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PERKIN-ELMER BOLLER & CHIVENS DIVISION
DESIGN STUDY
of the
ACCESSIBLE FOCAL PLANE TELESCOPE
For Shuttle

FINAL REPORT
April 1976

CONTRACT NAS 9-14684
Mod. No. 2

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lyndon B. Johnson Space Center
Houston, Texas
FORWARD

This final report on the design study of the Accessible Focal Plane Telescope (AFPT) was prepared by the Boller & Chivens Division of The Perkin-Elmer Corporation, South Pasadena, California for the NASA Johnson Space Center (JSC), under Contract No. NAS9-14684 Modification No. 2.

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INTRODUCTION

In July of 1974, in response to the Announcement of Opportunity for Spacelab UV-Optical Telescope (SUOT), a group of astronomers at Johnson Space Center headed by Dr. Karl G. Henize proposed an accessible-focal-plane telescope attached to the aft end of a Spacelab pressurized module. The rationale for such a configuration was to provide a facility for astronomy which met the fundamental concept of Shuttle sortie operation - a facility in which astronomers could operate low-cost manually-controlled instruments in a shirt-sleeve environment.

Although the proposal was successful and Dr. Henize eventually was appointed Team Leader of the SUOT Facility Definition Team, the Team and its sponsors concluded that several possible disadvantages (mainly the complex interface with the Spacelab module) had not been sufficiently studied and chose to focus their attention on a pallet-mounted configuration. In the meantime a study was initiated at Johnson Space Center to further investigate the feasibility of an accessible-focal-plane configuration. As a result, the Boller & Chivens Division of The Perkin-Elmer Corporation was contracted to perform a design study of an Accessible-Focal-Plane Telescope (AFPT) in sufficient detail to establish the feasibility of such a concept as well as its probable cost. The result of this design study is presented in this report.
TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>FORWARD</th>
<th>PAGE NO.</th>
</tr>
</thead>
<tbody>
<tr>
<td>INTRODUCTION</td>
<td>ii</td>
</tr>
</tbody>
</table>

I. General Description of the Accessible Focal Plane Telescope
   A. Physical Arrangement | 1 |
   B. Telescope Tube | 1 |
   C. Telescope Mount | 1 |
   D. Airlock and Aft Plate | 2 |
   E. Instrument Adapter | 2 |
   F. Instrument Package | 3 |

II. Modification to the SUOT Tube Design
   A. Modifications to the Existing Design | 4 |
   B. Tertiary Mirror Mount | 4 |
   C. Telescope Tube Modifications | 6 |
   D. Finder Telescope | 6 |

III. Telescope Mounting
   A. General Configuration | 7 |
   B. Interface with Spacelab Module | 8 |
   C. Launch and Re-entry Support Structure | 8 |
   D. Construction | 9 |
   E. Structural Considerations | 9 |
   F. Performance | 10 |
# TABLE OF CONTENTS (Cont’d)

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>IV.</td>
<td>Airlock and Aft Plate</td>
<td>12</td>
</tr>
<tr>
<td>V.</td>
<td>Instrument Adapter</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>A. General Configuration</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>B. Rotating Joint</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>C. Direct Field Viewing System</td>
<td>14</td>
</tr>
<tr>
<td></td>
<td>D. Image Motion Compensation System</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td>E. Optical System Correctors</td>
<td>18</td>
</tr>
<tr>
<td></td>
<td>F. Retention Mechanism</td>
<td>18</td>
</tr>
<tr>
<td>VI.</td>
<td>Echelle Spectrograph</td>
<td>19</td>
</tr>
<tr>
<td></td>
<td>A. Configuration</td>
<td>19</td>
</tr>
<tr>
<td></td>
<td>B. Instrument Adapter Interface</td>
<td>23</td>
</tr>
<tr>
<td></td>
<td>C. Detectors</td>
<td>23</td>
</tr>
<tr>
<td></td>
<td>D. Controls</td>
<td>23</td>
</tr>
<tr>
<td></td>
<td>E. Performance</td>
<td>24</td>
</tr>
<tr>
<td>VII.</td>
<td>Acceptance and Qualification Test Plan</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>A. Purpose</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>B. Applicable Documents</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>C. Test Configuration and Interface Fixture</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>D. Acceptance and Qualification Tests</td>
<td>27</td>
</tr>
<tr>
<td>VIII.</td>
<td>AFPT Costing</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>References</td>
<td>32</td>
</tr>
</tbody>
</table>

**APPENDIX A**
Design Analysis

**APPENDIX B**
Effects of Crew Motors and Vernier Thrusters on Tracking and Image Stabilization
I. General Description of the Accessible Focal Plane Telescope

A. Physical Arrangement

The AFPT design resulting from this study may be divided into five general sections. The telescope tube, the telescope mounting, the airlock plus spacelab module aft plate, the instrument adapter, and the instrument package. These sections combine to make a system which allows access to the image plane with instrumentation that can be operated directly by a scientist in a shirt-sleeve environment inside a spacelab module. See Fig. 1.

B. Telescope Tube

The basic optical and mechanical design of the telescope is adapted from the Ball Brothers SUOT design. (ref. 1, 2) Modifications consist of the removal of the instrument package, the addition of a third mirror to divert the last 1.5 meters of the optical path at a right angle into the spacelab module, and redesign of a portion of the telescope tube in order to interface it with the mount. The resulting configuration is shown in Fig. 2, 3 & 4.

C. Telescope Mount

The mounting interfaces with the telescope tube and the spacelab module and provides the required coarse pointing and guiding motions. It also provides support for the telescope tube during launch and re-entry. The mounting is joined directly to the spacelab module and does not require pallet support. In addition, the interface of the mount with the module has been designed so that support from the end cone and aft plate is not required.
REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR
REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR
I. C. (cont'd)

No stressing of these sections occurs at any time during a shuttle flight due to the presence of the telescope mount. This is important during launch and re-entry.

D. Airlock and Aft Plate

The airlock is mounted to the interior side of the spacelab module aft plate and is the same design used on Skylab. The aft plate is a modified version of the design. (ref. 5). It contains mechanisms for locking the telescope in its stowage support cradle for launch and re-entry as well as mechanisms for locking the telescope mounting rigidly, relative to the aft plate once orbit has been achieved.

E. Instrument Adapter

The instrument adapter, which attaches to the airlock, provides a universal mount for a wide variety of different instruments. Special provisions have been made to accommodate an electrographic direct imaging camera. The primary function of the instrument adapter is to de-rotate the image and to provide all the required error signals for tracking and stabilizing the image by means of a focal plane guiding system using focal plane detectors. This includes control of the secondary mirror for final image stabilization to ±.03 arc sec. Secondary functions of the instrument adapter include a direct field viewing system for the purpose of target identification and alignment and providing a mounting surface for the Gascoigne corrector and field flattener. The instrument adapter has a quick-release retention mechanism for mounting scientific instruments. This mechanism is similar to the one on the airlock.
I. F. **Instrument Package**

For the purpose of this study, one instrument has been designed for use with the system. It is an echelle spectrograph which, with the interchanging of four sets of detectors, echelle and cross disperser gratings, can cover the spectral range between 1200Å and 1.03 micrometers.
II Modification to the SUOT Tube Design

A. Modifications to the Existing Design

The modifications necessary to adapt the SUOT tube to the AFPT mount are shown in Fig. 2. They consist primarily of the removal of the instrument package, the addition of a third mirror, and redesign of the graphite epoxy tube structure to accommodate a tilt axis support ring.

In the AFPT concept most instrumentation is located inside the spancelab module. However, it is highly desirable for two types of instruments, polarimeters and far UV (900-1100A) spectrographs to be located in the normal Cassegrain position in order to avoid the effects of the tertiary mirror (ref. 3). Such a concept may be incorporated into the AFPT concept since there is a 1.4 meter clearance at the rear of the telescope. Such an arrangement would require only minor modification of the present AFPT design to allow the tertiary mirror assembly to fold back out of the optical path, much like removal of Coude mirror systems on conventional ground-based telescopes.

B. Tertiary Mirror Mount

The mounting arrangement for the third mirror is shown in Fig. 2. As can be seen, it is supported by a structure extending through the hole in the primary mirror. The material chosen for this structure is the same graphite epoxy used for the telescope tube. This results in good thermal stability for the position of the mirror along the optical axis. The mirror mounting is also provided with a mechanism for tilting it from its nominal position (45° to the axis) by an amount equal to one half the angle the telescope optical axis is tilted from its nominal
position, (90° to the axis of the spacelab module). See Fig. 3. This mechanism is required in order to maintain the axis of the last 1.5 meters of the optical path exactly in alignment with the instrument adapter which houses the Gascoigne correctors and field flattener. It is driven mechanically by the movement of the main telescope tube relative to the mounting. It is located at the side of the tube in the vicinity of one of the fork arms. The mirror is driven by a torque tube which is optically in line with one of the secondary spider vanes and exactly on axis with the tilt axis bearings.

The half angle tilt mechanism (Fig. 3) has been used by Perkin-Elmer in similar applications. The mirror tilts ±3.75° on a flexure mount which provides a preload for the mechanism. The linear translator is a ball bushing mechanism. Experience has shown that the smoothness and precision of motion of this mechanism is in the ±1.0 arc second range.

One area of concern relative to the implementation of the third mirror was its effect on the baffling of the telescope system. Since observations will be made in the day part of the orbit it is necessary to require that light from outside the field of view be reflected at least twice before reaching the image plane. The resulting configuration is shown in Fig. 5. The initial AFPT optical design couldn't be fully baffled because of the light striking the tertiary mirror as shown in Fig. 5, ray A. However, a working solution was achieved by extending the telescope tube 10cm and shortening the tertiary mirror.
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II. B. (Cont'd)

Some vignetting results, but only in the guiding portion of the field. No part of the data field suffers any vignetting. The worst case vignetting in the guiding field is about 14%.

C. Telescope Tube Modifications

The modifications necessary to connect the mounting fork to the telescope tube are shown in Fig. 2. The metering tube is divided into two sections with the addition of a structural support ring between the two sections. It is a hollow rectangle in cross-section and has been designed to be considerably stronger than the rest of the telescope tube. The material chosen for this structure is the same graphite epoxy used in the SUOT design. This results in the same thermal stability as in the original design.

D. Finder Telescope

A wide-field (3° diameter) finder telescope is necessary to facilitate the acquisition of fields. Although Orbiter can be oriented to within 0.1 degree using its startrackers, pointing misalignments between the Orbiter startrackers and the payload experiment mount can be as large as two degrees.

The optical system of the proposed finder telescope is a 100mm aperture, f/1.75 telescope which images a three-by-three degree field on view onto a 400-by-400 element charge coupled device (CCD). Such a system should be able to detect and track 10th magnitude stars accurate to two arc seconds. Such a finder telescope is being developed by a group headed by Mr. John McLauchlin at Jet Propulsion Laboratory (ref. 6).
III. Telescope Mounting

A. General Configuration

The general configuration of the telescope mounting is shown in Fig. 2. It is a fork type design with two duplex bearing sets supporting the telescope tube at the tilt axis and a single four point contact bearing supporting the mounting at the rotation axis. The tilt axis has a ±7.5° range of movement and the rotation axis has a ±90° range of movement. ±70° of the range is used for observing (no vignetting by the shuttle vehicle) with the remaining 10° being used to bring the telescope tube into its stowage position in the launch and re-entry support structure. Both axes are driven by torque motors with tachometers. The tilt axis motor is capable of producing a torque of 14 lb-ft which is quite adequate for handling the bearing friction which is expected to be on the order of 0.25 lb-ft. The rotation axis bearing, however, has quite a bit more friction; 44 lb-ft. In both cases, the torque available above that required to overcome the bearing friction is required to provide the accelerations necessary to follow angular accelerations of the shuttle vehicle caused by thruster firings. The bearing friction, particularly in the rotation axis, may vary as much as 20%, however, this is not a problem since it will occur on a very low frequency basis and can easily be handled by the servo drive system. A summary of the driving torques and resulting power requirements is tabulated in Appendix A. Each axis also has a 17 bit absolute encoder for pointing to ±0.2 arc minute. More precision than this is not required since the telescope need only be pointed accurately enough to acquire the target in the 3° field of the bore sight finder telescope mounted on the main telescope tube.
III. A. (Cont'd)
The tilt axis encoder is mounted directly to the shaft while the rotation encoder is mounted off axis and coupled to its shaft by a 2:1 gear train.

B. Interface with Spacelab Module
It is a basic guideline of the study to minimize the mechanical interface between the AFPT mount and the Spacelab module. This has been achieved by supporting the AFPT on a cone which covers the aft cone structure of the Spacelab module without touching it. The base of the support cone attaches directly to the structurally sound bolt ring at the aft end of the cylindrical section of the module. This ring also provides the structural tie between the module and the shuttle. During launch and re-entry the AFPT facility is attached only to the bolt ring. During orbital operations (when the mount is subjected to very little stress) it is necessary to directly connect the airlock in the module aft plate to the telescope mount in order to preserve precise optical alignment and focus during observation. Such a connection does not constitute a modification to the existing module aft plate since it is proposed that the existing aft plate be replaced by a special one tailored to accommodate the airlock and the above-mentioned connectors. Thus, the only interface with the module required by these modifications will be a replacement of the aft plate plus the attachment of this base of the support cone to the Spacelab's bolt ring.

C. Launch and Re-entry Support Structure
The mounting is also provided with a launch and re-entry support structure for the telescope tube, Fig. 6. It is mounted to the support cone which interfaces the mounting to the Spacelab module. The axis of the telescope is parallel to the Y axis of the shuttle vehicle during launch and re-entry. Locking of the telescope in this position is accomplished by driving it to the zero position and actuating a manual mechanical linkage from inside the Spacelab module. The linkage operates through a rotary vacuum feed-through in the aft plate.
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Support Cone

Support Ring

Fork Tine

Telescope

Launch & Re-Entr Support Structure

Actuating Handle (Inside Spacelab Module)
III. D. Construction

Construction of the mounting is largely of welded aluminum plate (2219-651). Some of the smaller sections are machined from solid stock. One concern with the aluminum structure is that parts of it retain steel bearings. Differences in the relative coefficient of expansion must be accounted for in the mounting design. As a result, it was decided to control the temperature of these sections by thermal heating and blanketing. The design calls for thermal blanketing of some parts of the dynamic portion of the mounting in the vicinity of the bearings and complete blanketing of the support structure which interfaces the mounting with the spacetlab module. The weight of the mounting and telescope tube is 995 kg (2194 lbs). The mounting by itself weighs 542 kg (1194 lbs).

E. Structural Considerations

The structural analysis of the telescope mount is detailed in Appendix A. The stiffness of the mounting is high resulting in natural frequencies of 50.5 Hz in the X-axis, 33.6 Hz in the Y-axis, and 41.7 Hz in the Z-axis. As a result of the telescope mount being relatively stiff, most of the stress levels in the structure itself are quite low, even considering the worst case conditions which occur during "solid rocket cut-off" for the X-axis, and "launch release" for the Y-axis and the Z-axis. Stressing of the spacetlab module due to the presence of the telescope mounting is also quite low. It is about one-eighth the stress caused by the internal pressurization of the module.
III. F. Performance

There does not appear to be any problem with the capability of the mounting to track under normal conditions to ±2 arc seconds. Calculations detailing the effects of crew motion and thruster firings on the tracking precision are tabulated in Appendix B. Indications are that both the crew motions and small thruster firings have about the same effect on the telescope pointing errors (ref. 4). The magnitude of the telescope pointing and final image stabilization errors are, of course, dependent upon the characteristics of the servo system employed.

It is recommended that a non-linear digital servo system controlled by a computer (mini or microprocessor) be used to control both the rate at which the main telescope mount is driven and the tilt angle of the secondary mirror. In such a servo system, the computer samples the angular error of the image from its nominal guiding position (as indicated by the focal plane tracker) at intervals equal to \( T \), where \( T \) equals the time constant of the trackers, and makes corrections by tilting the secondary mirror. The computer also makes corrections in the speed at which the main telescope mount is being driven according to the following algorithm:

\[
\Delta \omega_m = \frac{1}{T} (K \theta_T - \theta_{T-1})
\]

Where \( \Delta \omega_m \) equals an incremental drive speed correction (arc sec/sec), \( T \) (as mentioned above) equals the time constant of the focal plane tracker (sec.), \( \theta_T \) equals two times the angle the secondary is presently tilted from its nominal position (arc sec), \( \theta_{T-1} \) is the same value for the last sample, and \( K \) is a constant which determines the effective damping characteristic of the system. The system is critically damped when \( K = 2.0 \), over
III. F. Performance - (Cont’d)

damped when K is less than 2.0 and under damped when K is greater than 2.0.

