fab assy difficult.	8	64	Symmetry & large structure aids fab & assy.	10	80	Compact design makes fab & assy difficult	8	64			
Low Launch Shock & Vibration Sensitivity	2	Non-uniform bearing load.	7	14	Bearing loads uniform	10	20	Non-Uniform bearing load	7	14			
Pointing Accuracy	8	Support points close together.	7	56	Wide support points gives good angul. accuracy	10	80	Support points close together	7	56			
Maintenance Simplicity	5	Compact design makes access difficult.	9	45	Large structure makes access simple	10	50	Compact design makes access difficult	9	45			
Low Cost	2	Torque design plus torque tube adds cost	6	12	Symmetry & simplicity makes for low cost	10	20	Compact design requires more accuracy	8	16			
Adaptability to Larger Reflector Diameter	5	Single arm pickup with adapter for larger dia.	8	40	Long unwieldly yoke arms	6	30	Single arm pickup with adapter for large dia.	8	40			
High and Low Temperature Resistance	3	Torque tube exposed	9	27	Component protected	10	30	Components protected	10	30			
Low Acceleration at all Positions	5	Gimbal lock	8	40	Gimbal lock	8	40	Gimbal lock	8	40			
Low Vacuum Sensitivity	8	Torque tube opening	9	72	More openings to vacuum	8	64	Enclosed with minimum openings	10	80			
Low Bulk Space	4	Torque tube adds bulk	8	32	Long support arms	6	24	Most compact	10	40			
RF Plumbing Simplicity	5	Torque tube adds complex.	8	40	Long run with 2-axes	8	40	Short run with 2 rotary axes	10	50			
Low Bearing Load	10	Close support points & outside load	5	50	Wide support & midspan load makes less load	10	100	Close support points & outside load	5	50			
Low Thermal Distortion (Effect on Pointing Accuracy)	8	Short structure, less expansion & contraction	10	80	Long support arms	3	24	Short structure	10	80			
Hemispherical Coverage	8	Inherent in design	10	80	Inherent in design	10	80	Inherent in design	10	80			
TOTAL WT SC				798			862			841			

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2695 B(FRONT)

TABLE 4-22

Concept Tradeoff Results (Sheet 1 of 2)

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Decision Statement: SELECT A GIMBAL CONCEPT FOR USE AS AN ANTENNA POSITIONER

- What are we trying to decide?
- What is the decision to be made?
- What decisions must have already been made?

Decision Analysis Worksheet

OBJECTIVES / ALTERNATIVES

MUST (– Is it absolutely essential? – Is it measurable?)

MUST	A 3-Axis Coplanar Axes INFO	GO/NO	B 3-Axis Offset Axes INFO	GO/NO	C INFO	GO/NO	D INFO	GO/NO
Hemispherical Coverage								
Ten Year Life Capability								
Vacuum Operation (1 x 10^{-13} Torr)								
Operation in Expected Temp. Range (-200°F to +200°F)								
Pointing Accuracy Better Than .05 Degree								

WANT (– All objectives that are not "musts" – "Musts" we wish to minimize or maximize)

WANT	WT	A INFO	SC	WT SC	B INFO	SC	WT SC	C INFO	SC	WT SC	D INFO	SC	WT SC
Low Inertia	10	Off axis load about 3-axes	7	70	Off axis load about 3-axes (X-axis offset)	6	60						
Lightweight	5	Third axis adds weight	6	30	Third axis adds weight	6	30						
Design Simplicity	4	Third axis adds complex.	8	32	Third axis adds complex.	8	32						
Fab and Assembly Simplicity	8	Third axis adds complex.	6	48	Third axis adds complex.	6	48						
Low Launch Shock & Vibration Sensitivity	2	Bearing loads uniform but higher than 360° gimb	9	18	Bearing loads uniform but higher than 360°gimb.	9	18						
Pointing Accuracy	8	Support points not as far apart as 360° gimbal	8	64	Support points not as far apart as 360° gimbal	8	64						
Maintenance Simplicity	5	Third axis makes access difficult	8	40	Third axis makes access difficult	8	40						
Low Cost	2	Third axis adds cost	8	16	Third axis offset adds cost	7	14						
Adaptability to Larger Reflector Diameter	5	Spaced pickup arms most adaptable	10	50	Spaced pickup arms most adaptable	10	50						
High and Low Temperature Resistance	3	Components protected	10	30	Components protected	10	30						
Low Acceleration at all Positions	5	No gimbal lock	10	50	No gimbal lock	10	50						
Low Vacuum Sensitivity	8	More openings to vacuum	7	56	More openings to vacuum	7	56						
Low Bulk Space	4	Third axis bulk	7	28	Third axis bulk	7	28						
RF Plumbing Simplicity	5	Third axis adds complex.	6	30	Third axis adds complex.	6	30						
Low Bearing Load	10	Midspan load makes less load	9	90	Midspan load makes less load	9	90						
Low Thermal Distortion (Effect on Pointing Accuracy)	8	Short structure but third axis added	8	64	Short structure but third axis added	8	64						
Hemispherical Coverage	8	Less than 180° coverage	5	40	180° Coverage	10	80						
TOTAL WT SC				756			784						

WEIGHTING
- Which of our "Want" objectives is the most important? (10)
- What is the relative importance of all other "Want" objectives?
- Does the weight assigned to each objective accurately indicate the contribution that the objective should make in the ultimate decision?

SCORING
- How well do all alternatives under consideration satisfy each specific objective?
- Which alternative provides the greatest satisfaction? (10)
- What is the relative satisfaction provided by all other alternatives?

FORM LMSC 2695 B(FRONT)

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TABLE 4-22

Concept Tradeoff Results (Sheet 2 of 2)

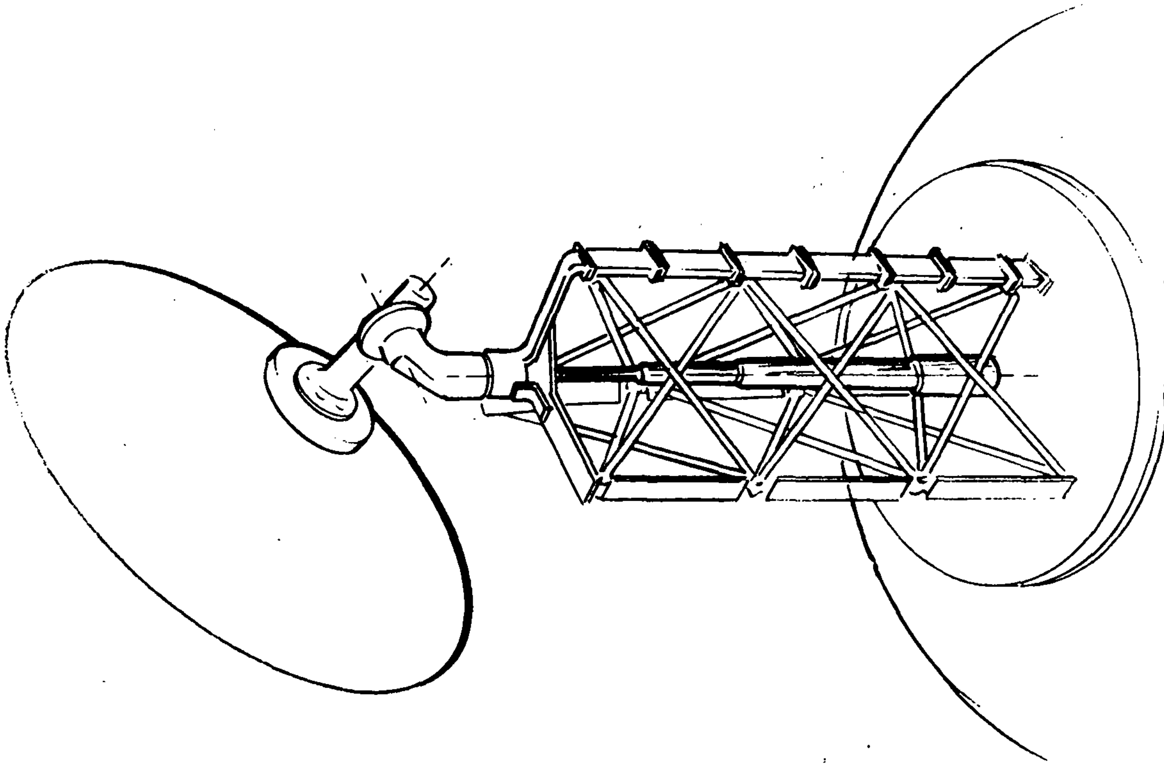
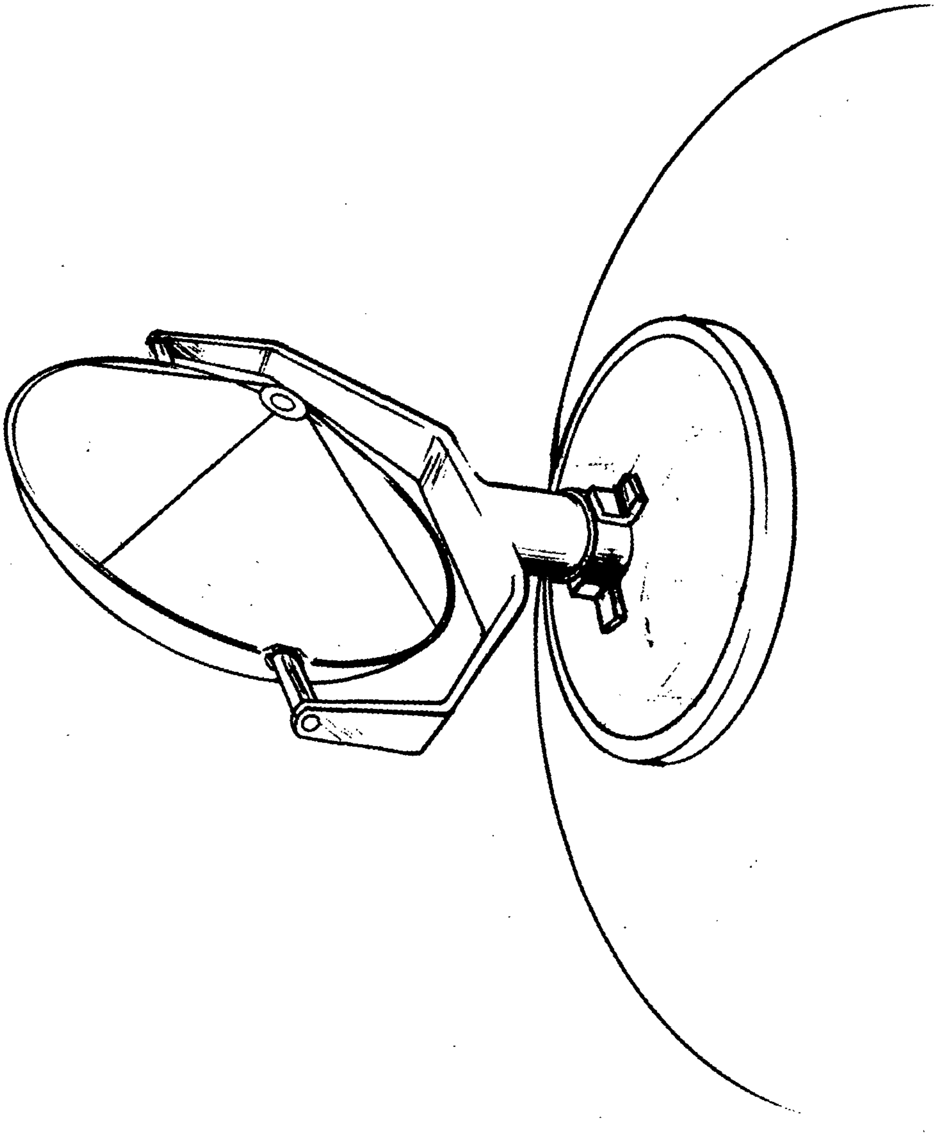
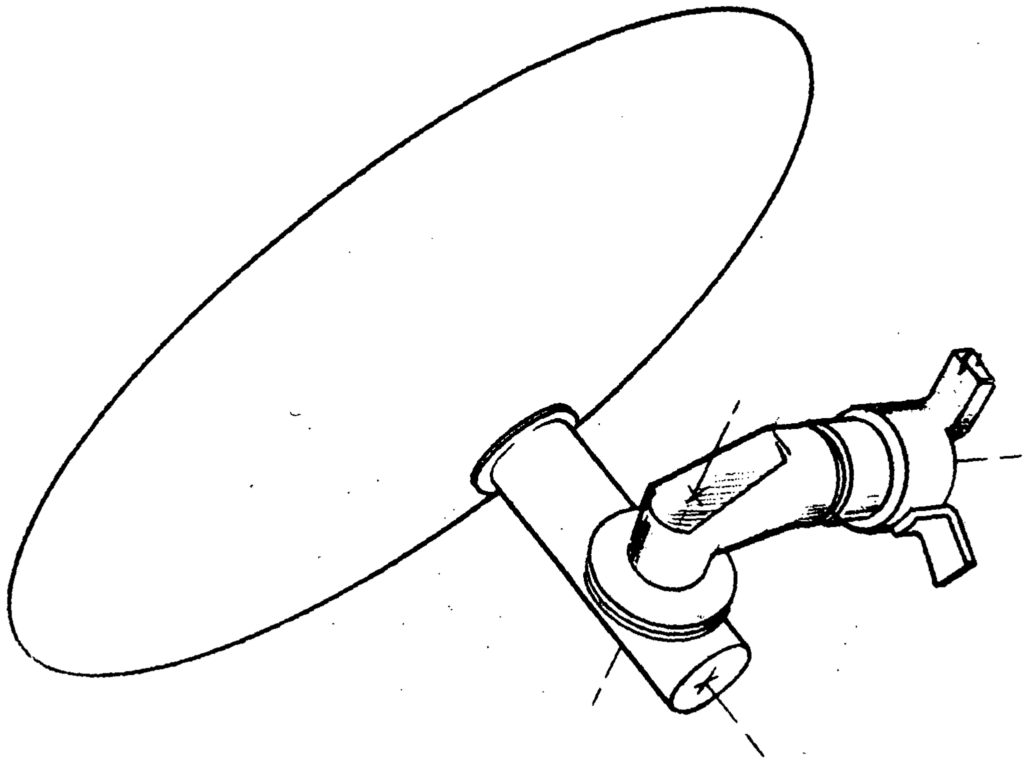


Figure 4-6
TORQUE TUBE CONCEPT

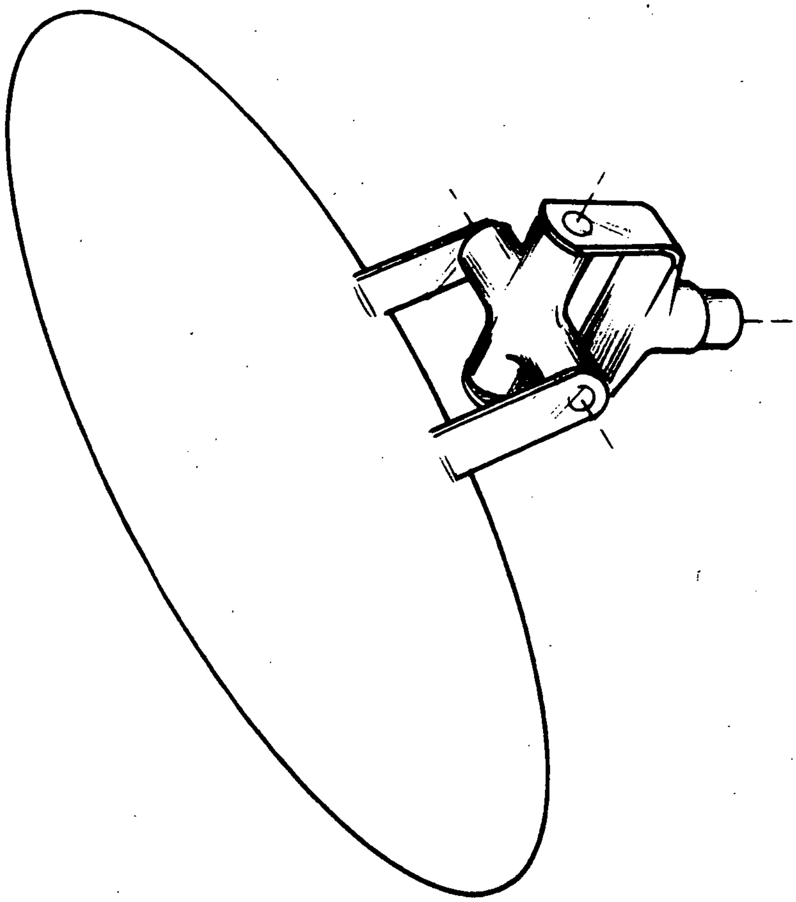


2-AXIS, 360°/AXIS CONCEPT
Figure 4-7



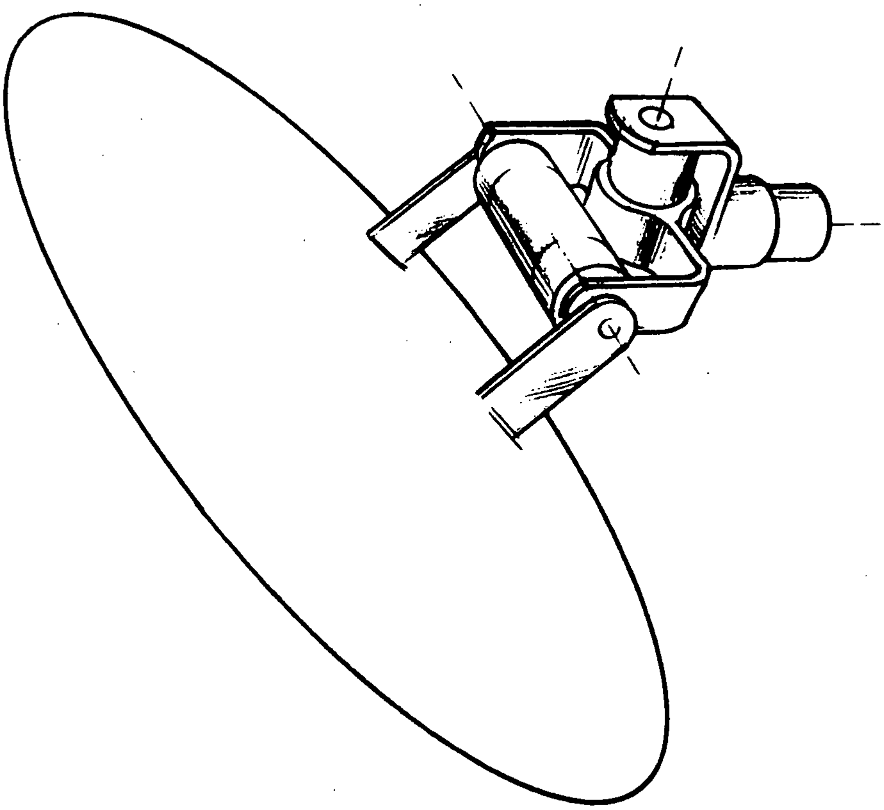
AZIMUTH-ELEVATION 2-AXIS

Figure 4-8



COPLANAR CONCEPT - 3 AXIS

Figure 4-9



STACKED CONCEPT - 3 AXIS
Figure 4-10

Low Bearing Load (10). Although it appears that this WANT is somewhat redundant when low inertia was also considered, it is included as a want because of the higher bearing loads imposed by one type of gimbal arrangement, that is, the arrangement with the load (reflector) outside of the supporting bearings results in higher bearing loads than with the load midway between the bearings. Since bearing load again affects life significantly, this want is given a weight of ten (10).

