STRESS ANALYSIS

FRACTIONAL WATT
VUILLEUMIER CYROGENIC REFREIGERATOR PROGRAM
ENGINEERING NOTEBOOK

74-9896-2
January 1974

Prepared under Contract No. NAS 5-21715

for

National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, Maryland
FOREWORD

Under NASA Contract NAS 5-21715, the AiResearch Manufacturing Company, a Division of The Garrett Corporation, developed a 659K Vuilleumier (VM) cryogenic refrigerator for the NASA Goddard Space Flight Center (GSFC), Greenbelt, Maryland. During the program, thermal analysis and stress analysis notebooks were compiled for submittal to GSFC. This two-volume document contains (or references) all material compiled during the course of the thermal and stress analyses. In certain instances, copyrighted reference material was used during the analytical work and will not be reproduced in this document.

Volume 1, identified as AiResearch document 74-9896-1, presents the detailed thermal analysis that was conducted during the program.

Volume 2, identified as AiResearch document 74-9896-2 presents the detailed stress analysis that was conducted during the program on various component parts/assemblies of the VM refrigerator.
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INTRODUCTION
SECTION I

INTRODUCTION

A structural analysis was performed on the 1/4-watt Vuilleumier (VM) cryogenic refrigerator developed for the NASA Goddard Space Flight Center. The analysis covered the complete assembly except for the cooling jacket and mounting brackets. Maximum stresses, margins of safety, and natural frequencies were calculated for structurally loaded refrigerator components shown in assembly drawing 852522.

The stress analysis indicates that the VM design is satisfactory for the specified vibration environment, and the proof, burst, and normal operating loads.

The organization and content of the Stress Analysis Engineering Notebook is summarized below:

Section 2 Stress Analysis Summary - Overviews and summarizes the stress analysis effort performed during design and development of the VM refrigerator.

Section 3 Design Requirements - Summarizes design requirements specified by NASA GSFC that were the basis for VM refrigerator preliminary design.

Section 4 Material Properties - Reproduces AiResearch material specifications and information used in design/stress analysis of VM. References copyrighted vendor materials specifications.

Section 5 Crankshaft Assembly - Stress analysis of the complete shaft assembly including wristpin, and connecting rod, and critical speed analysis of the assembly.

Section 6 Crankshaft Housing - Analysis covers the spherical dome-cylinder intersection, sump cross tube intersection, motor inner housing, motor outer housing, and flanged connections.

Section 7 Hot-End Pressure Shell - Analysis covers shell stresses and deflections, flange connection, and vacuum jacket.

Section 8 Cold-End Pressure Shell - Analysis covers shell stresses and deflections, flange connection, and vacuum jacket.
Section 9  Hot-End Displacer - Analysis covers collapsing pressure calculation and shell stresses and deflections.

Section 10  Cold-End Displacer - Analysis covers collapsing pressure calculation and shell stresses and deflections.
SECTION 2
STRESS ANALYSIS SUMMARY
INTRODUCTION

This section summarizes the structural analysis performed on the 1/4-watt Vuilleumier (VM) cryogenic refrigerator developed for the NASA Goddard Space Flight Center. The analysis covered the complete assembly except for the cooling jacket and mounting brackets. Maximum stresses, margins of safety, and natural frequencies were calculated for structurally loaded refrigerator components shown on the final assembly drawing 852-522.

The structural analysis indicates that the VM design is satisfactory for the specified vibration environment, and the proof, burst, and normal operating loads.

STRESS ANALYSIS

Stress analysis performed on the 1/4-watt Vuilleumier (VM) cryogenic refrigerator covered the complete assembly except for the cooling jacket and mounting brackets. Maximum stresses, margins of safety, and natural frequencies were calculated for structurally loaded refrigerator components shown in the final assembly drawing (Figure 2-1).

VM Description

A brief description of the VM refrigerator is given here to aid in understanding the analytical summary which follows. A cross-sectional drawing of the VM refrigerator is presented in Figure 2-1.

The VM refrigerator is a two-stage, refrigeration machine with a nominal design speed of 400 rpm and a continuous operation design life of two years. The moving components within the machine consist of a crankshaft assembly which drives two displacers (one hot, one cold) attached to the crankshaft with connecting rods. Crankshaft throws are 90 degrees apart with the hot displacer leading. The displacers travel inside of packed-bed regenerators that are enclosed in pressure shells joined to the crankshaft housing. Thus, the entire assembly forms a pressure-tight enclosure. The operating gas in the enclosure is helium. The crankshaft is driven by an electric motor.
Figure 2-1. Fractional Watt Willaumier Cryogenic Refrigerator
For analytical purposes, various parts groupings were considered separately. Physical descriptions of these groups are given below.

2. Crankshaft Assembly

The crankshaft assembly is an eccentric shaft on which the cold end crank and the bearing journal are stacked. A bolt threaded into the eccentric shaft secures the parts to the shaft. The hot end crank is an integral part of the shaft.

3. Crankshaft Housing

The group of structural elements surrounding the crankshaft are designated as the crankshaft housing and consist of: the spherical section and flanged crosstubes of the sump housing; the bearing sleeve; the closure cap; the retaining ring; and the inner and outer motor housings. The retaining ring and closure cap form an enclosure on one side of the crankshaft housing, and the inner motor housing form the enclosure on the other side. The ends of the crankshaft housing are connected to the hot and cold end pressure shells by threaded flange connections.

4. Refrigerator Mounting

The refrigerator is mounted by brackets that attach to the sump structure. The brackets support the machine and transmit external environmental loads (i.e., shock and vibration) to the sump mounting points.

5. Hot-End Pressure Shell

The hot-end pressure shell is a cylinder with a flange on one end and a dome at the other. The dome supports the bearing, which supports the hot end of the hot displacer. A vacuum jacket completely surrounds the hot-end pressure shell assembly.

6. Cold-End Pressure Shell

The cold-end pressure shell is a cylinder with a flange on one end and a closure cap on the other end. A vacuum jacket completely surrounds the cold-end pressure shell.

7. Hot-End and Cold End-Displacers

Both displacers are cylindrical in shape, enclosed by domes on the outboard ends. The cylindrical inboard end contain rod bearings for the connecting rods that attach the displacers to the crankshaft. Linear bearings support the reciprocating displacers. The hot-end displacer is supported at both ends. The cold end displacer is supported at the inboard end only.
Stress Analysis Summary

Stress analysis was performed on major sections of the VM refrigerator. Design load considerations and criteria for material allowable stresses are overviewed below, followed by results of the stress analysis.

1. Loads and Material Stress Criteria

a. Loads

Pressure throughout the enclosure is nearly uniform except for very low pressure differentials (between $6.895 \times 10^3$ and $3.447 \times 10^4$ N/m$^2$ (1 and 5 psi) caused by flow friction. Although the pressure is essentially uniform within the machine, the pressure varies between the approximate limits of $6.088 \times 10^6$ N/m$^2$ (883 psia) and $6.895 \times 10^6$ N/m$^2$ (1000 psia) during each crankshaft revolution.

The components subjected to working pressures were analyzed for:

- Operating Pressure = $8.62 \times 10^6$ N/m$^2$ (1250 psia)
- Proof Pressure = $1.67 \times$ Operating Pressure = $1.420 \times 10^7$ N/m$^2$ (2060 psig)
- Burst Pressure = $2.25 \times$ Operating Pressure = $1.931 \times 10^7$ N/m$^2$ (2800 psig)

The maximum temperature at the end of the hot end pressure shell is 9220K (12000°F). Temperature at the end of the cold displacer is 650K (-343°F).

Dynamic loads resulting from the reciprocating motion of the displacers, and environmentally induced vibratory loads were also considered in the stress analysis.

b. Material Stress Criteria

Allowable stresses for various materials used in the refrigerator were generally established as follows:

- Components subjected to working pressure to withstand proof pressure without permanent deformation and burst pressure without rupture.
- For components subjected to high temperatures, the allowable creep stress is that value at which 0.2 percent creep occurs in 20,000 hours, (2.28 years).
- For components subjected to cyclic loads, the allowable stress for steel components will be based on a life requirement of $10^6$ cycles.
2. Analytical Summaries

a. Crankshaft Assembly Analysis

Both connecting rods simultaneously transmit three types of loads to the crankshaft: (1) dynamic loads; (2) gas pressure loads; and (3) vibratory loads. Dynamic loads are caused by linear acceleration of displacer masses; the gas pressure load is caused by pressure drop across the regenerative bed; and the vibratory load corresponds to the specified random vibration environment. These combined loads were imposed on the shaft, which was considered as an overhung beam on two supports. Shear and moment diagrams were drawn to identify maximum loads and determine maximum stresses.

A summary of the analysis performed on the crankshaft assembly is presented in Table 2-1.

b. Crankshaft Housing Analysis

The crankshaft housing was analyzed for \(8.62 \times 10^6\) N/m\(^2\) (1250 psi) maximum operating pressure and \(1.931 \times 10^7\) N/m\(^2\) (2800 psi) burst pressure. The motor outer housing was analyzed for loads due to random vibration. Portions of the crankshaft housing were included in the hot end pressure shell model and the cold end pressure shell model.

A summary of the analysis performed on the crankshaft housing is presented in Table 2-2.

c. Hot-End Pressure Shell Analysis

The hot end pressure shell was analyzed for \(8.62 \times 10^6\) N/m\(^2\) (1250 psi) maximum operating pressure and \(1.931 \times 10^7\) N/m\(^2\) (2800 psi) burst pressure. Stresses due to a vibratory load of 87 g (random vibration) were also determined. In addition, a temperature gradient was imposed on the shell, with \(922^\circ K\) (1200°F) at the dome and \(333^\circ K\) (140°F) at the flange.

A summary of the analysis performed on the hot-end pressure shell is presented in Table 2-3.

d. Cold-End Pressure Shell Analysis

The cold-end pressure shell was analyzed for internal pressure and vibratory loading. The analysis was performed for \(2 \times 10^6\) N/m\(^2\) (1250 psi) operating pressure and \(1.931 \times 10^7\) N/m\(^2\) (2800 psi) burst pressure.

A summary of the analysis performed on the cold-end pressure shell is presented in Table 2-4.