The excursion of the image from its nominal position in the focal plane due to vehicle disturbances is very dependent on the time constant of the CCD focal plane tracker. With a time constant of 0.1 sec, crew motions can cause the image to be outside the desired ±0.03 arc sec for short periods of time. However, if the tracker can be operated with a 0.05 sec time constant, the image could be kept within the ±0.03 arc sec at all times. Minimum errors, of course, will result when the system is critically or slightly under damped.
IV. Airlock and Aft Plate

A. As a cost saving measure it was directed that the design of the airlock be the same as that of the existing Scientific Air Lock (SAL) of Skylab. It is a fortuitous coincidence that its 20cm x 20cm aperture is just adequate to transmit the converging optical beam.

To accommodate the SAL as well as the mechanisms for locking the aft plate to the telescope mount and for locking the telescope into its launch cradle the aft plate has been redesigned.

The mechanism for locking the aft plate and mounting together is shown in Fig. 2. It is a push-pull coaxial screw arrangement which is actuated with three rotary knobs from inside the space-lab module. One knob extends a part which engages a conical socket on the telescope mount. A second knob locks the extended part and conical socket together. The third knob locks the extension thread eliminating all freedom of movement between the aft plate and telescope mount. The mechanism has been designed to be vacuum tight. Triple Viton O-ring seals are used on all rotating parts of these mechanisms. It should be noted that the aft plate is indirectly locked to a dynamic part of the mounting via a four point contact bearing. All freedom of motion of the end plate relative to the dynamic portion of the telescope mount is restricted except rotation. It would have been somewhat more desirable to lock the aft plate directly to a rigid static part of the mounting. However, no space was available for implementation of such a structure. This is the result of the fairly limited distance (1.5 meters) between the telescope axis at the tertiary mirror and the image plane. The back focus of the telescope could have been changed to increase this distance. However, it was a basic guideline that the existing SUOT optical design be used for the AFPT.
V. Instrument Adapter

A. General Configuration

The configuration of the instrument adapter is shown in Fig. 7. Mechanically, the instrument adapter is divided into two main sections, which are connected by a rotating vacuum joint. The front section connects to the SAL by means of a square flange. The rear section contains a direct field viewing system, sensors for the image motion compensation (IMC) system, mountings for two optical correctors, and a retention mechanism to which scientific instruments may be attached.

B. Rotating Joint

The rotating joint is driven by two torque motors with tachometers. Each torque motor is mounted off axis and drives the joint through a 10:1 gear train. One torque motor drives the joint, while the other controls the backlash in the gear train by applying a reverse torque. The rotating joint is also provided with a 10 bit encoder which is also mounted off axis and driven by means of two gear meshes of 10:1 and 1:10. A summary of the torque and electrical power requirements is outlined in Appendix A. The maximum speed at which this joint needs to be driven is 4.6°/min which is the orbital rate plus the free drift rate of the vehicle. Slewing is accomplished manually. The limit of movement is ±360°.

The rotating joint is provided with a 10 bit encoder. The purpose of the encoder is to indicate the relative rotation orientation of the rear section of the instrument adapter to the telescope mount.
Foldout Frame

- 400x400 CCD array
- Knob (fine focussing)
- X-control knob & counter
- Y-control knob & counter
- Rear nest
- Retractable field viewer
- Actuating lever
- Torque motor with 10:1 gear box
- Encoder with 10:1 gear box, tachometer at opposite corner
V. Instrument Adapter (Cont’d)

B. Rotating Joint (Cont’d)

This information is required in order to translate the direction of image motion sensed by the IMC detectors into telescope tilt and rotation corrections. The output of this encoder must be fed both into the telescope control computer, as well as into the servo system controlling the secondary mirror.

Several vacuum seals were considered for the rotating joint. A summary of the drive and power requirements is detailed in Appendix A. The drive and power requirements tabulated are based on using lubricated double Viton O ring seals. There is no doubt that the Viton O ring seals can be qualified, but they have an undesirably high friction—about 10.8 lb-ft each. A considerable reduction in the drag could be realized if either Bal-seals or Omniseals could be used. Their drag is only 1.25 lb-ft for the Bal-seal and 1.6 lb-ft for the Omniseal. However, it remains to be seen whether such seals can be space qualified.

C. Direct Field Viewing System

Also contained in the rear section is a direct field viewing system. It is used for field identification and fine alignment of the image. Viewing is accomplished by means of a 75mm focal length eyepiece and a diagonal mirror which can be moved in or out of the optical path. The field diameter is 11.0 arc minutes. The design can accommodate interchangeable eye pieces so that very high power viewing of images is also possible. For example, a 15mm focal length eye piece may be
V. Instrument Adapter (Cont'd)

C. Direct Field Viewing System (Cont'd)

used to give a power of 1000. Such an arrangement would allow the eye to achieve an angular resolution of about 0.1 arc sec. Such a capability would be useful both scientifically (e.g., for visual observations of very close double stars), and technically (e.g., for monitoring image quality and mirror alignment).

D. Image Motion Compensation (IMC) System

The IMC System consists of a pair of focal plane detectors, and a servo system which corrects image motion by tilting the secondary mirror. The servo system and the drive for the secondary mirror are beyond the scope of this study and it is assumed that they are identical to those which were proposed by BBRC for SUOT. However, since the focal plane sensing system is an integral part of the AFPT instrument adapter, their nature and their positioning systems are included in this study.

The sensors for the IMS system consists of two independent optical systems and detectors, each of which can access any part of a semi-field (See Fig. 8). Together, the whole field can be covered. The detector in each system is a 400 x 400 CCD array with elements on 23µm centers. The device is currently being developed by JPL and is approximately 9mm square. It presently can detect image movements in the 2µm range. Its spectral response extends from 4000Å to 9000Å. In the focal
FOLDOUT FRAME

OPTICAL AXIS

PROBE MIRROR
MOVEABLE IN X AND Y DIRECTIONS

0.8" (14 MRAD) FIELD

DISTANCE VARIES DUE TO FIELD CURVATURE

DIAGONAL

DEPTH OF FOCUS @ 400nm = .36mm (RAYLEIGH LIMIT)

G'FL COLLIMATING LENS
MOVEABLE IN X AND Y DIRECTIONS
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<thead>
<tr>
<th>Item No.</th>
<th>Description</th>
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<th>Material or Note</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>G. FL. LENS</td>
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<tr>
<td></td>
<td>WITH MANUAL FINE FOCUS</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>400 x 400 ARRAY</td>
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<tr>
<td></td>
<td>MOVEABLE IN Y DIRECTION</td>
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<td></td>
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<tr>
<td>3</td>
<td>Diagonal Mirror</td>
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V. Instrument Adapter (cont'd)

D. Image Motion Compensation (IMC) System (cont'd)

plane tracking system the CCD is capable of guiding with ±0.03 arc seconds tracking error on stars as faint as 13th magnitude. The mounting for the CCD detector is stationary which allows thermoelectric cooling by a heat exchanger that is located on the outside of the instrument adapter housing. No plumbing for cooling fluids is needed since the exchanger is simply a metal heat sink device with fins.

The optical system for each guider probe consists of a pick-off mirror, a collimator, a diagonal mirror, and a focusing lens. Movement of the pick-off mirror, collimator, and diagonal mirror, by means of mechanical controls on the outside of the housing, allows each CCD detector to optically view any part of a 100mm x 200mm area in the image plane. The pick-off mirror, collimator, and diagonal mirror all move as a unit. The distance between the collimator and pick-off mirror is fixed. The distance between the collimator and diagonal mirror varies depending upon where the pick-off mirror is located in the image plane. Positioning of the pick-off mirror is bases on an X-Y coordinate system with independent controls for X and Y. The position is indicated to 1.0mm by digital readouts.

Optically, the CCD is located behind the Gascoigne corrector but in front of the field flattener. As a result of this and the fact that the CCD is statically mounted, the image plane is slightly tilted when working with parts of the image plane other than the center. At the very edge this amounts to 11.0 arc minutes. This is not a problem, however, since the CCD works best on slightly defocused images which extend over at least a 2 x 2 element subarray. Since the telescope will
V. Instrument Adapter (cont'd)

D. Image Motion Compensation (cont'd)

produce near diffraction-limited image diameters of about 20µm, defocusing is required to produce images with approximately 40µm diameters. The defocusing effect of the tilted image plane is small compared to images this size. Current tests at JPL indicate that the CCD is capable of detecting displacements of such images as small as 2µm (i.e. 0.03 arc seconds).

Diameter of the stellar image on the CCD is adjustable by manual controls on the outside of the instrument adapter housing which vary the distance between the focusing lens and the CCD. Whatever position the focus is adjusted to is automatically maintained as the effective viewing position of the guider in the image plane is varied. This is necessary because the pick-off mirrors are before the field flattener in the optical path of the telescope.

Alignment of each of the CCD arrays is such that it is parallel to the X-Y coordinate system for its effective optical location in the image plane as well as the slit of the spectrograph when it is in use with the system. The reason for this is that the null position for guiding on the CCD is shifted digitally to accomplish widening.

Ordinary guiding is accomplished by using one of the CCD trackers to control movement of the main telescope mount motions (tilt and rotation) and the secondary mirror, while the second is used to control the rotation of the instrument adapter. Either tracker can serve either function. In addition, either tracker by itself can serve both functions to a limited extent.
V. Instrument Adapter (cont'd)

E. Optical System Correctors

A Gascoigne corrector and a field flattener are mounted in the instrument adapter. The Gascoigne corrector is used in two versions which can be interchanged. One has a hole in it for use with the spectrograph. In this mode the corrector provides the image quality required by the IMC detectors, yet passes a central beam to the spectrograph which is free of absorption from the corrector material. The field flattener is also removable for on-axis spectroscopy. Since spectroscopy uses only the center of the field, correction for astigmatism and field curvature is not needed.

F. Retention Mechanism

The retention mechanism for connecting an instrument to the adapter is placed slightly ahead of the field flattener so that an electrographic direct imaging camera can be used with the adapter. This provides clearance for the magnet which extends out in front of this type of device. The retention mechanism is identical to the one on the SAL and allows mating and demating in a matter of seconds once the SAL is closed.
VI. Echelle Spectrograph

A. Configuration

The spectrograph is a versatile instrument, providing high spectral resolution over a wide spectral range. The design is based on spectrographs used at ground based observatories, but modified to take advantage of the unique benefits of space such as the extended ultraviolet response and the diffraction limited telescope images. A single slit is provided for observations of stellar objects, and multiple slits, covering a 5mm (75 arc second) square format, for study of extended, emission line objects.

Like virtually all astronomical spectrographs, this echelle spectrograph has a data channel consisting of a slit, collimator, dispersing elements, a camera, a detector, and auxiliary systems including the comparison light system, a viewing system and order blocking filters.

In order to operate efficiently over the entire spectral range from 1200\(\text{Å}\) to 10300\(\text{Å}\), this range has been divided into four spectral regions. For each region, a specific echelle grating, cross dispersing grating, blocking filter, comparison source and detector are used. Table 1 lists these spectral regions and the required units.

Figure 9 is an optical diagram of the spectrograph and Fig. 10 is an outline drawing of the exterior configuration. The design has interchangeable slit assemblies consisting of several
### TABLE 1

**ECHELLE SPECTROGRAPH CHARACTERISTICS**

1. Spectral resolution ($\lambda/\Delta \lambda$); 1200/ (detector linear resolution (mm))
2. Dispersion (Å/mm); $\lambda_c$ (Å)/1200
3. Length of any order (mm); 1200/spectral order number
4. Format, 25mm square
5. Separation of orders; 1 to 3.1 mm
6. Number of spectral regions; 4
7. Exposures required to cover any one spectral region; 3
8. Blaze angles
   a. Echelle, 63.435° (R-2)
   b. Cross disperser, 11°
9. Spectral data

<table>
<thead>
<tr>
<th>Order</th>
<th>Length (mm)</th>
<th>Separation (mm)</th>
<th>Region 1 (Å)</th>
<th>Region 2 (Å)</th>
<th>Region 3 (Å)</th>
<th>Region 4 (Å)</th>
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<tr>
<td>48</td>
<td>25</td>
<td>3.1</td>
<td>2100</td>
<td>3570</td>
<td>6060</td>
<td>1.03µ</td>
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<tr>
<td>50</td>
<td>24</td>
<td>2.8</td>
<td>2016</td>
<td>3427</td>
<td>5818</td>
<td>9888</td>
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<td>55</td>
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<td>2.3</td>
<td>1833</td>
<td>3116</td>
<td>5289</td>
<td>8989</td>
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<td>2856</td>
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<td>8240</td>
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<tr>
<td>65</td>
<td>18.5</td>
<td>1.7</td>
<td>1551</td>
<td>2636</td>
<td>4475</td>
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<td>70</td>
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<td>1.4</td>
<td>1440</td>
<td>2448</td>
<td>4155</td>
<td>7063</td>
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<tr>
<td>75</td>
<td>16</td>
<td>1.3</td>
<td>1344</td>
<td>2285</td>
<td>3878</td>
<td>6592</td>
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<tr>
<td>80</td>
<td>15</td>
<td>1.1</td>
<td>1260</td>
<td>2142</td>
<td>3636</td>
<td>6180</td>
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<tr>
<td>84</td>
<td>14.3</td>
<td>1</td>
<td>1200</td>
<td>2040</td>
<td>3463</td>
<td>5886</td>
</tr>
</tbody>
</table>

10. Grating Data (G/mm)
   a. Echelle 177.5 104.4 61.6 36.2
   b. Cross Disperser 2300 1360 800 470
11. Spectrum in one exposure (Å) 350 600 1010 1720
12. Blocking Filter
   a. Type MgF₂
   b. Short side block (Å) Corning 9-58 0-54 3-66
   c. Second order of cross disperser overlaps at (Å) 2200 3600 6040 1.12µ
NOTES:
1. ALL OPTICS: CERAMIC, ALUMINISED
2. COLLIMATOR TO SLIT CAMERA TO FOCAL PLANE, MOUNTED IN NITRILE VITON.
3. SCHLIERE DISPERSION = 1/1000.
FOLDOUT FRAME

SHUTTER CONTROL
VI. Echelle Spectrograph

A. Configuration (Cont'd)

single slits of different but fixed widths and several multiple slits. All the slits are of a fixed width to simplify the instrument and enhance its reliability. A pair of comparison prisms are used to introduce comparison light into the ends of the slit (or the ends of the central slit when multiple slits are used). The comparison prism spacing is adjustable. The upper surface of the slit is highly reflective to reflect the telescopes field into the slit viewing periscope. With the comparison prisms retracted, the viewing field is a rectangle of about 10mm x 25mm (2.3 x 5.7 arc minutes).

An off-axis parabolic mirror is used to collimate the beam. The diameter of the collimated beam for a point source is 20mm. At the corner of the field, where the images are the largest, the comatic image has a spread of 10µm (0.14 arc seconds) in the direction of the echelle dispersion and 20.3µm (0.28 arc seconds) along the direction of the projected slit.

The high dispersion grating is an R-2 echelle grating (tangent of blaze angle = 2). It has a ruled area of 2.5cm x 6cm and a blank size of 2.7cm x 5.4cm x 1cm. Four echelles, one for each spectral region, are required.
First order gratings are used to separate the echelle orders. The minimum separation is 1mm at the short wavelength limit of the spectral region and the separation increases to 3.1mm at the longer wavelength limit. The ruled area of the cross dispersers is 2.3cm x 8.5cm with an 11° blaze angle. Four cross dispersers, one for each spectral region, are also required. Table 1 lists the number of grooves per millimeter for both dispersers, for each of the four spectral regions.

A 300mm all reflecting, off-axis, folded Schmidt camera is used to image the spectrum on the detector. Although the field is curved, the visual light images formed on a flat detector, will be within the Rayleigh limit except for the extreme corners. The corrector is figured to produce a point image at the center of the field. At the corners of the field, the image spread, due to the camera's aberrations, is considerably smaller than the diffraction image in visible light. In visible light, the camera is diffraction limited, with the previously noted exception of the extreme corners of the field.

The spectrograph has several auxiliary systems.

1) Slit Viewer - Light reflected from the slit jaws enters a slit viewing periscope which is used to acquire the object under study and, in the case of extended objects, to assist in guiding the telescope. An image intensifier can be added to permit the viewing of very faint objects.
VI. A. (Cont'd)

2) **Spectrum Comparison System** - Several spectrum comparison sources may be provided. A final selection was not made as part of this study. The conventional iron-argon hollow cathode and neon glow lamp sources can be used for the near IR red, blue, and near UV regions. A thorium hollow cathode source may be a good selection for the UV region.

A filter holder will be placed in the comparison system.

3) **Shutter** - A blade type shutter, mounted shortly behind the slit, is used to interrupt the beam to limit the period of the exposures.