Low Cost (2). This want is given a relatively low weight because of the replacement or down time costs that would result in case of failure or equipment deterioration. These replacement and/or down time costs will exceed the expected difference in cost significantly.

After scoring the WANTS for each of the five alternatives and taking the sum of the weighted scores, it was found that they scored as shown below:

Alternative	Weighted Score
2-Axis 360^{o}/Axis	862
2-Axis Az-El	841
2-Axis Torque Tube	798
3-Axis Offset	784
3-Axis Coplanar	756

Discussion of Results. It is evident from the weighted scores that the 2-axis gimbals, as a group, are better for this application than the 3-axis.

In reviewing the requirements for a two or three-axis gimbal, the selection is based on the gimbal lock[1] that is inherent in a two-axis gimbal at certain positions. If the gimbal lock can be placed so that it does not interfere with the function of the antenna, or if the gimbal lock can be tolerated, then it appears that the 2-axis, 360^{o}/axis gimbal concept would be the most desirable.

1. Eggert, Dennis, Study of a Low Altitude Satellite Utilizing a Data Relay Satellite System, Hughes Aircraft Company, August 1969.

The limitation of this gimbal concept would be that the yoke arms, to straddle a large diameter, would be unwieldly if the reflector diameter were increased significantly from the 5-1/2 feet that is presently considered. It will not be necessary to support the reflector at the center of gravity since any other convenient position close to the center of gravity offers a lower mass moment of inertia than either of the other 2-axis concepts considered.

If a three-axis gimbal were found necessary, the Coplanar and the Offset axis are both approximately equal in weighted score (784 to 756) and therefore both should be considered. (See Table 4-22).

Assumptions made in the analysis were as follows:

1. The mass moment of inertia about the second axis is lower for the Coplanar than the Offset concept.

2. All concepts can be fabricated from identical material and therefore, the weight is a function of its physical size.

3. Launch shock and vibration will be absorbed by a "lock-out" device and therefore sensitivity is not a heavily weighted want.

One deficiency in the Coplanar design is that the axis closest to the reflector (Figure 4-9) does not have a plus or minus 90 degree coverage. The reflector support arms interfere with the yoke arms at or close to the travel extremes. At reflector "look" angles parallel to the X-axis and with the X axis rotated to the extreme position (plus or minus 90°), the support arms interfere with the yoke. However, since this limited coverage may be tolerable, this gimbal concept cannot be discarded and is included in the KTA Analysis.

The 2-axis gimbals must have plus or minus 90 degrees of travel for hemispherical coverage, whereas the 3 axis gimbals can have hemispherical coverage by utilizing the third axis and not necessarily have plus or minus 90 degrees of travel in the Y axis.

This WANT was given a weight of 8 since the desired coverage is relatively important in achieving hemispherical coverage; that is, if the desired coverage is not available, then the third axis would have to be programmed to rotate the reflector when the Y and X axis reaches a certain pointing angle.

After using a score of 5 for the Coplanar and 10 for the Offset axis and adding the total weighted scores, the Offset axis appears as the preferred with a total weighted score of 784 as compared to 756 for the Coplanar.

In concluding the gimbal design concept study, the following conclusions have been reached:

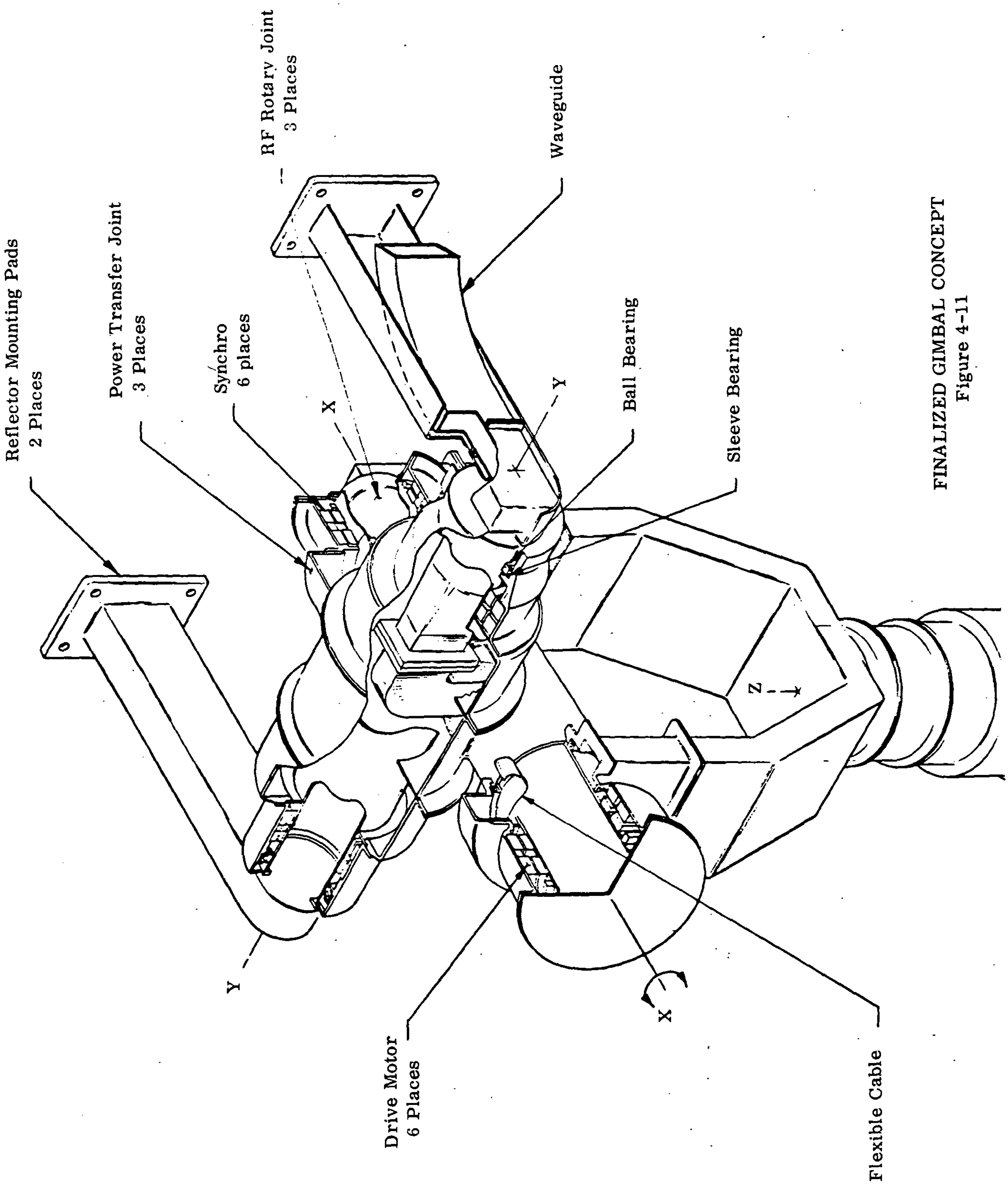
1. The two-axis gimbal concept utilizing a yoke is preferred, provided the gimbal lock problem can be tolerated. The choice of either the 360^{o}/axis or the yoke type X-Y axis can be made as the design parameters (reflector size and coverage) are defined.

2. The three-axis Offset X-Y axis gimbal concept is preferred if gimbal lock <u>cannot</u> be tolerated and it is necessary to have equal to or greater than 180 degree coverage for the Y axis at all X axis positions.

4.5 FINALIZED CONCEPT DEFINITION

Figure 4-11 depicts a gimbal with what is currently felt to be the most reliable components. The motors and position transducers are of the non-contacting brushless variety.

Sleeve bearings are shown as a backup to the ball bearings. The rationale for this is that ball bearings can be designed to last the full 10 year period without approaching technology limitations. But we know from experience that random failures do occur at times in even the most stringently designed and controlled systems. Our KTA study indicated that sleeve bearings were as good as ball bearings and, in fact, have a lower failure rate. Since the redundant bearing, whatever it might be, would be stationary until the torque of the innermost set increased to a point that it was greater than that of the outer set,



FINALIZED GIMBAL CONCEPT
Figure 4-11

it was felt brinelling of the race and/or balls might occur if it were a ball bearing. This will not occur with a journal bearing.

The only point of contact between moving and stationary points in any of the axes is the bearings and the flex cable.

Consideration was given to the desirability of waveguide or coaxial transmission line through the gimbal from a mechanical standpoint and it was concluded that a coaxial line would be more desirable because of the space required for waveguide and waveguide rotary joints. A more compact gimbal assembly can result from the use of coaxial lines through the gimbal. The coaxial lines can be enclosed within the gimbal to partially shield it from the space environment. Non-contacting rotary joints (waveguide or coaxial) would be enclosed within the gimbal shell to protect them in the same manner. The waveguide shown in this concept was sized for 6.5 GHz. It was realized that at this frequency, coax would probably be used. However, it represents a worst case from a mechanical envelope standpoint, since either an increase or decrease in frequency would result in a smaller transmission line.

The effect of a simulated one-g load on the gimbal design was considered from the standpoint of increased load on the motor, taking the worst case loading as one, with one-g acting at the antenna center of gravity and perpendicular to the lever arm.

Assumptions made in determining the motor load were as follows:

Reflector f/D	=	.35
Reflector Thickness	=	.050 inch
Feed and Support Structure Weight	=	3 lb.

The reflector and feed weight with these assumptions add to 22 pounds, with the center of gravity approximately 7 inches from the vertex of the paraboloid.

Using the lever arm as 14 inches (based on preliminary layout), the torque required to raise the antenna in a one-g field would be 308 pound-inches, which is a sizeable torque requirement when compared to torque imposed by accelerations in the order of 5-10 degrées per second squared.

The torque required due to an acceleration of 10 degrees per second squared without the simulated g is approximately 3 pound-inches, or about 100 times less than that required in raising the antenna in a one-g field. Therefore, it appears that the penalty in increased motor, and thus gimbal size, is considerable.

4.6 RELIABILITY MODEL

The following illustration (Figure 4-12) is the reliability model for the chosen baseline concept. This is a block diagram of the various elements of the antenna system showing whether the elements are in a series or parallel relationship and whether redundant elements are in standby or active redundancy. Additionally, the model provides a convenient means of identifying areas where additional redundancy could and should be provided. The assignment of individual element reliabilities to each element permits the conversion of a Reliability Model into a Reliability Mathematical Model.

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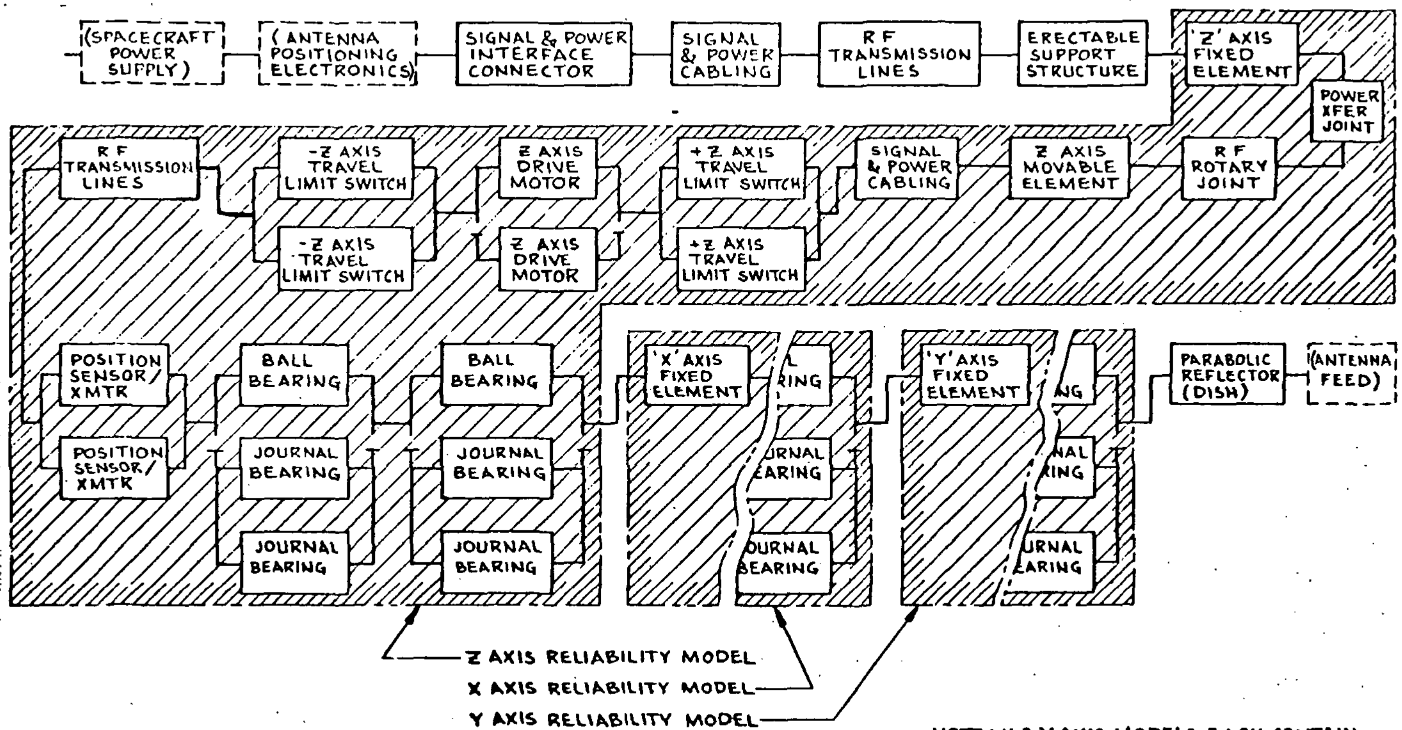
HIGH GAIN SYSTEM RELIABILITY MODEL



NOTE: X & Y AXIS MODELS EACH CONTAIN THE SAME NUMBER AND TYPE OF ELEMENTS AS THE Z AXIS MODEL

DO5958

Figure 4-12

Section 5
TEST DESIGN

To paraphrase a quote, "*to test or not to test - that is the question*". The answer depends a great deal on the technical background of the person being asked. For example, when this question was put to engineers primarily concerned with testing as a vocation, they were amazed that the question even needed to be asked. "Of course!", was their collective reply.

On the other hand when the question is asked of a mechanical design engineer, and when he sees the relatively low environmental levels to which the design is to be subjected, he is hard put to justify a "functional life test" solely on the basis of the 10 year life requirement. "After all", he reasons, "time in and of itself is not a detrimental element".

It is to this difference of "opinion" which we will address ourselves in this section.