e. Hot-End and Cold-End Displacer Analysis

The hot displacer was analyzed for external pressure and temperature loading using Kalnin's shell analysis program. The analysis was performed at maximum...
<table>
<thead>
<tr>
<th>Detail Assy/Part</th>
<th>Design Requirement</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crankshaft stacked-parts, cold crank, bearing journal, and bolt</td>
<td>(a) Stacked-part interfaces must not separate under bending load.</td>
<td>Bolt preload sufficient to preclude separation of stacked-parts.</td>
</tr>
<tr>
<td></td>
<td>(b) Combined axial prestress and bending stresses shall not exceed allowable stress.</td>
<td>Stresses due to preload and vibration less than allowable stresses. Tensile, bending, and thread shear stresses computed.</td>
</tr>
<tr>
<td></td>
<td>(c) Shear pins and/or friction between stocked-parts must transmit torque.</td>
<td>Forces transmitted to crankshaft convert to torque due to crank eccentricity. Shear pins between stacked parts was found sufficient to transmit torque.</td>
</tr>
<tr>
<td>Overhang section</td>
<td>Maximum bending and shear stresses must be endured.</td>
<td>Calculated stresses low; margin of safety adequate.</td>
</tr>
<tr>
<td>Critical speed analysis</td>
<td>Critical speed must be higher than maximum operating speed.</td>
<td>AiResearch computer program V0245 determined first rigid-body mode to be significantly higher than 400 rpm maximum operating speed.</td>
</tr>
<tr>
<td>Crankshaft rotary unbalance</td>
<td>Determine unbalance</td>
<td>Reciprocating linear unbalance was determined as well as rotary unbalance. Results indicated that correction for rotary unbalance not required because: 1) total linear unbalance cannot be corrected; 2) VM is low-speed machine; 3) envelope limitation; 4) past experience with this problem.</td>
</tr>
<tr>
<td>Connecting rod, wrist pin and retainer</td>
<td>Parts must endure displacer forces transmitted to crankshaft.</td>
<td>Calculated stresses using displacer forces indicate adequate safety margin.</td>
</tr>
<tr>
<td>Detail Assy/Part</td>
<td>Design Requirement</td>
<td>Comments</td>
</tr>
<tr>
<td>-----------------</td>
<td>---------------------</td>
<td>----------</td>
</tr>
<tr>
<td>Housing at cross tube intersection</td>
<td>Must withstand maximum pressure.</td>
<td>Particular attention given to fact that sump housing is weakened by intersection of the cross tubes. Analysis showed the reinforcement around the holes are adequate to resist the pressures.</td>
</tr>
<tr>
<td>Spherical dome and cylinder intersection (cold end)</td>
<td>Must withstand pressure and vibratory loading.</td>
<td>Stresses due to the vibratory loads were added to pressure load stresses and were found to be at safe levels.</td>
</tr>
<tr>
<td>Flanged connections</td>
<td>Installation torque must be sufficient to prevent joint separation. Must withstand preload plus applied load.</td>
<td>Installation torque specified on drawing sufficient to prevent joint separation. Stresses are low.</td>
</tr>
<tr>
<td>Tubular section, cold end</td>
<td>Must withstand pressure and vibratory loading.</td>
<td>Combined stresses are at safe levels.</td>
</tr>
<tr>
<td>Motor inner housing</td>
<td>Must withstand internal pressure.</td>
<td>Shell analysis and flange analysis show safe stress level.</td>
</tr>
<tr>
<td>Motor outer housing</td>
<td>Must withstand vibratory loads.</td>
<td>Analysis of housing and fasteners show acceptable stress levels for 98 g's parallel and 68 g's perpendicular to the motor centerline.</td>
</tr>
<tr>
<td>Detail Ass'y/Part</td>
<td>Design Requirement</td>
<td>Comments</td>
</tr>
<tr>
<td>------------------</td>
<td>--------------------</td>
<td>----------</td>
</tr>
<tr>
<td>Hot-end pressure shell; included hot end flange and adjacent tubular portion of sump housing</td>
<td>Must have adequate strength to support pressure, temperature, and vibratory loading.</td>
<td>Hot end flange and adjacent tubular portion of sump housing were included in the shell analysis model. Analysis was performed with Kalnins' shell program. Pressure and temperature loads were imposed on the shell. Resulting meridional stresses were added to stresses from vibratory loading. Pressure shell strength was found to be adequate to support pressure, temperature, and vibratory loading.</td>
</tr>
<tr>
<td>Flanged connection</td>
<td>Installation torque must be sufficient to prevent joint separation. Must withstand preload plus applied load.</td>
<td>Installation torque specified on drawing sufficient to prevent joint separation. Stresses are low.</td>
</tr>
<tr>
<td>Shell/displacer deflections</td>
<td>Must be no interference between the shell and displacer under vibratory loading.</td>
<td>Deflections were calculated to determine if interference between the displacer and the heat exchanger sleeve would occur. The computer deflections were found to be less than the clearances.</td>
</tr>
<tr>
<td>Hot bearing shaft</td>
<td>Must withstand vibratory loads transmitted to dome, in combination with temperature/pressure loads.</td>
<td>Maximum bending moment at root of hot bearing shaft is transmitted to the dome, generating meridional and hoop stresses. These stresses were manually calculated and added to pressure and temperature stresses. The combined stress did not exceed the allowable stress.</td>
</tr>
<tr>
<td>Hot end</td>
<td>Vacuum jacket must be capable of resisting collapsing pressure</td>
<td>Calculations show that collapsing pressure is much higher than actual external pressure.</td>
</tr>
<tr>
<td>Detail Assy/Part</td>
<td>Design Requirement</td>
<td>Comments</td>
</tr>
<tr>
<td>-----------------</td>
<td>--------------------</td>
<td>----------</td>
</tr>
<tr>
<td>Pressure shell</td>
<td>Must have adequate strength to withstand pressure, temperature, and vibratory loading.</td>
<td>Pressure shell analyzed with Kalnins' shell program for internal pressure and temperature loading. The temperature was 333\degree K (140\degree F) at the sump and 65\degree K (-343\degree F) at the cold end. Stresses due to vibratory loads were computed and combined with shell stresses due to max operating pressure. Computed stresses were below the allowable stresses.</td>
</tr>
<tr>
<td>Flanged connection</td>
<td>Installation torque must be sufficient to prevent joint separation. Must withstand preload plus applied load.</td>
<td>Installation torque specified to prevent joint separation. Stresses are low.</td>
</tr>
<tr>
<td>Cold-end pressure shell and displacer</td>
<td>Determine whether clearance is sufficient to preclude contact between the shell and displacer under vibratory loads.</td>
<td>Analysis indicates contact will occur between the shell and displacer during the application of the vibratory loads. Such contact, however, is not detrimental because: 1) the deflections are elastic and momentary, and 2) the probable consequence of shell displacer contact would be a momentary stalling of the engine, after which the engine will continue to operate without performance degradation.</td>
</tr>
<tr>
<td>Cold end displacement due to self-induced vibration must not exceed 0.000200 inches.</td>
<td>AiResearch computer program V0245 was used to determine the displacement at the cold end due to self-induced vibration. At max operating speed (400 rpm), displacement at cold end was found to be less than the allowable displacement.</td>
<td></td>
</tr>
<tr>
<td>Vacuum jacket</td>
<td>Must be capable of resisting external pressure.</td>
<td>Analysis indicates that vacuum jacket will withstand external pressure with adequate margin of safety.</td>
</tr>
</tbody>
</table>
operating pressure and at burst pressure. The temperature gradient was $922^\circ K$ (1200°F) at the hot end and $333^\circ K$ (1400°F) at the inboard bearing. The external collapsing pressure were also determined. A summary of the analysis in the hot-end and cold-end displacers is presented in Table 2-5.

**TABLE 2-5**

**HOT END AND COLD END DISPLACER STRESS ANALYSIS SUMMARY**

<table>
<thead>
<tr>
<th>Detail Assy/Part</th>
<th>Design Requirement</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacer</td>
<td>Must endure collapsing pressure</td>
<td>Calculated collapsing pressure was found to be much higher than the actual external operating pressure.</td>
</tr>
<tr>
<td></td>
<td>Must endure combined pressure and temperature loads</td>
<td>Hot displacer was analyzed with Kalnins' shell program. A model of the entire displacer and part of the bearing journal section was prepared. Pressure and temperature loads were imposed on the model. Maximum stresses from computer output were less than the allowable stresses. Cold displacer stresses, computed manually, are less than the allowable stresses.</td>
</tr>
<tr>
<td>Hot displacer bearing journal</td>
<td>Tip deflection must not cause interference under vibratory loading.</td>
<td>Maximum deflection at the tip was calculated to determine if interference would occur between the journal and the bearing under vibratory loads. The calculated radial displacements were found to be less than the minimum running clearance.</td>
</tr>
</tbody>
</table>
SECTION 3
DESIGN REQUIREMENTS
INTRODUCTION

The Vuilleumier cryogenic refrigerator has been designed to meet requirements specified by NASA Goddard Space Flight Center (GSFC). This section summarizes NASA GSFC design requirements that served as a basis for the original VM layout drawings.

NASA GSFC PRELIMINARY DESIGN REQUIREMENTS

The NASA GSFC preliminary design requirements are summarized in Table 3-1. These include operational performance requirements and the qualification and operational environments that the machine must endure.

STRESS ANALYSIS

The stress analysis and thermal analysis were based on the layout drawing (Figure 2-1). As a result of these analyses and subsequent design iterations, the drawing was changed to update the basic design.

The vibration requirements for the fractional watt VM refrigerator are given in the following figures:

Figure 3-1. Sinusoidal Vibration in Lateral Direction
Figure 3-2. Sinusoidal Vibration in Longitudinal Direction
Figure 3-3. Random Vibration for Each Axis.
TABLE 3-1
PRELIMINARY DESIGN REQUIREMENTS
VUILLEUMIER CRYOGENIC ENGINE

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Cooling capacity at 65°K after 2 yrs</td>
<td>1/4 watt min.</td>
</tr>
<tr>
<td>Thermal power input</td>
<td>80 watts max.</td>
</tr>
<tr>
<td>Electrical power for drive motor</td>
<td>10 watts max.</td>
</tr>
<tr>
<td>Temperatures</td>
<td></td>
</tr>
<tr>
<td>Thermal power input</td>
<td>Not specified</td>
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<tr>
<td>Heat rejection</td>
<td>180°F</td>
</tr>
<tr>
<td><strong>Weight</strong></td>
<td></td>
</tr>
<tr>
<td></td>
<td>18 lbs. max.</td>
</tr>
<tr>
<td></td>
<td>Not specified</td>
</tr>
<tr>
<td><strong>Size</strong></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Not Specified</td>
</tr>
<tr>
<td><strong>Spacecraft Qualification Environment</strong></td>
<td></td>
</tr>
<tr>
<td>Shock</td>
<td></td>
</tr>
<tr>
<td>Humidity</td>
<td></td>
</tr>
<tr>
<td>Temperature</td>
<td>30 ±2°C</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>90 ±3%</td>
</tr>
<tr>
<td>Vibration</td>
<td></td>
</tr>
<tr>
<td>Sinusoidal</td>
<td>See Figure 3-1</td>
</tr>
<tr>
<td>Lateral (X-X and Y-Y)</td>
<td>See Figure 3-2</td>
</tr>
<tr>
<td>Longitudinal (Z-Z)</td>
<td>See Figure 3-3</td>
</tr>
<tr>
<td>Random</td>
<td></td>
</tr>
<tr>
<td>Prelaunch environment</td>
<td></td>
</tr>
<tr>
<td>Temperature</td>
<td>75 ±5°F</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>50% max.</td>
</tr>
<tr>
<td>Duration</td>
<td>3 weeks</td>
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<tr>
<td>Design load</td>
<td></td>
</tr>
<tr>
<td>Ultimate static load factor</td>
<td>30 g</td>
</tr>
<tr>
<td>Storage temperature (up to 3 yrs)</td>
<td>0 to 130°F</td>
</tr>
<tr>
<td>Pressure (up to 3 yrs.)</td>
<td>1 atm to 10⁻¹⁰ torr</td>
</tr>
<tr>
<td><strong>Operating Environment</strong></td>
<td></td>
</tr>
<tr>
<td>Shock</td>
<td></td>
</tr>
<tr>
<td>Acceleration</td>
<td></td>
</tr>
<tr>
<td>Level</td>
<td></td>
</tr>
<tr>
<td>Duration</td>
<td></td>
</tr>
<tr>
<td>Vibration</td>
<td></td>
</tr>
<tr>
<td>Sinusoidal</td>
<td>See Figure 3-1</td>
</tr>
<tr>
<td>Lateral (X-X and Y-Y)</td>
<td>See Figure 3-2</td>
</tr>
<tr>
<td>Longitudinal (Z-Z)</td>
<td>See Figure 3-3</td>
</tr>
<tr>
<td>Random</td>
<td></td>
</tr>
<tr>
<td>Temperature</td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td></td>
</tr>
</tbody>
</table>
Figure 3-1. Sinusoidal Vibration in Lateral Direction
Figure 3-2. Sinusoidal Vibration in the Longitudinal Direction.

SINUSOIDAL VIBRATION
LONGITUDINAL Z AXIS
SWEEP RATE: 2.0 OCT/MIN

DOUBLE AMPLITUDE - IN.

FREQUENCY - HZ

0.48

0.1

0.01

0.001

0.0001

0.00001

1

10

100

1000

200 CPS.

0.00001
Figure 3-3. Random Vibration Requirements for Each Axis
SECTION 4
MATERIAL PROPERTIES
SECTION 4

MATERIAL PROPERTIES

This section references/reproduces material properties information that was used during stress analysis performed on the NASA Goddard 1/4 watt Vuilleumier cryogenic refrigerator. Certain material properties catalog information was copyrighted and will not be reproduced here. In this case, the publication and its source are listed.
4. MATERIAL PROPERTIES

4.1 INCONEL 718
4.2 PH 13-8 Mo
4.3 7075 ALUM ALLOY
4.4 TITANIUM ALLOY Ti-6Al-4V
4.5 TUNGSTEN-CARBIDE, CA 310
4.1 INCONEL 718

The specifications pertaining to Inconel 718 which were used during the stress analysis are listed below.

Aerospace Material Specifications (by Society of Automotive Engineers)

AMS5662B
Issued 9-1-65
Rev. 11-1-67
Alloy Bars, Forgings, and Rings, Corrosion and Heat Resistant

AMS5663B
Issued 9-1-65
Rev. 11-1-67
Alloy Bars, Forgings, and Rings, Corrosion and Heat Resistant

AMS5596C
Issued 1-31-64
Rev. 11-1-68
Alloy Sheet, Strip, and Plate, Corrosion and Heat Resistant

Other Specifications

Catalog T-39
Huntington Alloy Products Division; Inconel 718
International Nickel Corp.
Catalog T-39, pages 13, 14, 15, and 16
4.1 MECHANICAL AND PHYSICAL PROPERTIES OF INCONEL 718

Alloy: Inconel 718, AMS 5662, AMS 5663, AMS 5596

Form: Bars, Forgings, Sheet and Plate; Consumable Electrode or Vacuum Induction Melted

Condition: Solution and Precipitation Heat Treated

Mechanical Properties at Room Temperature (Minimum):

- Ultimate Tensile Strength, KSI: 180.0
- Yield Strength (0.2% offset), KSI: 150.0
- Elongation, %: 10.0

Physical Properties:

- Modulus of Elasticity, $10^6$ psi: 29.6
- Density, lb/cu. in.: .297
- Thermal Conductivity, BTU/hr/ft$^2$/°F/ft: 6.3 at 70°F
- Mean Coefficient of Expansion, $10^{-6}$in/in/°F: 7.6 (70°F to 400°F)
CONDITION:
1750°F FOR 1 HR., AIR COOLED
1325°F FOR 8 HR., FURNACE COOLED
1150°F FOR 10 HR., AIR COOLED

EFFECT OF TEMPERATURE ON TENSILE PROPERTIES OF INCONEL 718 BAR
LARSON - MILLER PARAMETER

\[(T + 460)(20 + \log_{10} t) \times 10^{-3}\]

WHERE \(T = \text{TEMPERATURE, } ^\circ\text{F}\)
\(t = \text{TIME, HOURS}\)

TYPICAL STRESS-RUPTURE CURVE
FOR INCONEL 718

200
100
90
80
70
60
50
40
30
20
10

30 31 32 33 34 35 36 37 38 39 40 41 42 43

STRESS, KSI

RUPTURE

\(.2\% \text{ CREEP}\)

HEAT TREATMENT
1750°F - 1 HR. - AIR COOLED
1325°F - 8 HR. - FURNACE COOLED
1150°F - 10 HR. - AIR COOLED
TYPICAL PHYSICAL PROPERTIES OF INCONEL 718
4.1 INCONEL 718

CREEP AND CREEP RUPTURE VS. TEMPERATURE

\[ LM = \text{LARSON-MILLER PARAMETER} \]
\[ = (T + 460)(\log_{10}\frac{t}{T}) \times 10^3 \]

WHERE: \( T = \text{Temp.} \), \(^\circ\text{F}\)
\( t = \text{Time, hrs.} \)