4) **Order Separating Filters** - A filter is placed behind the slit to eliminate the overlapping second order of the cross disperser. The selection of filters is shown in Table 1.

5) **Exposure Meter** - A photon-counting exposure meter could be a useful accessory to verify that the starlight is passing through the slit and to indicate the amount of accumulated exposure. Several sources of light are available including the light scattered into other orders at the echelle or the cross disperser, or a flip mirror that reflects a small amount of light into the meter.

All of the optics are mounted on an "optical table," probably made of Invar. This table is kinematically mounted so that it cannot be distorted by pressure or temperature changes in the
VI. A. (Cont'd)

space lab. The plate is defined in all three coordinates near
the detector, and in two coordinates near the slit. By the use
of an Invar table, and zero expansion mirrors, the optical
alignment and focus is not sensitive to temperature.

A vacuum enclosure is used to enclose the entire optical path.
This housing has a large removable plate for laboratory
alignment purposes. Smaller access ports are provided for
use in changing gratings and the order separating filters in
orbit.

All the mirrors and diffraction gratings will be aluminized and
overcoated with magnesium fluoride.

B. Instrument Adapter Interface

A rectangular flange is used to mount the spectrograph to the
instrument adapter. A concentric pilot is provided to insure
alignment of the slit at the center of the telescope's field.

C. Detectors

Several different detectors may be used, including photographic
plates, electrographs, image intensifiers, CCD's, CID's, linear
arrays and more conventional television type detectors. A
common interface is used for all the detectors. Again, a con-
centric pilot will maintain alignment and an 'O' ring will seal
the joint.

D. Controls

Mechanical controls are provided on the outside of the vacuum
housing to operate the shutter, comparison prisms, cross dis-
perser angle, the collimator focus and comparison source
selector. Electrical switches are provided to operate the
comparison sources.
VI. D. (Cont'd)

The angle of the echelle will be adjusted in the laboratory and need not be adjusted again. To change the wavelength range within a given spectral region, the angle of the cross disperser is changed. Tables or charts can be prepared to aid in the selection of the proper angle.

When detectors with an electronic readout or image intensifiers are used, the observer can adjust the focus while viewing the output. If a photographic plate or an electrograph is used, he may either accept the laboratory calibration of focus or confirm the focus by making a focus plate and develop it aboard the spacecraft.

E. Performance

Table 1 lists many of the performance characteristics of the spectrograph including the dispersion, spectral resolution, length of orders and the separation of orders.

At f/15, the diameter of the Airy disk varies from 4.4µm at 1200 Å to 38µm at 1.03µm. In the center of the visible range, the diameter is 20µm. Photographic plates can resolve 10 to 15µm. Most photoelectric detectors resolve from 20 to 40µm. In general, the resolution of the spectrograph is limited by the linear resolution of the detector, not the angular resolution of the optics.

In Section A it was pointed out that the collimator produces diffraction limited images for objects at the center of the 5mm square slit field. At the corners of the field, the image spread amounts to 10µm in the direction of the echelle dispersion and 20µm in the direction of the projected slit.
VI. E. (Cont'd)

The aberrations of the camera were also discussed in Section A. With the exception of the far corners of the field, the images are within the visual Rayleigh limit. At shorter wavelengths, the detector limits the resolution.
VII. Acceptance and Qualification Test Plan

A. Purpose

The purpose of this section of the report is to outline and identify the equipment and tests necessary to verify the acceptance and qualification of the AFPD version of the SUOT and associated equipment for flight on shuttle. The tests are outlined in two sections; acceptance tests on all experiment hardware, and qualification tests to be applied only to the qualification test unit.

B. Applicable Documents

Levels and other applicable data for the above outlined tests were determined by use of "Spacelab Payload Accommodation Handbook" marked "Preliminary" and dated May 1975 (SLP/2104). Much of the required data is in very preliminary form or TBD. Levels and other data included in this document are best estimates based on SKYLAB experience.

C. Test Configuration and Interface Fixture

Many of the tests require the complete experiment to be assembled in an operational configuration. This will require a test fixture which simulates the end cone structure of the Spacelab Module. The test fixture is mounted to a suitable adapter plate which is capable of supporting the entire AFPT facility including the modified telescope tube, the telescope mounting, the redesigned aft plate, the airlock, the instrument adapter, and the spectrograph. This test fixture should be designed to simulate all the required interfaces and should be compatible with the requirements of all the environmental tests.
VII. Acceptance and Qualification Test Plan (Cont'd)

C. As presently envisioned, the above mentioned test fixture would be a stainless steel ring structure the diameter of the Spacelab Module. The structure would be designed to support the experiment hardware through vibration and thermal testing. The materials shall be qualified to be compatible with vacuum and thermal vacuum tests. Separate fixtures shall be made to separately qualify sub-assemblies such as the spectrograph, instrument adapter and airlock during leakage and vacuum tests.

D. Acceptance and Qualification Tests

The following tests are considered required or desirable for the AFPT hardware. Actual requirements could vary depending on future definition and completion of shuttle payload environmental tests requirements documentation.

1) Vibration

The AFPT hardware shall be subjected to vibration tests in an assembled non-operating configuration. That means the telescope will be locked into its launch and re-entry support structure and the airlock will be blocked off with its cover plate. In addition, the spectrograph and instrument adapter will be in their respective stowage structures. The environmental levels are defined by SLP/2104 Paragraph 5.1.

2) Shock

Mounted in the configuration described in D 1), the experiment hardware shall be subjected to shock tests as outlined in Section 5.1.3 of SLP/2104. This includes pyro shock TBD, loading shock as detailed in table 5-2 of SLP/2104, and crash safety shock which will be covered by static
VII. Acceptance and Qualification Test Plan (Cont'd)

D. Acceptance and Qualification Tests (Cont'd)

stress analysis. This last requirement is for equipment survival only in a non-operating condition with no loose parts.

3) Acceleration

The experiment hardware shall be subjected to acceleration tests to simulate a nominal and emergency sequence of orbital mission operations as detailed in table 5-3 of SLP/2104.

The experiment hardware shall also be subjected to the environments detailed in table 5-4 of SLP/2104 while in the free operational configuration.

4) Temperature

All airlock and airlock experiment hardware shall be subjected to temperature tests of 15° C. to 45° C. mounted in a suitable test chamber. Gas compositions and pressure is defined in table 5-6 of SPL/2104.

5) Thermal Vacuum

The telescope experiment package shall be mounted in a suitable thermal vacuum chamber in an unstowed operational configuration. All equipment control heaters shall be on during operation tests. The pressure shall be reduced to \(1 \times 10^{-6}\) torr with a non-contaminating pumping system. The thermal environment is defined by tables 5-9 and 5-10 of SPL/2104.
D. Acceptance and Qualification Tests (Cont'd)

6) **EMI**

To insure compatibility with the shuttle spacecraft and its sub-systems, the AFPD hardware in the operational mode shall be qualified in accordance with the Space Shuttle Amendment to MIL-STD-461A.

7) **Bonding**

All equipment shall have suitable coatings and bonding straps to insure bonding resistance paths not to exceed 2.5 milli ohms.

8) **Optical Performance Tests**

Using an ultraviolet collimator, several diverter mirrors, and a focal plane photometer, all the following optical operation parameters shall be verified in a vacuum environment of $1 \times 10^{-6}$ torr.

   a) Collimation
   b) Resolution
   c) Mount guidance to ±2 arc sec
   d) Secondary mirror guidance to ± .03 arc sec
   e) Slew rate of 3°/sec
   f) Absolute pointing to ±20 arc sec
VIII. AFPT Costing

Table 2 summarizes the expected costs for two possible AFPT funding programs. Under the first program, a single complete set of prototype, qualification test, and flight unit. Under the second program, two complete sets of hardware would be built. One set would serve as a combined prototype and qualification test unit, while the second would be used as the flight hardware.

The cost figures shown in Table 2 include all the necessary engineering and qualification testing required to produce the final flight hardware.
## TABLE 2

### AFPT COST ANALYSIS

<table>
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<th>CONSTRUCTION CONCEPTS</th>
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<td>Protosflight ($1000)</td>
<td>Unit</td>
<td>Prototype &amp; Flight Unit ($1000)</td>
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<td>1919</td>
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<td>Telescope Mount</td>
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<td>Scientific Airlock</td>
<td>16</td>
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<td>Aft Plate</td>
<td>612</td>
<td></td>
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<td>Instrument Adapter</td>
<td>794</td>
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<td>1302</td>
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<tr>
<td><strong>AFPT Total</strong></td>
<td>5825</td>
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<td>9803</td>
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</table>

| Spectrograph          | 899                    |          | 1475     |
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   NASA and ESRO

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   March 6-7, 1975, Jet Propulsion Laboratory
   JPL SP 43-21 (p 31).
APPENDIX A INDEX

<table>
<thead>
<tr>
<th>SECTION</th>
<th>TITLE</th>
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<tbody>
<tr>
<td>SUM</td>
<td>SUMMARY</td>
</tr>
<tr>
<td>SUM-1</td>
<td>DEFLECTION AND SLOPES</td>
</tr>
<tr>
<td>SUM-1</td>
<td>NATURAL FREQUENCY</td>
</tr>
<tr>
<td>SUM-2</td>
<td>MAX. STRESS LEVELS</td>
</tr>
<tr>
<td>SUM-4</td>
<td>CREW MOTION DISTURBANCE</td>
</tr>
<tr>
<td>SUM-7</td>
<td>WEIGHTS AND CENTER OF GRAVITY</td>
</tr>
<tr>
<td>SUM-8</td>
<td>DRIVE MOTOR SUMMARY</td>
</tr>
<tr>
<td>SUM-9</td>
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SUMMARY

DEFLECTION AND SLOPE AT TELESCOPE TUBE
(FROM SUPPORT CONE, 4-PT BEARING & FORK)
ADDITIONAL SUPPORT FROM THE IN-FLIGHT HARD MOUNTING TO THE CENTER RING OF THE SPACELAB ART-CONE HAS NOT BEEN INCLUDED

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<tr>
<th></th>
<th>DEFLECTION AT TELESCOPE</th>
<th>SLOPE AT TELESCOPE</th>
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<tr>
<td></td>
<td>X-AXIS</td>
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<td>FORK TIMES</td>
<td>2.5</td>
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<tr>
<td></td>
<td>(0.099)</td>
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<td>FORK RING</td>
<td>0</td>
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<td>(2.2)</td>
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<td>4-PT BEARING</td>
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<td></td>
<td>(0.088)</td>
<td>(0.33)</td>
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<td>SUPPORT CONE</td>
<td>95.5</td>
<td>14.58</td>
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<tr>
<td></td>
<td>(3.8)</td>
<td>(0.57)</td>
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<td>TOTAL EFFECT ON TELESCOPE</td>
<td>100.24</td>
<td>208.12</td>
</tr>
<tr>
<td></td>
<td>(3.9)</td>
<td>(8.2)</td>
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</tbody>
</table>

*FORK TIMES DEFLECT AS A PARALLELGRAM THUS HAVING ZERO SLOPE AT TELESCOPE

NATURAL FREQUENCY OF SUPPORT SYSTEM (CONE + BEARING + FORK)

X-AXIS \( f_n = 50.5 \, \text{Hz} \)

Y-AXIS \( f_n = 34.5 \, \text{Hz} \)

Z-AXIS \( f_n = 43.5 \, \text{Hz} \)
SUMMARY (CONT)

MAXIMUM STRESSES IN TELESCOPE SUPPORT SYSTEM

Since the telescope tube support has been designed for stiffness, the nominal stress levels are very low. For this analysis, the following extreme conditions have been used to determine maximum stresses:

1. No auxiliary cradle provided to support the telescope tube during launch and re-entry

2. Dynamic accelerations are occurring at the resonant frequency of the support system (~40 Hz) with a mechanical amplification factor of Q=20. (Low damping)

3. Random vibrations are occurring at the resonant frequency of the support system (~40 Hz) with a mechanical amplification factor of Q=20. (Low damping)

MAX LOADS OCCUR DURING THE FOLLOWING MISSION PHASES:

- X-axis max load = 41 g during "solid rocket cut-off"
- Y-axis max load = 10.7 g during "launch release"
- Z-axis max load = 58.1 g during "launch release"

Updated to 20.1 g

Note: the above load levels exceed the 9 g crash landing load in the +X direction.

(CONT)
SUMMARY (CONT)

MAX. STRESSES IN TELESCOPE SUPPORT SYSTEM (CONT)

Max Combined Stress in: \[ \sigma = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2} \]

1. SUPPORT CONE \[ \sigma_x = \frac{1.14 \times 10^8 \text{ N/m}^2}{(10,500 \text{ psi})} \]

\[ \begin{align*}
\Sigma x &= 41.9 \\
\Sigma y &= 32.5 \\
\Sigma z &= 8.19 \\
\text{Margin of Safety} &= 1.29
\end{align*} \]

2. A.F.T CONTACT BEARING

Axial Load \[ F_a = \frac{2.19 \times 10^5 \text{ N}}{(49,000 \text{ lb})} \]

\[ \text{M.S.} = 3.05 \]

Radial Load \[ F_r = \frac{3.14 \times 10^5 \text{ N}}{(70,935 \text{ lb})} \]

\[ \text{M.S.} = 0.02 \] (UPDATED TO +1.59)

Moment Load \[ M = \frac{2.81 \times 10^5 \text{ N.m}}{(2,480,000 \text{ lb-in})} \]

\[ \text{M.S.} = -0.38 \] (UPDATED TO +0.82)

3. FORK RING

\[ \sigma_x = 12.5 \]

\[ \begin{align*}
\Sigma y &= 16.7 \\
\Sigma z &= 58.19 \\
\text{M.S.} &= 1.66
\end{align*} \]

4. FORK TIMES

\[ \begin{align*}
\Sigma x &= 12.5 \\
\Sigma y &= 16.7 \\
\Sigma z &= 58.19 \\
\text{M.S.} &= 0.17
\end{align*} \]

\[ \sigma_x = \frac{2.71 \times 10^8 \text{ N/m}^2}{(39,240 \text{ psi})} \]

\[ \text{DURING "LAUNCH RELEASE"} \]
Crew Motion Disturbance - Resulting Pointing Error From Deflection of Teles. Mount Only

Average Force Due to Crew Movement \( \frac{F}{2} = 50 \text{ N} \) (11.24 lbs)

Located 16m Forward of C.G. in X-Axis

Moment of Inertia of Orbiter + Spacelab

\[ I_{xx} = 1.034 \times 10^6 \text{ kg} \cdot \text{m}^2 \] (7.76 suit ft²)

\[ I_{yy} = 7.643 \times 10^6 \text{ kg} \cdot \text{m}^2 \] (5.6 suit ft²)

\[ I_{zz} = 7.838 \times 10^6 \text{ kg} \cdot \text{m}^2 \] (5.8 suit ft²)

<table>
<thead>
<tr>
<th>( T )</th>
<th>Orbiter Forced Acceleration</th>
</tr>
</thead>
<tbody>
<tr>
<td>50 \times 10^{-3} \text{ m}</td>
<td>( \alpha_{xx} = \frac{T}{I_{xx}} = \frac{50 \times 10^{-3}}{1.034 \times 10^6} \text{ rad/\text{sec}^2} )</td>
</tr>
<tr>
<td>( T )</td>
<td>( \alpha_{yy} = \frac{T}{I_{yy}} = \frac{50 \times 10^{-3}}{7.643 \times 10^6} \text{ rad/\text{sec}^2} )</td>
</tr>
<tr>
<td>( T )</td>
<td>( \alpha_{zz} = \frac{T}{I_{zz}} = \frac{50 \times 10^{-3}}{7.838 \times 10^6} \text{ rad/\text{sec}^2} )</td>
</tr>
</tbody>
</table>

(cont)
Crew Motion Disturbance - (Cont)

Crew motion disturbance in the X-Z plane will produce deflections of the telescope tube support structure and thus pointing errors. (For small tilt errors the IMC will make necessary corrections)

Pointing Error from Crew Motion in X-Z Plane

![Diagram of telescope tube and pointing error]

\[ a = 10 \alpha_y = 4 \times (1.05 \times 10^{-4}) = 4.2 \times 10^{-4} \text{ rad/sec}^2 \]

\[ \alpha_y = 1.05 \times 10^{-4} \text{ rad} \]

ACCELERATION OF TELESCOPE TUBE

\[ \theta = \text{Pointing Error (0.0017)} \]

\[ \Delta = \text{gross image displacement from pointing error (} \theta \text{)} = \text{Plate Scale} \times \theta \]

\[ = 4.35 \times 10^{-6} \text{ rad} \times \frac{\text{sec} \times \text{mm}}{0.3727} \times 0.0017 = 0.0012 \text{ mm} (12 \mu) \]

\[ Y = \text{net image displacement from pointing error} = 0.0012 \text{ mm - 0.00006} = 0.0011 \text{ mm} \]

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR.
Crew Motion Disturbance (Cont)

Image Displacement from Crew Motion X-Y Plane

Crew motion disturbance in the X-Y plane will produce parallelogram deflection of the telescope tube fork times and slight bending of the support structure rotation axis.

\[ \delta' = \text{deflection from 1g load (see deflection calcs)} = 2.19 \times 10^{-4} \text{ m} (0.086 \text{ in}) \]

\[ \delta = \text{deflection from crew motion (1000, 0429 g)} = 4.29 \times 10^{-5} \text{ rad} \]

\[ \theta' = \text{rotation from 1g load (see deflection calcs)} = 9.99 \times 10^{-5} \text{ rad} \]

\[ \theta = \text{rotation from crew motion (1000, 0429 g)} = 4.29 \times 10^{-5} \left( \frac{9.99 \times 10^{-5} \text{ rad}}{g} \right) = 4.28 \times 10^{-9} \text{ rad} (0.000887) \]

\[ \Delta = \text{gross image displacement} = \theta' \times 1.5 \text{ m} = 6.42 \times 10^{-9} \text{ m} (2.53 \times 10^{-7} \text{ in}) \]

\[ y = \text{net image displacement} = \Delta - \delta = 6.42 \times 10^{-9} - 9.39 \times 10^{-9} = -2.97 \times 10^{-9} \text{ m} (0.03 \text{ in}) \]

(NEGLC)
WEIGHTS & C.G.