5.1 TEST OBJECTIVES AND PHILOSOPHY

5.1.1 Objectives

Before we consider the actual test design we must first define what any test program should prove, or in other words, "What is the test objective?". One objective is, of course, to demonstrate the capability of a particular component or subassembly to survive the environment. But, the word survive can take on many meanings. For example, a bearing over a period of time operating within *its design limits will have a gradual but usually predictable increase in resistive* torque. As long as this torque increase is within the capabilities of the drive system to overcome it should not be considered detrimental. If, however, at the same time, the motor torque drops significantly below its nominal capability because of a decrease in the field strength of a permanent magnet (which can happen under thermal cycling) we would, indeed, have a failure in the form of lockup. In this case, without other diagnostic information, we would probably surmise the bearings had caused the failure when they were really doing just what had been expected of them.

This brings us to a second objective of "functional life test", failure diagnostics. If a component is going to fail, how do we detect this failure in advance? How do we know, for instance, if the torque on a bearing is increasing, if the motor torque capability is decreasing, or if some other frictional effect (rubbing surfaces) has come into play? Without historical data of each component, the "health" of the system cannot be accurately ascertained and corrective measures can not be taken without a physical inspection, which would involve extra vehicular activity.

A third objective which would come into play under a subassembly level test is determination of material compatibility. As an example of the types of problems which could occur, it was discovered that silicone oils, normally felt to be an excellent lubricant for space applications, combined with residual (absorbed) oxygen in the presence of an electrical arc to form $Si\,0_2$ which is common beach sand. This compound, in turn, migrated under zero-g to the bearings causing catastrophic failure. In our particular design we are not recommending silicone oil as a lubricant and the motor will be a brushless torquer and therefore we should not see this problem, but we cannot at this juncture rule out all other possibilities.

5.1.2 Philosophy

In considering the design of tests to prove a 10 year capability, the second step is to establish the basic philosophy to be used.

1. The test must simulate the actual environment to which that particular component will be subjected.
2. Once the test has begun, it must be conducted "hands off" through the planned duration. Any interruption of the test, i.e., loss of environment, would invalidate the test results.

3. The components were designed for representative sizes, materials, and capabilities of the proposed actual flight configuration.
4. Automatic controls will be used wherever possible to reduce cost and eliminate the "human factor".
5. Components will be tested in such a way that failure of one specimen under test or failure of the test set-up will not invalidate the remainder of the test.
6. Incipient failure detection techniques will be employed to provide historical information for future reference and in-flight failure prediction.

In reviewing the component studies performed to date, there seems to be no justification for testing any particular component to show that it will last 10 years. We can, with present technology design a bearing/lubrication system that is capable of surviving the environment with little or no cause for concern, barring some manufacturing process deficiency.

The flex cable being used for motor power transfer can be analyzed and tested on an accelerated flex test with no cost or schedule impact on the overall space station program. It, therefore, can be classified "non-critical" by definition.

The motor can be designed with present technology and currently available materials so that it will outlive the design life.

One might be tempted at this point to say that there is no reason to run a life test because each of the components is fully capable of surviving the environment. This brings us to the illustration alluded to previously, that of a motor torque capability decreasing while the resistive torque of the bearings and flex cabling increases to a point that the inertial torque requirement could not be met.

To ensure that this combination of events does not occur, torque signature tests will be required of each of the three components. These curves would then be superimposed on the inertial torque curve, as in Figure 5-1, and a critical time determined. A factor of safety can then be applied to this time and the overall confidence in the system increased significantly.

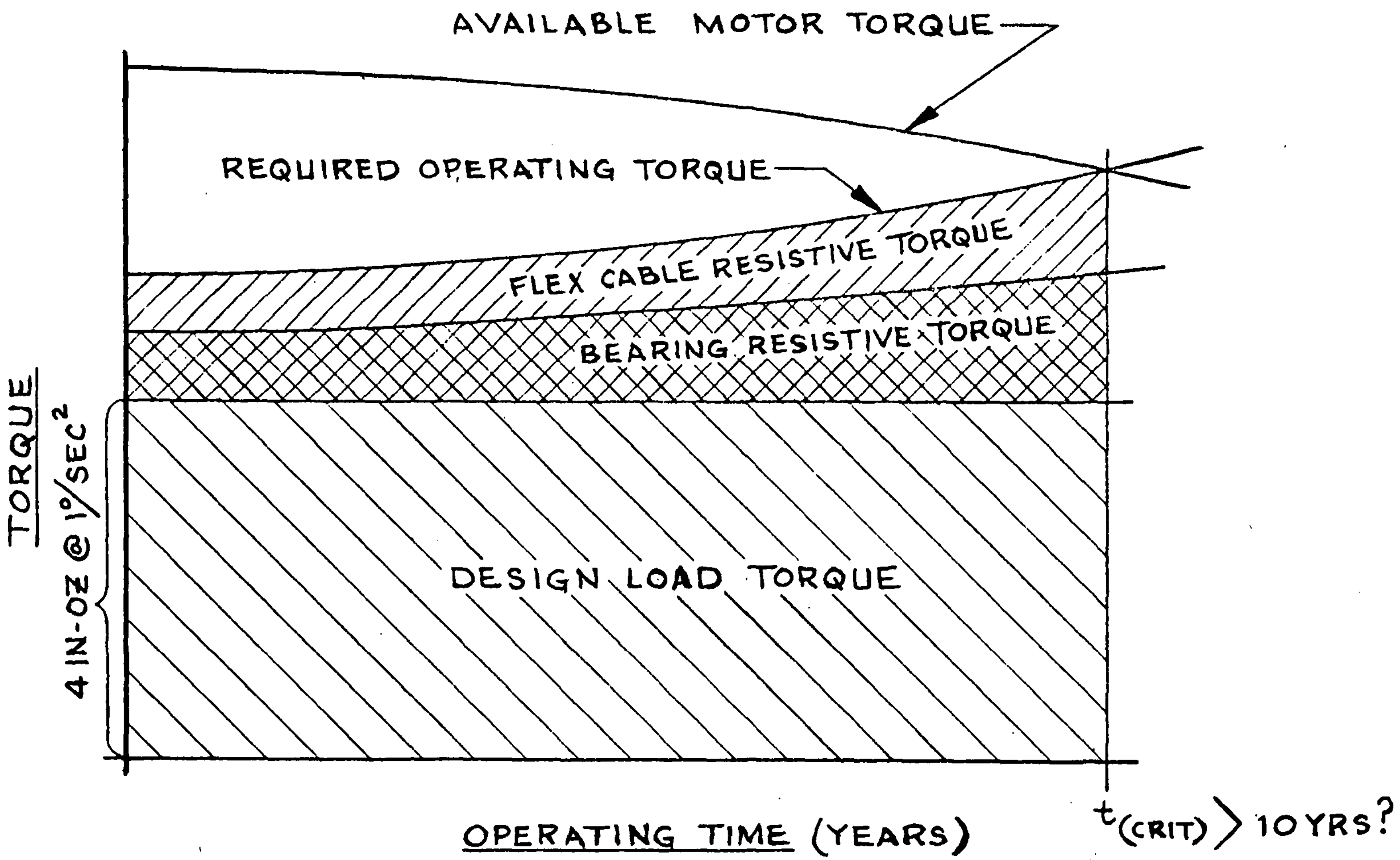
5.2 DETAILS OF TEST SETUP

5.2.1 Bearing Testing

From all indications, the only conceivable failure that could occur in the bearings would be due to a manufacturing deficiency or space borne contamination. The manufacturing problem areas have been discussed at length in Section 4.3.1. It was brought out that these are very real problems and warrant a whole study program on their own merits. The problem of contamination from elements within the gimbal can, however, be handled more readily with certain basic assumptions.

1. Materials outgassing rate can be measured and the long term effects can be extrapolated.
2. The recondensibles of a particular material (usually a small percentage of the whole) will attach themselves to the bearings.
3. Any chemical reaction which might take place between the outgassed molecules and the lubricant or bearing material can be predicted or detected.

With these assumptions in mind, we can conceive a test that would cycle (both mechanically and thermally) a bearing in an atmosphere of outgassed molecules at a pressure probably around 10^{-6} Torr. This pressure is based on measurements made during past flights. It is difficult to achieve pressures lower than 10^{-6} Torr in the near vicinity of a spacecraft even though the ambient pressure is 10^{-9} Torr or below.



Component Torque Capability

Figure 5-1

Gimbal Bearing Rates. The gimbal's angular excursion and nominal angular velocity have been calculated under the computer simulation study currently going on at LMSC. Assuming an orbital altitude of 250 N.M. the angular excursion is ± 90 degrees and the angular velocity is given by Figure 5-2.

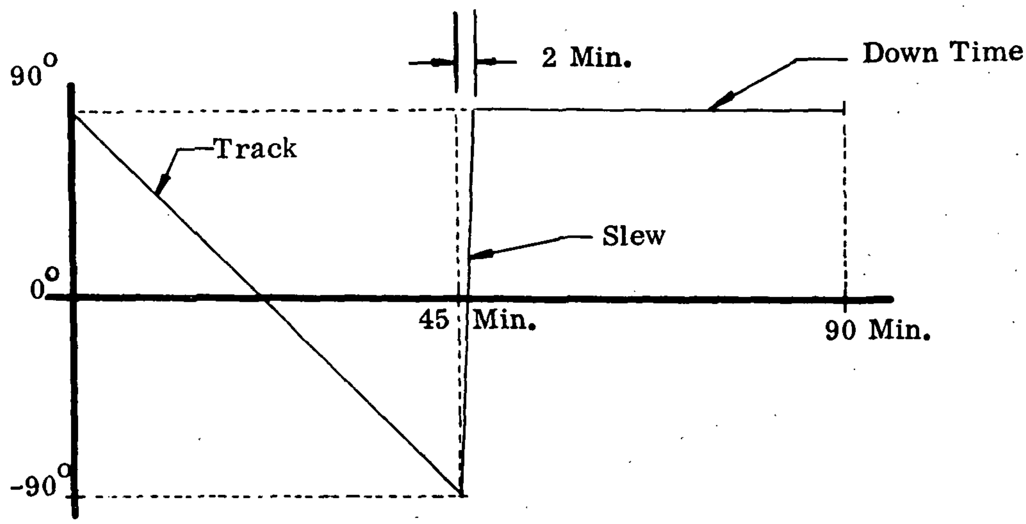


Figure 5-2
Gimbal Bearing Rate - Angular Velocity

If we assume we can accelerate this cycle by a factor of 10, which from all indications is well below the point that dynamic effects begin to appear, the cycle then is represented by Figure 5-3. The complete 10 year mission could be simulated in one year at this scale factor. It would be possible to accelerate the cycle to a factor of 20 if necessary, but beyond this the slew rate would have to be modified to reduce the dynamic effects.

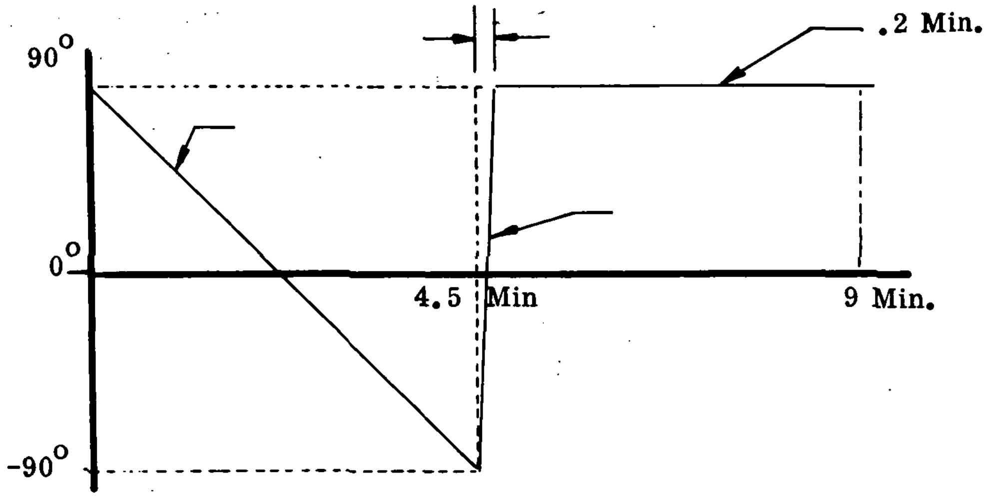


Figure 5-3

Gimbal Bearing Rate - Duty Cycle

During the evaluation of the accelerated testing duty cycle many test reports were reviewed and testing techniques were investigated. In all cases, the tests were on relatively high speed applications with limited operational duty cycles for relatively short design lives. There was agreement between authors that there is a speed below which dynamic effects become insignificant. The exact speed at which this occurs is dependent on a number of parameters; load (thrust and radial), lubrication, temperature, materials (ball and race), and bearing size. This speed limit is arrived at though empirical rather than theoretical analysis. Because of the lack of empirical information on the present configuration certain intuitive decisions had to be made in order to conduct the first test. Through this test, data will be obtained which will either substantiate or refute the original assumptions and more refined judgments can then be made to evaluate the accelerated duty cycle.

In assigning the acceleration factor the cônsiderations and assumptions made were:

- The slew, on orbit would require 2 minutes for 180^{o} (0.25 RPM).
- A factor of 100 (25 RPM) was too high to obtain valid test results.
- A factor of 10 (2.5 RPM) was reasonable and could be performed within the time frame allowed.

It may turn out that with the aid of test results, this factor could be revised to some higher rate and thus reduce the overall testing time.

Because of concern about gimbal oscillation, an investigation was performed to determine how much oscillation would be present in the proposed design. The results of this study indicate there will be very little. The exact number is dependent on

(1) Space Station bending modes

(2) Antenna boom bending modes

(3) Orbital attitude corrections.

The space station modal frequency is expected to be extremely low, less than 0.1 Hz.

The antenna boom bending modal frequency will be relatively high, above 3 Hz. Both of these frequencies are far enough removed from the servo response that with proper filter design they should have little affect on gimbal bearing oscillation. Because of the high mass of the space station, orbital attitude corrections are expected to be slow and infrequent. If the proposed design was a different configuration such that it

(1) contained a geared stepper motor,

(2) contained intentional dither due to extremely tight tracking requirements,

or

(3) was design improperly with non-linearities,

there would be a much higher number of cycles impressed on the bearings. Since none of these are contained in this design, it was concluded that the duty cycle as defined previously is still valid.

5.2.2 Flex Cable Testing

Test Design Goals. Various design concepts were evaluated for the bearing and flex cable tests.

1. The concept that was chosen consists of each test article having its own test chamber (for molecular isolation) containing a predetermined amount of type of out-gassing materials.
2. The chamber will be evacuated through the sequential use of a mechanical roughing pump, an oil diffusion pump with LN_2 cold trap, and finally an Ion pump to maintain pressure below 1×10^{-6} Torr (Ref: Pump similar to Varian - V-11402 Vaclon Pump with a V-1304 power supply).

The bearing and flex cable test will be operated from the same power source which will be a motor geared to .11 RPM. This motor will drive a cam which will in turn drive a rack and pinion operating a magnetic coupling. The magnetic coupling has an external and internal disc, thus no rotary seal is required. This would ensure that the reliability of the test set up was higher than that of the test specimen. With this design a test hardware failure can be repaired without effecting the test specimen.

Rejected Design Concepts. Concepts which were evaluated included:

(1) A design in which there was a sealed unit containing the test bearings mounted on a shaft fixed in place. The outer race would be restrained from rotating with respect to the shaft using a weighted pendulum. This would require cycling the entire chamber. The flex cable would be tested using a Rolimite*.

*Technical Memorandum 33-382, Proceedings of the 3rd Aerospace Mechanisms Symposium, October 1, 1968.

This again would require cycling the entire chamber. It was determined that this concept would be difficult to set up and possibly more difficult to calibrate.