SEE PAGE 4-6 FOR CREEP AND CREEP RUPTURE VS. LM

FOR \( t = 20,000 \), HRS,

<table>
<thead>
<tr>
<th>( T ) ((^\circ\text{F}))</th>
<th>LM [\times 10^3]</th>
<th>1.2% CREEP (ksi)</th>
<th>CREEP RUPTURE (ksi)</th>
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ENDURANCE, CREEP, AND CREEP RUPTURE ALLOWABLES VS. TEMPERATURE FOR INCONEL 718
4.2 PH 13-8 Mo

The specification sheets pertaining to ARMCO PHI3-8MO used during the stress analysis are listed below:

Catalog S-33b Armco PHI3-8MO, Precipitation-Hardening Stainless Steel Bar, Wire, Plate, and Forging Armco Steel Corp. Billets, Pages 4, 6, and 12

From MIL-HDBK-15, PH 13-8 Mo in condition H1000 have:

\[ F_{tu} = 205,000 \text{ psi} \]
\[ F_{ty} = 197,000 \text{ psi} \]

The analysis will use:

\[ F_{tu} = 205,000 \text{ psi} \]
\[ F_{ty} = 190,000 \text{ psi} \]
GOODMAN DIAGRAM

PH-13-8 Mo
CONDITION H-1000
F_{ah} = 200,000 psi
F_{as} = 130,000 psi

REF: ARMCO PH-13-8 Mo "PRELIMINARY"
HARDENED STAINLESS STEEL
BAR, WIRE, BOLT AND FORGING
BULLETS, PRODUCT DATA 5-278
4.3 7075 Alum. Alloy

7075-T73 per QQ-A-250/12

From MIL-HDBK-5,

\[ F_{tu} = 67,000 \, \text{PSI} \]
\[ F_{ty} = 56,000 \, \text{PSI} \]

Stress corrosion "threshold" stress = \( \frac{47,000 \, \text{PSI}}{\text{(MIN.)}} \) (short transverse)
MECHANICAL AND PHYSICAL PROPERTIES OF TITANIUM ALLOY TI-6Al-4V

Alloy: TI-6Al-4V, AMS 4928
Form: Bars and Forgings
Condition: Annealed

Mechanical Properties at Room Temperature:

- Ultimate Tensile Strength, KSI: 130.0 minimum
- Yield Strength (0.2% offset), KSI: 120.0 minimum
- Elongation, % in 4D: 10.0 minimum

Physical Properties:

- Modulus of Elasticity, 10^6 psi: 16.0
- Density, lb/cu.in.: 0.160
- Thermal Conductivity, BTU/hr/ft^2/°F/ft: 4.2 at 70°F
- Mean Coefficient of Expansion, 10^-5 in/in/°F: 5.3 (70°F to 200°F)
- Specific Heat, BTU/lb/°F: 0.130 at 70°F
HIGH-TEMPERATURE TENSILE PROPERTIES OF Ti-6Al-4V ALLOY
4.5 TUNGSTEN-CARBINE, CABIO

The specification sheets pertaining to tungsten carbide used during the stress analysis are listed below:

Catalog 300 Series High Strength Microcarbides for Tools and Dies,
Copyright 1969 4-page specification
Carmet Co.

DENSITY: 14.6 g/cm³
TRANSVERSE RUPTURE STRENGTH: 450,000 PSI
HARDNESS: 92 Rc

INFORMATION RECEIVED FROM:
CARMET CO., STANTON, CALIF.
PHONE: AC 714 - 828-5820
MR. LARRY WILCASTER INFORMED THAT THE SHEAR
STRENGTH OF THIS MATERIAL IS NEARLY EQUAL TO
ITS TRANSVERSE RUPTURE STRENGTH.

\[ \tau_{tr} = 450,000 \text{ PSI} \]

\[ \tau_{w} = \frac{\tau_{tr}}{2.5} = 180,000 \text{ PSI} \]
SECTION 5
CRANKSHAFT ASSEMBLY STRESS ANALYSIS
5. CRANKSHAFT ASSY.

5.1 TASKS

5.2 CRITICAL SPEED ANALYSIS

5.3 SHAFT ANALYSIS

5.3.1 VIBRATORY LOADS

5.3.2 PRESSURE LOADS

5.3.3 DYNAMIC (SELF-INDUCED) LOADS

5.3.4 COMBINED LOADS

5.3.5 STRESS ANALYSIS

5.3.6 BOLT PRELOAD

5.3.7 SHEAR PINS

5.4 CONNECTING ROD, HOT DISPLACED

5.5 CONNECTING ROD, COLD DISPLACED
5.1 TASKS

THE CRANKSHAFT ASSY ANALYSIS CONSISTS OF THE FOLLOWING TASKS:

- DETERMINE SHAFT CRITICAL SPEED
- DETERMINE SINUSOIDAL AND RANDOM VIBRATORY LOADS, PRESSURE LOADS, AND DYNAMIC (SELF INDUCED) LOADS, AND COMBINE AS APPLICABLE.
- PERFORM STRESS ANALYSIS OF SHAFT AND CONNECTED COMPONENTS
- DETERMINE CRANKSHAFT BOLT AXIAL PRELOAD AND TORQUE REQUIRED.
FIG. 5-1 CRANKSHAFT ASSY.
5.2 CRITICAL SPEED ANALYSIS

The critical speed was determined by using Airesearch computer program V0245.

The mass model and stiffness model used are shown in Figures 5-2 and 5-3.

The first mode (critical) frequency was found to be 3770 RPM (62.8 cps).

Compared to the 400 RPM maximum operating speed, the crankshaft critical speed is satisfactory.
FIG. 5-2 CRANKSHAFT MASS STATIONS
FOR CRITICAL SPEED ANALYSIS
5.7 CRANKSHAFT CRITICAL SPEED

UNBALANCE AT LAT END CRANK

\[ W_H = W_1 + W_2 + \frac{1}{r} \left( W_3 + W_D \right) \]

\( W_H \) = UNBALANCE AT CRANK RADIUS.
\( W_1 \) = WT. OF CRANK + BRG. (85256)
\( W_2 \) = WT. OF CONNECTING ROD (85256)
\( W_D, W_D = WT. OF CONNECTING ROD "DUMP-DELIC" AT CRANK BRG. (B) AND Wrist Pin (D), respectively.
\( W_3 \) = WT. OF DISPLACED, Wrist Pin Retainer, Wrist Pin, Wrist Pin BRG.

\( W_1 = \pi \left( 33.8^2 \right) \left( .50 \right) \left( .297 \right) + .14 \times 0.193 = 0.193 \text{ LBS.} \)

\( W_2 = 0.04 \times 0.193 \)

\( W_3 = 1.12 + 0.01 + 0.02 + 0.01 = 1.22 \text{ LBS.} \)

Assume \( W_D = W_2 = 0.02 \)

\( W_H = .193 + .02 + \frac{1}{r} \left( 1.22 + 0.02 \right) = 0.033 \text{ LBS.} \)

\( r = 0.24 \)

\( F_4 = 0.033 \times 0.24 \times 16 = 3.20 \text{ IN.-OZ.} \)

NOTE: * VALUES FROM WEIGHT REPORT, APPENDIX A.
5.2 CRANKSHAFT CRITICAL SPEED

UNBALANCE AT COLD END CRANK

REFER TO EQUATION AND TERMINOLOGY ON PRECEDING PAGE

\[ W_c = W_1 + W_c + \frac{1}{c} (W_3 + W_4) \]

\[ W_1 = \pi (3.38) (3.97) + 0.14 = 0.193 \text{ lbs} \]

\[ W_c = 0.04 \text{ lbs} \]

\[ W_3 = 0.16 + 0.05 + 0.02 + 0.01 = 0.24 \text{ lbs} \]

Assume \( W_c = W_4 = 0.02 \text{ lbs} \)

\[ W_c = 0.193 + 0.02 + \frac{1}{c} (0.24 + 0.02) = 0.343 \text{ lbs} \]

\[ r = 0.22 \]

\[ F_{u+} = 0.343 \times 14 \times 0.22 = 1.21 \text{ in-lb} \]

TOTAL UNBALANCE

\[ U = F_{u+} + F_c = 3.42 \text{ in-lb} \]

\[ \theta = 20.7^\circ \]
5.2 CRANKSHAFT CRITICAL SPEED

INPUT DATA FOR VO245

STATION 7

\[ x_7 = 4.855 + \frac{4}{3} \times 50 = 5.105 \text{ in.} \]

\[ m_7 = \frac{1.833}{0.00215803} = 863 - \text{sec}^{-2} / \text{in.} \]

\[ i_{p7} = \frac{w}{q} = \frac{\pi (1.338)(0.50)(1.977) \times (1.338)^2}{32} = 1.887 \times 10^{-5} \text{ lb}-\text{sec}^{-2} / \text{in.} \]

\[ i_{dc7} = \frac{w}{3} \times \left( 3 \times 1.338^2 + 0.50 \right) = 6.8202 \times 10^6 \text{ lb} \cdot \text{sec}^{-2} / \text{in.} \]

STATION 8

\[ x_8 = 5.355 + 2.5 = 5.605 \text{ in.} \]

\[ m_8 = \frac{348}{32} = 0.00088860 \text{ lb} \cdot \text{sec}^{-2} / \text{in.} \]

\[ i_{p8} = i_{p7} = 1.887 \times 10^{-5} \text{ lb} \cdot \text{sec}^{-2} / \text{in.} \]

\[ i_{dc8} = i_{dc7} = 6.8202 \times 10^6 \text{ lb} \cdot \text{sec}^{-2} / \text{in.} \]

ORIGINAL PAGE IS OF POOR QUALITY.
### V.L. CYROGENIC ENGINE CRANKSHAFT CRITICAL SPEED ANALYSIS

##### Mike Morthyny Jan 74

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**Model:** 04-07-14

**Date:** 12-24-74

**Part No:** 572447

**Calc No:** 67263

**Sheet No:** 5-11

**Prepared by:** M. Millard

**Checked by:** W. W. M.

**Drawn by:** J. A. K.

**Airsec Company:** 2.07910.04 CALIFORNIA
### Table: VIBRATIONS - CRANKSHAFT CRITICAL SPEED ANALYSIS

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### Table: AVG PHASES

| STATION | X | Phase | Y | Phase | AVG | | |
|---------|---|-------|---|-------|-----|---|
| 5       | 2,1222E-06 | 1,8000E-02 | 0,0000 | 0,0000 | 0,0000 |
| 9       | 6,2721E-05 | 1,8000E-02 | 0,0000 | 0,0000 | 0,0000 |

### Table: ZK PHASES

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SEARCHING COMPLETED FROM 0,0 TO 6,2832E-03 RAD/SEC

PREPARED BY: MIKE MORIYAMA

DATE: JAN 74

PAGE 5-12
5.3 SHAFT ANALYSIS

5.3.1 VIBRATORY LOADS

The sinusoidal vibration spectrum frequency range is 5 to 200 cps, and the random vibration spectrum frequency range is 20 to 2000 cps. (Ref. Section 3)

The critical speed analysis yielded:

First mode frequency, \( f_1 = 62.8 \text{ cps} \) \( \text{Ref. 74 5-11} \)

Second mode frequency, \( f_2 = 580 \text{ cps} \)

The sinusoidal vibratory loading, based on an assumed magnification factor of 10, is:

\[
\text{L.F.} = \frac{7.1 \times 10}{1.0} = 71.
\]

(Ref. Section 3)

The random vibratory loading, based on an assumed magnification factor of 10, is:

\[
\text{L.F.} = \left( \frac{\pi^2}{4} \times \text{PSD} \times T \times f_1 \right)^{\frac{1}{3}}
\]

At \( f_1 = 62.8 \text{ cps} \), \( \text{PSD} = 0.010 \) (Ref. Section 3)

\[
\text{L.F.} = 3 \left( \frac{\pi^2}{4} \times 0.010 \times 1.0 \times 62.8 \right)^{\frac{1}{3}} = 9.42 \text{ g}^3 \text{ (3 sigma response)}
\]

At \( f_2 = 580 \text{ cps} \), \( \text{PSD} = 0.045 \) (Max Value)

\[
\text{L.F.} = 3 \left( \frac{\pi^2}{4} \times 0.045 \times 1.0 \times 580 \right)^{\frac{1}{3}} = 30.7 \text{ g}^3 \text{ (3 sigma response)}
\]
5.3 SHAFT ANALYSIS

5.3.1 VIBRATORY LOADS

The loading from the random vibration spectrum at 580 GFS is greatest.

\[ L.F. = 60.7 \text{ GFS} \]

The loads on the crankshaft are:

\[ P = (\text{Wt. of displacer, wrist pin retainer, wrist pin, wrist pin drag, connecting rod, connecting rod drag & crank}) \times (\text{load factor}) \]

\[ P = (1.12 + .07 + .02 + .01 + .04 + .14 + .053) \times (60.7) \]

\[ = 88.2 \text{ LBS.} \]

\[ P_e = (0.16 + .05 + .02 + .01 + .04 + .14 + .053) \times (60.7) \]

\[ = 28.7 \text{ LBS.} \]

5.3.2 PRESSURE LOADS

\[ \delta_{\text{max}} = 5.356 \text{ (assumed for drag only)} \]

\[ P_{\text{H}} = \frac{\pi}{4} \times 2.25 \times 5. = 20. \text{ LBS.} \]

\[ P_{e} = \frac{\pi}{4} \times .40 \times 5. = 0.628 \text{ LBS.} \]
5.3 SHAFT ANALYSIS

5.3.3 DYNAMIC LOADS

The self-induced dynamic loading at the cranks are:

\[
P_x = \frac{\omega^2}{2} \left[ (W_1 + W_{bc}) R + W_{bc}^2 R + W_1 D_1 \right] \cos \theta + (W_1 + W_{bc}) l \left( \frac{D_1}{R} \right) \cos 2\theta
\]

\[
P_x = \frac{\omega^2}{2} \left[ W_1 D_1 + W_{bc}^2 \right] \sin \theta
\]