<table>
<thead>
<tr>
<th>ITEM</th>
<th>NAME</th>
<th>WEIGHT (KG)</th>
<th>LB</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Telescope Tube</td>
<td>453.6</td>
<td>1000</td>
</tr>
<tr>
<td>2</td>
<td>Tilt Drive + BRG</td>
<td>6.8</td>
<td>15</td>
</tr>
<tr>
<td>3</td>
<td>Tilt Bearing</td>
<td>4.5</td>
<td>10</td>
</tr>
<tr>
<td>4</td>
<td>Tine</td>
<td>15.9</td>
<td>35</td>
</tr>
<tr>
<td>5</td>
<td>Tine</td>
<td>15.9</td>
<td>35</td>
</tr>
<tr>
<td>6</td>
<td>Fork Ring</td>
<td>99.8</td>
<td>220</td>
</tr>
<tr>
<td>7</td>
<td>Moment BRG.</td>
<td>22.7</td>
<td>50</td>
</tr>
<tr>
<td>8</td>
<td>Encoder Assy</td>
<td>4.5</td>
<td>10</td>
</tr>
<tr>
<td>9</td>
<td>Tachometer</td>
<td>3.6</td>
<td>8</td>
</tr>
<tr>
<td>10</td>
<td>Motor</td>
<td>6.8</td>
<td>15</td>
</tr>
<tr>
<td>11</td>
<td>Bearing</td>
<td>5.4</td>
<td>12</td>
</tr>
<tr>
<td>12</td>
<td>Housing</td>
<td>67.6</td>
<td>149</td>
</tr>
<tr>
<td>13</td>
<td>Aft Cone Head</td>
<td>34.0</td>
<td>75</td>
</tr>
<tr>
<td>14</td>
<td>Support Cone</td>
<td>254.0</td>
<td>560</td>
</tr>
<tr>
<td></td>
<td></td>
<td><strong>595.1</strong></td>
<td>2194</td>
</tr>
</tbody>
</table>
# Drive Motor Summary

## Figure

<table>
<thead>
<tr>
<th></th>
<th>Rotation Drive</th>
<th>Tilt Drive</th>
<th>Instrument Adapter Drive</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Motor Peak Torque</strong></td>
<td>2 @ 22 = 44</td>
<td>14 lb-ft</td>
<td>200 oz-in</td>
</tr>
<tr>
<td><strong>Bearing and/or Seal Friction</strong></td>
<td>1G lb-ft</td>
<td>0.25 lb-ft</td>
<td>6270 oz-in</td>
</tr>
<tr>
<td><strong>Max Tracking Rate</strong></td>
<td>4°/min</td>
<td>4°/min</td>
<td>0.077°/sec</td>
</tr>
<tr>
<td><strong>Power to Track</strong></td>
<td>188 watts</td>
<td>0.081 watts</td>
<td>15 watts</td>
</tr>
<tr>
<td><strong>Ave Power Req'd. to Accel. to Track Rate (Approx.)</strong></td>
<td>444 watts</td>
<td>127.5 watts</td>
<td>67.5 watts</td>
</tr>
<tr>
<td><strong>Max Slew Rate</strong></td>
<td>5°/sec</td>
<td>5°/sec</td>
<td></td>
</tr>
<tr>
<td><strong>Power to Slew</strong></td>
<td>190 watts</td>
<td>0.11 watts</td>
<td></td>
</tr>
<tr>
<td><strong>Ave Power Req'd. to Accel. to Slew Rate (Approx.)</strong></td>
<td>445 watts</td>
<td>127.5 watts</td>
<td>67.5 watts</td>
</tr>
<tr>
<td><strong>Max Accel. Available</strong></td>
<td>8.3°/sec²</td>
<td>4.6°/sec²</td>
<td>1363°/sec²</td>
</tr>
</tbody>
</table>

**Note:**

- Direct Drive Motor
- Direct Drive Motor
- 100:1 Gear Ratio
**Rotation Drive**

**Inertia Torque About X-Axis**

<table>
<thead>
<tr>
<th>ELEMENT</th>
<th>NAME</th>
<th>WEIGHT (LB)</th>
<th>$K^2$ (ft$^2$)</th>
<th>$WK^2$ (lb-ft$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Telescope Tube</td>
<td>1000</td>
<td>5.50</td>
<td>5500</td>
</tr>
<tr>
<td>2</td>
<td>Tilt Drive+Bag</td>
<td>15</td>
<td>6.04</td>
<td>91</td>
</tr>
<tr>
<td>3</td>
<td>Tilt Bearing</td>
<td>10</td>
<td>5.25</td>
<td>53</td>
</tr>
<tr>
<td>4</td>
<td>Tine</td>
<td>35</td>
<td>3.20</td>
<td>112</td>
</tr>
<tr>
<td>5</td>
<td>Tine</td>
<td>35</td>
<td>3.20</td>
<td>112</td>
</tr>
<tr>
<td>6</td>
<td>Fork Ring</td>
<td>220</td>
<td>1.41</td>
<td>310</td>
</tr>
<tr>
<td>7</td>
<td>Moment Bag (½)</td>
<td>25</td>
<td>2.54</td>
<td>64</td>
</tr>
<tr>
<td>8</td>
<td>Encoder + BEK</td>
<td>10</td>
<td>1.00</td>
<td>10</td>
</tr>
</tbody>
</table>

Moment of Inertia

$$I_{xx} = \frac{WK^2}{32.2} = \frac{6258}{32.2} = 194 \text{ lb-ft-sec}^2$$

Acceleration

From Table 5-4 Ref. 1: Max Accel. from Large Thruster Firing

$$\alpha = 1.46 \text{% sec}^{-2} (0.0255 \text{ rad/sec}^2)$$

Torque to Accelerate

$$T = I\alpha = 194 \text{ lb-ft-sec}^2 \times 0.0255 \text{ rad/sec}^2 = 4.95 \text{ lb-ft}$$
**Rotation Drive (Cont.)**

**Moment Bearing Friction Torque**

<table>
<thead>
<tr>
<th>Description</th>
<th>Friction Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Preload Only (1000 lb-in)</td>
<td>15 lb-ft</td>
</tr>
<tr>
<td>2. Moment + Radial (1000 lb, 24,000 lb-in)</td>
<td>16 lb-ft</td>
</tr>
<tr>
<td>3. Axial Only (1000 lb)</td>
<td>8.4 lb-ft</td>
</tr>
</tbody>
</table>

*From: Kaydon Bearing Co. For 4 pt. Bearing No. 6684-1*
Elevation Drive Motor

2 Motors Used in Parallel
Magnetic Technology Model 12250-160 22-lb-ft (Ea)
Peak Torque \( T_p = 21.875 \text{ lb-ft/Motor} \)
Motor Constant \( K_m = 1.17 \frac{\text{lb-ft}}{\text{Watt/Motor}} \)
Resistance \( R_m = 1.7 \text{ ohms/Motor} \)
Voltage at \( T_p \) \( V_p = 24.5 \text{ Volts/Motor} \)
Torque Sensitivity \( K_t = 1.52 \text{ lb-ft/Amp/Motor} \)
Back EMF \( K_b = 2.0 \text{ Volts/rad/sec/Motor} \)
Power at Peak Torque \( P_p = 350 \text{ Watts/Motor} \)

Max Friction Torque \( = 16 \text{ lb-ft} \)

Max Tracking Rate \( = 4^\circ/\text{min} = 0.0116 \text{ rad/sec} \)
Max Slew Rate \( = 5^\circ/\text{sec} = 0.872 \text{ rad/sec} \)

Torque Available to Accelerate \( T_a = 44 - 16 = 28 \text{ lb-ft} \)
Max Acceleration Available \( \alpha_{max} \)

\( \alpha_{max} = \frac{28}{194 \text{ lb-ft/sec}^2} = 0.144 \text{ rad/sec}^2 \)
\( \Rightarrow 8.3^\circ/\text{sec}^2 \)

Power Required to Track at \( 4^\circ/\text{min} \) (Overcome Brg Friction)

Back EMF \( = 0.0016 \text{ rad/sec} \times 2 \text{ Volts/rad/sec} = 0.0032 \text{ Volts} \)
Current Req \( 16 \text{ lb-ft} \)
\( I_a = \frac{16}{1.52 \text{ lb-ft/Amp}} = 10.53 \text{ Amps} \)
Voltage \( V = I_a \times R_m = 10.53 \times 1.7 = 17.89 \text{ Volts} \)
Total Voltage \( V_t = 100.23 + 17.89 = 118 \text{ Volts} \)
Power \( = 10.53 \text{ Amps} \times 17.89 \text{ Volts} = 188 \text{ Watts} \)

Power Required to Slew at \( 5^\circ/\text{sec} \) (Overcome Brg. Friction)

Back EMF \( = 0.0872 \text{ rad/sec} \times 2 \text{ Volts/rad/sec} = 0.174 \text{ Volts} \)
Current Req \( 16 \text{ lb-ft} \)
\( I_a = \frac{16}{1.52 \text{ lb-ft/Amp}} = 10.53 \text{ Amps} \)
Voltage \( V = I_a \times R_m = 10.53 \times 1.7 = 17.89 \text{ Volts} \)
Total Voltage \( V_t = 0.174 + 17.89 = 18.06 \text{ Volts} \)
Power \( = 10.53 \text{ Amps} \times 18.06 = 190 \text{ Watts} \)

(cont)
Rotation Drive Motor (cont)

Average Power Required to Accelerate to Track Speed

Peak Torque Power $P_p = 2 @ 350$ Watts = 700 Watts
(2 motors @ Stall)
Power Required to Track at 4°/min = 188 Watts

Average Power to Accel. $P_{ave} = \frac{700 - 188}{2} + 188 = 444$ Watts
(Approx)

Average Power Required to Accelerate to Slew Speed

Power Required to Slew at 5°/sec = 190 Watts

Average Power to Accel. $P_{ave} = \frac{700 - 190}{2} + 190 = 445$ Watts
(Approx)

Minimum Time to Accelerate to Slew Speed (5°/sec)

Max Acceleration Rate $a_{\text{max}} = 8.3°/\text{sec}^2$ (see preceding)

Min. Time to Accel. $t_{\text{min}} = \frac{w}{a} = \frac{5°/\text{sec}}{8.3°/\text{sec}^2} = 0.6$ sec
TILT DRIVE

\[ I_{yy} = \frac{53 \text{ lb-ft-sec}^2}{(34 + 19) \text{ M}^2} \]

\[ I_{yy} = \frac{102 \text{ lb-ft-sec}^2}{(24 + 78) \text{ M}_o \text{ M}^2} \]

INERTIA TORQUE

Total \( I_{yy} \approx 171 \text{ lb-ft-sec}^2 \)

\[ \text{ACCEL}(\alpha) = 2.05^\circ/\text{sec}^2 \left(0.0358 \text{ rad/sec}^2 \right) \] (Table 5-4)

VEL. = 4^\circ/\text{MIN.}

TORQUE REQD = I\alpha

= 6.1 \text{ lb-ft}

WT. (TOTAL) = 1000 \text{ lb}

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR
TILT DRIVE MOTOR

INLAND TORQUE MOTOR
T = 579.5 14.16-ft

PEAK TORQUE
T_P = 14 LB-FT

Motor Constant
K_m =

Resistance
R_m = 1.4 Ω

Voltage at T_P
V_P = 18.9 Volts

Sensitivity
K_T = 1.04 lb-ft/amp

Back EMF
K_B = 1.41 V/rad/sec

Power & Peak Torque
P_P = 255 Watts

 Bearing Friction
C = .25 lb-ft

Inertia
J_I = 171 lb-ft/sec^2

Max Track Rate
U_T = 4°/min (.0016 rad/sec)

Max Slew Rate
U_S = 5°/sec (.0872 rad/sec)

Torque Available to Accelerate
T = 14 - .25 = 13.75 lb-ft

Max Accel Available
a = T/J_I = 13.75/171 lb-ft/sec^2 = .080 rad/sec^2

Power Required to Track at 4°/min (Overcome Friction)

Back EMF
V_B = .0016 x 1.41 V/rad/sec = .0016 Volts

Current for 25 LB-FT
I_A = 125 LB-FT / 1.04 lb-ft/amp = .24 AMP

Voltage
V_A = I_A R_m = .24 x 1.4 = 3.36 Volts

Total Voltage
V_T = V_B + V_A = .0016 + .336 = .338 Volts

Power
P = V_T x I_A = .338 x .24 = .081 Watts

Power Required to Slew at 5°/sec (Overcome Friction)

V_B = .0872 rad/sec x 1.41 V/rad/sec = .123 Volts

I_A = .25 LB-FT / 1.04 lb-ft/amp = .24 AMP

Voltage
V_A = I_A R_m = .24 x 1.4 = 3.36 Volts

Total Voltage
V_T = V_B + V_A = .123 + .336 = .459 Volts

Power
P = V_T x I_A = .459 x .24 = .11 Watts

(cont.)
TILT DRIVE MOTOR (CONT)

**Ave Power Required to Accelerate to Track Speed (4%/min)**

**Peak Torque Power** \( P_p = 255 \text{ WATTS} \)

**Power Required to Track at 4%/min** \( = 0.81 \text{ WATTS} \) (See preceding page)

**Ave Power to Accel** \( \text{Ave} = \frac{255 - 0.81}{2} + 0.81 = 127.5 \text{ WATTS} \) (approx.)

**Ave Power Required to Accelerate to Slew Speed (5%/sec)**

**Power Required to Slew at 5%/sec** \( = 11 \text{ WATTS} \)

**Ave Power to Accel** \( \text{Ave} = \frac{255 - 11}{2} + 11 = 127.5 \text{ WATTS} \) (approx. 1.1 sec) (See below)

**Minimum Time to Accelerate to Slew Speed (5%/sec)**

**Max. Accel. Rate** \( \alpha_{\text{max}} = 4.6 \text{ %/sec}^2 \) (See preceding page)

**Min. Time to Accel** to Slew Rate \( \tau_{\text{min}} = \frac{W}{\alpha} = \frac{5.0 \text{ %/sec}}{4.6 \text{ %/sec}^2} = 1.1 \text{ sec} \)
INSTRUMENT ADAPTER DRIVE MOTOR

MOTOR - INLAND "TORQUE-TACH" NO. TT-1810

PEAK TORQUE \( T_p = 200 \text{ oz-in} \)

POWER @ \( T_p \) \( P_p = 120 \text{ watt} \)

MOTOR CONSTANT \( K_m = 18.3 \text{ oz-in/watt} \)

VOLTAGE AT \( T_p \) \( V_p = 27.5 \text{ volts} \)

RESISTANCE \( R_m = 6.4 \text{ ohms} \)

TORQUE SENSITIVITY \( K_T = 47 \text{ oz-in/amp} \)

BACK EMF \( K_B = 0.332 \text{ V/rad/sec} \)

NO LOAD SPEED \( \omega_m = 210 \text{ rad/sec} \) \((3.66 \text{ RPG} = 220 \text{ RPM})\)

CONTINUOUS STALL TORQUE \( T_s = 72 \)

ADAPTER FRICTION TORQUE

SEALS \((3 \text{ O-rings} \times 100 \text{ in-lb ea.)} \) \( T_S = 3 \times 100 \times 16 = 624 \text{ oz-in} \)