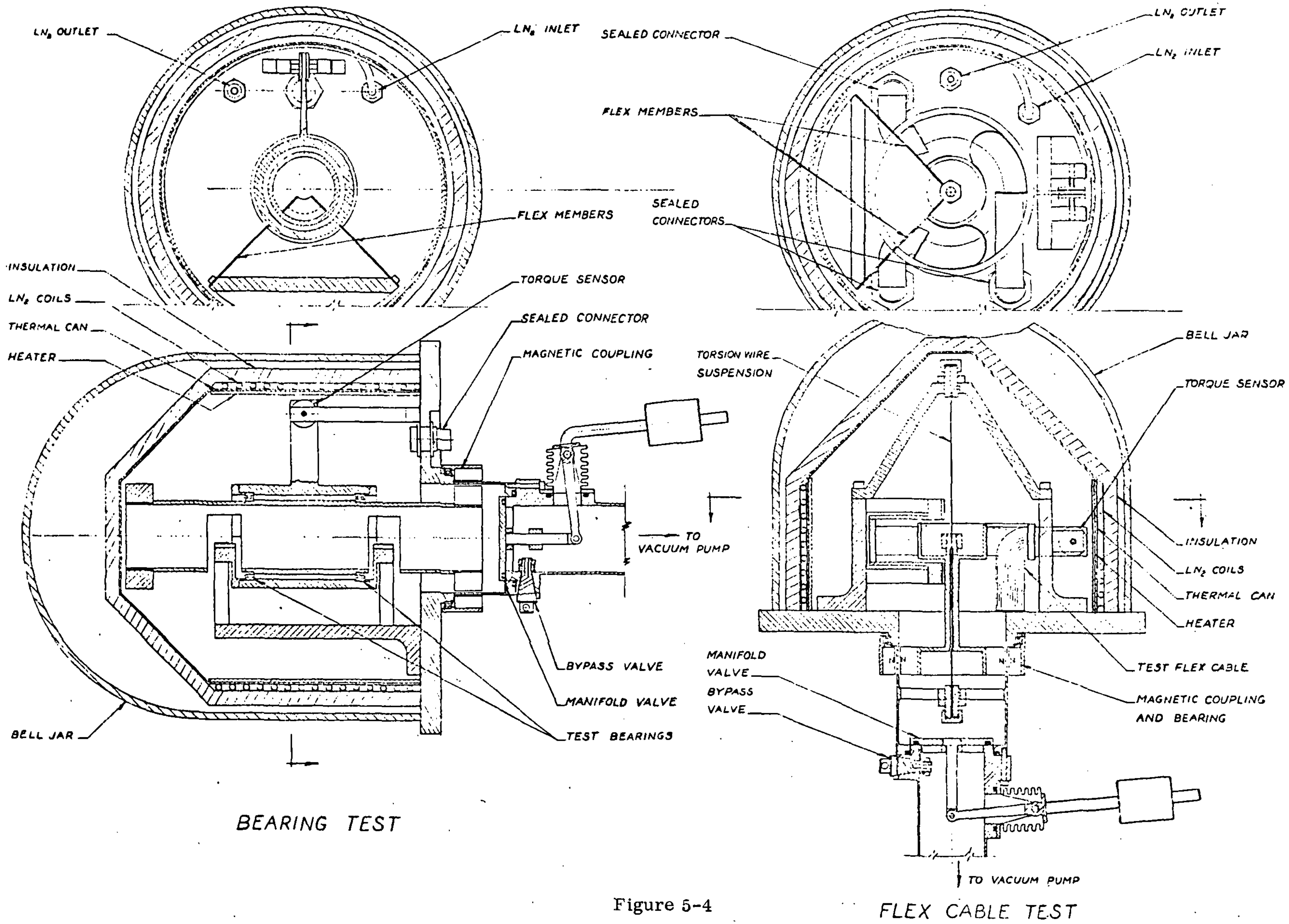
(2) Another concept required an external power source, but it would be connected to the test bearing and would require a dynamic seal such as ferrometic rotary feed thru seal or a hermetically sealed rotary flex seal. Both of these seals have a history of unreliable operation.

Bearing/Flex Cable Test Design. Figure 5-4 is a copy of the preliminary layout for the bearing and flex cable tests.

Flex Cable Testing. Work hardening of flexible cables has been shown to be the primary concern in this type device. Outgassing of the insulation can cause a certain amount of stiffening of the insulation. The net result of both these is the increase in resistive torque. A flex test can be set up, in a vacuum under temperature cycling to measure the change in stiffness of a particular configuration and this, in turn, correlated to its effect on torque. The flexing rate will be handled the same as the bearings.

5.2.3 Motor Torque Test

The testing apparatus for the motor is a bit more difficult to conceive. We would like to test the motor torque as it moves a shaft through an angle. The shaft would have to be mounted on bearings for this type of test, but we do not want the bearings to influence the motor torque measurements. Another more direct method for obtaining the desired information is through the measurement of stall torque. The rotor would be mounted on a fixed shaft. The shaft would be instrumented either with strain gages or a deflectometer and this information calibrated to give the torque. This would not be simulating the exact duty cycle,



LN2 OUTLET
LN2 INLET
FLEX MEMBERS
INSULATION
LN2 COILS
THERMAL CAN
HEATER
TORQUE SENSOR
SEALED CONNECTOR
MAGNETIC COUPLING
TO VACUUM PUMP
BYPASS VALVE
MANIFOLD VALVE
TEST BEARINGS
BELL JAR
BEARING TEST
SEALED CONNECTOR
FLEX MEMBERS
SEALED CONNECTORS
LN2 OUTLET
LN2 INLET
TORSION WIRE SUSPENSION
BELL JAR
TORQUE SENSOR
INSULATION
LN2 COILS
THERMAL CAN
HEATER
TEST FLEX CABLE
MAGNETIC COUPLING AND BEARING
MANIFOLD VALVE
BYPASS VALVE
TO VACUUM PUMP
FLEX CABLE TEST

Figure 5-4

since the motor generally will be operating at something less than stall torque on orbit and we can expect higher heat rates in the coil windings during the test. The test cycle rate would have to be modified to account for this heating effect.

It will be necessary to subject the motor to vibration and/or shock after torque calibration and before the start of the torque test to get a true picture of the torque history as it will appear on orbit.

The torque motor, an angular position transducer (resolver) and a signal generator are to be mounted on a common shaft supported on two ball bearings. An inertial load disc will be mounted on the upper end of the vertical shaft. Torque sensors will be installed on the motor and ball bearings. The entire test assembly will be enclosed in a bell jar for evacuation. Electrical leads will pass thru a hermetically sealed connector in a manifold base of the bell jar. (See Figure 5-5).

The signal generator will provide commutation angular information for energizing the proper stator coils for controlled rotation. The resolver will provide angular position information for closed loop positional follow-up for the programmed duty cycle.

The torque sensors will provide continuous torque values in the form of electrical signals. Separate sensors will be used for the bearings so that the effects of bearing torque can be isolated from those of the motor. The electrical signals vs torque relationship may be calibrated by applying known torsional loads to the motor or bearings and recording the resultant electrical values of the torque sensors.

The motor torque/current relationship will be measured and recorded at the beginning of the Life Test, See Figure 5-6. As the test progresses this relationship is expected to change as a function of time and number of duty cycles. In addition, torque requirements of the bearings may increase as a function of time, number of duty cycles and contamination. The motor current is expected to increase gradually as the motor is commanded to follow the programmed duty cycle. A sensing device set at a pre-determined failure

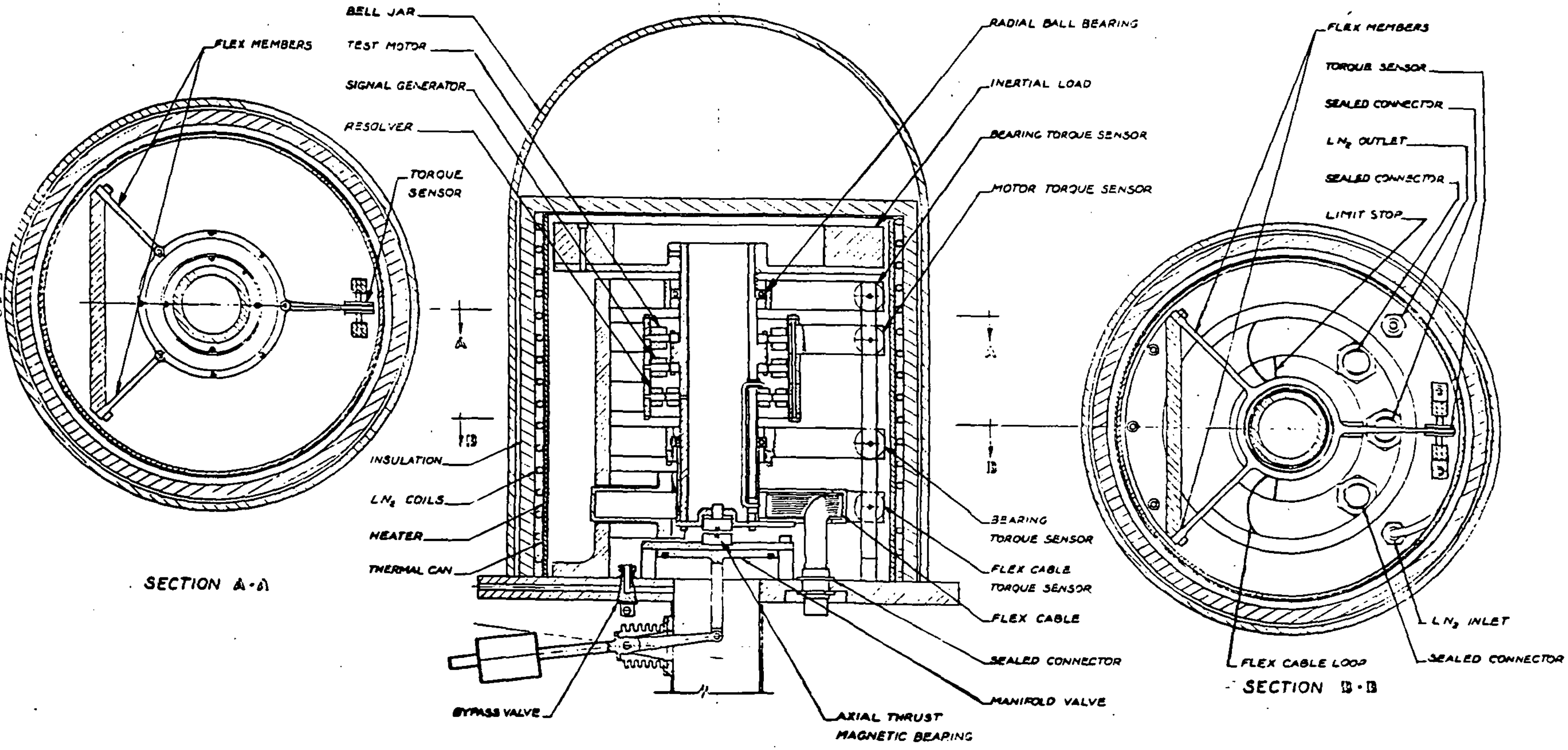


Figure 5-5
Torque Motor Test

level of current flow will terminate the test if this value is reached.

Test Requirements - Equipment (Figure 5-6)

1. Digital Voltmeter
2. Current Shunt
3. Current Monitor (Meter)
4. Strip Recorder
5. Stroboscope or Tachometer
6. Motor Pulleys, Weights, etc.
7. Variable Load (Weights)
8. Variable Power Supply (Voltage)

A switch time delay for the motor is needed to start the recorder sufficiently ahead of the motor start to assure steady operation of the recorder.

The power supply is to be capable of operating the motor on start with no more than 1% change in voltage.

1. Calibrate (Figure 5-6). Load weight platforms with 1 ounce weights each side - Power and record on (length of record to be sufficient to assure record of motor RPM in stady state-up to speed).

Load in 2 ounce increments - one ounce each weight platform.

-Load to stall-

2. Calibrate (Alternate Method (Figure 5-7). Substitute springs and tension load cell for weight platforms.

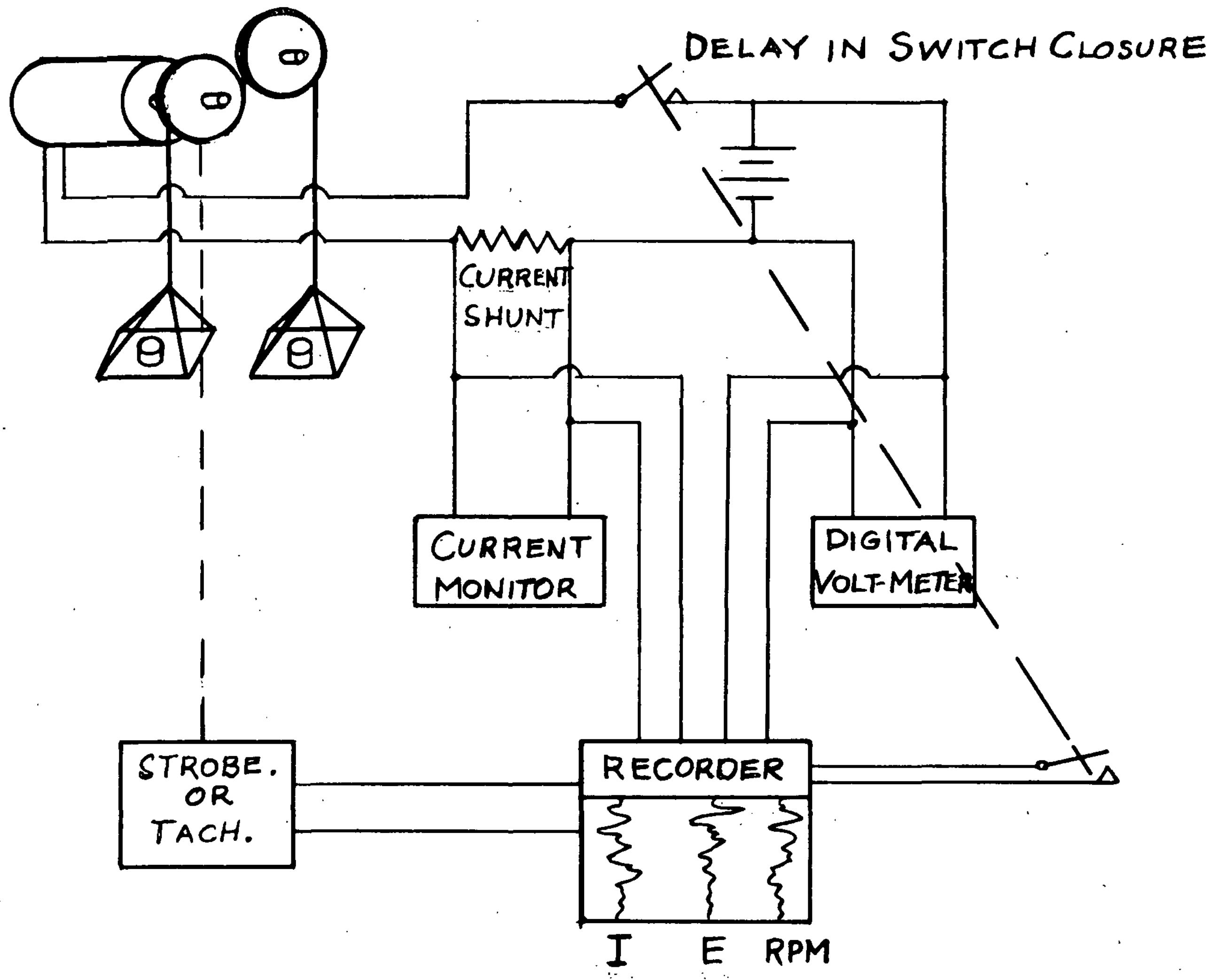
Apply power until motor stalls, making a continuous record of all parameters, including BOTH load cells.

5.2.4 Test Instrumentation

5.2.4.1 Torque Measurement

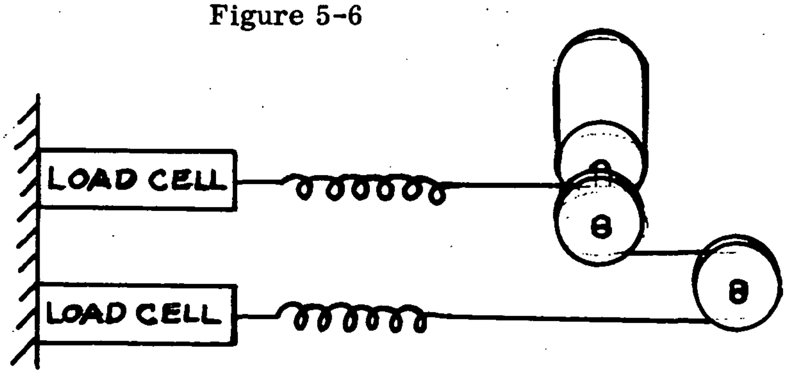
Torque sensing for the bearing and flex cable test and for the bearings and flex cable in the motor torque test will be accomplished via a capacitive transducer as illustrated in Figure 5-8.