WHERE:
- \( W_1 \) = WT. OF CRANK AND BRAZING
- \( W_{bc} \) = WT. OF CONNECTING ROD
- \( W_{bc}^2 \) = WT. OF CONNECTING ROD "DUMB-BELLS" AT CRANK BRAZING, AND DISPLACEMENT RESPECTIVELY
- \( W_3 \) = WT. OF DISPLACEMENT, Wrist Pin Retainer, Wrist Pin
- \( R \) = CRANK RADIUS
- \( D_1 \) = C.G. OF CRANK

ORIGINAL PAGE IS OF POOR QUALITY.
5.3 SHAFT ANALYSIS

5.3.3 DYN. LOADING - HOT DISPLACER

REF. PG. 5-1 FOR WEIGHTS

\[ W_1 = 193 \text{ lbf} \]
\[ W_2 = 0.04 \text{ lbf} \]
\[ W_3 = W_4 = W_5 = 0.02 \text{ lbf}. \text{ (assumed)} \]
\[ W_3 = 1.22 \text{ lbf} \]
\[ R = 0.24 \text{ in.} \]
\[ P_1 = P_4 = 0.24 \text{ in.} \]
\[ L = 1.75 \text{ in.} \]
\[ W = 400, \text{ RPM} = 41.89 \text{ rad/sec} \]

\[ W_1 + W_2 = 1.10 \]
\[ P_2 = \frac{41.89}{384} \left[ (1.24 \times 2.4 + 0.02 \times 2.4 + 1.93 \times 2.4) \cos \theta \right. \\
\[ + (1.24 \times 2.4^2) \cos 2\theta \right] \]
\[ \frac{1.75}{1.75} \]
\[ = 1.9530 \cos \theta + 0.04081 \cos 2\theta \]

\[ P_3 = \frac{41.89}{384} \left( 1.93 \times 2.4 + 0.02 \times 2.4 \right) \sin \theta \]
\[ = 0.23239 \sin \theta \]

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</table>
5.3 SHAFT ANALYSIS

5.3.4 COMBINED LOADING

The loads due to vibration (random) and pressure will be combined. The dynamic (self induced) loads will be neglected; \( P_{\text{max}} = 1.63 \text{ LBS} \) and \( P_{c_{\text{max}}} = 0.587 \text{ LBS} \).

To determine the vibratory loads, the max. acceleration will be applied to the shaft and attached components.

AIRESEARCH COMPUTER PROGRAM V0245 will be used to obtain the shear and moment along the crankshaft. An acceleration of 40.7 g's will be applied to the masses, at stations 7 and 8, displaced dynamic loads and pressure loads were applied also. The computer output and shear and moment diagrams are shown on the following page.
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<thead>
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5.3 SHAFT ANALYSIS

5.3.4 COMBINED LOADS

SHEAR DIAGRAM

\[
\begin{align*}
\text{SH.} & : 2.89 \\
\text{STA.} & : 1, 2, 3, 4, 5, 6, 7, 8, 9
\end{align*}
\]

\[
\text{MOMENT DIAGRAM}
\]

\[
\text{MOMENT (14-18)} = 438
\]

107, 130, 170, 854, 384, 132
5.3 SHAFT ANALYSIS

5.3.5 STRESS ANALYSIS

<table>
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<tr>
<th>X (in)</th>
<th>M (in-lb)</th>
<th>R (in)</th>
<th>$f_b$ (psi)</th>
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<td>.734</td>
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<td>.094</td>
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MAX $f_b$ (40,000, psi) occurs in the bolt.

Mat'1 is cast iron 13-5 1000 condition H 1000.

$F_{u}$ = 205,000, psi

$F_{y}$ = 190,000, psi

The bolt will be torqued to 45 to 50 in-lbs.

$P = \frac{F K}{(0.8 + 0.45)}$

$P_{max} = \frac{50 \times 37}{(0.83 + 1.76)} = 3700, LBS$ (Found at X = 5.555 in.)

$F_{c} = \frac{P}{A} = \frac{3700,000}{\pi \times .094} = 133,000, PSI$ (At X = 5.555 in.)

$F_{b} + F_{c} = 173,000, PSI$

M. S. = \frac{190,000}{173,000} = \frac{1.10}{173,000}$
5.3 SHAFT ANALYSIS
5.3.5 STRESS ANALYSIS

ITEM 1 of Dwg. 852567 - CHECK HOOP TENSION DUE TO BOLT THREAD LOAD
- CHECK SHEAR STRESS IN THREADS

MATERIAL: STRESS PH 13-8 M. CONDITION: H 1000
Ftu = 205 ksi  Fty = 190 ksi
Fty = 154 ksi

HOOP STRESS

\[ U = \frac{D_0}{D_m} \left[ \frac{D_0}{D_m} \right]^2 \left( D_m - D_0 \right) \]

\[ D_m = \text{MAX. MAJOR DIA.} = 0.197 \]
\[ D_0/D_m = 1.72 \quad c_2 = 0.56 \]

\[ \text{Assume } c_2 = 1.5 \quad n = 1.5 \times 171 \times 32 = 8 \]

\[ \text{Therefore} \]
\[ \text{MAX } P = 3700, 1200 \text{ (see preceding page)} \]

\[ \frac{F_t}{c_2} = \frac{1.21 F_t}{(0.44)(0.44)} \]
\[ = 0.56 \times 32 \times 3700 \times \frac{51,900 \times 53,700 \times 95,400 \times 755}{9} \times (0.335 + 1.377) \]
\[ M.S. = 190,000, -1 = \frac{51,900}{3.66} \]

\[ \text{THREAD SHEAR} \]

\[ \text{For equal load distribution: } \]
\[ F_s = \frac{F_t}{4} = \frac{3700 \times 2}{4 \times 171 \times 1.5 \times 171} \]
\[ \approx 35,700, \text{psi} \]

\[ \text{For 1st thread carrying } 22.2\% \text{ of the total load } (3/2 + \text{th}) \]
\[ F_s = \frac{0.222 \times 53,700 \times 95,400 \times 755}{1.25} \]
\[ M.S. = 0.19 \]
5.3 Shaft Analysis

5.3.4 Bolt preload required

The mating surfaces of the hot displaced crank and cold displaced crank will be treated as a "center" with O.D. of "a" in. and I.D. of .188 in. (bolt dia.),

\[ d = 2r = (x^2 + y^2)^{\frac{1}{2}} \]
\[ x = .24 \]
\[ y = .22 \]
\[ r = .335 \]

\[ A_c = \frac{\pi}{4} (\text{.341 in.} - .188 \text{ in.}) = .6636 \text{ in.}^2 \]

\[ V = 49.2 \text{ lbf } (x = 5.355 \text{ in.}) \text{ REF. PG. 5-20} \]

Required axial load assuming all shear transmitted by friction:

\[ P = \frac{V}{\mu} = \frac{49.2}{.10} = 492 \text{ lbf} \]

To prevent separation under moment loading

\[ M = 24.1 \text{ in.-lbf} \text{ (REF. PG. 5-21)} \]
\[ I = \frac{\pi}{4} (\text{.341 in.} - .188 \text{ in.})^{4} \]

\[ f_b = \frac{M \mu}{I} = \frac{24.1 \times 172}{6.02 \times 10^{-4}} = 7,500 \text{ lbf} \]

\[ P_f = 7,500 \times .0636 = 477 \text{ lbf} \]

Applying a factor of safety of 2.75

\[ P_f = 1070 \]

Use \[ P = 1100 \text{ lbf} \]
5.3 SHAFT ASSY

5.3.4 BOLT PRELOAD (PN 852697)

THE THREADS WILL BE "MOLY COATED", BASED ON M = .5.

\[ \mu = 0.05 \text{ TO } 0.15 \]

Axial load load reqd. \( P = 1100 \text{ LBS.} \)

\[ T = \frac{p}{\mu} \left( \frac{d_1 + d_2}{K} \right) \]

\[ = \frac{1100 \times (0.33 + 0.17)}{12.5} \]

\[ = 550 \text{ IN-LBS.} \]

For \( \mu = 0.15 \) , \( K = 12.5 \)

\[ T = 44 \text{ IN-LBS.} \]

Use \( T = 45 \text{ TO } 50 \text{ IN-LBS.} \)

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5.3 SHEFT ANALYSIS

5.3.7 SHEAR PINS, COLD END CRANK TO HOT END CRANK

Since it was assumed that the vibratory inertia loads and the pressure loads occur at any point of shaft rotation, the max torque occurs where the load is 1.

To the line containing the crank, E's, neglect friction (conservative).

\[ T = P.e \]

\[ P_e = 29.7 + 4 = 29.3 \text{ LBS} \]
\[ (\text{Ref. PG 5-14}) \]

\[ T = 29.3 \times 1.63 = 47.8 \text{ in-LB} \]

\[ \frac{P_f}{T} = \frac{13.1}{1.63} = 13.1 \text{ LBS} \]

\[ P_f = \frac{P_e}{2} = 14.7 \text{ LBS} \]

\[ P_f = (13.1 + 14.7) = 19.7 \text{ LBS} \]

For 0.063 dia pin (4566-015-9005)

\[ A = \frac{\pi}{4} (0.063)^2 = 0.00312 \text{ in}^2 \]

\[ f_s = 19.7 = 6300, \text{ psi} \]

\[ F_{su} = 50,000, \text{ psi} \quad F_{ty} = 30,000, \text{ psi} \quad F_{tu} = 75,000, \text{ psi} \]

For 301 stainless,

Assuming \( F_{sy} = 60 \text{ ksi} \), \( F_{ty} = 18,000, \text{ psi} \)

\[ M_S = 18,000, -12 = 6300, \text{ in-psi} \]
5.4 CONNECTING ROD, HOT DISPLACER

LOAD

\[ P = \text{VIBRATION} + \text{PRESSURE} = 88.2 + 20.0 = 108.2 \text{ LBS} \]

\( \text{REF. PG 5-14} \)

BOLT

\[ d = 0.10, \quad a = \frac{\pi}{4} (10)^2 = 0.00785 \text{ in}^2 \]

\[ f_s = \frac{108.2}{100785} = 13,800 \text{, psi} \]

\[ M = 108.2 \times 0.067 = 7.25 \text{ in.-lb} \]

\[ \tau = \frac{7.25}{0.00785} = 0.000982 \]

\[ f_b = \frac{7.25}{0.000982} = 73,200 \text{, psi} \]

\[ R_s = \frac{13,800}{114,000} = 0.121 \]

\[ R_b = \frac{73,200}{114,000} = 0.308 \]

\[ M.S. = \frac{1}{(0.121 + 0.308^2)^{0.5}} = 1.46 \]

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5.4 Connecting Rod, Hot Displacer

Wrist Pin

\[ P = 108 \text{ LBS, (see preceding page)} \]
\[ d = 0.25'' \quad \pi \left( \frac{0.25}{2} \right)^2 = 0.49 \text{ in}^2 \]
\[ f_1 = \frac{108 \text{ LBS}}{0.49} = 220 \text{ psf} \]
\[ M = \frac{108 \times 0.25}{2} \]
\[ = 13.5 \text{ in-lbs, (Conservative)} \]
\[ r = \frac{\pi}{3} \left( \frac{0.25}{2} \right)^3 = 0.00153 \text{ in}^3 \]
\[ f_b = \frac{13.5}{0.00153} = 8500 \text{ psi} \]

Material: Tungsten-Carbide CA 310
\[ F_{su} = 450,000 \text{ psi} \]
\[ M.S. = 12 \text{ lb} \]

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5.4 Connecting Rod, Hat Displacer

**Wrist Pin Retainer**

\[ P = 108 \text{ lbs.} \]  
(Ref. Pg 5-26)

**Shear Stress at Threads**

\[ \sigma_s = \frac{M}{A} \left( \frac{d}{2} - t \right) = 0.147 \text{ psi} \]

\[ f_c = \frac{108}{1.147} = 93.9 \text{ psi} \]

**Preload Required to Prevent Separation of the Wrist Pin Retainer and Displacer**

\[ P_c = P \left( \frac{k_c}{k_c + k_e} \right) \]

\[ k_c = \text{Stiffness of compression element} \]

\[ k_e = \text{Stiffness of tensile element} \]

\[ k_{shn} = \frac{0.256 \times 76 \times 10^6}{0.060} = 172.8 \text{ x } 10^6 \text{ lb/in}^2 \]

\[ k_{disp} = \frac{0.296 \times 76 \times 10^6}{2.53} = 30.4 \text{ x } 10^6 \text{ lb/in}^2 \]

\[ k = \frac{k_{shn} + \frac{1}{k_{disp}}}{k_{shn} + \frac{1}{k_{disp}}} = 29.6 \text{ x } 10^6 \text{ lb/in}^2 \]

\[ k_t = \text{Stiffness of "Stud"} \]

\[ A_t = \frac{\pi}{4} x 25 = 0.0491 \text{ in}^2 \]

\[ L = 0.313 \text{ (From shoulder to mid point of threads)} \]

\[ E = 29.4 \text{ x } 10^6 \text{ psi} \text{ (Material: P-16-8 Mo)} \]

\[ k_t = 0.049 \times 29.4 \times 10^6 = 461 \text{ x } 10^6 \text{ lb/in}^2 \]

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5.4 CONNECTING ROD, HOT DISPLACER

WEIST PIN RETAINER

PRELOAD (CONTD.)

\[
P_c = P \left( \frac{K_c}{K_1 + K_c} \right) = 108 \left( \frac{74.6}{4.61 + 74.6} \right)
= 91.0 \text{ LB}
\]

TIGHTENING TORSION

THREAD DATA: \( T_c = 24 \text{ UNF} \)

\[\frac{P_c}{P} = .340 \quad \text{ FOR NO = 6.0}, \text{ F = 25} \]

\[T_1 = P \left( \frac{P_c}{P} \right) \frac{d}{2} \]

\[= 108 \cdot (.340) \left( \frac{.255}{2} \right) = 5.24 \text{ IN-LB} \]

\[T_2 = \text{ NUT FACE FRATION TORSION = } P \cdot T \quad \text{ R = MEAN READING OF SHIM = .274} \]

\[= 108 \cdot .25 \times .274 = 7.40 \text{ IN-LB} \]

\[T = T_1 + T_2 = 12.6 \text{ IN-LB} \]

USE \( T = 60 \), 1IN-LB

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CONNECTING ROD, CALD DISPLACER

LOAD

P = 29.3 lbs.  (REF. PG. 5-25)

BOLT

O.K. BY COMPARISON TO H/D DISPLACER BOLT. SMALLER LOAD FOR SAME SIZE BOLT.