4-PT BEARING \((\text{KAYDON KD09XPA}) \) \( T_B = 30 \text{ oz-in} \)

MAX TRACK RATE \( (\text{combine free & orbital drift}) \)

\( .01 \text{ %sec} \) FREE DRIFT

\( .067 \text{ %sec} \) ORBITAL DRIFT

\( .077 \text{ %sec} \) \((= .000214 \text{ deg/sec} = .0128 \text{ RPM} = .00134 \text{ rad/sec})\)

GEAR RATIO \( = 100:1 \)

MOTOR TORQUE REQUIRED TO OVERCOME FRICTION

\[
T_f = \frac{6240 + 30}{100 \times .96^3} = \frac{6270}{100 \times .88} = 70.9 \text{ oz-in}
\]

MOTOR TORQUE AVAILABLE TO ACCELERATE

\[
T_a = \frac{200 - 70.9}{100 \times .96^3} = 129 \text{ oz-in at motor}
\]

\[
= 129 \times 100 \times .96^3 = 11,352 \text{ oz-in at adapter center line}
\]
INSTRUMENT ADAPTER DRIVE MOTOR (CONT)

Max. Accel. From Avail. Torque

\[ J = \frac{80 \text{ lb-ft}^2}{32.2} = 2.48 \text{ lb-ft} \cdot \text{sec}^2 \]

Torque at Adapter \( \phi \) 

\[ T_e = 11352 \text{ oz-in} \]

Accel. 

\[ \alpha = \frac{T}{J} = \frac{11352}{477} = 23.8 \text{ rad/sec}^2 \text{ (1363 \text{ \%}/\text{sec}^2) } \]

Max Motor Speed @ Track Rate = 0.077\%/sec

\[ U_m = 100 \times 0.077\% = 7.7 \text{ \%} = \frac{0.021}{\text{RPM}} \]

\[ \text{Max Allow. Motor Speed} = 210 \text{ rad/sec} \]

\[ = \frac{0.134}{\text{sec}} \]

Power Required To Track @ 0.077\%/sec

Neglect Back EMF, Track Speed \( \propto \) Stall Speed

Motor Constant @ Stall \( K_m = 18.3 \frac{\text{oz-in}}{\text{Watt}} \)

\[ WATT = \left( \frac{\text{oz-in}}{18.3} \right)^2 \]

\[ \text{Power} = \left( \frac{20.9 \text{ oz-in}}{18.3} \right)^2 = 15 \text{ Watts} \]

Ave. Power To Accel. To Track Speed (Approx)

\[ P_{\text{ave}} = \frac{\text{Peak Torque Power} - \text{Track Power}}{2} + \text{Track Power} \]

\[ = \frac{120 - 15}{2} + 15 = 52.5 + 15 = 67.5 \text{ Watts} \]

Min Time To Accel. To Track Speed (0.077\%/sec)

\[ t = \frac{U}{\alpha} = \frac{0.077\%}{1363 \text{ \%}/\text{sec}^2} = 0.05 \text{ sec} \]
LOADING IN X-AXIS

F_x

+Y

X

-Y

F_x

NASA / JSC

AFPT FOR SWOT

DEF - X - 1

PERKIN-ELMER
BOLLER & CHIVENS DIVISION
916 MERIDIAN AVENUE
SOUTH PASADENA, CALIFORNIA 91030
TELEPHONE: (213) 682-3391

DESIGN ANALYSIS

DATE 2-27-76

BY D. R. McLaughlin

JOB No. 59060

SHEET... OF

TELEPHONE: (213) 682-3391
LOADING IN X-AXIS

Fork Tines

\[ \delta_4 = 2.50 \times 10^{-6} \text{ m} \quad (0.000,094 \text{ in}) \]

Fork Ring

\[ \delta_3 = \text{NEGligible} \]

Bearing

\[ \delta_2 = 2.24 \times 10^{-6} \text{ m} \quad (0.000,088 \text{ in}) \]

Support Cone

\[ \delta_1 = 95.5 \times 10^{-6} \text{ m} \quad (0.0038 \text{ in}) \]

\[ F_x = 4448 \text{ N} \quad (1000 \text{ LB}) \]

\[ F_x+ = 5340 \text{ N} \quad (1200 \text{ LB}) \]

**TOTAL DEFLECTION AT LOAD**

\[ \delta_x = \sum \delta_{i,j} = (95.5 + 2.24 + 0 + 2.50) \times 10^{-6} = 100.26 \times 10^{-6} \text{ m} \]

\[ (0.0039 \text{ in}) \]

**NATURAL FREQUENCY**

\[ f_n = \frac{1}{2\pi} \sqrt{\frac{9.8 \text{ m/s}^2}{\frac{g}{\delta_x}}} = \frac{1}{2\pi} \sqrt{\frac{317 \text{ m/s}^2}{\frac{9.8 \text{ m/s}^2}{1.022 \times 10^{-6} \text{ m}}}} = 50.5 \text{ Hz} \]
SUPPORT CONE DEFLECTION
LOADING IN X-AXIS

\[ F_x = 5340 \text{ N (1200 lb)} = 1 g \]

\[ a = 2.04 \text{ m (80.45 in)} \]
\[ b = 0.65 \text{ m (25.6 in)} \]
\[ t = 0.0035 \text{ m (0.138 in)} \]
\[ h = 0.80 \text{ m (31.5 in)} \]
\[ E = 7.31 \times 10^{10} \text{ N/m}^2 \]

Consider cone as a Belleville spring without benefit of mounting flanges, top and bottom, and restricting support of the Space Lab mounting flange (this would be a "worst case" condition).

Axial Deflection of Cone

\[ \delta_x = \frac{F_x a^2 (1-\nu)}{E h^2 t} \quad \text{for} \quad \delta \ll h \]

\[ \nu = \text{Poisson's Ratio} = 0.33 \]
\[ M = \text{Constant} = 0.79 \quad \text{for} \quad \rho = 3.14 \]

\[ = \frac{5340 (1.0)(2.04^2)(1-0.33^3)}{7.31 \times 10^4 (0.8^2)(0.0035)} = 9.55 \times 10^{-5} \text{ m (0.0038 in)} \]
MOMENT BEARING (4-POINT) - LOADING IN THE X-AXIS

OUTER BEARING - KAYDON 6684-1

AXIAL LOAD \( F_x = 5340 \text{ N (1200 LB)} \)

BEARING AXIAL DEFLECTION (FROM KAYDON FORMULA DEVELOPED FOR THIS BEARING WITH 0.0005 IN. PRELOAD)

\[
\Delta_x = 7.353 \times 10^{-8} \text{ in/lb (1200 lb)} = 8.82 \times 10^{-5} \text{ in} = 2.24 \times 10^{-6} \text{ m}
\]
Fork Ring Deflection - Loading in the X-Axis

Ring Rotation at Fork Tine (if tines were free to move laterally)

\[
\beta = \frac{\pi r T_0 (1 + \lambda)}{B EI_y} = \frac{\pi (1.3)(972)(1 + .97)}{B (7.31 \times 10^6)(4.49 \times 10^5)} = \frac{68.73 \times 10^{-6}}{\text{rad}} \quad (14^\circ) \quad \text{(Ref Only)}
\]

Lateral Tine Movement (if free to move) at Telescope Tube

\[
\Delta_y = 0\beta = 0.762 (68.73 \times 10^{-6}) = 5.24 \times 10^{-5} \text{ m} \quad (0.002 \text{ in})
\]

Resisting Force &\ 90° of Telescope Tube at Fork Tine

\[
F_y = \frac{r F_x}{a} = \frac{2225 \times 437}{0.762} = 1276 \text{ N} \quad (287 \text{ lb})
\]

Conclusion:

If tines are restrained laterally (F_x = 1276 N) by the telescope tube, there will be no axial displacement of the telescope tube from fork ring deflection.
Fork Deflection - Loading in X-Axis

Telescope Tube Reaction

\[ R_y = \frac{F_x \cdot \alpha}{\lambda} = \frac{2224 \cdot 0.356}{0.762} = 1276 \text{ N (287 lb)} \]

Fork tine will act as a column with the telescope tube reacting to the lateral force produced by the offset, \( \lambda \).

Axial Tine Deflection

\[ \delta_x = \frac{F_x \cdot \alpha}{A \cdot E} = \frac{1276 \cdot 0.356}{0.0053 \cdot 7.31 \times 10^{10}} = 2.5 \times 10^{-6} \text{ m} = 9.88 \times 10^{-3} \text{ in} \]
LOADING IN Y-AXIS

\[ F_y \]
DEFLECTION & NATURAL FREQUENCY

LOADING IN THE Y-AXIS

\[ F_y = 4448 \text{ N (1000 lb)} \]
\[ F_y^+ = 5340 \text{ N (1280 lb)} \]
\[ g = 9.8 \text{ m/s}^2 \]

**Support Cone**

**Telescope Tube**

**Bearing**

**Fork Ring**

**Fork Tines**

**Total Deflection at Load:**

\[ \delta_y = \Sigma \delta = (14.58 + 8.44 + 58.1 + 127) \times 10^{-6} = \frac{208.12 \times 10^{-6}}{0.208 \text{ mm}} \]

**Natural Frequency**

\[ f_y = \frac{1}{2\pi} \sqrt{\frac{g \Sigma F_y}{2.2 \pi \delta_y}} = \frac{1}{2\pi} \sqrt{\frac{9.8 \text{ m/s}^2}{208.12 \times 10^{-6}}} = \frac{34.5}{17.7} \text{ Hz} \]

**Scope**

\[ \theta_y = \Sigma \theta = (4.65 + 4.98 + 16.3 + 0) \times 10^{-6} = 35.93 \times 10^{-6} \text{ rad} (17.7 \text{ m}) \]

**Assume Telescope Does Not Rotate Because of Parallelogram Motion**
Support Cone Deflection
Loading in the Y-Axis

\[ R_0 = 0.65 \text{ m} \ (25.6 \text{ in}) \]
\[ R_L = 2.04 \text{ m} \ (80.45 \text{ in}) \]
\[ L = 0.80 \text{ m} \ (31.5 \text{ in}) \]
\[ t = 0.0035 \text{ m} \ (0.138 \text{ in}) \]
\[ F_y = 5340 \text{ N} \ (1200 \text{ lb}) \]
\[ L = 0.95 \text{ m} \ (37.5 \text{ in}) \]
\[ M = F_L = 5086 \text{ N m} \]

From End Moment, \( M \)

\[ \text{Slope} \quad \phi_x = -\frac{M L (R_L + R_0)}{2 E t_c R_0 R_L^2} = \frac{5086 (0.8) (2.04 + 0.65)}{2 (7.31 \times 10^{10}) (0.0035) (3.14) (2.04) (0.65)^2} = -3.87 \times 10^{-6} \text{ RAD} \ (\theta 97 \text{ m}) \]

\[ \text{Deflection} \quad y_x = \frac{M L^2}{2 E t_c R_0 R_L^2} = \frac{5086}{2 (7.31 \times 10^{10}) (0.0035) (3.14) (0.65) (2.04)^2} = 7.48 \times 10^{-7} \text{ m} \ (0.000029 \text{ in}) \]

From End Load, \( F_y \)

\[ \text{Slope} \quad \phi_y = -\frac{F}{2 E t_c R_0} \left( \frac{L}{R_L} \right)^2 = \frac{5340}{2 (7.31 \times 10^{10}) (0.0035) (3.14) (0.65) (2.04)} = -785 \times 10^{-6} \text{ RAD} \]

\[ \text{Deflection} \quad y_y = \frac{2 F}{3 E t_c} \left( \frac{L}{R_L + R_0} \right)^3 \left[ 1 + \frac{3}{4} \left( 1 + \frac{R_0}{R_L} \right)^2 \right] \]
\[ = \frac{2 (5340)}{3 (7.31 \times 10^{10}) (0.0035) (3.14) (2.04 + 0.65)} \left[ 1 + \frac{3}{4} \left( 1 + \frac{0.65}{2.04} \right)^2 \right] \]
\[ = 2.68 \times 10^{-7} \text{ m} \ (0.000010 \text{ in}) \]

(CONT)
Support Cone Deflection (Cont) (Y-Axis Loading)

Telescope Tube Rotation from Cone Deflection

\[ \delta_T = \delta_m + \delta_c = 3.87 \times 10^{-6} + 0.78 \times 10^{-6} = 4.65 \times 10^{-6} \text{ rad.} \]

Telescope Tube Deflection from Cone Deflection + Rotation

\[ \delta_y = \delta_T L = 4.65 \times 10^{-6} (1.95) = 4.42 \times 10^{-6} \text{ m} \]

\[ \delta_{FM} = Y_m + Y_c = 7.48 \times 10^{-7} + 2.68 \times 10^{-7} = 10.16 \times 10^{-7} \text{ m (0.000040 in)} \]

\[ \delta_y = \delta_T + \delta_{FM} = 14.58 \times 10^{-6} \text{ m (0.00057 in)} \]
Moment Bearing (4-Pr) - Loading in the Y-Axis

Outer Bearing

Moment Loading

\[ F_y = 5340 \text{ N} \quad (1200 \text{ LB}) \]
\[ L = 0.89 \text{ m} \quad (35 \text{ in}) \]
\[ M = F_L = 4747 \text{ N} \cdot \text{m} \quad (42000 \text{ LB-IN}) \]

Bearing Deflection from Moment Loading:

\[ \theta_m = \frac{M \times C_m}{W_m} \quad \text{where} \quad C_m = 1.187 \times 10^{-10} \frac{\text{rad}}{\text{in}^2} \text{ from graph developed by Kaydon Bearings for 0.005 in. preloaded 4-Pr bearing} \]

\[ \theta_m = 42000 \left( 1.187 \times 10^{-10} \frac{\text{rad}}{\text{in}^2} \right) = 4.98 \times 10^{-6} \text{ rad} \quad (1^\circ) \]

Bearing Deflection from Radial Loading

\[ \Delta_r = F_y C_r \quad C_r = 1.316 \times 10^{-7} \frac{\text{in}}{\text{lb}} \]

\[ = 1200 \text{ LB} \times 1.316 \times 10^{-7} \frac{\text{in}}{\text{lb}} = 0.000158 \text{ in} \quad (4.01 \times 10^{-6} \text{ m}) \]

Deflection at Telescope Tube from Moment Load:

\[ \delta_y = \theta_m \times L = 4.98 \times 10^{-6} \times 0.89 = 4.43 \times 10^{-6} \text{ m} \quad (0.0017 \text{ in}) \]

Total Deflection at Telescope Tube from Mom. & Radial Loads

\[ \delta_y = \delta_r + \delta_y = 4.01 \times 10^{-6} \text{ m} + 4.43 \times 10^{-6} \text{ m} = 8.44 \times 10^{-6} \text{ m} \quad (0.0033 \text{ in}) \]
FORK RING DEFLECTION Y-AXIS LOADING

Consider the fork ring under equal and opposite twisting moments from loading of the fork tines in the Y direction.

Case 1 will be used to determine the rotation angle.

**Case 1**

Actual Loading

Case 1 has zero slope at the mid position ($\theta = 90^\circ$). By method of superposition, one half the ring can be inverted and aligned with the end points and mid-position on the ring center line plane.

\[
K = 1.217 \times 10^{-4} \text{ m}^4
\]

\[
r^2 = .3 \text{ m}
\]

\[
I_Y = 4.49 \times 10^{-5} \text{ m}^4
\]

\[
E = 7.31 \times 10^{10} \text{ N/m}^2
\]

\[
G = 2.76 \times 10^{10} \text{ N/m}^2
\]

\[
\lambda = \frac{E I_Y}{K} = .97
\]

\[
F_1 = 2225 \text{ N (500 lb)}
\]

\[
I_Y = .762 \text{ m (30 in)}
\]

\[
T_{0Y} = F_1 l_Y = 1695.5 \text{ Nm}
\]

(Cont)
FORK RING DEFLECTION (CONT)

\[
\Delta_Y = \frac{r^2 T_{ov} (1 + \lambda) (\pi - 2)}{8EI} = \frac{1.3^2 (1694.5)(1.97)(1.144)}{8(7.31 \times 10^9)(4.49 \times 10^{-5})} = 13.05 \times 10^{-6} \text{ m (0.00051 in)}
\]

\[
\sigma_Y = \frac{\Delta_Y}{\rho} = \frac{13.05 \times 10^{-6}}{0.03} = 4.35 \times 10^{-5} \text{ rad}
\]

\[
\beta_Y = \frac{\pi r^2 T_{ov}(1 + \lambda)}{8EI} = \frac{\pi (1.3)(1694.5)(1.97)}{8(7.31 \times 10^9)(4.49 \times 10^{-5})} = 11.98 \times 10^{-5} \text{ rad}
\]

RING ROTATION @ FORK TINE:

\[
\phi_Y = \beta - \alpha = 7.63 \times 10^{-5} \text{ rad (15.7 m)}
\]

TELESCOPE TUBE DEFLECTION AT END OF FORK TINE

\[
\delta_Y = \phi_Y \cdot \Delta_Y = (7.63 \times 10^{-5}) (0.762) = 5.81 \times 10^{-5} \text{ m (0.0022 in)}
\]
Fork Deflection - Loading in Y-Axis

\[ F_y = 2224 \text{ N (500 lb)} \]

*\( \text{Ave } I_{Y,y} = 9.3 \times 10^{-5} \text{ m}^4 \) (223 in\(^4\))

*\( \text{Ave } I_{Z,z} = 4.6 \times 10^{-5} \text{ m}^4 \) (110 in\(^4\))

*\( K = 1.53 \times 10^{-4} \text{ m}^4 \) (367 in\(^4\))

*See Sits for Fork Time Properties

Consider Time a Simple Cantilever with Effective Length of Curved Area, \( L \).