The torque output of the motor will be transduced by measuring the angular acceleration of the rotor. The scheme recommended is to differentiate



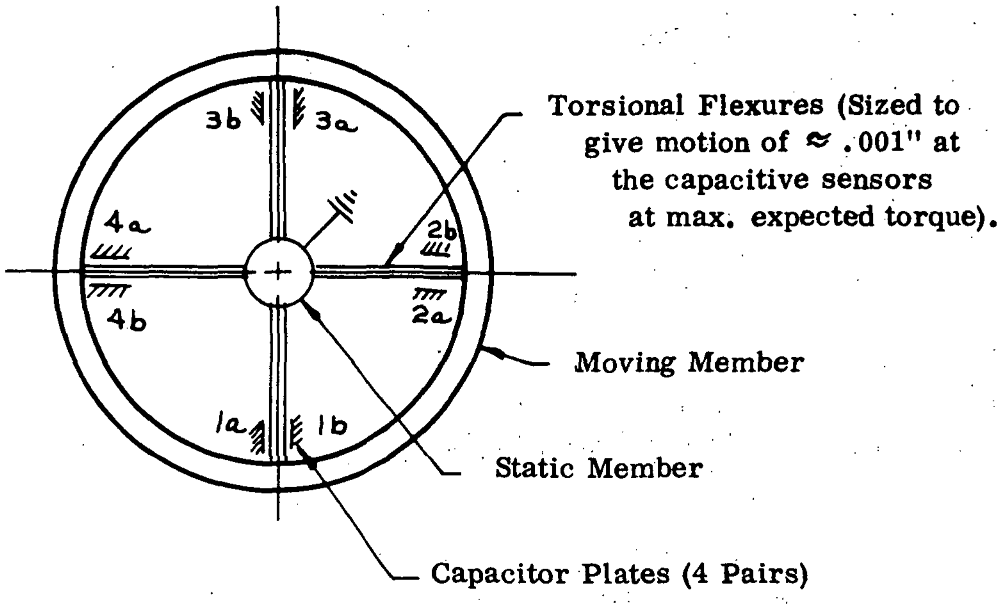
Motor Calibration

Figure 5-6

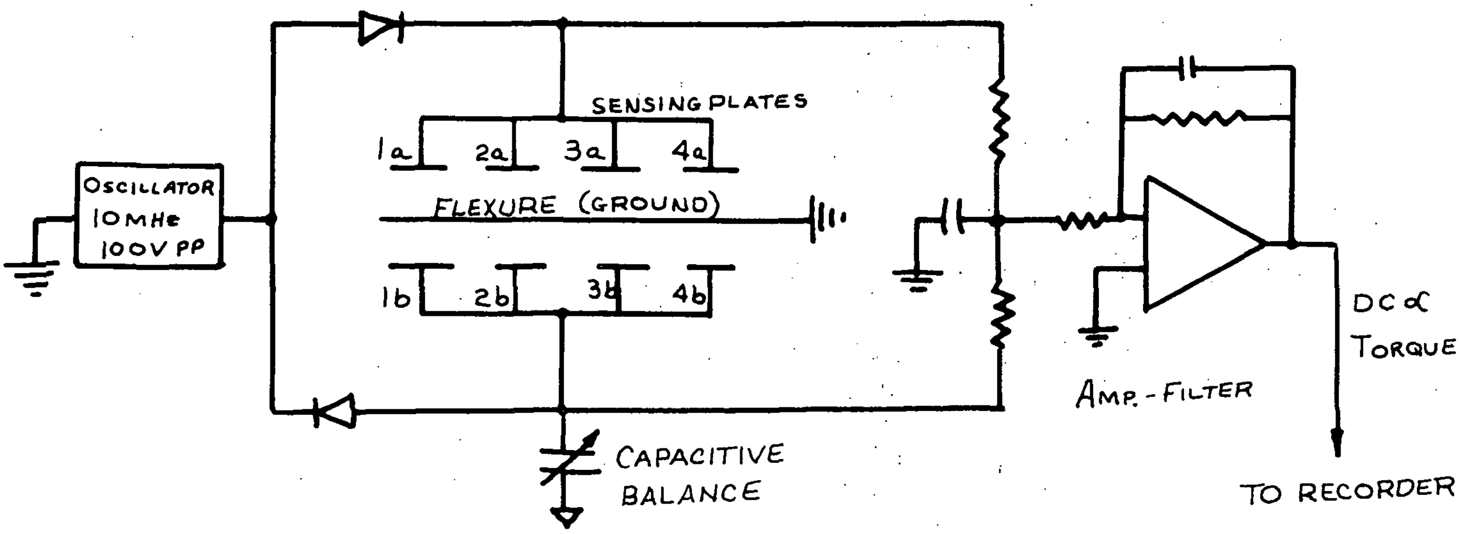


Alternate Motor Calibration

Figure 5-7



Capacitive Transducer Circuit (From Ref. 1)



Expected Resolution - .001 x Maximum Expected Torque.

Figure 5-8

Ref. 1 Folduari, T. L. and Lion, K. S.; Capacitive Transducers, Instruments & Control Systems, Nov. 1964, pp 77-85.

the position transducer (resolver) output twice to obtain the required acceleration signal.

The instrumentation block diagram, which includes relating the torque to the current is as follows

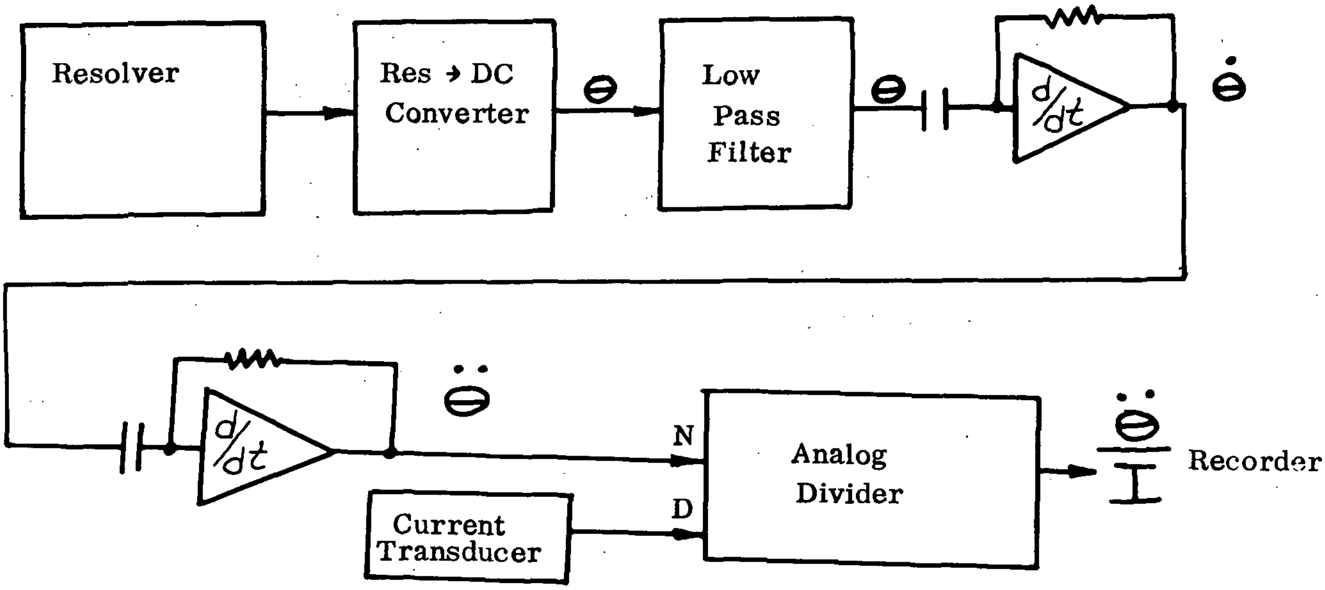


Figure 5-9

Motor Torque Measurement Diagram

5.2.4.2 Incipient Failure Detection.

Acoustic emission (ultrasonic stress wave emission) may be used to advantage in both the bearing test and the cabling test. In essence, the point of the method is to detect and quantify the ultrasonic noise produced by a test item during cycling. As mechanical failure is approached the noise level will generally rise considerably long before eminent failure can be detected by other means. In general, both the level and spectral content will change.

For this test series, it is felt that a very simple approach to analyses is applicable. This is to count the number of times a set stress wave level is exceeded during a cycle of the system. The block diagram for such an instrumentation system is:

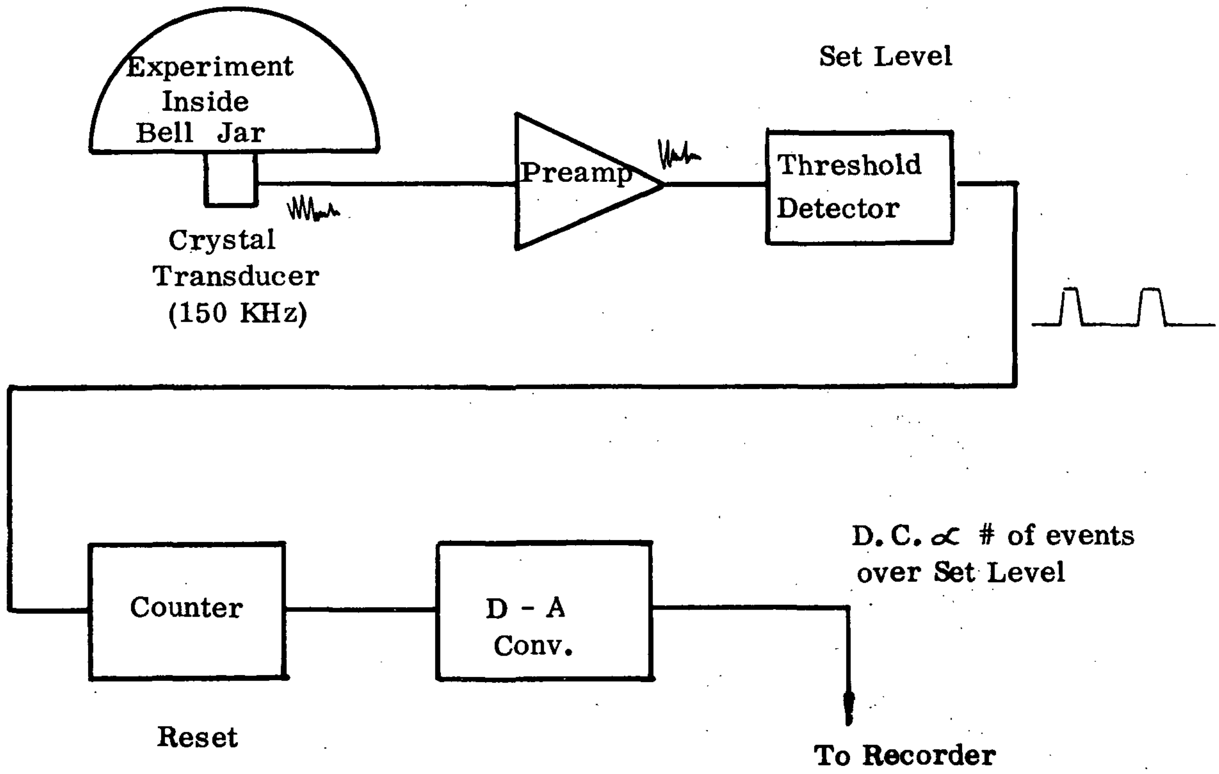


Figure 5-10

Acoustic Emission Instrumentation

By using different sources for the reset pulse the following may be determined.

1) Number of events per 1 or N loading cycles
2) Number of events per portion of the cycle
3) Number of events per unit time.

It is probable that (1) is the most useful and straightforward measurement, but the others will be tried as well.

An alternate method to incipient failure detection in the flex cable is time domain reflectometry (TDR). This technique, used by telephone companies to detect line anomalies and determine their position, measures the reflected energy of an impressed signal and also measures the time for the signal to be reflected. In this manner, the anomaly can be detected, located, and analyzed as to type, i.e., resistive, capacitive or inductive. A breadboard test was set up to check out the concept using a standard TDR and flat flexible cabling. The results of this test proved the feasibility.

Incipient failure detection for the motor will be handled in the following manner.

Plot. (a) Starting current peaks vs torque load, (b) Steady state current vs torque load, and (c) Torque vs voltage, current, RPM, peak start current.

At the failure level, the volt sensor will trip, giving a warning of incipient failure (torque requirements increasing to the point of motor stalling). Volt sensor (Figure 5-12) will only trip if time interval "B" (see Figure 5-11) is present. A lesser time interval will not cause a trip - no warning.

The time intervals (at the failure level) will effectively prevent pseudo-warnings caused by voltage spikes of the same level as the failure level, provided that these spikes are of less duration than "B".

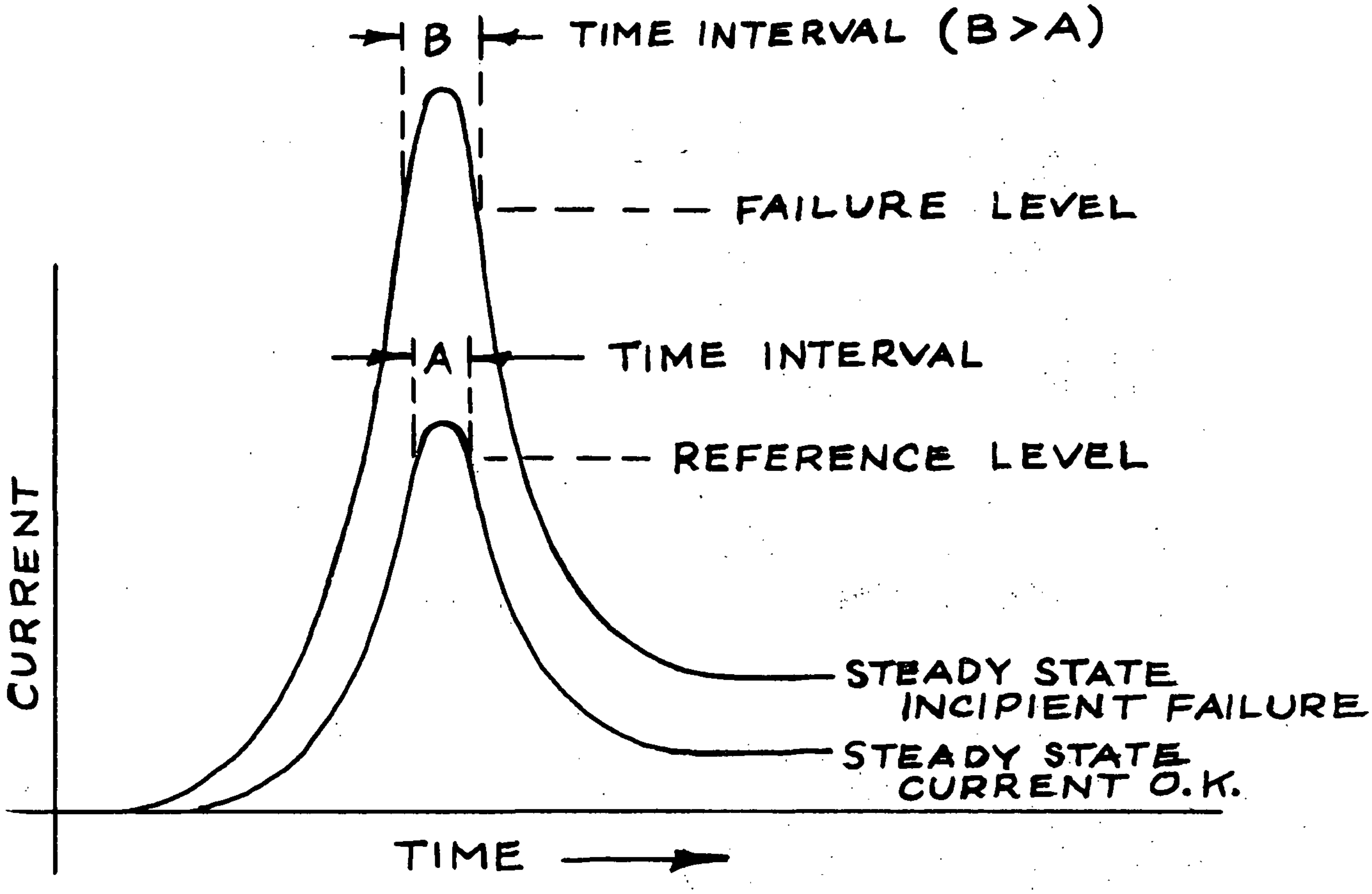


Figure 5-11

Torque Motor Incipient Failure Detection

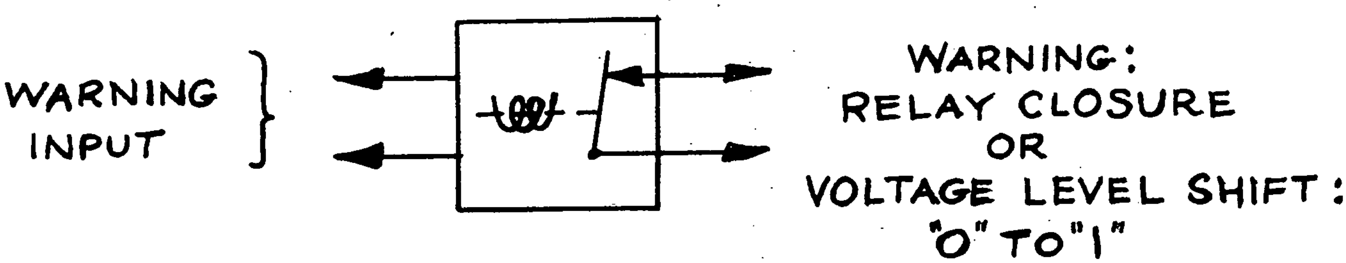


Figure 5-12

Volt Sensor with Time Delay

5.2.4.3 <u>Thermal Cycling.</u> The effect of thermal cycling in accelerated long life testing will be a major factor contributing to potential failures. Two cyclical variations in the orbital environment can be identified.

- Fluctuations in each orbit period due to the spacecraft passing in and out of the earth's shadow.
- Long term environmental changes associated with changes in the orbit plane (solar incidence beta angle).

Since the components in this particular design are enclosed within an outer housing and have a moderate thermal mass, temperature excursions, due to the orbital period are expected to be small with proper selection of thermal control surfaces and/or insulation. Temperature excursions due to beta angle shift (seasonal changes) are caused by a combination of the precession of the orbit plane about the earth's axis and the earth's motion about the sun.

A computer run was made to determine the 10 year beta angle history for the orbit under consideration (55 degrees inclination, 250 nautical mile circular orbit). Figure 5-13 illustrates the beta angle variation for a typical one-year period. The following years will closely resemble this pattern. A 63 day cycle is superimposed on a 365 day cycle. The total range of the beta angle is $\pm$ 76 degrees. The gimbal components are expected to attain their coldest temperatures when the beta angle is approximately 0 degrees since about 40 percent of the orbit is in the earth's shadow (see Figure 5-14). The beta angles near $\pm$ 76 degrees will result in higher temperature levels due to the fact that there will be less shadowing by the earth.