WRIST PIN

O.K. BY INSPECTION.

WRIST PIN RETAINER

TIGHTENING TORSION: USE T = 60 in- lbs. SAME AS ON H/D DISPLACER
SECTION 6
CRANKSHAFT HOUSING STRESS ANALYSIS
6. HOUSING ASSY

6.1 TASKS

6.2 HOUSING AT CROSS TUBE JUNCTION

6.3 HOUSING AT SPHERE/CYLINDER JUNCTION

6.4 HOUSING AT TUBULAR SECTION, COLD END

6.5 FLANGE ASSY

6.6 INNER HOUSING, DRIVE MOTOR

6.7 OUTER HOUSING, DRIVE MOTOR

6.8 FLANGE, OUTER HOUSING
6.1 TASKS

The analysis of the housing assy consists of the following tasks:

- Determine strength of housing at cross tube junction
- Determine stresses at sphere/cylinder junction due to pressure and vibratory loading
- Determine stresses in tubular section, cold end due to pressure and vibratory loading
- Determine nut preload required and stress at flange assy
- Determine stresses in inner housing, drive motor due to pressure loading
- Determine stresses in outer housing, drive motor due to vibration
- Determine stress at welded section of flange for outer housing of drive motor

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FIG. 2-1  HOUSING ASSY.
G.2 Supply / Cross Tube Interconnection

THE AVAILABLE PRESSURE FOR THE HOUSING WILL BE BASED ON THE METHOD OF REF. 2, PAGE 67.

\[ P_H = 1.250 \]
\[ L_H = 0.04 \]
\[ P_B = 1.15 \]
\[ L_B = 0.125 \]

\[ F_{H1} = 90,000 \text{ psi} \]

Determine \( P_H \) --hole pressure for intact housing.

\[ \frac{L_H}{P_H} = \frac{P_H \cdot R_H}{F_{H1} + 0.9 \cdot T_H} \]

\[ F_H = \frac{L_H \cdot F_{H1}}{R_H - 0.4 \cdot L_H} \]

\[ F_H = \frac{5760 \cdot 0.064 \times 90,000}{1.25 - 0.4 \times 0.064} = 4700, \text{ psi} \]

PRESSURE REDUCTION RATIO \( \frac{P}{P_H} \)

\[ \frac{P}{P_H} = \frac{(R_H / F_H) - 0.4}{0.8 \cdot R_H \cdot (R_B - L_B) + R_B \cdot L_B - 0.4 \cdot T_H \cdot (R_B - L_B) + T_B + 0.2 \cdot \omega} \]

\[ = \frac{(1.250 / 0.064) - 0.4}{0.8 \times 1.250 \times (1.15 \times 0.125) + 1.15 \times 0.125} \]

\[ = 0.70750 \]

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6.25MPG (CROSS TUBE INTERSECTION)

ALLOWABLE PRESSURE

\[ P = 0.70750 \times 4700 = 3325 \text{ psi} \]

Based on Fatigue Allowable:

\[ = \frac{3325 \times 150,000}{70,000} = 5550 \text{ psi} \]

Based on Yield:

\[ = 6650 \text{ psi} \]

MARGIN OF SAFETY

OPERATING, \( M/S = \frac{3325}{1750} = 1.89 \)

PROOF, \( M/S = \frac{5550}{2060} = 2.69 \)

BURST, \( M/S = \frac{6650}{2800} = 2.4 \)
6.3 SPHERE / CYLINDER JUNCTION

\[
\begin{align*}
R_5 &= 1.25B / 1.299 \\
\text{t}_5 &= 0.068 / 0.064 \\
R_{m5} &= 1.254 - \\
\text{t}_{m5} &= 0.069 / 0.069 \\
\end{align*}
\]

\[
\begin{align*}
\rho &= \frac{Y_5}{R_5} \left( \frac{R_5 - \text{t}_5}{\text{t}_5} \right) = \frac{0.625}{1.254} \left( \frac{1.254}{0.064} \right) = 2.20 \\
\frac{\text{t}_5}{R_5} &= 1.39 \\
\frac{f_{\text{max}}}{Y_5/2\times t_5} &= 3.1 \\
\frac{f_{\text{max}}}{Y_5/2\times t_5} &= 3.1 \times 1.254 \times 1250 = 38,000 \text{ psi at operating} \\
\end{align*}
\]

STRESS DUE TO SHEAR LOAD

The stress due to shear load is negligible for values of \( \rho > 1.0 \), see Fig. 3.14, Ref 8.
6.3 SPHERE / CYLINDRICAL JUNCTION

THE SUPPORT REACTIONS WILL BE COMPUTED FOR THE MOUNTING SCHEME SHOWN IN THE DRAWING.

STATIC REACTIONS

\[
R_1 = \frac{1 \times 16.65 \times \left(3 \times 4.416 \times 7.50 - 4.416^2\right)}{7.50^3} \times 2.15 = 6.957 \text{ ft}
\]

\[
R_2 = 16.65 - 6.957 = 9.691 \text{ lbs}
\]

\[
M = \frac{1}{2} \times 16.65 \left(4.416^2 + 2 \times 4.416 \times 7.50 - 3 \times 4.416 \times 7.50 \right) = 21.33 \text{ in.-lb, STATIC}
\]

VIBRATION LOAD

FROM THE ANALYSIS OF THE COLD END SHELL, THE SECOND MODE FREQUENCY WAS 1120 cps (REF SECTION 8).
AT 1120 cps, THE RANDOM VIBRATION 30% RESPONSE, WITH AN ASSUMED AMPLITUDE FACTOR OF 10, IS

\[
LF = 3 \left( \frac{\pi}{2} \times 0.45 \times 10 \times 1120 \right)^\frac{1}{2} = 84.4 \text{ g}
\]
6.3 SPHERE/CYL. JUNCTION

STRESS DUE TO M

\[ \text{MOMENT} = \text{l.f.} \left[ R_1 (7.50 - 1.75) \right] \]
\[ - w (4.414 - 1.75) \]

\[ M = 84.4 \left( 6.759 \times 5.75 - 16.65 \times 2.666 \right) \]
\[ = 369, 120 \text{ lb-ft} \]

Using the method of Ref. 8,

\[ \frac{f_{max}}{M} \left( \frac{R_1}{t_1} \right)^k = 1.8 \]
from Fig. 3.12, pg. 104, Ref. 8,

\[ f_{max} = 1.8 \times 369 \times \frac{1.254 \times 0.064}{\pi \times 0.625 \times 0.064 \times 0.064} = 37,400 \text{ psi} \]

TOTAL STRESS DUE TO OPERATING PRESSURE AND VIBRATION

\[ f_T = 32,000 \text{ psi} + 37,400 \text{ psi} = 75,400 \text{ psi} \]
6.3 Sphere/Cylinder Junction

MARGIN OF SAFETY

LIFE

\[ F_2 = 90,000 \, \text{psi} \]

\[ \text{M.S.} = \frac{50,000}{75,400} - 1 = 0.19 \]

Proof Pressure (w/o vibr. stress)

\[ f = 2000, \frac{(38,900)}{1250} = 62,600 \, \text{psi} \]

\[ \text{M.S.} = \frac{150,000}{62,600} - 1 = 1.40 \]

Burst Pressure

\[ f = 2800, \frac{(38,000)}{1250} = 85,100 \, \text{psi} \]

\[ \text{M.S.} = \frac{175,000}{85,100} - 1 = 1.02 \]

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6.4 TUBULAR SECTION, COLD END

The moment at the mounting face will be obtained from the model used in the cold end shell analysis (Section 9). The moment will be interpolated from the moments at stations 8 and 9 of the cold end shell model (Ref. PG 8-42). The moments on page 8-53 are for 39.2 G5. Random vibration leading is 35.0 G5.

\[ M = \left[ 163.46 + \left( \frac{8.3125 - 7.325}{2} \right) \left( 181.89 - 163.46 \right) \right] \times 35.0 \]

\[ = 160.11 \text{ in-lbs} \]

\[ I = \frac{\pi}{24} (D_o^4 - D_a^4) = \frac{D_a^4}{24} \left( 1.249^4 - 1.057^4 \right) = 0.057714 \text{ in}^4 \]

\[ S_b = \frac{160.11 \times 1.249}{0.0577} = 1700 \text{ psi} \]

**ORIGINAL PAGE IS OF POOR QUALITY.**
6.4 TUBULAR SECTION, COLD END

\[
f_t = \frac{1.824 \times \frac{pR}{2}}{t} = \frac{1.824 \times 1.250 \times 1.577}{1.075} = 13,800 \text{ psi - operating}
\]

\[
f_c = \frac{pR}{2t} = \frac{1.250 \times 1.577}{2 \times 0.075} = 3,500 \text{ psi - operating}
\]

\[
f_t + f_c = 17,300 \text{ psi}
\]

STRESS DUE TO OPERATING PRESSURE PLUS VIBRATION

\[
f = 1,700 + 17,300 = 19,000 \text{ psi}
\]

FATIGUE ALLOWABLE IS 9,000 PSI FOR 10^6 CYCLES.

\[
\text{MIS.} = \frac{9,000}{19,300} = 0.466
\]

DEFLECTION AT OUTER BEARING, \( \delta \)

USING THE DEFLECTION AND SLOPE OF STATION 3, COLD END SHELL MODEL (REF PG 8-42),

\[
y = 0.000296 \times 35.0/39.2 = 0.000264 \text{ in.}
\]

\[
\theta = 0.000289 \times 35.0/39.2 = 0.000258 \text{ rad.}
\]
6.5 FLANGE ASSY

NUT PRELOAD

JOINT FLEXIBILITY

A. Pin 852525, CAP, CLOSURE, JUMP
B. Pin 852525, HSG ASSY, MACHINED
C. Pin 852532, RING, RETAINING, JUMP

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<td>.250</td>
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<td>.411</td>
<td>1.679</td>
<td>8.437 x 10^-3</td>
</tr>
</tbody>
</table>

\[ \Delta P_{\text{Allow}} = \frac{5.095 \times 10^{-3}}{2.143 \times 10^{-6}} = 238 \]

\[ P_{\text{Allow}} = P_{\text{Preload}} + 0.238 P_{\text{Applied}} \]

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6.5 FLANGE ASSY

NUT PRELOAD

\[ \frac{F_A}{T} = \frac{D_p}{(1 - f_s \sin \lambda)} \left( \frac{\tan \lambda + f_t \cos \lambda}{\frac{1}{R}} \right) + \frac{f_n D_m}{\cos \phi} \]

\[ F_A = \text{AXIAL LOAD, LB} \]
\[ T = \text{TORQUE, IN- LBS} \]

For \( 2.6875 - 20 \left( \text{N BUSH -3} \right) \) THREAD

\[ \lambda = \text{HEX ANGLE} = .05 \text{RAD} = 2.8648^\circ \]
\[ \phi = \text{THREAD PRESSURE ANGLE} = 7^\circ \]

\[ R = (1 + \tan \lambda + \tan \phi) \frac{K}{K} = 1.01758 \frac{K}{K} = 1.00875 \]
\[ f_t = \text{THREAD FRICTION FACTOR} = .12 \]
\[ f_n = \text{NUT FRICTION FACTOR} = .12 \]
\[ D_m = \text{MEAN CONTACT DIAM. AT NUT FACE} = 2.223 \text{ IN.} \]
\[ \phi = (180^\circ - \text{COUNTERDINK ANGLE}) \div 2 = 0 \]
\[ D_p = 2.6875 \text{ IN.} \]

\[ \frac{F_A}{T} = \frac{1}{.08599 D_p + .06 D_m} = 2.744 \]

\[ F_A = 2.744 \times T \] LBS
\[ T \] IN LBS
G.5 Flange Assy

Nut Preload

Applied Loads

Consider seal o.d. as pressurized area.