**Deflection**

\[ \delta_y = \frac{F_y L^3}{3 E I_z} = \frac{2224 (0.83)^3}{3 (7.31 \times 10^7) (4.6 \times 10^5)} = 1.27 \times 10^{-4} \text{ m} \ (0.005 \text{ in}) \]

**Slope**

\[ \theta_y = \frac{F_y L^2}{2 E I_z} = \frac{2224 (0.83)^2}{2 (7.31 \times 10^7) (4.6 \times 10^5)} = 228 \times 10^{-6} \text{ rad} \ (47^\circ) \]

Note: Tines Will Approximately Rotate As A Parallelogram System and Thus Produce No Rotation of The Telescope Tube
LOADING IN Z-AXIS
LOADING IN Z-AXIS

\[ F_z = 4448 \text{ N (1000 lb)} \]

\[ F_{z1} = 5340 \text{ N (1200 lb)} \]

**Total Deflection at Load:**

\[ \delta_x = \sum \delta_x = (14.58 + 8.44 + 41.66 + 66.6) \times 10^{-6} = 131.28 \times 10^{-6} \text{ m (0.0052 in)} \]

**Natural Frequency**

\[ f_x = \frac{1}{2\pi} \sqrt{\frac{3 \sum F_{z1}}{2 \delta_x}} = \frac{1}{2\pi} \sqrt{\frac{9.8 \text{ m/s}^2}{1.313 \times 10^{-6} \text{ m}}} = 43.5 \text{ Hz} \]

**Slope**

\[ \theta_x = \sum \theta_x = (4.65 + 4.98 + 54.67 + 118) \times 10^{-6} = 182.30 \times 10^{-6} \text{ rad} \]
Support Cone Deflection

Loading in the Z-Axis

\[ F \]

\[ R_0 = 0.65 \text{ m (25.6 in)} \]
\[ R_L = 2.04 \text{ m (80.45 in)} \]
\[ L = 0.80 \text{ m (31.5 in)} \]
\[ t = 0.0035 \text{ m (0.138 in)} \]
\[ F = 5340 \text{ N (1200 lb)} \]
\[ L = 0.95 \text{ m (37.5 in)} \]
\[ M = F L = 5086 \text{ N}\cdot\text{m} \]

From End Moment, \( M \)

Slope \[ \phi_s = -\frac{M L (R_L + R_0)}{2 E t C \epsilon R_L R_0^2} = \frac{5086 (0.8) (2.04 + 0.65)}{2 (7.31 \times 10^4) (0.0035) (3.14) (2.04^2) (0.65)^2} \]
\[ = -3.87 \times 10^{-6} \text{ rad (0.79 m)} \]

Deflection \[ y_s = \frac{M L^2}{2 E t C \epsilon R_L R_0^2} = \frac{5086}{2 (7.31 \times 10^4) (0.0035) (3.14) (0.65) (2.04)} \]
\[ = 7.48 \times 10^{-7} \text{ m (0.000029 in)} \]

From End Load, \( F \)

Slope \[ \phi_F = -\frac{F L}{2 E t C R_0} \left( \frac{R_L}{R_0} \right)^2 = \frac{-5340}{2 (7.31 \times 10^4) (0.0035) (3.14) (0.65) (2.04)} \]
\[ = -7.85 \times 10^{-6} \text{ rad} \]

Deflection \[ y_F = \frac{2 F}{3 E t C} \left( \frac{L}{R_L} \right) \left( \frac{R_L + R_0}{R_L} \right)^{\frac{3}{2}} \left[ 1 + \frac{3}{4} \left( \frac{R_L + R_0}{R_L} \right)^2 \left( 1 + \frac{R_L}{R_0} \right)^2 \right] \]
\[ = \frac{2 (5340)}{3 (7.31 \times 10^4) (0.0035) (3.14) (2.04 + 0.65)} \left[ 1 + \frac{3}{4} \left( \frac{0.65}{2.04} \right)^2 \right] \]
\[ = 2.68 \times 10^{-7} \text{ m (0.0000104 in)} \]

(Cont)
Support Cone Deflection (Cont)

(2-axis loading)

Telescope Tube Rotation from Cone Deflection

\[ \theta_t = \theta_m + \theta_e = 3.87 \times 10^{-6} + 0.18 \times 10^{-6} = 4.65 \times 10^{-6} \text{ rad.} \]

Telescope Tube Deflection from Cone Deflection + Rotation

\[ \delta_y = \theta_t l = 4.65 \times 10^{-6} \times 0.95 = 4.42 \times 10^{-6} \text{ m} \]

\[ \delta_{FM} = y_m + y_e = 7.48 \times 10^{-7} + 2.68 \times 10^{-7} = 10.16 \times 10^{-7} \text{ m (0.000040 in)} \]

\[ \delta_z = \delta_y + \delta_{FM} = 14.58 \times 10^{-6} \text{ m (0.00057 in)} \]
MOMENT BEARING (4-PT) - LOADING IN THE Z-AXIS

OUTER BEARING

MOMENT LOADING

\[ F_x = 5340 \text{ N} \ (1200 \text{ lb}) \]
\[ l = 1.89 \text{ m} \ (3.5 \text{ in}) \]
\[ M = F_x \times l = 5340 \times 1.89 = 4747 \text{ N.m} \ (42,000 \text{ lb-in}) \]

BEARING DEFLECTION FROM MOMENT LOADING:

\[ \theta_m = \frac{M \times C_m}{E \times I} \]
\[ \text{WHERE } C_m = 1.187 \times 10^{-10} \text{ rad in-lb from graph developed by Keydon Bros.} \]
\[ \text{for .005 in. preloaded 4-PT Brg.} \]
\[ \theta_m = 42,000 \ (1.187 \times 10^{-10}) = 4.98 \times 10^{-6} \text{ rad.} \ (1") \]

BEARING DEFLECTION FROM RADIAL LOADING:

\[ \delta_r = \frac{F_x \times C_r}{E \times I} \]
\[ \text{WHERE } C_r = 1.316 \times 10^{-7} \text{ in/lb} \]
\[ = 1200 \times 1.316 \times 10^{-7} = 0.000150 \text{ in.} \ (4.01 \times 10^{-6} \text{ m}) \]

DEFLECTION AT TELES. TUBE FROM MOMENT LOAD:

\[ \delta_m = \theta_m \times l = 4.98 \times 10^{-6} \times 1.89 = 4.43 \times 10^{-6} \text{ m} \ (0.00017 \text{ in}) \]

TOTAL DEFLECTION AT TELES. TUBE FROM MOM. & RADIAL LOADS:

\[ \delta_z = \delta_r + \delta_m = 4.01 \times 10^{-6} + 4.43 \times 10^{-6} = 8.44 \times 10^{-6} \text{ m} \]
\[ (0.00033 \text{ in}) \]
Fork Ring Deflection Z-Axis Loading

\( \alpha = \text{Fork Tine Rotation} \)

The fork ring is considered to be free standing with moment loads at the mid-section and simple supports at the extreme sections. (Ring is actually supported around its full diameter)

By Method of Superposition, the ring is taken apart an rotated to conform to Case 3 of Ref. 2. The ring deflec. and thus the tine rotation can then be determined.

(Conservative Analysis). The lateral load and reaction have been neglected as they do not contribute to tine rotation.

\( F_z = 2225 \text{ N (500 lb)} \)
\( l = .762 \text{ m (30 in)} \)
\( M = 2225(.762) = 1695.5 \text{ N.m} \)
\( R = \frac{2M}{\pi} = \frac{1695.5}{.3} = 5652 \text{ N} \)

- \( K = \text{Torsional Shape Factor} = 1.217 \times 10^{-4} \text{ m}^4 \)
- \( I_y = \text{Moment of Inertia About Neutral Axis} = 4.49 \times 10^{-5} \text{ m}^4 \)
- \( E = \text{Modulus of Elasticity} = 7.31 \times 10^6 \text{ N/m}^2 \)
- \( G = \text{Modulus of Rigidity} = 2.76 \times 10^5 \text{ N/m}^2 \)
- \( \lambda = \text{Ratio of Flexural to Torsional Rigidity} = \frac{E}{GI} = .97 \)

(cont)
FORK RING DEFLECTION - Z-AXIS LOADING (CONT)

Solve for Deflection $y$ at Reaction Points

$$y = \frac{Rr^3}{6EI} \left( 2.28 + 0.564\lambda \right) = \frac{5652(1.3^3)}{8(7.31\times10^{10})(4.49\times10^{-5})} \left[ 2.28 + 0.564(1.97) \right]$$

$$= 1.64 \times 10^{-5} \text{ m} \quad (0.00065 \text{ in})$$

$$\alpha = \frac{y}{r} = \frac{1.64 \times 10^{-5}}{0.3} = 54.67 \times 10^{-6} \text{ rad}$$

$$\beta = \frac{Rr^2}{6EI} (1+\lambda)(2-\pi) = \frac{5652(1.3^2)}{8(7.31\times10^{10})(4.49\times10^{-5})} \left[ 1 + 1.97 \right] (-1.14)$$

$$= 43.57 \times 10^{-6} \text{ rad} \quad \text{(REF ONLY)} \quad \text{(TWIST AT REACTION POINT)}$$

RING ROTATION AT FORK TINE:

$$\theta_2 = \alpha = 54.67 \times 10^{-6} \text{ rad} \quad (0.27^\circ)$$

TELESCOPE TUBE DEFLECTION AT END OF FORK TINE

$$\delta_z = \theta_2 L = 54.67 \times 10^{-6} \times 1.962 = 41.66 \times 10^{-6} \text{ m} \quad (0.0016 \text{ in})$$

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR
**Fork Tine Deflection - Loading in Z-Axis**

Consider Tine as Curved Cantilever Beam (See Ref 5, Case 15)

**Deflection at Load**

\[
\delta_z = \frac{FE}{4E I_Y} (B_1 + \lambda B_2)
\]

For \(\theta = 65^\circ\), \(\theta = 0^\circ\) (at load):

\[ B_1 = 1.508 \quad B_2 = 0.314 \]

\[ F = 2224 \text{ N} \]

\[ L = 0.74 \text{ m} \]

\[ \lambda = 0.33 \]

\[ I_Y = 2.31 \times 10^{-5} \text{ m}^4 \]

\[ E = 2.06 \times 10^5 \text{ N/m}^2 \]

\[ \delta_z = \frac{2224 (0.74)^3}{4 (2.31 \times 10^{-5}) (0.33)} = 6.66 \times 10^{-5} \text{ m} (0.0026 \text{ in}) \]

**Slope at Load** (Consider Tine as Simple Cantilever of Length, \(L\))

\[ \theta_z = \frac{F E L^2}{2 E I_Y} \]

\[ L = 0.85 \text{ m} \quad (33 \text{ in}) \]

\[ \theta_z = \frac{2224 (0.85)^2}{2 (2.31 \times 10^{-5}) (0.33)} = 118 \times 10^6 \text{ rad (24°)} \]
STRESS ANALYSIS

LOAD CONDITION 1
(LIMIT LOAD FACTORS TABLE 5-3)

LOAD CONDITION 2
(INCLUDES MECH Q=20 + RANDOM VIB.)

LAUNCH RELEASE

SOLID ROCKET CUT-OFF SEPARATION

RE-ENTRY

CRASH LANDING
STRESS ANALYSIS

DESIGN LOAD CONDITIONS

BY INSPECTION OF TABLE 5-3, THE FOLLOWING THREE LOAD CONDITIONS WERE FOUND TO PRODUCE MAXIMUM SINGLE AXIS AND COMBINED AXIS STRESSES:

1. LAUNCH RELEASE
2. SOLID ROCKET MOTOR CUT-OFF-Separation
3. RE-ENTRY

VIBRATION LOADING

STEADY-STATE ACCELERATION HAS BEEN COMBINED WITH DYNAMIC SINUSOIDAL AND RANDOM VIBRATION ACCELERATIONS. THE DYNAMIC ACCELERATIONS OF TABLE 5-3 ARE REDUCED TO RMS VALUES AND THE "SUM OF THE SQUARES" TAKEN TO FIND THE COMBINED SINUSOIDAL AND RANDOM VIBRATION RMS ACCELERATIONS.

RANDOM VIBRATION ACCELERATIONS

FROM TABLE 5-7, THE ACCELERATION SPECTRAL DENSITY HAS BEEN PLOTTED AND ANALYSED (SEE SEPARATE GRAPH) FOR ACCELERATION OF A 40 Hz RESONANT SYSTEM OPERATING WITHIN A 10 Hz BANDWIDTH.

\[ g = 0.003 \, \frac{g^2}{Hz} \]

\[ B = 10 \, Hz \]

\[ g = 0.17 \text{ RMS} \]

MECHANICAL AMPLIFICATION FACTOR, \( Q \).

A MECHANICAL AMPLIFICATION FACTOR, \( Q \), OF 20 IS ASSUMED FOR THE LIGHTLY DAMPED STRUCTURAL SYSTEM.
STRESS ANALYSIS

NATURAL FREQUENCY

THE SUPPORT STRUCTURE NATURAL FREQUENCIES FOR THE THREE AXES OF CONCERN:

X-AXIS \( f_n = 50.5 \) CPS
Y-AXIS \( f_n = 34.6 \) CPS
Z-AXIS \( f_n = 43.5 \) CPS

DESIGN LOADS (ACCELERATIONS)

ASSUME A "WORST CASE CONDITION" WITH THE DYNAMIC ACCELERATION (RANGE BETWEEN 0-60 Hz) HAPPENING AT THE APPROXIMATE RESONANT FREQUENCY (40 Hz) OF THE SUPPORT STRUCTURE

\[ Q = 20 \]

\[ g_{\text{RMS}} = 0.17 \text{ g rms} \]

1. LAUNCH RELEASE

<table>
<thead>
<tr>
<th>STEADY STATE</th>
<th>( \sqrt{(1002_{\text{rms}})^2 + 9_{\text{rms}}^2} \times Q )</th>
<th>TOTAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>X- DIRECTION</td>
<td>(-1.5)</td>
<td>+ (\sqrt{1.5^2 + 0.17^2} \times 20 = .72 (20) = 14)</td>
</tr>
<tr>
<td>Y- DIRECTION</td>
<td>(\pm 0.1)</td>
<td>+ (\sqrt{1.5^2 + 0.17^2} \times 20 = .53 (20) = 10.6)</td>
</tr>
<tr>
<td>Z- DIRECTION</td>
<td>(\pm 0.1)</td>
<td>+ (\sqrt{2.05^2 + 0.17^2} \times 20 = 2.9 (20) = 58)</td>
</tr>
</tbody>
</table>

2. SOLID ROCKET CUT-OFF - SEPARATION

| X- DIRECTION | \(-1.0\) | + \(\sqrt{2.1^2 + 0.17^2} \times 20 = 2.1 (20) = 42\) | = \(\pm 41.1\) g (max) |
| Y- DIRECTION | \(\pm 0.1\) | + \(\sqrt{2.1^2 + 0.17^2} \times 20 = 2.7 (20) = 5.4\) | = \(\pm 5.5\) g |
| Z- DIRECTION | \(\pm 0.3\) | + \(\sqrt{3.5^2 + 0.17^2} \times 20 = 3.9 (20) = 7.8\) | = \(\pm 8.1\) g |

3. RE-ENTRY

| X- DIRECTION | \(+1.0\) | + \(\sqrt{3.5^2 + 0.17^2} \times 20 = 3.9 (20) = 7.8\) | = \(\pm 8.8\) g |
| Y- DIRECTION | \(\pm 0.4\) | + \(\sqrt{2.1^2 + 0.17^2} \times 20 = 2.7 (20) = 5.4\) | = \(\pm 5.8\) g |
| Z- DIRECTION | \(+3.0\) | + \(\sqrt{1.7^2 + 0.17^2} \times 20 = 1.72 (20) = 14\) | = \(+17.0\) g |