For purposes of this test, the thermal cycling will be consistent with the beta angle cycles. Each peak on the beta curve will be regarded as a hot condition with the cold condition occurring at beta = 0.

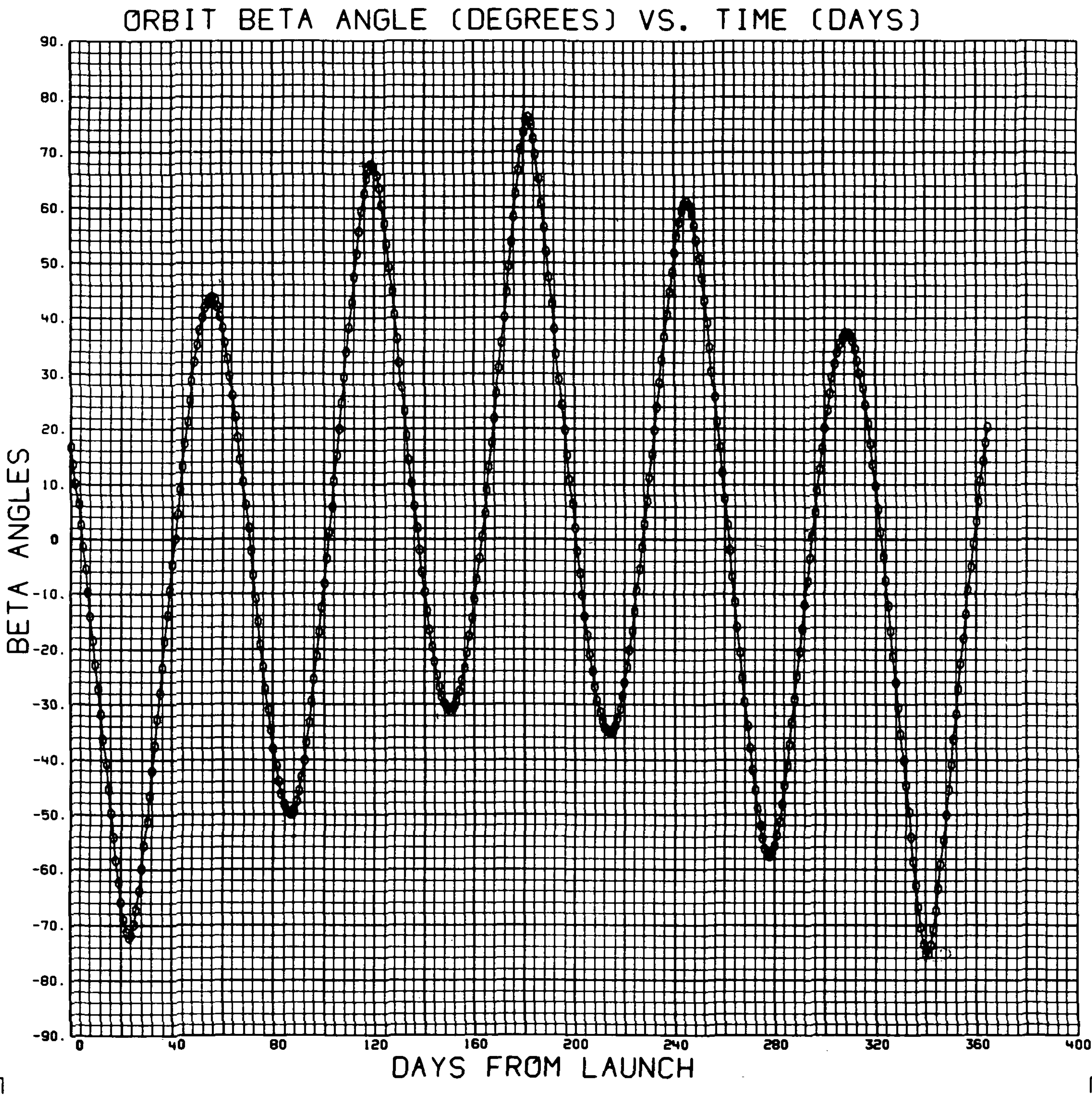

ORBIT BETA ANGLE (DEGREES) VS. TIME (DAYS)
BETA ANGLES
DAYS FROM LAUNCH
90.
80.
70.
60.
50.
40.
30.
20.
10.
0
-10.
-20.
-30.
-40.
-50.
-60.
-70.
-80.
-90.
0
40
80
120
160
200
240
280
320
360
400

Figure 5-13

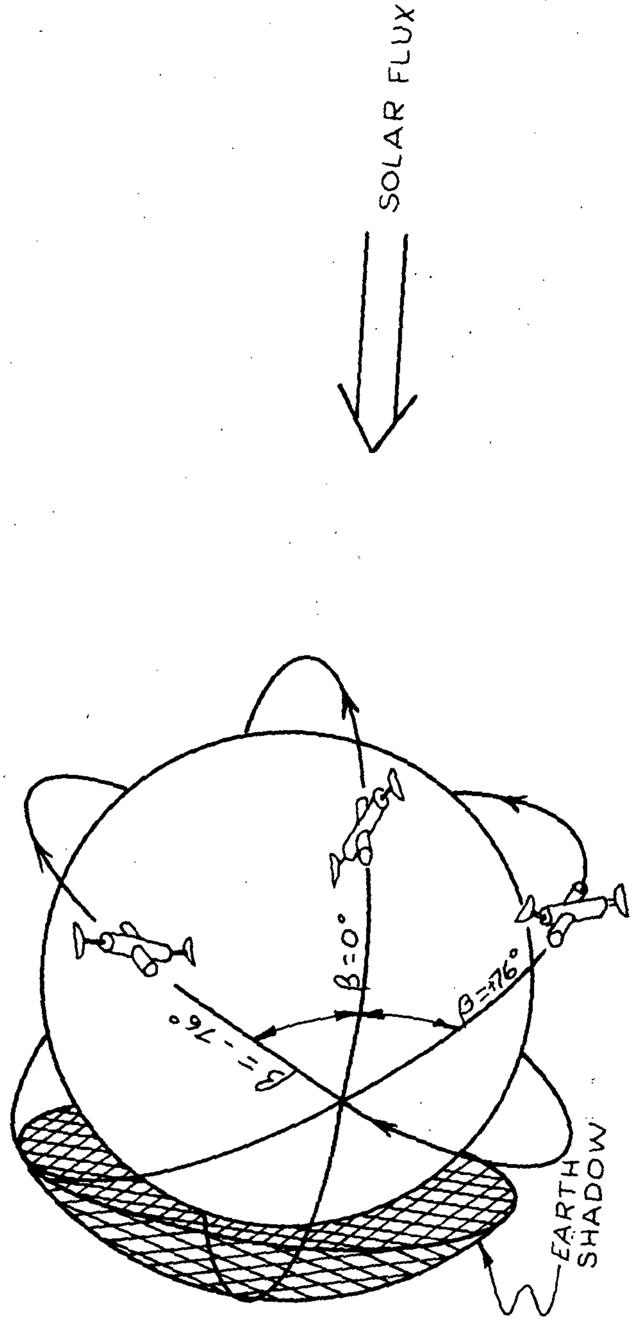


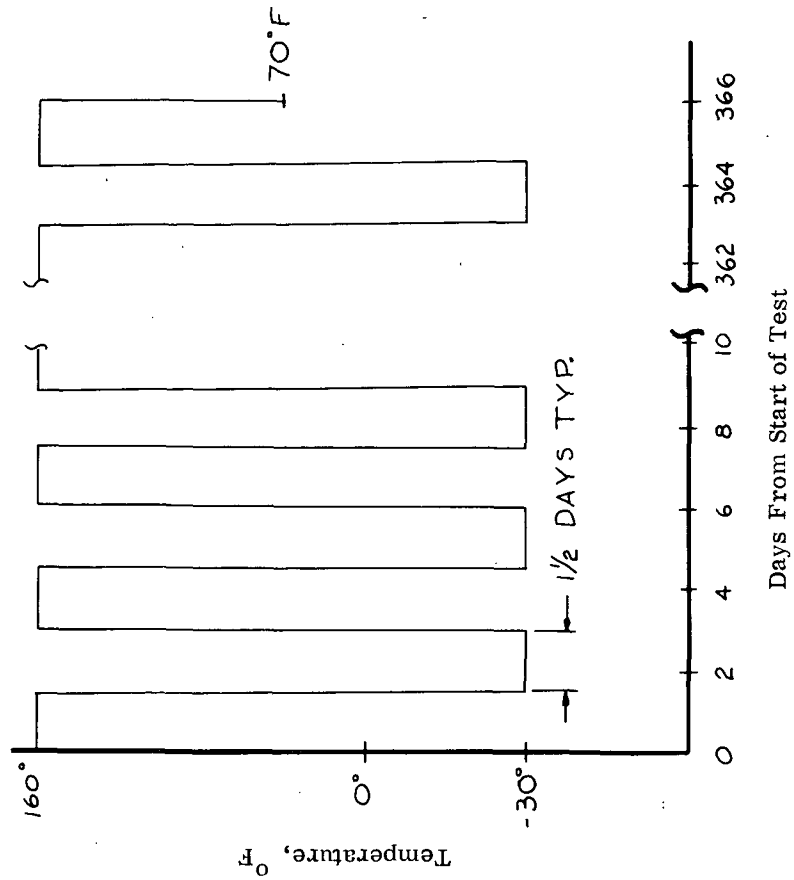
Figure 5-14

This approach is more conservative than the actual case since not all beta angle peaks are equal in magnitude to the maximum.

Compressing the 10 year life into one year requires a complete thermal cycle every 76 hours. By reducing this to 72 hours not only do we increase the total number of cycles the units will see but monitoring of the test will become much simpler in that each cycle will be completed at the same time of day, thus staggered work shifts will not be necessary.

The temperature levels shown in Figure 5-15, based on experience in testing for similar orbital conditions on previous programs, will be -30°F to +160°F. Conventional thermal control surfaces are adequate to maintain the temperatures within these limits.

Figure 5-4 and 5-5 show the position and size of the heat exchangers that will be used. The bell jars will be mounted on insulating blocks to increase the thermal efficiency of the test. The high temperature will be achieved using electrical heaters. Liquid nitrogen will be pumped through the shroud for the low temperature condition.



Component Temperature Cycles
Figure 5-15

Section 6

RECOMMENDED PROJECTS

Throughout the study, there were specific areas of technology for which there was little information, additional hardware development is required, or changes in current techniques were necessary. This section discusses some of these areas and recommends the type effort required.

6.1 BEARING CONTAMINATION CONTROL

The technology evaluation phase of this study uncovered a number of potential problem areas which were directly related to the manufacturing and quality control processes currently in use in the industry. In certain cases, even though stringent quality control measures were used and detail instructions were provided, failures had occurred which were subsequently traced back to the vendor processes.

In a recent program at LMSC, bearing failures began to occur in a tape recorder designed for space applications. The bearings were removed from the recorders and investigated. The results of this investigation were literally astonishing. Out of 112 bearings inspected the following observations were noted:

- Varnish was present in 24.1%
- Contamination was present in 9.8%
- Frosty appearance 31.2%
- Inadequate lubricant 39.3%
- Wrong lubricant 5.35%

Only 60.7% of the bearings contained adequate lubrication. It should be pointed out that the indication of one symptom did not necessarily exclude the occurrences of other symptoms.

On another program, which was concerned with development of an acoustical emission detection device to be used on rotating devices, similar results were found. Five bearings were ordered to help check out the device once it was assembled. All five of the bearings were found to be contaminated as received from the vendor. One of the bearings was found to have a scratch in the race. These anomalies were detected with the acoustical technique Mechanical Signature Analysis.

It is obvious from these examples that current quality assurance techniques are not doing what the name implies and considerable effort should be expended to rectify the situation.

One method that would certainly improve the quality of bearings as received from the vendor would be to apply the Mechanical Signature Analysis (MSA) technique at the vendor's location just prior to packaging for shipping. The bearings would then be hermetically sealed and placed in a frangible container for shipping. Upon receipt, the containers would be inspected for damage to ensure they were not broken, indicating excessive shock or vibration during shipment. The bearings would be stored in this container until they were to be installed. At that point MSA would again be applied both to the bearings before installation and to the assembly after installation.

It is recommended that a trial run be initiated using this format to evaluate its feasibility.

The program might include ordering two lots of bearings. Subject one lot of these bearings to the MSA/frangible container sequence and allow the other lot to follow currently used processes. Each lot would then be subjected to a high speed ($>$ 5000 RPM) test to failure and the results evaluated.

With a program of this type, the effects of contamination should become quite evident and the overall confidence of the system raised significantly.

6.2 ACCELERATED TESTING SCALE FACTORS

In determining the scale factor for the accelerated tests described in Section 5, certain intuitive judgements were made. There was no analytical tool found in the literature which could be applied to these particular tests. An attempt was made at one point to use energy methods, i.e., a Bernoulli energy equation, to predict the expected life of a component.

The approach was similar to that used in the nuclear testing discipline. The assumptions made there are that nuclear radiation energy dosage can be increased to determine the long term effects of a lower dosage. The analogy for a mechanical system undergoing an accelerated life test seemed at least plausible. Equations for bearings were derived,taking into account such things as friction, ball/race deflection, and kinetic energy passing across the ball/race interface. The losses, ultimately resulting in thermal energy radiated to space, were assumed to be a measure of the deterioration rate of the bearings. For the case of the bearings on the antenna gimbal, the losses as calculated were so low (less than 1 BTU for the full 10 year period) the technique was abandoned as not being applicable.

Even though this approach ended seemingly non-fruitful, it is strongly recommended that additional effort be expended in this area to arrive at an analytical rather than intuitive justification for accelerated testing rates.

6.3 DYNAMIC SEALS

There is no seal available which will provide a perfect or near perfect seal around rotating shafts over a large temperature range or for extended periods of time. While the design concept proposed under this study does not contain a seal, other than the normal labyrinth effect due to bearing shielding and small shaft/housing clearances, it is recognized that some rotating seals will probably be necessary on the space station for other systems.

There are two seals which appear to have the potential capability required; the magnetic fluid seal developed by Ferrofluidics Corp., and the bellows flex seal.

The bellows flex seal is fabricated by electroforming an extremely thin (0.0005 in.) layer of stainless steel. With proper linkages, rotary motion is changed to oscillatory motion at the seal and back to rotary motion on the opposite side of the seal. The bellows itself is sealed with conventional methods such as dip brazing or electron beam welding. The main disadvantage to this type seal is the limited number of mechanical cycles to which the bellows can be subjected — usually less than 2000.

The other seal which shows promise, the magnetic fluid seal, uses extremely finely ground magnetic particles (100 Å) in suspension in a carrier fluid. This fluid is then placed in a magnetized labyrinth arrangement thus forming a wall through which molecular migration cannot occur.

The disadvantage of this seal is that the carrier fluid must be tailored to fit the particular temperature range at which it will be used and this temperature must be controlled. The carrier fluid tends to evaporate, leaving behind a higher viscosity suspension. This effect is quite gradual and could probably be accounted for in the initial design if proper information is available early in the design phase.

Each of these seals has the potential to create an almost zero leak rate when their particular problems are solved. It is therefore recommended, because of the 10 year life requirement on the space station, that a development program be initiated to advance the state-of-the-art of these rotary seals.

6.4 TESTING RECOMMENDED TO PROVIDE FALL BACK POSITION

6.4.1 Test Objectives

Section 4 of this report identifies an optimum gimbal design concept, based upon the best knowledge now available relative to its mission and performance requirements. Assumptions have been made in a number of areas where detail information will not be available until after the life testing is done. For example, the requirements for gimbal stiffness, compliance, pointing accuracy, and drive torque may be different than we now think that they will be when analyses are done in these areas. It may be necessary at that time to modify what now looks like an optimum design choice.

With this thought in mind, a test plan has been prepared that includes the critical components of the selected optimum design and also provides testing of several alternate design features that stand a reasonable possibility of being needed during the final hardware design phase.

The test plan recognizes the fact that there is some risk that the preferred design and components will fail during life testing. Alternate designs and materials are recommended for life testing where this risk is significant.

It is also recognized that the life testing must be performed on an accelerated schedule. The recommended tests compress 10 years of operational life into 12 months of testing. Where possible to do so, testing is provided beyond the equivalent of 10 years of life to determine the wear-out failure point.

Five separate tests are included in the plan. It is recommended that all be performed. Some of the tests can be eliminated in the interest of economics, with a reduction in test information and in life-proven components and materials to select from when the detail gimbal hardware design is performed.

6.4.2 Life Test Candidates

Table 6-1 lists the components that are recommended for life testing. This includes not only the basic components and materials that would be used in the optimum design as discussed in Section 4, but also the various alternate components and materials that should be tested as discussed above in Para. 6.4. Index letters "A" through "U" have been assigned for convenience in identifying the various test items.

The actual hardware design for the alternate tests would be similar to those in Section 5. Figure 6-1, however, is a schematic representation of how the alternates might be tested in order to arrive at additional information.