Seal: AP 20-2313-7-5P
O.D. = 2.316 (max)

\[ P_a = \frac{\pi \times 2.316 \times 1250}{4} = 5770.125 \text{ at max. oper. press.} \]

\[ = 4.213 \times 2080 = 8760 \text{ lbs, at proof pressure} \]

\[ = 4.213 \times 2800 = 11800 \text{ lbs, at burst pressure} \]

Nut Load

\[ T = 210 \text{ ft-lb} = 2520 \text{ in-lb} \text{ (min.)} \]

\[ P_{\text{preload}} = 2.744 T = 6910 \text{ lbs,} \]

At proof pressure

\[ P_{\text{nut}} = P_{\text{preload}} + 0.238 P_{\text{ applied}} \]

\[ = 6910 + 0.238 \times 8760 = 9000 \text{ lbs.} \]

As shown on the diagram (next page), the joint will not separate at proof pressure.
6.5 FLANGE ASSY

\[ T = 210, \text{ FT-LBS, (MIN.)} \]
\[ P_{\text{NUT}} = P_{\text{PRELOAD}} + 0.38 \times P_{\text{APPLIED}} \]
\[ P_{\text{PRELOAD}} = 2.744 T \text{ (IN-LB)} \]
6.5 FLANGE ASSY

THREAD SHEAR

CHECK SHEAR OF EXTERNAL THD AT MINOR DIA. OF INTERNAL THD

\[ 2.6875 - 20 = (N) \text{ BOLT - 3 THREAD} \]

ASSUMING \( G = 0.045 \) (REF. NBS HDBK 148)

\[ D_o = \text{MAJOR DIA. OF EXT. THD} = 2.683 \text{ in.} \]

\[ d_m = \text{MINOR DIA. OF INT. THD} = D_{\text{basic}} - 1.2 \times 0.05 \]

\[ = 2.6875 - 1.2 \times 0.05 = 2.628 \text{ in.} \]

THICKNESS OF EXT. THD. AT 1.314 RADIUS

\[ t = \frac{(D_o - d_m)}{2} \tan 7^\circ + \frac{F}{2} + \frac{(D_o - d_m)}{2} \tan 45^\circ \]

\[ = \frac{(2.683 - 2.628)}{2} \tan 7^\circ + \frac{1.6316 \times 0.05 + 0.0275}{2} \]

\[ = 0.0390 \text{ in.} \]

SHEAR AREA PER THD = \( A_s = 0.0390 \times \pi \times 2.628 = 0.322 \text{ in}^2 \)

NO. OF ENGAGED THD = \( 1.58 \times 20 = 3 \) (1/2 THD CARRIES 50% OF THE LOAD)

SHEAR STRESS AT BURST

\[ f_s = \frac{0.56 \times 11,800}{0.322} = 18,300.975 \text{ psi} \]

For Inco 718, \( F_t = 120,000 \text{ ksi} \)

\[ F_t = 0.6 \times F_t = 108,000 \text{ ksi} \]

\[ MS = \frac{108,000}{18,300} = 6 \]

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HIGH
G.6 INNER HOUSING, DRIVE MOTOR

ECLIPSE hardened CYLINDER JUNCTION

\[ f_h = \frac{1.128 \times \ell \cdot \sigma}{t} \]

\[ f_h = 1.128 \times 617 \times 1250 = 72,500 \text{ psi} \text{ (Operating)} \]
\[ = 119,000 \text{ psi} \text{ (Proof)} \]
\[ = 162,000 \text{ psi} \text{ (Burst)} \]

\[ M.S. = \frac{180,000}{162,000} = 0.11 \]

CYLINDER - AT OUTER EDGE OF STATOR

\[ f_b = 1.824 \times \frac{t R}{I} \text{ (Assuming Rigid at edge)} \]
\[ = 1.824 \times 2800 \times 7 = 119,000 \text{ psi} \text{ (Burst)} \]

\[ f_t = \frac{t R}{2t} = 33,600 \text{ psi} \]

\[ f_b + f_t = 153,000 \text{ psi} \text{ (at burst)} \]

\[ M.S. = \frac{180,000}{153,000} = 0.12 \]
C.6 INNER HOUSING, DRIVE MOTOR

CYLINDER - JUNCTION IF \( t = 0.008 + t = 0.030 \) SECTIONS

\[
\begin{align*}
\delta_1 &= \frac{p_1 R_i}{E_1 t_1} - \frac{V_0}{2 D_1 \lambda_1} - \frac{M_o}{2 D_1 \lambda_1}, \\
\theta_1 &= -\frac{M_o}{\lambda_1 D_1}, \\
\theta_2 &= \frac{M_o}{\lambda_2 D_2}, \\
\delta_1 &= 0, \quad \delta_2 = 0, \quad \theta_1 = \theta_2.
\end{align*}
\]

\[
\begin{align*}
\frac{V_0}{2 D_1 \lambda_1} + \frac{M_o}{2 D_1 \lambda_1} &= \frac{p_1 R_i}{E_1 t_1}, \\
\frac{V_0}{2 D_1 \lambda_1} + M_o \left( \frac{1}{D_1 \lambda_1} - \frac{1}{D_2 \lambda_2} \right) &= 0, \\
V_0 + \lambda_1 M_o &= \frac{p_1 R_i}{E_1 t_1} \left( 2 D_1 \lambda_1^2 \right), \\
V_0 + 2 D_1 \lambda_1 M_o \left( \frac{1}{D_1 \lambda_1} - \frac{1}{D_2 \lambda_2} \right) &= 0, \\
V_0 + M_o \left( \frac{\lambda_1}{2} - 2 D_1 \lambda_1 \frac{D_2 \lambda_2}{D_2 \lambda_2} \right) &= 0.
\end{align*}
\]
6.6 INNER HOUSING, DRIVE MOTOR

CYLINDER - t = 0.008 ft; \( t = 0.030 \) JUNCTION

\[
\frac{2D_1x_1}{M_0} \left( \frac{1}{D_1x_1} - \frac{1}{D_2x_2} \right) + \frac{x_1}{M_0} = \frac{P_1t_1}{E_1t_1} \left( 2D_1x_1 \right)
\]

\[M_0 \left( \frac{3}{2} - \frac{2D_1x_1}{D_2x_2} \right) = \frac{P_1t_1}{E_1t_1} x_2D_1x_2
\]

\[D_1 = \frac{E_1t_1^3}{12(1-u^2)}
\]

\[\lambda_1 = \frac{1.285}{(R_1t_1)^{1/2}}
\]

\[\lambda_2 = \frac{1.65}{R_1t_1}
\]

\[
\frac{D_1x_1}{D_2x_2} = \frac{E_1t_1^3}{12(1-u^2)} \times \frac{1.285}{(R_1t_1)^{1/2}} \times \frac{1.285}{(R_1t_1)^{1/2}} \times \frac{(R_2t_2)^{1/2}}{1.285}
\]

\[
= \left( \frac{t_2}{t_1} \right)^{3/2} \times \frac{(R_2t_2)^{1/2}}{(R_1t_1)^{1/2}} = \left( \frac{t_1}{t_2} \right)^{3/2} \left( \frac{R_2t_2}{R_1t_1} \right)^{1/2}
\]

\[
\frac{R_1t_1^3}{E_1t_1} \times \frac{x_1}{1.65} \times \frac{1.65}{12(1-u^2)} \times \frac{1.65}{R_1t_1} = \frac{P_1t_1t_1}{E_1t_1} \frac{1.65}{12(1-u^2)}
\]
INNER HOUSING DRIVE MOTOR

Cylinder: $t = 0.008$ & $t = 0.030$ junction

$$
M_0 \left[ 1 - \frac{2x(0.030)}{3(0.088)(0.685 \times 0.030)} \right]^{\frac{3.53}{3}} = \frac{\pi x (0.088 \times 0.030 \times 1.285)}{3 \times 0.6(1 - 1.5)}
$$

$$
M_0 (17.352306) = 10020729 \, \pi
$$

$$
M_0 = 0.0011946 \, \pi
$$

$$
= -1.334 \, \text{in-lbs}
$$

$$
M = \frac{M_0}{\pi}\frac{2}{3}
$$

$$
V_0 + \lambda_1 M_0 = \frac{P_1 R_1}{E_1 t_1} (2.2, 1.3)
$$

$$
V_0 + \frac{1.285}{(0.685 \times 0.030)^{1.5}} x (-334) = \frac{P_1 R_1 t_1 (1.45) \times 1.285}{6(1 - 1.5)} \left( \frac{R_1}{t_1} \right)^{1.5}
$$

$$
= \frac{P_1 R_1 x (0.088 \times 0.030 \times 1.45) x 1.285}{6 x 0.91 (0.685 \times 0.030)^{1.5}}
$$

$$
= 0.555647 \, \pi
$$

$$
V_0 = 0.555647 \times 2800 + 2.9374 = 158.9 \, \text{lb/in.}
$$

ORIGINAL PAGE IS OF POOR QUALITY
6.6 INNER HOUSING, DRIVE MOTOR

CYLINDER - t = 0.008 in., portion

This portion of the cylinder is restrained from radial expansion by the stator.

\[ f_t = \frac{7200 \times 0.008}{2} = 28800 \times 0.008 = 123,000 \text{ psi} \]

\[ f_b = \frac{61200 \times 33}{0.008} = 31,300 \text{ psi} \]

\[ f_b + f_t = 154,300 \text{ psi} \]

\[ M.S. = \frac{180,000}{154,000} = 1.17 \]

The clearance between the pressure dome & stator is 15,000 / 10,000 on the diameter.

\[ \gamma = \frac{7200}{10000} = 0.7 \]

\[ f_h = \frac{0.005 \times 30 \times 10^4 \times 1000}{0.70} = 122,150 \text{ psi} \]

\[ f_h (at OD = 30000) = \frac{0.005 \times 30 \times 10^4}{0.70} = 15,000 \text{ psi} \]
6.6 INNER HOUSING, DRIVE MOTOR

**FLANGE**

Ref. 11, pg. 142:

\[
M = \frac{P (R_{Ec} - R_c)}{1 + 0.05 + (1 - 0.05) (\frac{R_c}{R_c})^{0.5} (\frac{R_c}{R_c})^{0.5}}
\]

\[
R = 0.05, \quad R_c = 0.25
\]

\[
t_c = 1.00, \quad R_c = 1.65
\]

\[
P = \frac{P}{2 R_c} = \frac{2800 \times 1025}{2 \times 1650} = 1465 \text{ lb/in of cros., at}
\]

\[
inner radius (R_c)
\]

\[
\frac{\beta}{(R_c t_c) t_c} = \frac{1.25}{(1.65 \times 1000)} = 6.51
\]

\[
M = 1465 \times \left(1.25 - 1.65 \right)
\]

\[
1 + 0.15 \times 1090 + 0.9 \times 1090 \times 1.65 \times 1650
\]

\[
= 2000 \text{ in-lb/in}
\]

\[
M_{net} = M - (\text{moment due to } P \text{ acting on flange face, M')}\]

\[
M' = M \times \frac{0.25}{1.25} = M \times 0.20 \text{ in-lb/in}
\]

\[
M_{net} = 465 M
\]

\[
= 93.0 \text{ in-lb/in}
\]
6.6 INNER HOUSING, DRIVE MOTOR

FLANGE

CYLINDER STRESSES

\[ f_b = \frac{6 \times 93.0}{1,060} = 550,000 \text{ psi} \] (by burst)

\[ f_t = \frac{2800 \times 3.5}{2 \times 1,060} = 15,000 \text{ psi} \]

\[ f_b + f_t = 170,000 \text{ psi} \]

\[ \text{H.S.} = \frac{186,000}{170,000} = 1.06 \]
6.7 OUTER HOUSING, DRIVE MOTOR

VIBRATORY LOADS

The frequency estimate will consider the flexibilities of the outer housing and the crankshaft housing flange.

**Frequency, Lateral Vibration**

\[
\omega_L = \frac{3.13}{\delta_L}
\]

\[
\delta_L = \frac{wL^3}{3EI_L} + \delta_L \quad \text{where} \quad \delta_L = \frac{M}{\alpha EL^2}
\]

\[
= \frac{wL^3 + ML}{3EI_L} \quad \text{where} \quad \delta_L = \frac{M}{\alpha EL^2}
\]

\[W = \text{Static Wt} = 50.0 \, \text{lb} \]

\[L = 3.40 - 1.55 = 1.85 \, \text{in.} \]

\[I = \frac{\pi}{64} \left( 3.013 - 2.895 \right) = 1.607 \, \text{in.}^4 \]

\[E_t = 10 \times 10^6 \, \text{psi - Aluminum Alloy} \]

\[t = \text{Flange Thickness} = 0.088 \]

\[E_s = 29.3 \times 10^6 \, \text{psi, Steel 718} \]

\[a = 1.397 \, \text{in.} \]

\[b = 1.052 \, \text{in.} \]

\[\alpha = 156. \quad \text{Ref. No.} \quad 8525.85 \]

\[\delta_L = \frac{5.1 \times 1.85^3}{3 	imes 7.0 \times 10^{-6} 	imes 50.7} + \frac{5100 \times 1.85 \times 1.85}{152 \times 23.5 \times 10^{-6} 	imes 0.088} \]

\[= 7.14 \times 10^{-6} \, \text{in.} \]

\[\omega_L = 1170 \, \text{cps} \]

*ORIGINAL PAGE IS OF POOR QUALITY*
6.7 OUTER HOUSING, DRIVE MOTOR

VIBRATIONAL LOADS

RANDOM VIBRATION 3.0 RESPONSE

$$\dot{X} = 3 \left( \frac{\pi}{2} \times 0.85 \times 10 \times 11.70 \right) = 86.3 \text{ g's} \quad \text{(axial)}$$

Lateral deflection at c.g. due to 86.3 g's

$$\dot{Y} = 0.0006 \text{ in.}$$

FREQUENCY, AXIAL VIBRATION

$$f_A = \frac{\pi L + \alpha W \omega_n}{AE_1} \frac{1}{E_2 + \tau^2}$$

$$\alpha = 0.0021 \quad \text{For} \quad \frac{\alpha}{E_2} = 1.33$$

$$A = \frac{\pi}{4} (0.15 - 2.893) = 0.557 \text{ in}^2$$

$$f_A = \frac{5.00 \times 1.85 + 0.0021 \times 5.00 \times 137.7}{0.557 \times 10 \times 10^2} + \frac{29.8 \times 16^2 \times 0.083^3}{0.557 \times 10 \times 10^2}$$

$$f_A = 1920. \text{ c.p.s.}$$

RANDOM VIBRATION 3.0 RESPONSE

$$\dot{X}_A = 3 \left( \frac{\pi}{2} \times 0.85 \times 10 \times 1920 \right) = 111. \text{ g's} \quad \text{(axial)}$$
G.7 OUTER HOUSING, DRIVE MOTOR