(Note: Re-entry loads are less than launch or rocket cut-off)
Stress Analysis

Stress Levels for Operational Loads
Extreme load conditions considered: resonant vibration at Q = 20

1. Launch Release

\[ \Sigma X - \text{load} = 12.5 \text{g} \quad (\text{steady-state + sin vibration + random vibration}) \]
\[ \Sigma Y - \text{load} = 10.7 \text{g} \quad (\text{do}) \]
\[ \Sigma Z - \text{load} = 58.1 \text{g} \quad (\text{do}) \]

1.1 Support Cone, Alum 2219-T851
\[ f_y = 46,000 \text{ psi} \]

Bending + Axial Stress

\[ f_x = 12.5 \text{g} \times 2.78 \times 10^6 \text{N/m}^2 = 3.47 \times 10^7 \text{N/m}^2 \]
\[ (403 \text{ ksi}) \quad (5037 \text{ GPa}) \]

\[ f_y = 10.7 \text{g} \times 2.22 \times 10^5 \text{N/m}^2 = 2.37 \times 10^6 \text{N/m}^2 \]
\[ (32 \text{ ksi}) \quad (342 \text{ GPa}) \]

\[ f_z = 58.1 \text{g} \times 2.22 \times 10^6 \text{N/m}^2 = 1.29 \times 10^7 \text{N/m}^2 \]
\[ (32 \text{ ksi}) \quad (1,859 \text{ GPa}) \]

Combined Bending Stress

\[ f_b = \sqrt{f_x^2 + f_y^2 + f_z^2} = 3.71 \times 10^7 \text{N/m}^2 \]
\[ (5380 \text{ ksi}) \]

Margin of Safety

\[ M.S. = \frac{46,000}{5380} - 1 = 7.55 \]
**Stress Analysis**

**Stress Levels for Operational Loads**

**Extreme Load Conditions**

1. **Launch Release (Cont.)**

1.2 **4-PT Contact Bearing**

\[ \Sigma X - \text{load} = 12.5 \text{ g} \quad \text{Axial Only} \]
\[ \Sigma Y - \text{load} = 10.7 \text{ g} \quad \text{Radial & Moment} \]
\[ \Sigma Z - \text{load} = 58.1 \text{ g} \quad \text{Radial & Moment} \]

**Updated to 261 g**

**Total Axial Load**

\[ F_x = \frac{12.5 \times 65340}{1200} = \frac{6673 \times 10^4 \text{ N}}{15,000 \text{ LB}} \]

**Y-Axis Radial Load**

\[ F_y = 10.7 \times \frac{5340}{1200} = \frac{5714 \times 10^4 \text{ N}}{12,840 \text{ LB}} \]

**Z-Axis Radial Load**

\[ F_z = 58.1 \times \frac{5340}{1200} = \frac{3103 \times 10^4 \text{ N}}{64,720 \text{ LB}} \]

**Total Radial Load**

\[ F_0 = \sqrt{F_y^2 + F_z^2} = \frac{3.16 \times 10^5 \text{ N}}{70,935 \text{ LB}} \]

**Moment Load Y-Axis**

\[ M_y = 10.7 \times \frac{4747}{42000 \text{ LB-LIN}} = \frac{5.08 \times 10^4 \text{ N-M}}{431,000 \text{ LB-LIN}} \]

**Moment Load Z-Axis**

\[ M_z = 58.1 \times \frac{4747}{42000 \text{ LB-LIN}} = \frac{2.76 \times 10^5 \text{ N-M}}{1,339,000 \text{ LB-LIN}} \]

**Total Moment Load**

\[ M = \sqrt{M_y^2 + M_z^2} = \frac{2.81 \times 10^5 \text{ N-M}}{2.48 \times 10^6 \text{ LB-LIN}} \]

**Margin of Safety**

(See Next Sht for Updated M.S.)

\[ \text{M.S. (Axial)} = \frac{199430}{15,000} - 1 = 12.29 \]
\[ \text{M.S. (Radial)} = \frac{49,540}{10,935} - 1 = -0.02 \]
\[ \text{M.S. (Moment)} = \frac{154,610}{2,483,000} - 1 = -0.38 \]

**Note:** See next sheet for updated design loads and reduced stress levels.
1.2 4-PT CONTACT BEARING (CONT)

NOTE: Updated Design accelerations have been received subsequent to completion of this stress analysis.

The following correction to the bearing load analysis is included to reflect the reduced load in the Z-axis (produces neg. Margin of Safety).

**New: Design Acceleration in Z-Direction**

**Launch Release**: ± 0.1 g Steady State
(Z-Direction)

\[ \frac{q}{g} = 20 \]

\[ f_e = 43.5 \text{ Hz} \]

**Design Loads - Z-Direction**

\[ Z = \pm 0.1 + \sqrt{\left(0.1\right)^2 + \left(0.1\right)^2} \times 20 = \pm 20.1 \text{ g (WAS 5.81 g)} \]

**Z-Axis Radial Load** \( F_z = 5340 \text{ N} \times 20.19 = 10.73 \times 10^4 \text{ N} \) (24/20 k)

**Total Radial Load** \( F = \sqrt{F_z^2 + F_e^2} = \sqrt{(5.714 \times 10^4)^2 + (10.73 \times 10^4)^2} = 1.26 \times 10^5 \text{ N} \)

**Z-Axis Moment Load** \( M_z = 474.7 \frac{\text{N} \cdot \text{m}}{g} \times 20.19 = 9.54 \times 10^4 \frac{\text{N} \cdot \text{m}}{g} \)

**Total Moment Load** \( M = \sqrt{M_z^2 + M_e^2} = \sqrt{(5.02 \times 10^4)^2 + (9.54 \times 10^4)^2} = 1.08 \times 10^5 \frac{\text{N} \cdot \text{m}}{g} \)

**Margin of Safety Radial Load** \( M.S. = \frac{69.540}{27.318} - 1 = 1.54 \) (OK)

**Moment Load** \( M.S. = \frac{1540.1}{844.2} - 1 = \pm 1.82 \) (OK)
Stress Analysis

Stress Levels for Operational Loads

Extreme Load Conditions

1. Launch Release (Cont)

1.3 Fork Ring

Alum 2219-T851

\[ f_y = 46,000 \text{ lb/in}^2 \]

\[ \Sigma X-Load = 12.5g \]

\[ \Sigma Y-Load = 10.7g \]

\[ \Sigma Z-Load = 58.1g \]

Bending Stress

\[ f_x = 12.5g \times 1.147 \times 10^6 \text{ N/m}^2 = 1.43 \times 10^7 \text{ N/m}^2 \]

\[ f_y = 10.7g \times 2.80 \times 10^6 \text{ N/m}^2 = 2.44 \times 10^7 \text{ N/m}^2 \]

\[ f_z = 58.1g \times 2.00 \times 10^6 \text{ N/m}^2 = 1.16 \times 10^8 \text{ N/m}^2 \]

Combined Bending Stress

\[ f_b = \sqrt{f_x^2 + f_y^2 + f_z^2} = \frac{1.19 \times 10^8 \text{ N/m}^2}{(17,261 \text{ lb/in}^2)} \]

Margin of Safety

\[ M.S. = \frac{46,000}{17,261} - 1 = 1.66 \]
STRESS ANALYSIS

STRESS LEVELS FOR OPERATIONAL LOADS

EXTREME LOAD CONDITIONS

1. LAUNCH RELEASE (CONT)

1.4 FORK TINE  ALUM 2219-T851

\[ \Sigma X - LOAD = 12.59 \]

\[ \Sigma Y' - LOAD = 10.79 \]

\[ \Sigma Z - LOAD = 58.19 \]

BENDING STRESS + AXIAL STRESS

\[ f_x = 12.59 \times 4.2 \times 10^5 \text{ N/m}^2 = 5.25 \times 10^6 \text{ N/m}^2 \]

\[ f_y = 10.79 \times 5.51 \times 10^4 \text{ N/m}^2 = 5.90 \times 10^7 \text{ N/m}^2 \]

\[ f_z = 58.19 \times 4.54 \times 10^6 \text{ N/m}^2 = 2.64 \times 10^8 \text{ N/m}^2 \]

COMBINED TENSILE STRESS

\[ f_\sigma = \sqrt{f_x^2 + f_y^2 + f_z^2} = \frac{2.71 \times 10^8 \text{ N/m}^2}{39,240 \text{ lb/in}^2} \]

MARGIN OF SAFETY

\[ M.S. = \frac{46,000}{39,240} - 1 = 0.17 \]
STRESS ANALYSIS

STRESS LEVELS FOR OPERATIONAL LOADS

EXTREME LOAD CONDITIONS CONSIDERED: RESONANT VIBRATIONS
AT Q = 20

2. SOLID ROCKET CUT-OFF: SEPARATION

2.1 SUPPORT CONE

ALUM 2219-T851

\[ f_{\text{eq}} = 46,000 \text{ ksi} \]

\[ \Sigma X \text{- LOAD} = 419 \text{ g} \]
\[ \Sigma Y \text{- LOAD} = 5.59 \text{ g} \]
\[ \Sigma Z \text{- LOAD} = 8.19 \text{ g} \]

BENDING + AXIAL STRESS

\[ f_x = \frac{419 \text{ g @ } 2.28 \times 10^4 \text{ N/m}^2}{16,500 \text{ ksi}} = 1.14 \times 10^8 \text{ N/m}^2 \]
\[ f_y = \frac{5.59 \text{ g @ } 2.22 \times 10^6 \text{ N/m}^2}{16,500 \text{ ksi}} = 1.21 \times 10^6 \text{ N/m}^2 \]
\[ f_z = \frac{8.19 \text{ g @ } 2.22 \times 10^6 \text{ N/m}^2}{16,500 \text{ ksi}} = 1.78 \times 10^6 \text{ N/m}^2 \]

COMBINED TENSILE STRESS

\[ f_t = \sqrt{f_x^2 + f_y^2 + f_z^2} = 1.14 \times 10^8 \frac{\text{N/m}^2}{16,500 \text{ ksi}} = 1.14 \times 10^8 \text{ N/m}^2 \]

MARGIN OF SAFETY

\[ M.S. = \frac{46,000}{16,500} - 1 = 1.79 \]
STRESS ANALYSIS

STRESS LEVELS FOR OPERATIONAL LOADS

EXTREME LOAD CONDITIONS

2.0 SOLID ROCKET CUT-OFF-SEPARATION (CONT)

2.2 4-PT CONTACT BEARING (62100 std)

Σ X-LOAD = 419g
Σ Y-LOAD = 5.59
Σ Z-LOAD = 8.19

TOTAL AXIAL LOAD

\[ F_x = \frac{419 \times 5340}{9.81} \times 10^5 = 2.19 \times 10^8 \text{ N} \]

RADIAL LOADS Y-AXIS

\[ F_y = \frac{5.59 \times 5390}{9.81} = 2.94 \times 10^4 \text{ N} \]

\[ F_z = \frac{8.19 \times 5390}{9.81} = 4.32 \times 10^4 \text{ N} \]

TOTAL AXIAL LOAD

\[ F_R = \sqrt{F_y^2 + F_z^2} = 5.23 \times 10^4 \text{ N} \]

MOMENT LOAD Y-AXIS

\[ M_y = \frac{5.59 \times 4747}{9} = 2.61 \times 10^4 \text{ Nm} \]

\[ M_z = \frac{8.19 \times 4747}{9} = 3.85 \times 10^4 \text{ Nm} \]

TOTAL MOMENT LOAD

\[ M = \sqrt{M_y^2 + M_z^2} = 4.65 \times 10^4 \text{ Nm} \]

MARGIN OF SAFETY

\[ M.S. \ (AXIAL) = \frac{199,800}{44,200} - 1 = 3.05 \]

\[ M.S. \ (RADIAL) = \frac{69,540}{11,758} - 1 = 4.91 \]

\[ M.S. \ (MOMENT) = \frac{1,540,000}{10,448} - 1 = 146.4 \]
Stress Analysis

Stress Levels for Operational Loads

Extreme Load Conditions

2.0 Solid Rocket Cut-off Separation (cont)

2.3 Fork Ring

\[ \Sigma X - \text{Load} = 41g \]
\[ \Sigma Y - \text{Load} = 5.5g \]
\[ \Sigma Z - \text{Load} = 8.1g \]

Bending Stress

\[ f_x = 41g \times 1.147 \times 10^6 \frac{N}{m^2} = 4.70 \times 10^7 \frac{N}{m^2} \]
\[ f_y = 5.5g \times 2.00 \times 10^6 \frac{N}{m^2} = 1.10 \times 10^7 \frac{N}{m^2} \]
\[ f_z = 8.1g \times 2.00 \times 10^6 \frac{N}{m^2} = 1.62 \times 10^7 \frac{N}{m^2} \]

Combined Bending Stress

\[ f_b = \sqrt{f_x^2 + f_y^2 + f_z^2} = 5.09 \times 10^7 \frac{N}{m^2} \]

\[ f_b = \frac{5.09 \times 10^7 \frac{N}{m^2}}{7384 \frac{lb/in^2}} \]

Margin of Safety

\[ M.S. = \frac{46,000}{7384} - 1 = 5.23 \]
STRESS LEVEL FOR OPERATIONAL LOADS
EXTREME LOAD CONDITIONS
2.0 SOLID ROCKET CUT-OFF-SEPARATION (CONT)

2.4 FORK TINE

Alum 2219-T851

\[ \sigma_y = 46,000 \text{ lb/in}^2 \]

\[ \Sigma X \text{- LOAD} = 41g \]
\[ \Sigma Y \text{- LOAD} = 5.5g \]
\[ \Sigma Z \text{- LOAD} = 8.1g \]

**BENDING AND AXIAL STRESS**

\[ f_x = 41g \times 4.2 \times 10^5 \text{ N/m}^2 = 1.72 \times 10^7 \text{ N/m}^2 \]

\[ f_y = 5.5g \times 5.51 \times 10^6 \text{ N/m}^2 = 3.03 \times 10^7 \text{ N/m}^2 \]

\[ f_z = 8.1g \times 4.54 \times 10^6 \text{ N/m}^2 = 3.68 \times 10^7 \text{ N/m}^2 \]

**COMBINED TENSILE STRESS**

\[ f_z = \sqrt{f_x^2 + f_y^2 + f_z^2} = \frac{5.07 \times 10^7 \text{ N/m}^2}{(7.347 \text{ lb/in})} \]

**MARGIN OF SAFETY**

\[ M.S. = \frac{46,000}{7,347} - 1 = 5.26 \]
STRESS ANALYSIS - LOADING IN X-AXIS - 1g LOAD

Support Cone
See deflection calc's for loading and geometry

Max Stress
(as a Belleville spring) (worst case)

\[ \sigma = \frac{E SC_h}{(1-\nu^2)Ma^2} \]

\[ \nu = \text{Poisson's ratio} = 0.33 \]
\[ M = \text{constant} = 0.74 \text{ @ } \frac{h}{b} = 3.14 \]
\[ C_f = \text{constant} = 1.46 \text{ @ } \frac{h}{b} = 3.14 \]
\[ \delta = 9.55 \times 10^{-5} \text{m (from deflection calc's)} \]

\[ \sigma = \frac{(7.31 \times 10^6)(9.55 \times 10^{-5})(1.46)(0.80)}{(1-0.33^2)(0.79)(2.04^2)} = 2.78 \times 10^6 \text{ N/m}^2 \text{ (403 lb/in}^2) \text{ @ 1g.} \]
STRESS ANALYSIS - LOADING IN X-AXIS - 1g LOAD

**Outer Bearing**
Kaydon No 6684-1 with 0.005 in Preload
Axial preload relieved at 4800 lb

Axial Load \( F_x = 5340 \text{ N} \) (1200 lb)

\[ M.S. = \frac{4800}{1200} - 1 = 3.0 \]
STRESS ANALYSIS - LOADING IN X-AXIS - 1G LOAD

FORK RING

(SEE DEFLECTION ANALYSIS FOR CONFIGURATION)

\[ \text{Moment } M = \frac{T \cdot \sin \theta}{2} = \frac{972(1)}{2} = 486 \text{ Nm} \quad \theta = 90^\circ \]

\[ \text{Torque } T = \frac{G \cdot \theta}{2} = \frac{972(1)}{2} = 486 \text{ Nm} \quad \theta = 0^\circ \]

**Max Bending Stress**

\[ f_b = \frac{Mc}{I_y} = \frac{486 \cdot 106}{4.49 \times 10^{-5}} = 1.147 \times 10^6 \text{ N/m}^2 \quad (166 \text{ ksi}) \text{ @ 1g} \]