6.5 NEW DEVELOPMENTS

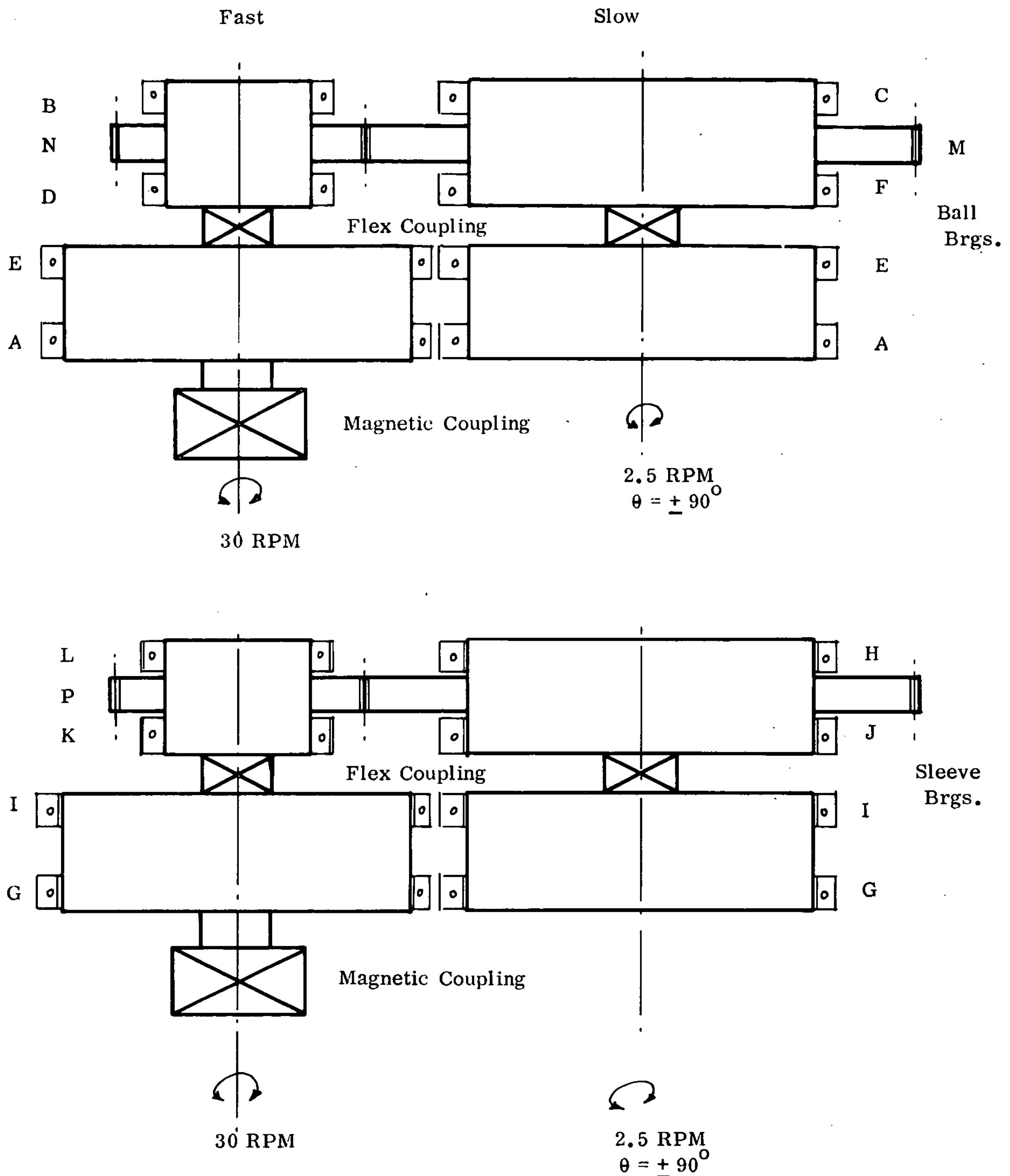
Two particular new concepts show promise in increasing the life expectancy of mechanical drives.

One of these concepts is a new gear tooth form. According to the vendor, Rolling Contact Gear Co., Santa Rosa, California, gears with the tooth form per their patented system transmit power with no sliding. In tests performed at their facility, a 20:1 service life edge over involute tooth form gears was observed. A 40% weight/power advantage was also reported.

The other concept which deserves development effort was a direct result of the efforts on this study. It has been estimated that 50% of all bearing failures are a result of retainer wear-out. This wear is a result of sliding contact of the balls on the retainer. The proposed concept eliminates this failure mode by providing idler elements (hourglass shaped rollers) as separators. Each of the idlers, fabricated from MoS_2 impregnated polyimide, would run on its own shaft and provide a rolling rather than sliding contact on the balls. Lubricant replenishment for the balls would be provided in much the same manner as is used in gearing where an idler gear is provided. With this device, a significant improvement in the bearing reliability should result.

TABLE 6-1 LIFE TEST CANDIDATES

Item	Description	Size (Approx)	Lubricant	Supplier (Tentative)	Basic or Alternate Candidate	Critical Failure Modes
A	Single Row Ball Bearing	2.500 ID x 3.250 OD x .375	Sb_2O_3/MoS_2	Open	Basic	Mechanical Wear
B	Single Row Ball Bearing	0.500 ID x 1.000 OD x .250	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
C	Single Row Ball Bearing	2.500 ID x 3.250 OD x .375	Bar-Temp	Barden Corp.	Alt.	Mechanical Wear
D	Single Row Ball Bearing	0.500 ID x 1.000 OD x .250	Bar-Temp	Barden Corp.	Alt.	Mechanical Wear
E	Duplex Ball Bearing	2.500 ID x 3.250 OD x .375	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
F	Duplex Ball Bearing	2.500 ID x 3.250 OD x .375	Bar-Temp	Barden Corp.	Alt.	Mechanical Wear
G	Sleeve Bearing (for use w/Items A or C)	-	Sb_2O_3/MoS_2	Open	Basic	Mechanical Wear
H	Sleeve Bearing (for use w/Items A or C)	-	Rulon A	Dixon Corp.	Alt.	Mechanical Wear
I	Sleeve Bearing (for use w/Items E or F)	-	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
J	Sleeve Bearing (for use w/Items E or F)	-	Rulon A	Dixon Corp.	Alt.	Mechanical Wear
K	Sleeve Bearing (for use w/Items B or D)	-	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
L	Sleeve Bearing (for use w/Items B or D)	-	Rulon A	Dixon Corp.	Alt.	Mechanical Wear
M	Bull Gear, 32 DP, 216T	6.750 PD x .625 W	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
N	Pinion Gear, 32 DP, 18 T	.562 PD x .750 W	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
O	Bull Gear, 32 DP, 216T	6.750 PD x .625 W	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
P	Pinion Gear, 32 DP, 18T	.562 PD x .750 W	Sb_2O_3/MoS_2	Open	Alt.	Mechanical Wear
Q	Torque Motor - Brushless D.C.				Basic	Thermal Cycling; Change in Magnet Properties
R	Commutation Signal Generator				Basic	Thermal Cycling; Change in Magnet Properties
S	Resolver - Brushless				Basic	Thermal Cycling; Change in Magnet Properties
T	Flexible Cable			LMSC	Basic	Fatigue Due To Thermal And/Or Mechanical Cycling



Examples of Bearing/Gear Test Combinations

Figure 6-1

Section 7

BIBLIOGRAPHY

LMSC DOCUMENTS

1. Ball Bearing Analysis -- Type 29 Tape Recorder

2. Evaluation of Space Station Solar Array Technology, LMSC A981486.

3. Gears, Bearings and Lubricants for Aerospace Applications.

4. Lubricants and Self-Lubricating Materials for Spacecraft Mechanisms, LMSD 894812.

5. Lubrication Evaluation MRI 503.01.

6. Lubrication Evaluation MRI 503.02.

7. Space Materials Handbook, 1965, Lockheed Missiles & Space Company, Inc., Sunnyvale, California.

8. Standard Electronic Parts Configuration Analysis.

9. Survey of Flat Conductor Cable Technology, LMSC A951795B, December 19, 1969, Lockheed Missiles & Space Company, Inc., Sunnyvale, California.

10. Test Evaluation Plan For The Space Station Solar Array Technology Evaluation Program, Contract NAS9-11039, 1971, LMSC A984133.

ANTENNA DRIVE MARKET SURVEY

1. Aeroflex Labs, Incorporated
   Torque Motor Products Division
   South Service Road
   Plainview, New York 11803
   Attn: J. H. Lewis

2. American Electronics, Inc.
   Control Components Div.
   1600 E. Valencia Street
   Fullerton, Calif.
   Attn: Sales Manager

3. Beckman Instruments, Inc.
   2500 Harbor Blvd.
   Fullerton, Calif. 92634
   Attn: Sales Manager

4. Bowmar Instrument Corp.
   8000 Bluffton Road
   Fort Wayne, Indiana 46809
   Attn: C. Jackson

5. Cedar Division
   Control Data Corp.
   5806 West 36th St.
   Minneapolis, Minn. 55416
   Attn: C. Teichert

6. Clifton Division
   Litton Precision Products, Inc.
   5050 State Road
   Drexel Hill, Pa.

7. Enercon Motors
   Hickory Street
   Grafton, Wisconsin
   Attn: Sales Manager

8. General Electric Company
   ESCO 1 River Rd.
   Schenectady, New York 12305
   Attn: Sales Manager

9. IMC Magnetics Corp.
   570 Main St.
   Westbury, New York 11590
   Attn: G. Egan

10. Inland Motor Corp. of Virginia
    501 First St.
    Radford, Va. 24141
    Attn: L. Davis

11. Killsman Motor Corp.
    Mill Street
    Dublin, Pa. 18917
    Attn: J. Kendrick

12. Magnetic Technology
    Div. of Vernitron
    21001 Kittridge St.
    Canoga Park, Calif. 91303
    Attn: W. G. Osborn

13. MPC Products Corp.
    4200 Victoria Avenue
    Chicago, Illinois 60646
    Attn: Sales Manager

14. Singer-General Precision, Inc.
    Kearfoot Division
    1150 McBride Avenue
    Little Falls, New Jersey 07424
    Attn: W. Quigley

15. Superior Electric Co.
1000 Middle Street
Bristol, Conn. 06010
Attn: Sales Manager

16. United States Shoe Corp.
1658 Herald Ave.
Cincinnati, Ohio 45212
Attn: Sales Manager

17. Uriel Corporation
53 Union Avenue
Ronkonkoma, New York 11779
Attn: M. Kasindorf

18. Vernitron Corp.
Control Components Division
2440 W. Carson St.
Torrance, Calif. 90501
Attn: S. W. Silverman

BOOKS

1. AGMA Standard 110.03, Nomenclature of Gear Tooth Wear and Failure, BS 12-1964.

2. D. W. Dudley, Gear Handbook, McGraw-Hill Book Co., New York, 1962.

3. D. W. Dudley, Practical Gear Design, McGraw-Hill Book Co., New York, 1954.

4. E. E. Bisson & W. J. Anderson, Advanced Bearing Technology, National Aeronautics & Space Administration, Washington, D. C., 1964.

5. S. P. Timoshenko & J. N. Goodier, Theory of Elasticity, McGraw-Hill Book Co., New York.

TECHNICAL PUBLICATIONS

1. G. A. Bernier and D. E. Kline, Dynamic Mechanical Behavior of a Polyimide, Journal of Applied Science, 1968.

2. T. G. Bryant & J. A. Weiss, Parameters of Microstrip Transmission Lines and of Coupled Pairs of Microstrip Lines, IEEE Transactions of Microwave Theory and Techniques, Vol. MTT-16, No. 12, December 1968.

3. E. R. Bunker, Jr., Voltage Breakdown in Electronic Equipment at Low Air Pressures, Jet Propulsion Laboratory, Dec. 15, 1966.

4. M. Campbell, et al, Solid Lubricant Handbook for Use in the Space Industry, Midwest Research Institute, 1971.

5. J. R. Carden, Flat Conductor Cable for Limited Rotary or Linear Motion, NASA TM-X-53960, NASA/George C. Marshall Space Flight Center, Alabama.

6. W. Coleman, NASA SP-237 - Interdisciplinary Approach to the Lubrication of Concentrated Contacts, July 15, 1969.

7. R. F. Crawford, Final Report - Lunar Antenna Mast-Engineering Model, Astro Research Corporation, June 1970.

8. J. L. Frarey, Development of Gear Failure Detection Methods, Mechanical Technology Inc., Latham, N. Y., Dec. 1970.

9. W. E. Fromm, E. G. Fubini & H. S. Keen, A New Microwave Rotary Joint, Airborne Instruments Laboratory, Inc., Mineola, New York.

10. J. A. Gatlin, A. G. Buckingham & W. H. Pleasants, Satellite Attitude Control Using a Torqued, 2-Axis Gimballed Boom as the Actuator, Spacecraft Journal, Vol. 6, No. 9, September 1969.

11. R. W. Gibb, et al, Flat Conductor Cable Termination Development Program, Final Report, North American Rockwell Corp., Sept. 15, 1970.

12. A. R. Giddis & T. P. Jordan, Comparative Evaluation of Mechanically and Electronically Despun Antennas for Spacecraft, International Astronautical Congress, Vol. 1, XVIII, 1967.

13. M. Granzow, et al., Development of Mechanical Antenna Despin System, Dornier-Werke G.m.b.lt., Germany, Feb. 1970.

14. H. T. Haynie, Rotary Relay for Space Power Transfer, The Boeing Company, Seattle, Washington, Presented at the 5th Aerospace Mechanism Symposium, June 1970.

15. E. Heer & J. M. Yang, Optimization of Space Antenna Structures, NASA Technical Memorandum 33-472, Jet Propulsion Laboratory, Pasadena, California, April 1, 1971.

16. J. P. Hirschfeld & B. Irrgang, The Measurement of Various Voltages in Vibration-Stressed Coaxial Cables, Naval Research Laboratory, NRL Translation No. 1097, January 1968.

17. V. Hopkins, R. D. Hubbell, M. T. Lavik, Improved High Temperature Solid Film Lubricants, Midwest Research Institute, Kansas City, Mo., April 1971.

18. E. S. Keats, Preliminary Investigation of Navigation Satellite Techniques for Achieving Accuracies in Position of the Order of Ten Metres, Westinghouse Electric Corporation, Baltimore, Maryland.

19. W. G. King, Fabrication Feasibility Study of a 30 Watt/Pound Roll-Up Solar Array, Quarterly Report, 27 June - 30 Sept. 1967, Fairchild Hiller Corp., Oct. 15, 1967.

20. L. L. Lewis, "Kapton" Polyimide Film, A Thin Wall High-Temperature Insulation for Wire and Cable, E. I. DuPont De Nemours & Co., (Inc.), Wilmington, Delaware, December 1965.

21. E. L. Marsh, The Attitude Control of a Flexible Solar Electric Spacecraft, Jet Propulsion Laboratory, Pasadena, California, Presented at the 8th AIAA Electric Propulsion Conference, August 31-September 2, 1970.

22. H. E. Newell & J. L. Sloop, Planning Space Technology for the 1970's, National Aeronautics and Space Administration, Presented to the National Security Industrial Association, Washington, D. C., June 11, 1969.

23. A. Owyoung, Communications Elements Research, JPL, Space Programs Summary 37-59, Vol, III.

24. A. J. Parks, et al, Space Environment Seals Test Program, Final Report, Pratt and Whitney Aircraft, Dec. 1965.

25. M. B. Peterson, Literature Search and Critical Review of Mechanical Failure Technology, Mechanical Technology, Incorporated, Report No. MTI 70TR48, July 1970.

26. T. Rees, Design Report to ESRO for a Despin Assembly, Hawker Siddeley Dynamics, Hatfield, England, October 1970.

27. E. E. Sabelman, Advanced Projects Development, JPL Space Programs Summary 37-60, Vol. III.

28. J. F. Saraceno, Nimbus Solar Array Drive Subsystem, Final Report, TRW Systems, Feb. 6, 1970.

29. J. Savides, Antenna Pattern Shaping, Sensing and Steering Study, Final Report, Dec. 1968-Nov. 1969, Philco-Ford Corp., July 9, 1970.

30. D. K. Snediker, et al., The Development and Evaluation of Lubricating Systems for Reusable Space Vehicles, Quarterly Report, 1 Oct.-31 Dec. 1970, General Electric Co.

31. H. S. Swan & J. M. Davis, Problems with Electrical Connectors and Insulated Wire in the Aerospace Industry, General Electric Company, Daytona Beach, Florida.

32. C. L. Tai & M.M.H. Loh, Planar Motion of a Rotating Cable-Connected Space Station in Orbit, Spacecraft Journal, Vol. 2, No. 6, Nov.-Dec. 1965.

33. A. I. Tucker, Gear Design:Dynamic Loads, Mechanical Engineering, Vol. 93, No. 10, October 1971.

34. R. D. Wagner, Electrical Connection Study, The Bendix Corporation, Kansas City, Missouri, September 1970.

35. G. C. Winston, R. B. Tatge & J. J. Lombardo, Diagnostic Procedures for Antenna Hydraulic Drive Systems, Goddard Space Flight Center, Greenbelt, Maryland, September 1969.

36. F. Yeaple, Metal-to-Metal Traction Drives Now Have a New Lease on Life, Product Engineering, Vol. 42, No. 15, October 1971.