HOLD-DOWN SCREWS

\[ V_{vc} = \left( \frac{2 \times \cos 180^\circ}{N} \right) M \]

\[ N = 8 \text{ (#6-32 Screws)} \]
\[ R = 1.70 \text{ in.} \]

LATERAL LOAD FACTOR of 86.3 G\( \frown \)

\[ M = 86.3 \times 5.00 \times 1.85 = 798 \text{, lbs.} \]

\[ P_{max} = \frac{2 \times \cos \left( \frac{180^\circ}{8} \right) \times 798}{8 \times 1.70} = 108 \text{, lbs.} \]

\[ F_{tu} = 4.23 \text{, lbs.} \text{ (Ref. MIL-HDBK-5, pp 9-63)} \]

ASSUMING \( F_e = 0.5 \)

\[ F_{tu} = 211 \text{, lbs.} \]

\[ M.S. = 211 \times 1 = \frac{108}{108} \]

AXIAL LOAD FACTOR of 111 G\( \frown \)

\[ P = \frac{5.00 \times 111}{8} = 69 \text{, lbs.} \text{, O.K.} \]
6.7 OUTER HOUSING, DRIVE MOTOR

FLANGE

\[ M = 117 \times 0.194 = 22.7 \text{ in.-lbs.} \]

\[ f_b = \frac{6M}{bd^2} \]

\[ = \frac{6 \times 22.7}{0.50 \times 0.090} \]

\[ = 33600 \text{ psi} \]

To determine local stresses in cylinder due to flange load, assume bending occurs along an arc of 0.25 inch radius.

\[ f_b = \frac{6M}{bd^2} \]

\[ = \frac{6 \times 22.7}{\pi (2.5/2) (0.060)} \]

\[ = 48200 \text{ psi} \]

\[ f_c = \frac{117}{0.50 \times 0.060} = 3900 \text{ psi} \]

\[ f_b + f_c = 52100 \text{ psi} \]

For 7075-T73 aluminum alloy

\[ F_t = 68 \text{ ksi}, \ F_y = 52 \text{ ksi} \]

\[ M_{ij} = \frac{52}{52} - 1 = 0.077 \]

REF. MIL-HDBK-5
6.7 OUTER HOUSING, DRIVE MOTOR

PRESS FIT STRESS

PRESS FIT BETWEEN THE HOUSING AND SHAFT IS 0.0015/0.0025 IN.

AT 0.0025 IN. PRESS

\[ \sigma = \frac{AD}{D} = \frac{0.0025 \times 10 \times 10^6}{1.95} \]

\[ = 17,200 \text{, PSI} \]

THIS STEADY STRESS WILL BE COMPARED TO THE "THRESHOLD" STRESS FOR STRESS CORROSION (Fsc).

\[ Fsc = 43 \text{, ksi (MIN.) for 7075 - T73 REF. MIL-HDBK-5 pg 3-10.} \]

\[ M.S. = \frac{43}{17.2} = 1.50 \]
6.7 OUTER HOUSING, DRIVE MOTOR

DEFLECTION AT END (FLAT HEAD)

σ = 15,000 psi EXTERNAL

USE METHOD FROM REF 12, TO COMPUTE DISCONTINUITY STRESS (σd).

From σd, compute M = \( \frac{f_s \sigma_d}{6} \)

Then apply M to flat plates.

\( \frac{T}{E} = 1.0 \quad , \quad R = 1.45 \)

\( D = \frac{2 \times 1.45}{E} = 37.7 \)

\( t = \frac{2 \times 1.45}{E} = 37.7 \)

\( I_o = 18 \quad , \quad \text{REF. 12, pA 4.7.2-11} \)

\( f_{\text{max}} = I_o \cdot \frac{R}{t} = 18 \times 1.5 \times 1.45 = 5100 \text{, psi} \)

\( M = \frac{5100 \times 37.7}{18} = 5.02 \text{ in-cf/in,} \)

\( \sigma_{\text{due to } \sigma} = 0.00368 \text{ in (see next page)} \)

\( \sigma_{\text{due to } M} = \frac{6(R-1)(R^2-1)}{E \cdot t^3} \left( \frac{M}{9} \right) = \frac{6 \times 2 \times 1.45 \times 5.02}{3 \times 10 \times 10^6 \times 0.77^3} \)

\( = 0.00462 \text{ in, (opposite to } \sigma_i) \)

\( \delta = \delta_i - \delta_{\sigma} = 0.0051 \text{ in.} \)
6.7 OUTER HOUSING DRIVE MOTOR

DEFLECTION AT END DUE TO \( P = 15 \text{ psi} \)

**SIMPLY SUPPORTED**

\[
\delta = 3 \frac{W (m-1)(5m+1)}{16 \pi E I} \frac{R^2}{L^3}
\]

\( m = \frac{1}{6} \) (revolution) \]

\( W = 8 \pi R^2 \hbar \)

\[
\delta = \frac{3 \times 1.45^4 (2 \times 1.5)}{0.77^3 \times 10^{-6}} = \frac{1.45^4}{0.77^3 \times 10^{-6}}
\]

\( = 0.00928 \text{ in.} \) (SIMPLE SUPPORT)

**FOR FIXED EDGES**

\[
\delta = \frac{0.00928 (m-1)}{(m-1)(5m+1)} \frac{8}{2 \times 16}
\]

\( \delta = 0.0242 \text{ in.} \)
6.8 FLANGE, OUTER HOUSING

**STRESS AT WELD**

1. **σ = 1.052**
2. **ε = 1.397**
3. **σ/ε = 0.753**
4. **Ref. 4, Par. 157 & 216.**

**σw** at weld joint, assume material is aluminum.

- **σ** = 125,000 psi
- **σy** = 114,000 psi

**σ = \( \frac{F}{A} \) = \( \frac{125,000 \times 798}{1.397 \times 1,088} \) = 25,500 psi

**M.S. = \( \frac{90 \times 1.6 \times 1}{75.6} \) = 2.88**
7. HOT END PRESSURE SHELL

7.1 TASKS

7.2 SHELL ANALYSIS

7.2.1 PRESSURE PLUS TEMPERATURE LOADING STRESSES

7.2.2 VIBRATORY STRESSES

7.2.3 COMBINED PRESSURE, TEMP., AND VIBRATORY STRESSES

7.2.4 SHELL AND DISPLACEMENT DEFLECTION

7.3 FLANGE ASSY

7.3.1 NUT PRELOAD CALCULATIONS

7.3.2 STRESS ANALYSIS

7.4 SHIELD ASSY, VACUUM
7.1 TASKS

THE ANALYSIS OF THE HOT END PRESSURE SHELL CONSISTS OF THE FOLLOWING TASKS:

- DETERMINE SHELL STRESSES DUE TO INTERNAL PRESSURE, TEMPERATURE LOADING, AND VIBRATION.
- DETERMINE SHELL AND DISPLACEMENT DEFLECTIONS AND COMPARE TO CLEARANCE.
- DETERMINE FLANGE ASSEMBLY PRELOAD REQUIREMENTS AND STRESS IN THE ASSEMBLY.
- DETERMINE COLLAPSING PRESSURE AND STRESSES OF THE VACUUM SHELL ASSEMBLY.
FIG. 7-1 HOT END ASSEMBLY
7.2 SHELL ANALYSIS

The hot end pressure shell was analyzed using a shell analysis program developed by A. KALVIN. The theoretical basis for this shell program is contained in Ref. (13).

The shell analysis model is shown on page 7-6.

The shell was analyzed for 1250 psi max operating pressure and 2800 psi burst pressure, plus a temperature gradient of 1200°F at the dome and 140°F at the flanges.

The stresses due to vibratory loads were determined and combined with stresses due to max operating pressure.

The stresses are tabulated on page 7-7 and plotted on page 7-8 (Fig. 7-3). The allowable stresses are also plotted in Fig. 7-3. The stresses due to burst pressure were not plotted beyond x = 4 inches because the stresses are very low. The max operating stresses were not plotted for x < 3.0 inches because the stresses, when compared to the creep allowable, is insignificant.

The max operating stress at the end of part 12 is 39.7 ksi which exceeds the creep allowable of 35 ksi. However, the shell is structurally adequate because:

1) At nominal design pressure (1000 psi), the stress reduces to 31.9 ksi with resulting M.S. 8.010.

2) A stress level of 39.7 ksi does not imply a structural failure but rather a reduction of exposure time and/or an increase above the 0.1% creep. The creep rupture stress is 49 ksi.
7.2 SHELL ANALYSIS

The shell frequency was estimated at 1180.0 cps. With an assumed amplification of 10, the 30th random vibration response was found to be 26.6 g's. The resulting vibratory stresses, which were unnoticeable, were combined with the stresses due to maximum operating pressure plus temperature loading. The maximum combined stress was 66,000 psi. The fatigue allowable for 10^8 cycles is 90,000 psi. This gives an M.S. of 0.35.
### 7.2.1 SHELL ANALYSIS - PRESSURE PLUS TEMPERATURE

The combined stresses will be computed from the computer output.

<table>
<thead>
<tr>
<th>Condition</th>
<th>X (in)</th>
<th>( f_L ) (ksi)</th>
<th>( f_T ) (ksi)</th>
<th>( f_{omr} ) (ksi)</th>
<th>( f_{F} ) (ksi)</th>
<th>M.S.</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>162.3</td>
<td>50.0</td>
<td>144.9</td>
<td>175.0</td>
<td>0.22</td>
<td></td>
</tr>
<tr>
<td>1.04</td>
<td>99.8</td>
<td>137.0</td>
<td>127.0</td>
<td>111.0</td>
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<td></td>
</tr>
<tr>
<td>1.07</td>
<td>99.7</td>
<td>128.6</td>
<td>110.4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.07</td>
<td>4.0</td>
<td>128.9</td>
<td>111.1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.51</td>
<td>4.2</td>
<td>125.0</td>
<td>105.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.03</td>
<td>6.1</td>
<td>121.1</td>
<td>105.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.55</td>
<td>31.0</td>
<td>121.0</td>
<td>105.0</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>3.05</td>
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<td>115.0</td>
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<td>3.92</td>
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<td>44.5</td>
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<td></td>
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<tr>
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<td>32.3</td>
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<td>4.47</td>
<td>17.1</td>
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</tr>
<tr>
<td>4.64</td>
<td>37.5</td>
<td>15.3</td>
<td>32.7</td>
<td>35.0</td>
<td>0.07</td>
<td></td>
</tr>
<tr>
<td>4.97</td>
<td>12.0</td>
<td>33.6</td>
<td>29.1</td>
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<tr>
<td>5.20</td>
<td>34.7</td>
<td>17.7</td>
<td>30.1</td>
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<tr>
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<td>17.4</td>
<td>4.1</td>
<td>15.7</td>
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<td></td>
</tr>
<tr>
<td>6.65</td>
<td>44.2</td>
<td>10.8</td>
<td>39.9</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Note:** \( f_{omr} = (f_L \sin \theta - 0.5 f_T \sin \theta)^{1/2} \)

**T = SHEAR = 0.**

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Page 7-7

**ORIGINAL PAGE IS OF POOR QUALITY**
Fig. 7-3  Stresses at max operating and burst pressures

Hot End Pressure Shell

\[ F_{tu} \]
\[ F_{ty} \]
\[ F_{burst} \]
\[ F_{operating} \]

Distance Along \( \phi \) (in)
7.2.1 SHELL ANALYSIS - PRESSURE PLUS TEMP.

The shell analysis computer output are included in this section as pages 7-16 thru 7-58.

Due to the modeling technique used, a fictitiously high stress level was computed at the end of part 10 (refer to fig. 7-2, pg 7-6). The high stress occurred because the generous fillet radius was not included at the end of part 10. However, the fillet was accounted for in part 11. Consequently, the stress at the beginning of part 11 was much lower and was used at the part 10/part 11 junction.
7.2.7. VIBRATORY STRESSES

**Frequency**

Estimate by finding the frequencies \( f_i \) of a uniformly loaded cantilever and \( f_r \) of a cantilever with an end load, and apply D'Anvers' formula.

\[
f_1 = \frac{3.52}{2\pi} \left( \frac{EI}{I^3} \gamma \right)^{1/2}
\]

\( l = 4.1'' \)

\( \delta_1 = \frac{\pi}{4} \left( 2.67 \cdot 2 - 2.57 \cdot 9.26 \right) = 0.114 \cdot 180 \)

\[
f_1 = \frac{3.52}{2\pi} \left( \frac{\gamma \cdot 10^4 \cdot 3.49}{1.14 \cdot 4 \cdot 29.18} \right)^{1/2} = 6570.0 \text{ cps}
\]

\[
f_r = \frac{1}{2\pi} \left( \frac{K_r}{W} \right)^{1/2} = 3.13 \left( \frac{1}{2} \right)^{1/2}
\]

\( W = \frac{K_r}{W} \text{ displaced pressure shell dome} + \text{Brax + Herter + Support + Flaw Distortion (tension)} = 1.67 + 1.3 + 0.94 + 17 + 12 + 11 = 1.55 \text{ lbs}
\]

\[
\delta = W \left[ \frac{L^3 + 1.5 L}{3EI} \right] = W L^2 \left( \frac{1}{3} + 1.5 \right)
\]

\[= \frac{4}{2 \pi} \left( 1.35 + 2.5 \right) \left( 1.56 \right) = 4.79 \times 10^6 \text{ in.}
\]

\[
f_r = 3.13 \left( \frac{1}{2} \right)^{1/2} = 1200 \text{ cps}
\]

\[
f_o = \left( \frac{1}{f_1} + \frac{1}{f_r} \right)^{1/2} = 1180 \text{ cps}
\]

**ORIGINAL PAGE IS OF POOR QUALITY**
7.2.2 Vibratory Stresses

Vibratory Loads

At 1180 op, the random vibration specification applies. The 3σ response, with an assumed amplification factor of 10, is: 15°.

\[ L.F. = 3 \left( \frac{\pi \times 0.045 \times 10 \times 1180}{2} \right)^{0.5} \]

\[ = 3 \left( \frac{\pi \times 0.045 \times 10 \times 1180}{2} \right)^{0.5} \]

\[ = 86.6 \text{ g's} \]
7.2.2 VIBRATORY STRESSES

The pressure seal supports half of the displacer weight (0.94x) at the outboard bearing plus the following items:

<table>
<thead>
<tr>
<th>Name</th>
<th>W</th>
<th>X</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell</td>
<td>1.04</td>
<td>11.71</td>
</tr>
<tr>
<td>Heated</td>
<td>0.12</td>
<td>13.25</td>
</tr>
<tr>
<td>Supporting</td>
<td>0.11</td>
<td>14.75</td>
</tr>
<tr>
<td>Bedring</td>
<td>0.17</td>
<td>14.75</td>
</tr>
<tr>
<td>Flow Duty</td>
<td>0.510</td>
<td>11.68</td>
</tr>
<tr>
<td>Screen hopper</td>
<td>0.43</td>
<td>10.50</td>
</tr>
</tbody>
</table>

\[ E = 2.38 \]
\[ EWX = 27.812 \]
\[ X = 11.686 \]

* * X is from fixed mount olung plane. See weight report, Appendix A.