**Max Torsional Stress**

\[ f_\tau = \frac{T}{2\pi A} = \frac{486}{2(0.013)(0.046)} = 4.06 \times 10^5 \text{ N/m}^2 \quad (59 \text{ ksi}) \text{ @ 1g} \]
Stress Analysis - Loading in X-Axis - 1g Load

Fork Tine (See Deflection Calc's for Geometry)

Column Load  \( F_x = 2224 \text{ N} \) (500 lb)

Axial Stress  \( \sigma_a = \frac{F_x}{A} = \frac{2224}{.0053} = 4.20 \times 10^5 \frac{\text{N}}{\text{m}^2} \) (6105 lb/in²) @ 1g
Stress Analysis - Loading in Y-Axis - 1g Load

Support Cone

\[ \begin{align*}
C_0 &= 3.14 \\
L &= 0.95 \text{m (37.5)} \\
L_0 &= 0.65 \text{m (25.6 in)} \\
L_c &= 2.04 \text{m (80.45 in)} \\
L &= 1.80 \text{m (31.5)} \\
L_c &= 1.00 \text{35 m (1331 in)} \\
F_y &= 4448 \text{N (1000 lb) @ 1g} \\
F_y^+ &= 5340 \text{N (1200 lb) @ 1g} \\
M &= F_y^+ l = 5086 \text{ N.m @ 1g}
\end{align*} \]

Bending Stress from Moment, \( M \) @ 1g

\[ \begin{align*}
\sigma_b &= \pm \frac{M}{C_0 t R_c^2} = \pm \frac{5086}{3.14(0.0035)(2.04^2)} = 1.112 \times 10^5 \text{ N/m}^2 \quad (16 \text{ kips/in}^2) \end{align*} \]

Bending Stress from End Load, \( F_y^+ \) @ 1g

\[ \begin{align*}
\sigma_b &= \pm \frac{F_y^+ R_c}{C_0 t R_c^2} = \pm \frac{5086}{3.14(0.0035)(2.04^2)} = 1.112 \times 10^5 \text{ N/m}^2 \\
\sigma_b &= 2.224 \times 10^5 \text{ N/m}^2 \quad (32 \text{ kips/in}^2) \end{align*} \]

Total Bending Stress

\[ \sigma_b = 2.224 \times 10^5 \text{ N/m}^2 \quad (32 \text{ kips/in}^2) \]
STRESS ANALYSIS - LOADING IN Y-AXIS - 1g LOAD

OUTER BEARING

0.005 IN. PRELOAD PREFERRED AT 162,000 LB-IN
RADIAL PRELOAD RELIEVED AT 31,000 LB

MOMENT LOADING

\[ M = F_Y L = 5340(18) = 9747 \text{ N.m} \]
\[ (42,000 \text{ lb-in}) \]

\[ @ 1g \]

\[ M.S. = \frac{162,000}{42,000} = 3.86 \]

RADIAL LOADING

\[ F_Y = 5340 \text{ N (1200 lb)} \]

\[ @ 1g \]

\[ M.S. = \frac{31,000 \text{ lb}}{1200 \text{ lb}} = 25.83 \]
STRESS ANALYSIS - LOADING IN Y-AXIS - 1g LOAD

FORK RING  (SEE DEFLECTION ANALYSIS FOR CONFIGURATION)

Moment  \( M = \frac{T \sin \theta}{2} = \frac{1645.5(1)}{2} = 827.75 \text{ N.m} \) @ \( \theta = 90^\circ \)

Torque  \( T = \frac{To \cos \theta}{2} = \frac{1645.5(1)}{2} = 827.75 \text{ N.m} \) @ \( \theta = 0^\circ \)

Max Bending Stress @ \( \theta = 90^\circ \)

\( f_b = \frac{Mc}{I_y} = \frac{827.75(1.06)}{4.49(10^{-3})} = 2.00 \times 10^6 \text{ N/m}^2 \) (290 ksi) @ 1g

Max Torsional Stress @ \( \theta = 0^\circ \)

\( f_t = \frac{T}{2\pi A} = \frac{827.75}{2(0.03)(0.04636)} = 7.03 \times 10^5 \text{ N/m}^2 \) (102 ksi) @ 1g
STRESS ANALYSIS - LOADING IN Y-AXIS 1-g LOAD

FORK TINES  (SEE DEFLECTION CALC'S FOR GEOMETRY)

BENDING STRESS

\[ \sigma_b = \frac{Mc}{I_z} = \frac{EzLz}{I_z} = \frac{2224(0.83)(1.37)}{4.6 \times 10^{-5}} = \frac{5.51 \times 10^6 \text{ N/m}^2}{(199.7 \text{ lb/in}^2)} \]

\( \sigma_b \approx 19 \)
Stress Analysis - Loading in Z-axis - 1g Load

Support Cone

Same Loading and Stress
As Loading in the Y-axis

Bending Stress \( f_b = 2.22 \times 10^5 \text{ N/m}^2 \) (32 kN/m²) @ 1g
Stress Analysis - Loading in Z-Axis - 1g Load

Outer Bearing

Same as loading in Y-Axis
Same Stress

Moment Loading

@ 1g M.S. = \[ \frac{162,000 \text{ lb-in}}{42,000 \text{ lb-in}} = 3.86 \]

Radial Loading

@ 1g M.S. = \[ \frac{31,000 \text{ lb}}{1200 \text{ lb}} = 25.83 \]
Stress Analysis - Loading in Z-Axis - 1g Load

Fork Ring

See Deflection Analysis for Configuration

Bending Moment @ \( \theta = 90^\circ \)

\[
M = \frac{Rr (\sin \theta - \cos \theta)}{2} = \frac{5652(0.3)(1-0)}{2} = 847.8 \text{ Nm}
\]

Twisting Moment @ \( \theta = 45^\circ \)

\[
T = \frac{Rr (\sin \theta + \cos \theta - 1)}{2} = \frac{5652(0.3 \times 2.07+2.07-1)}{2} = 351 \text{ Nm}
\]

Max Bending Stress @ \( \theta = 90^\circ \)

\[
f_b = \frac{MC}{I_y} = \frac{847.8(0.16)}{4.24 \times 10^5} = 2.00 \times 10^6 \text{ N/m}^2 \ (290 \text{ kgf/ in}^2) \leq 1 \text{ g}
\]

Max Torsional Stress @ \( \theta = 45^\circ \)

\[
f_T = \frac{T}{2\pi A} = \frac{351}{2(0.03)(0.046)} = 2.93 \times 10^5 \text{ N/m}^2 \ (42 \text{ kgf/ in}^2) \leq 1 \text{ g}
\]
Stress Analysis - Loading in Z-Axis - 1g Load

Fork Tines (See Deflection Calc's for Geometry)

Bending Stress

\[ f_b = \frac{M_e}{I_y} = \frac{F_x L}{I_y} = \frac{2224 (0.83)(0.229)}{9.3 \times 10^{-5}} = 4.54 \times 10^6 \text{ N/m}^2 \approx 1g \text{ (659 lb/in}^2) \]
MOUNTING BOLTS (ASSUMED) NO INFORMATION AVAILABLE

P.D. = 4.084 m (160.8 in)
N = 144 BOLTS (2.5° APART)
P = 87.1 mm (3.41 in)

Bolt Size ≈ 9 mm
Minor dia = 7.9
Stress Area ≈ 49 mm² (0.0759 in²)
Total Area A_t = 194 (49) = 9536 mm² (10.94 in²)

PRESSURE LOAD ON END CONE

\[ P = 15 \, \text{lb/in}^2 \left( \frac{100}{144} \right) = 102.36 \, \text{lb} \]

Bolt Stress \[ \approx \frac{301,592}{10.94} = 27,575 \, \text{lb/in}^2 \]
Assume Grade 12.9 Tensile \( = 170,000 \, \text{lb/in}^2 \) Yield \( = 153,000 \, \text{lb/in}^2 \)
Shear \( = 0.6 \times \text{Yield} = 91,800 \, \text{lb/in}^2 \)

MOMENT LOADING ON SPACELAB FROM TELESCOPE MOUNT

Max F_y = 2194 lb @ 10.7° During Launch Release

Max \( M_y = 2194 \times 10.7 \times 28 \text{ in} = 657322 \text{ lb-in} \)

SECTION MODULUS OF BOLT PATTERN

Bolt Linear Area \[ A_b = \frac{0.0759}{3.51} = 0.0216 \, \text{in}^2 \]
Effective \( (\text{CONT}) \)

TELESCOPE MOUNT = 995 kg (2194 lb)
Moment of Inertia of Bolt Pattern: (Pivot at Bottom Flange)
Assume Bolts Form Tube of 160.8 in Dia × 0.0216 in Thk

\[ I_1 = \frac{4}{3} \pi \times \frac{0.0216^2}{160.8} \times (160.8)^3 = 35,923 \text{ in}^4 \]
\[ I_2 = I_1 + \frac{A_0^2}{4} = 35,923 + \frac{10.91 (160.8)^2}{4} = 106,457 \text{ in}^4 \]

Sec. Modulus of Bolt Pattern (About Bottom of Flange)

\[ S = \frac{I}{D} = \frac{106,457}{160.8} = 662 \text{ in}^3 \]

Tensile Stress in Bolts From Moment Load (10.7 g\(^s\))

\[ f_b = \frac{M}{S} = \frac{657,322 \text{ lb-in}}{662 \text{ in}^3} = 1057 \frac{\text{lb}}{\text{in}^2} @ 10.7 g\(^s\) \]

Section Modulus of Spacelab Hull

\[ S = 8.2^2 \times 0.8 (160^3)(.20) = 4096 \text{ in}^3 \]

Stress in Hull From Moment Loading (10.7 g\(^s\))

\[ f_b = \frac{M}{S} = \frac{657,322 \text{ lb-in}}{4096 \text{ in}^3} = 160 \frac{\text{lb}}{\text{in}^2} @ 10.7 g\(^s\) \]
Axial Loading on Spacelab from Telescope Mount

Assume axial load at launch release = 12.5 g

This load will combine with moment load of 10.7 g's

\[ F_x = 2194 \text{ lb} \times 12.5 = 27425 \text{ lb} \]

Tensile Stress in bolts from axial load

\[ \sigma = \frac{F_x}{A_b} = \frac{27425 \text{ lb}}{10.44 \text{ in}^2} = 2607 \text{ lb/in}^2 @ 12.5 \text{ g's} \]

Tensile Stress in Hull from Axial Load

\[ \sigma = \frac{F_x}{A_h} = \frac{27425 \text{ lb}}{160(\pi/4)} = 273 \text{ lb/in}^2 @ 12.5 \text{ g's} \]

Combined Stresses at Launch Release from Telescope

\[ F_y @ 10.7 \text{ g's} \]
\[ F_x @ 12.5 \text{ g's} \]

Max Bolt Stress (Attachment to Hull)

\[ \sigma_b = 1.057 + 2.507 = 3.564 \text{ lb/in}^2 \]

(13% of Press Stress)

Bolt Stress from 15 psi pressure inside Hull

\[ \sigma = \frac{301.592 \text{ lb}}{10.44 \text{ in}^2} = 29.575 \text{ lb/in}^2 \]

Max Hull Stress

\[ \sigma_h = 160 + 273 = 433 \text{ lb/in}^2 \]

(14% of Press Stress)

Hull Stress from 15 psi pressure inside Hull

\[ \sigma = \frac{301.592 \text{ lb}}{100.5 \text{ in}^2} = 3.000 \text{ lb/in}^2 \]
**Fork Ring Properties**

**Moments of Inertia**

\[ I_{xy} = \sum I_{xy} + \sum A d_y^2 = \frac{44.9 \times 10^6 \text{mm}^4}{4.49 \times 10^{-5} \text{m}^4} = 107.9 \text{in}^4 \]

\[ I_{xx} = \sum I_{xx} + \sum A d_x^2 = \frac{98.4 \times 10^6 \text{mm}^4}{9.84 \times 10^{-5} \text{m}^4} = 236.4 \text{in}^4 \]

**Torsional Shape Factor**

\[ K = \frac{4A^2 \ell}{U} = \frac{4(46.360 \text{mm}^2)}{918} = 121.7 \times 10^6 \text{mm}^4 \]

\[ 1.217 \times 10^4 \text{mm}^4 \]
**Fork Tine - Properties**

![Diagram of fork tine properties]

**At Section A-A**

\[
I_{Y-Y} = \frac{0.006(229)^3}{12} + \frac{0.006(229)^3}{12} + 0.006(254)(0.152)(2) = 4.8 \times 10^{-5} + 2.2 \times 10^{-5} + 7.0 \times 10^{-6} = 12.4 \times 10^{-5} \text{ m}^4
\]

\[
I_{Z-Z} = \frac{0.006(229)^3}{12} + 0.006(229)(12)^2 + 0.006(254)(10)^2 = 1.2 \times 10^{-5} + 2.2 \times 10^{-5} + 2.8 \times 10^{-6} = 6.2 \times 10^{-5} \text{ m}^4
\]

**Average Moment of Inertia, Considering Taper in Last Section of Tine, Will Be Approximately 75% of That at Section A-A**

\[
\text{Ave. } I_{Y-Y} = 0.75 \times 12.4 \times 10^{-5} = 9.3 \times 10^{-5} \text{ m}^4 \quad (223 \text{ in}^4)
\]

\[
\text{Ave. } I_{Z-Z} = 0.75 \times 6.2 \times 10^{-5} = 4.6 \times 10^{-5} \text{ m}^4 \quad (110 \text{ in}^4)
\]

(Cont)
FORK TINES - PROPERTIES (CONT)

Torsional Shape Factor

\[ K = \text{Torsional Shape Factor} \]
\[ U = \text{Length of Median Boundary} = 458 + 254 + 229 + 259 = 1.195 \text{ m} \]
\[ A = \text{Area within Median Boundary} = \frac{458 + 229}{2}(254) = 0.08725 \text{ m}^2 \]
\[ t = \text{Wall Thickness} = 0.006 \text{ m} \]

\[ K = \frac{4A^2t}{U} = \frac{4(0.08725)(0.006)}{1.195} = 1.53 \times 10^{-4} \text{ m}^4 \]

\[ (507 \text{ in}^3) \]

Flexural to Torsional Rigidity Ratio

\[ \lambda_f = \frac{E I_f}{G K} = 1.60 \]
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APPENDIX B

EFFECTS OF CREW MOTIONS AND VERNIER THRUSTERS ON TRACKING AND IMAGE STABILIZATION

The following three pages show the effect of crew motion and small thruster firings on the tracking and image stabilization for the AFPT. The first curve plotted on each page shows the input disturbance. The second plot shows the resulting motion of the shuttle vehicle. The third plot shows the pointing error of the main telescope mount caused by the vehicle motion. The last plot shows the excursion of the image from its nominal position in focal plane. All the curves are based on a 0.1 second time constant and an effective overdamped condition for the servo system.
EFFECT OF CREW MOTION ON TRACKING
AND IMAGE STABILIZATION

CREW INPUT

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR

Force (N)

0 0.8 1.6 2.4 3.2 3.6
TIME (Sec)

Assumed Input

Rotation (Sec)

0 10 20 30

Shuttle Motion

Assumed Input

Error (Sec)

0 0.1 0.2 0.3

Telescope Mount Pointing Error

Accel. 24.3 Sec/sec²

K = 2.5
(UNDER DAMPED)

K = 2.0
(CRITICALLY DAMPED)

K = 1.5
(OVER DAMPED)

\[ \Delta \omega_m = \frac{1}{T} \left( k \Theta_T - \Theta_{T-1} \right) \]

T = guider detector time constant.

\[ \Theta_T = 2 \text{ (tilt c of secondary from nominal)} \]

\[ \Theta_{T-1} = 2 \text{ (last tilt c of secondary from nominal)} \]

Displacement (Sec)

0.10

Image Error

(1 Sec time const)

(Over Damped)

0

Time (Sec)

0.8 1.6 2.4 3.2 3.6

\[ k = 0.03 \text{ Sec} \]

-0.10
EFFECT OF SMALL THRUSTER (LONG BURN) ON TRACKING AND IMAGE STABILIZATION

**SMALL THRUSTER INPUT**

111 N (25 lb) for 1629 sec

**SHUTTLE MOTION**

\[ a = \frac{22.1 \text{ sec}}{\text{sec}^2} \]

**TELESCOPE POINTING ERROR**

\[ k = 1.5 \]

**IMAGE STABILIZATION**

\[ k = 1.5 \]

\[ +0.03 \text{ sec} \]

\[ -0.03 \text{ sec} \]
EFFECT OF SMALL THRUSTER (SHORT BURN) ON TRACKING AND IMAGE STABILIZATION

**Small Thruster Input**

111 N (25 lb) for .04 sec

**Shuttle Motion**

Reproducibility of the original page is poor

**Telescope Mount Pointing Error**

$k = 1.5$

**Image Stabilization Error**

$k = 1.5$