37. A Despin Assembly for the TACOMSAT Communications Satellite, 4th Aerospace Mechanisms Symposium.

38. Advanced Project Development; JPL Space Program Summary.

39. Back-Up Bearing Takes Over Alarmingly; Machine Design, Sept. 2, 1971.

40. Definition Study of an Antenna Despin Mechanism, Final Report, Dornier-Werke, Gmbh, W. Germany, Nov. 1970.

41. Flat Cable Applications Engineering Study Phase 3, Volume I, Final Report, Douglas Aircraft Company, Inc., July 1968.

42. Friction and Wear; IEEE Transactions on Parts, Hybrids, and Packaging, Vol. PH, p. 7, No. 2, June 1971.

43. Friction, Wear, and Evaporation Rates of Various Materials in Vacuum to 10^{-7} mmHg; ASLE Transactions 5,8-23 (1962).

44. High Temperature Ball Bearings for Operation in a Vacuum; Vestn. Mashinostr, N.2, Feb. 1970, pg. 11-13, in Russian.

45. Large Space Structure Experiments for AAP, Vol 1: Summary, Final Report, 15 Sept. 1966-15 Sept. 1967, General Dynamics/Convair, Nov. 30, 1967.

46. Lubricants for Space; Mechanical Engineering, April 1970.

47. Machine Design, 1970 Bearings Reference Issue, June 18, 1970.

48. Positive Deployable Solar Array Development Program Phase 3, Final Report, Fairchild Hiller Corp., July 1968.

49. Precision Lubricant Developed by Navy Offers Commercial Promise; Aerospace Daily, July 30, 1971.

50. Rotary Relay for Space Power Transfer; 5th Aerospace Mechanism, June 1970.

51. Space Transportation System Technology Symposium, Vol. 3: Structures and Materials, National Aeronautics & Space Agency, July 1970.

52. The Deep Space Network, Volume 3, Progress Report Mar.-April 1971, Jet Propulsion Laboratory, June 15, 1971.

53. The Solid Lubrication of Ball Bearings Under High Speed-High Load Conditions From -225° To $+1000^{\circ}$F; Lubrication Engineering, May 1971.

54. C. L. Harris, Vacuum Evaluation of Lubricants and Techniques for Space-Exposed Components, Elliott Bros. (London) LTD., Elliott Space and Weapons Research Laboratory, Firmley, Surrey, England.

55. C. R. Meeks, R. I. Christy, A. C. Cunningham, Accelerated Vacuum Testing of Long Life Ball Bearings and Sliprings, Hughes Aircraft Co., Space Systems Division, El Segundo, Calif.

56. C. E. Vest, P. A. Studer, Meeting The Challenge of a 50,000-Hour-Lifetime Requirement, NASA-Goddard Space Flight Center, Greenbelt, Maryland.

57. Friction Wear and Lubrication, M. J. Furey, Industrial and Engineering Chemistry, Vol. 61, No. 3, March 1969, p. 12.

58. Motor & Motor Generators - Miniature, Servo Motor, Vacuum Tests, Apollo Antenna Servo Motor, Rept. No. 532.29.50. 06-FL-02.

59. A. J. Haltner and C. S. Oliver, Effect of Water Vapor on The Friction of Molybdenum Disulfide, I & EC Fundamentals, Vol. 5, No. 3, Aug. 1966, p. 348.

60. J. Gansheimer, Influence of Certain Vapors and Liquids on the Frictional Properties of Molybdenum Disulfide, ASLE Transactions 10, p. 390, 1967.

61. D. Buckley, M. Swikert, R. L. Johnson, Friction Wear and Evaporation Rates of Various Metals in Vacuum to 10^{-7} mmHg., ASLE Transactions 5, p. 8, 1962.

62. Test Plan for Evaluation of the Effect of Vacuum Storage on Torques of Oiled Ball Bearings, Ball Brothers Research Corp., #38024, 1970.

63. Test Program Plan, Major Hardware ODAPT, Ball Brothers Research Corp., #38042, 1970.

64. Slip Rings for High Vacuum Operation, K. R. Zimmerman, Pacific Scientific Co., Anaheim, Calif. 1966.

65. Dependence of Thermal Control Coating Degradation Upon Electron Energy, The Boeing Co., Seattle, Wash. Under Contract to NASA-Goddard Space Flight Center, Contract No. NAS5-11164.

66. R. J. Benson, Which Instrument-Gear Materials Wear Least?, Product Engineering, 1961, p. 588.

67. J. D. Albert, R. H. Ingram, Environmental Problems in External Space Equipment Design, IEEE Spectrum, July 1965, p. 106.

68. E. W. Glossbrenner, Poly-Scientific's Role in Space, Poly-Scientific Division, Litton Precision Products, Inc., 1970.

69. I. Irving Pinkel, Space Shuttle Safety - A Hybrid Vehicle Breeds New Problems, Astronautics & Aeronautics, p. 28, December 1971.

70. Lubrication of D. C. Motors, Sliprings, Bearings, and Gears for Long-Life Space Applications, B. J. Perrin and R. W. Mayer, Technical Memorandum 33-382, 3rd Aerospace Mechanism Symposium, 1968.

71. Evaluation of Dry Lubricants and Bearings for Spacecraft Applications, D. L. Kirkpatrick and W. C. Young, Technical Memorandum 33-382, 3rd Aerospace Mechanism Symposium, 1968.

72. Controlled-Leakage Sealing of Bearings for Fluid Lubrication in a Space Vacuum Environment, H. I. Silversher, Technical Memorandum 33-382, 3rd Aerospace Mechanism Symposium, 1968.

73. Evaluation of a Spacecraft Antenna System, A. Kampinsky, TM 33-425, 4th Aerospace Mechanism Symposium, 1969.

74. The Design of Aerospace Mechanisms - A Customers Opinion, Major James C. McSherry, TM 33-345, 4th Aerospace Mechanism Symposium, 1968.

75. Evaluation of Mechanisms Returned From Surveyor 3, J. R. Jones, W. J. Quinn, K. G. Bingemann, Jr., 6th Aerospace Mechanism Symposium, 1971.

76. Pioneer F/G Feed Movement Mechanism, R. M. Acker, 6th Aerospace Mechanism Symposium, 1971.

77. The Intelsat IV Antenna Positions, Frank A. Glassnow, 6th Aerospace Mechanism Symposium, 1971.

78. Antenna Drive System for the Nimbus Satellite, G. J. Wedlake and J. D. Loudon, 6th Aerospace Mechanism Symposium.

79. Optical Telescope Technology, NASA SP 233, Marshall Space Flight Center, April 29-May 1, 1969.

80. Dynamic Response of Antenna Structures in Launch Environment, A.C.S. Hayden, Canadair Limited, Montreal, Canada, NASA TM X-52876, Vol. II, July 15-17, 1970.

81. James G. Biddle Co., Plymouth Meeting, Pa., Cable Fault Locating Equipment, Catalogue Bulletin 65a.

82. Bearing Technical Journal, FMC Corp., Link Belt Bearing Division, 1970.

83. A Stepper Motor for the Surveyor Spacecraft, Frank A. Glassow, Technical Memorandum 33-355, 2nd Aerospace Mechanism Symposium, 1967.

84. The Behavior of Lubrication System Components in a Vacuum Environment, Donald H. Buckley, Technical Memorandum 33-355, 2nd Aerospace Mechanism Symposium, 1967.

85. Lubrication As A Part of Total Design, Frank J. Clauss, Technical Memorandum 33-355, 2nd Aerospace Mechanism Symposium, 1967.

86. A Brushless Despun Drive and Control for a Communication Satellite Antenna, M. F. Flemming and D. D. Phinney, Technical Memorandum 33-382, 3rd Aerospace Mechanism Symposium, 1968.

87. Flexural Pivots for Space Applications, F. A. Seeling, 3rd Aerospace Mechanism Symposium, Technical Memorandum 33-382, 1968.

88. Product Engineering, April 1972, p. 11, "Compliant Steel Races Improve Linear Bearing Performance".

89. Machine Design, October 14, 1971, p. 136, "Beryllium on the Brink".

90. Machine Design, October 14, 1971, p. 140, "Bearing Orbit Analysis".

91. Machine Design, October 14, 1971, p. 154, "Hot Mesh Plots Bearing Failure".

92. Mechanical Engineering, March 1971, p. 20, "Ball Bearing: Cost VS Value".

93. MIL-P-13949 Plastic Sheet, Laminate D, Copper Clad (For Printed Wiring).

94. Machine Design, July 9, 1970, p. 108, "Lubricating Gears".

95. NASA Spec No. 345.40.55.00-FI-02 Gaskets & Seals - Labyrinth, Metal & Plastic Apollo Program, North American Aviation Inc., No. ME481-0008.

96. Machine Design, Sept. 1971, p. 8 "Sand: New Material for Motor Designers".

97. Design News, Oct. 18, 1971, "Electric Motors".

APPENDIX

K.T.A. SYSTEM OF DECISION MAKING

K.T.A. is best described by going through an example problem step-by-step. This technique was recently used by the Antenna Systems group to determine the optimum computer system for handling digital paper tapes.

The Antenna Systems Organization at LMSC processes a large number of digital paper tapes obtained on digital antenna pattern recording equipment. These tapes must be checked for accuracy and corrected when necessary. In some cases, calculations must be made to obtain derived data for engineering evaluation. Computer options considered were an off-site time-sharing computer system, a future in-house time-sharing computer, a mini-computer configured for the tape processing task, and various combinations of these alternatives. The objective was to find the best system for handling the tape processing work as well as for performing the scientific computations ordinarily programmed by the engineers. Combinations of the mini-computer and the in-house system were found to be best, followed by the mini-computer alone, the in-house system, and then the off-site system. Secondly, the mini-computer significantly improved the capabilities available when added to the in-house system. Most important of all, the analysis showed that adding certain frills to the mini-computer, such as extra core storage and a magnetic tape input/output, did not significantly improve the relative fit to requirements. Thus, a savings of almost 50 percent could be affected by obtaining a mini-computer of minimum capability rather than a more elaborate system. All of this insight was obtained in about two hours time using facts already available and applying systematic thinking to the comparison.

The principles of "decision analysis" can be applied to any tradeoff of alternate concepts. It is beyond the scope of this section to give a thorough description of the KTA "decision analysis" process, but a brief discussion sufficient in detail to depict the general principle will be presented in the following paragraphs.

Step 1. The first step in the decision analysis procedure is to make a decision statement. This states the purpose of the tradeoff analysis. A common format is: *to select an (option) for (purpose).* For example, in the computer case cited above, the decision statement was to select a computer system to handle the tape processing problem in addition to general scientific calculations. The decision statement defines the relevancy of alternatives, requirements, and desired characteristics to be considered in the analysis. For example, "to select a vehicle for personal transportation" would cause us to consider bicycles, motorcycles, and trucks, as well as automobiles. If an automobile is what is desired, the decision statement should be "to select an automobile for personal transportation".

Step 2. The second step is to establish a list of "musts" and "wants". Musts are absolute requirements, such that if an alternative does not satisfy a must, it would be rejected immediately, no matter what other characteristics are offered. For example, a must in the computer case was a standard teletype keyboard I/O device. Another was FORTRAN programming language. Any computer option which did not have a teletype I/O and FORTRAN language was immediately rejected. A want, on the other hand, is a desirable characteristic which can be satisfied to a degree. In the computer example a high speed tape read/punch capability was a want, and tape read speeds from 10 to 500 characters/second were considered.

In comparing antenna concepts gain might be either a must or a want. For example, the specification might require 32.0 dB gain but 31.9 dB might be acceptable if the antenna had other features, such as lower weight. Thus "32.0 dB gain" cannot be a "must" -- it must be a "want". But the gain figure could be set at some lower value, say 29 dB, if the system will not work at a lower gain figure. If "gain of at least 29.0 dB" is a "must" and "high gain" or "32.0 dB gain" is a "want", then any antenna with 28.9 dB gain or less would be immediately rejected without consideration of other characteristics. An antenna having a gain of 31.8 dB satisfies the "want" better than one having a gain of 30.5 dB.

Careful attention to the process of establishing "musts" and "wants" can speed up the analysis process. If "musts" are really musts, many competing concepts can be quickly eliminated without further investigation or consideration, thereby making additional effort available to consider choices which really have some prospect of being suitable. Establishing a comprehensive list of wants will make certain that all relevant aspects of performance are considered and will highlight areas where more information is needed about each of the competing concepts. Thus establishing the list of "musts" and "wants" defines the problem to be analyzed and plans the effort of the study program.

Step 3. Each of the "wants" must be given a relative weighting on a scale of 1 to 10. It is not necessary to weigh the musts, since measuring the competing concepts against the "musts" is a go-no go proposition. The most important "want" is given a weighting of 10. Other wants are compared against the most important one to determine relative weighting. In the computer case above, "high speed tape read capability" was the most important want with a weighting of 10. "High speed tape punch capability" was not considered to be nearly as important and was given a weighting of 5. "Low cost" was the second most important consideration and was rated 8. In essence high speed tape read capability was so important that it could offset a somewhat higher cost.

Step 4. Here all the competing concepts are measured against the "must" list. Any concept failing to satisfy the must is eliminated immediately. Following this step, the list of competing concepts has been narrowed down to a workable size (usually). Each of the remaining concepts has some chance of being the appropriate solution to the problem.

Step 5. In the study phase this step will be the bulk of the effort. Each of the competing concepts will be analyzed to determine how well it offers the desired characteristics in the "want" list. Some performance characteristics about some of the concepts may not be known. In this case, analysis and/or experiment will have to be undertaken to determine how well the competing concept can satisfy the want. It may be necessary to configure several different forms of a single concept to evaluate relative fit if the wants are specified and detailed. For example, three or four different size of reflector antennas might be compared since high gain and low weight will probably be among the wants. Where such parametric tradeoffs occur, a few embodiments of the basic concept will serve to highlight the relative fit to the want list.

When all the required information is obtained, the analysis chart is completed by making a comment on how well each alternative fits the established want.

Step 6. Once the analysis chart is completed, the concepts are scored. For each want the concept offering best performance is given a score of 10, the next best concept receiving a lower score and so on. The scores need not cover the entire range from 10 down to 0, however. If in response to a "low weight" want, one concept has a weight of 150 pounds and others range from 151 to 170 pounds, the scores may all range in the 9 to 10 region if the 20 pound difference is not significant. A concept which weighed 2000 pounds, however, might receive a score of 0.

Then the scores are multiplied by weightings to find the weighted scores. The weighted scores for each alternative are added to determine the total weighted score.

Step 7. Before the alternatives or competing concepts can be compared, the effect of "possible adverse consequences" must be assessed. For each alternative all possible adverse consequences are listed. Each adverse consequence is given a probability from 0 to 10 and a seriousness factor from 0 to 10. These two factors are multiplied and the products for each alternative are then totalled to obtain a risk factor for each considered alternative. In the computer example cited above, committing the tape processing problem to a dedicated mini-computer involved an element of risk in that the mini-computer would be subject to some down time leaving no way to process tapes. The seriousness factor was given a high rating of 9, but the probability of this happening was set at a low figure of 1, based on experience with this type of equipment. When the mini-computer was used in conjunction with other systems, the seriousness factor was somewhat reduced. A typical example of a possible adverse consequence for an antenna concept proposed for future application might be related to projected performance characteristics of future hardware development. For example, for a particular array concept to achieve certain projected performance capabilities may require the development of suitable solid state switching devices. Two possible adverse consequences must be considered: 1) that the devices will not have quite the performance capabilities anticipated and 2) that the devices may not be developed within the time-frame required.

Step 8. The final step is to evaluate the results of the comparison. First, the concepts are ranked in descending order of their total weighted scores. For the computer case the scores were 642, 637, 619, 575, 570, 392, 160. The total possible score was 720. The analysis procedure is not considered accurate enough to distinguish between the first two alternatives (5 points out of 720) and indeed distinguishing between the first five (72 points out of 720) may be open to some question. But clearly the last alternative fails to satisfy the established "wants" and the sixth ranked alternative with a score of 392 is significantly less desirable than the highest ranking alternative. The first five alternatives involved purchase or lease of the mini-computer and in some cases using it with the in-house system. The sixth alternative was the in-house system alone and the last ranked alternative was the off-site time-sharing computer. Thus it was found that obtaining the mini-computer significantly improved the capabilities available, even when the in-house system becomes available.

The final adjustment to the ranking must then be made on the basis of the risk factor score. If a high ranking alternative also has a high risk factor in comparison with other alternatives, this alternative would then be ranked lower than indicated by its total weighted score alone. In the computer case, the off-site computer had a relatively high risk factor, and all other alternatives had low and not significantly different risk factors. Thus no adjustment was necessary.

The foregoing outlines the formal decision analysis procedure which will be employed in conducting the tradeoff study. The steps defined above will actually be the sub-tasks of the tradeoff study and may be scheduled in relation to the milestones created by the procedure. For reference the steps are:

1. Making a decision statement -- (setting objectives)
2. Establishing a list of "musts" and "wants"
3. Weighting of the relative importance of "wants"
4. Measuring alternatives against "musts" to eliminate unsuitable alternatives
5. Assessing relative fit to "wants"
6. Scoring alternatives
7. Assessing possible adverse consequences
8. Evaluating results of the comparison.