At pressure seal cylinder / frame intersection, \[ x = 8.6875 \]

\[ M = 27.812 + 0.47 \times 14.75 - (0.75 + 0.75)(8.6875) \]

\[ = 9.750 \text{ lb-in.} \text{ at 1}^\circ \text{C} \]
7.2.2 VIBRATORY STRESSES

AT CYL/FLANGE JUNCTION

\[ D_1 = 2.589 \]
\[ t = 0.073 \text{ (min.)} \]
\[ I = \frac{\pi}{4} (2.726^4 - 2.589^4) = 0.501 \text{ in.}^4 \]
\[ f_b = \frac{9.750 \times 1.563}{0.501} = 26.5 \text{ psf at 1"} \]
\[ = 2302 \text{ psi at 86.6 ft} \]

CYLINDRICAL PART AT \( x = 9.03 \)

\[ D_2 = 2.589 \]
\[ t = 0.049 \]
\[ I = \frac{\pi}{4} (2.677^4 - 2.589^4) = 0.349 \text{ in.}^4 \]
\[ M = 9.750 - 2.85 \times 0.343 = 8.77 \text{ in.-lb at 1"} \]
\[ f_b = \frac{8.77 \times 1.339}{0.349} = 33.6 \text{ psi at 1"} \]
\[ = 2700 \text{ psi at 86.6 ft} \]
7.2.2 VIBRATORY STRESS

OUTBOARD BEARING JOURNAL (1521//4 IN) JUNCTION

Random VMA Load = P = \frac{1}{2} \text{displacement} \times N \times \text{G}

a = 1.319
b = 0.69 + k (0.65) = 0.74

R_s = \frac{a}{b} = \frac{1.319}{0.74} = 1.77

\epsilon_s = 0.065

R_c = 0.634
\epsilon_c = 0.036

\epsilon_c / \epsilon_s = 1.48

P = \frac{R_c}{R_s} \left( \frac{R_s}{\epsilon_s} \right)^{\epsilon_c} = \frac{0.634}{1.77} \left( \frac{2.57}{0.036} \right)^{0.065} = 1.67

\chi = (S.C.F.) \left( \frac{N}{\pi T} \right) \left( \frac{R}{T} \right)^{\epsilon_c} \quad \text{(Ref. 8, Section 3)}

M = P \epsilon_c = 0.37 \times 764 \times 0.75 = 30.5 \text{ in-lbs}

S.C.F. = 1.7 \quad \text{(Ref. 8, Section 3)}

\chi = 1.7 \left( \frac{30.5}{\pi \times 0.634 \times 0.036} \right) \left( \frac{2.57}{0.065} \right)^{0.036}

= 34200 \text{ psi}

ORIGINAL PAGE IS OF POOR QUALITY
7.7.3 COMBINED MAX OPERATING, TEMP, AND VIBRATION STRESSES

AT CYLINDER/FLANGE JUNCTION

\[
\sigma = 64,300 \, + \, 2,300 \, = \, 66,600 \, \text{psi}
\]

AT :34 in. FROM CYL/FLANGE JUNCTION

\[
\sigma = 54,500 \, + \, 2900 \, = \, 57,400 \, \text{psi}
\]

AT DOME/BEARING JOURNAL JUNCTION

\[
\sigma = 39,100 \, + \, 3,400 \, = \, 43,500 \, \text{psi}
\]

THE FATIGUE ENVIRONMENT AT 100 CYCLES IS 99,000 PSI

\[
M.J. = \frac{99,000}{66,600} = 0.35
\]
KALMINS SHELL STRESSES ANALYSIS... NOT END PRESSURE SHELL... MORIYAMA... JUN 75

STATIC ANALYSIS  PARTS= 14  BRANCHES= 0  NUMBER OF SUBCASES= 2

ANGLES OF ROTATION OF BOUNDARY CONDITIONS ARE

\[ \begin{align*}
\text{PART NO. 3} & \\
S1 & = 0.0000 \\
S2 & = 0.0000 \\
S3 & = 25000.00 \\
T1 & = 5 \\
\text{INSTRUMENT}\ 8 & \\
\text{SHELL TYPE} & = 0 \\
\text{LAYERS MLY} & = 1 \\
\text{GENERAL SHELL NO} & = 8 \\
K & = 0.0000 \\
1/RF & = 0.0000 \\
R & = 1.1930 \times 0.01 \\
\text{FILM} & = 90.000 \text{ DEG} \\
\end{align*} \]

\( \text{VARIABLE Z-COORDINATES OF THIS PART FOLLOW.} \)

\[ \begin{align*}
\text{Y COORDINATES} & = 2250.01 \quad -2250.01 \\
\text{X COORDINATES} & = 0.0000 \quad 2500.00 \\
\text{TOI LINEAR FUNCTION GENERATOR NO. 2 FROM 2 POINTS} \\
\end{align*} \]

\[ \begin{align*}
\text{Y COORDINATES} & = 2150.01 \quad 2250.01 \\
\text{X COORDINATES} & = 0.0000 \quad 2500.00 \\
\end{align*} \]

\[ \begin{align*}
\text{LAYER NO 1 FROM} & = 2250.01 \quad \text{TO} & = 2450.01 \\
\text{CONSISTS OF ISOTROPIC MATERIAL} & \\
\text{YOUNG'S MODULUS} & = 20000.00 \\
\text{POISSON'S RATIO} & = 0.3000 \times 0.00 \\
\text{COEFFICIENTS OF THERMAL EXPANSION} & = 0.2500.00 \\
\text{ATHETA} & = 7500.00 \times 0.05 \\
\text{MASS DENSITY} & = 0.0000 \\
\end{align*} \]

\[ \begin{align*}
\text{PART NO. 2} & \\
S1 & = 0.0000 \\
S2 & = 22010.01 \\
T1 & = 5 \\
\text{INSTRUMENT}\ 8 & \\
\text{SHELL TYPE} & = 2 \\
\text{NWP} & = 0 \\
\text{LAYERS MLY} & = 1 \\
\text{GENERAL SHELL NO} & = 8 \\
K & = 0.0000 \\
1/RF & = 0.0000 \\
R & = 1.1930 \times 0.01 \\
\text{FI} & = 90.000 \text{ DEG} \\
\end{align*} \]

\[ \begin{align*}
\text{LAYER NO 1 FROM} & = 2250.01 \quad \text{TO} & = 2250.01 \\
\text{CONSISTS OF ISOTROPIC MATERIAL} & \\
\text{YOUNG'S MODULUS} & = 20000.00 \\
\text{POISSON'S RATIO} & = 0.3000 \times 0.00 \\
\text{COEFFICIENTS OF THERMAL EXPANSION} & = 0.2500.00 \\
\text{ATHETA} & = 7500.00 \times 0.05 \\
\text{MASS DENSITY} & = 0.0000 \\
\end{align*} \]

\[ \begin{align*}
\text{PART NO. 3} & \\
S1 & = 0.0000 \\
S2 & = 25000.00 \\
T1 & = 5 \\
\text{INSTRUMENT}\ 8 & \\
\text{SHELL TYPE} & = 2 \\
\text{NWP} & = 0 \\
\text{LAYERS MLY} & = 1 \\
\text{GENERAL SHELL NO} & = 8 \\
K & = 0.0000 \\
1/RF & = 0.0000 \\
R & = 1.1930 \times 0.01 \\
\text{FI} & = 90.000 \text{ DEG} \\
\end{align*} \]

\( \text{VARIABLE Z-COORDINATES OF THIS PART FOLLOW.} \)

\[ \begin{align*}
\text{Y COORDINATES} & = 2250.01 \quad -2250.01 \\
\text{X COORDINATES} & = 0.0000 \quad 2500.00 \\
\text{TOI LINEAR FUNCTION GENERATOR NO. 4 FROM 2 POINTS} \\
\end{align*} \]
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS E = 39800+08, POISSON'S RATIO NU = 30000-00

COEFFICIENTS OF THERMAL EXPANSION

\[ \alpha = 500000 \] 
\[ \beta = 500000 \]
\[ \gamma = 500000 \]

PART NO 9

S1 = 00000, S2 = 44735+01, IPAR = 20, ING = 3, SHELL TYPE: NTP = 0, LAYERS = 1

GENERAL SHELL NO. 8, K = 00000, _T_R_F = 00000, R = 13290+01, FI = 90.000 DEG

VARIABLE ELASTIC PROPERTIES OF LAYER NO. 1 FOLLOW:

EVAR LINEAR FUNCTION GENERATOR NO. 5 FROM 5 POINTS

| Y COORDINATES | .2980+08 | .2980+08 | .2800+08 | .2800+08 |
| X COORDINATES | .0000 | .1250+50 | .1675+01 | .3825+01 |

ATHETA = .76000-05

RHO = .00000

SI = 44735+01, SX = 2800+08, IPAR = 15, ING = 2, SHELL TYPE: NTP = 0, LAYERS = 1

ELLIPSOIDAL SHELL NO. 5, K = 00000, A = 13290+01, B = .64300-00, DIRFCTN = 1.

LAYER NO. 1 FROM Z = .39000-01 TO Z = 26000-01

CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS E = 24400+08, POISSON'S RATIO NU = 30000-00

COEFFICIENTS OF THERMAL EXPANSION

\[ \alpha = 500000 \] 
\[ \beta = 500000 \]
\[ \gamma = 500000 \]
PART NO 11

SI = .00000  SX = .31752-00  IPAR = 6  ING = 1  SHELL TYPF = 6  NTP = 0  LAYERS MLY = 1

CONICAL SHELL NO 6  K = .00000  PHI = 104.120 DEGREES  A = -.21524+01

VARIABLE Z-COORDINATES OF THIS PART FOLLOW

T111 LINEAR FUNCTION GENERATOR NO. 10 FROM 2 POINTS

Y COORDINATES  -.6000-01  .2700-01
X COORDINATES  .0000  .3175-00

T011 LINEAR FUNCTION GENERATOR NO. 11 FROM 2 POINTS

Y COORDINATES  .6000-01  .2700-01
X COORDINATES  .0000  .3175-00

LAYER NO 1 FROM Z = .14600-00 TO Z = .25000-01
CONSISTS OF ISOTROPIC MATERIAL  YOUNG'S MODULUS E = 24400+08  POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .84000-05  ATHFTA = .84000-05  MASS DENSITY RHO = .00000

PART NO 12

SI = .00000  SX = .13300+01  IPAR = 6  ING = 3  SHELL TYPF = 2  NTP = 0  LAYERS MLY = 1

CYLINDRICAL SHELL NO 2  K = .00000  PHI = .60100-00  PHI = 90.000, DEGREES

LAYER NO 1 FROM Z = -.25500-01 TO Z = -.25500-01
CONSISTS OF ISOTROPIC MATERIAL  YOUNG'S MODULUS E = 24400+08  POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .84000-05  ATHFTA = .84000-05  MASS DENSITY RHO = .00000

PART NO 13

SI = .00000  SX = .29046-00  IPAR = 6  ING = 1  SHELL TYPF = 6  NTP = 0  LAYERS MLY = 1

CONICAL SHELL NO 6  K = .00000  PHI = 157.319 DEGREES  A = -.65137+00

VARIABLE Z-COORDINATES OF THIS PART FOLLOW

T113 LINEAR FUNCTION GENERATOR NO. 12 FROM 2 POINTS

Y COORDINATES  -.5000-01  .0000-01
X COORDINATES  .0000  .2905-00

T013 LINEAR FUNCTION GENERATOR NO. 13 FROM 2 POINTS

Y COORDINATES  .5000-01  .8000-01
X COORDINATES  .0000  .2905-00
LAYER NO 1 FROM Z=16000-00 TO Z=16000-00
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS E=244000+08
POISSON'S RATIO NU=0.30000-00
COEFFICIENTS OF THERMAL EXPANSION AF=0.84000-05 ATETA=0.84000-05 MASS DENSITY RHO=-0.0000

PART NO 1
S1=0.0000 SX=23300-00 IPAR=5 ING=2 SHELL TYPE 6 NTP=0 LAYERS MLY=1

CONICAL SHELL NO 6 K=0.0000 PHI=180.000 DEGREES AM=-9000-00

LAYER NO 1 FROM Z=-80000-01 TO Z=-80000-01
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS E=244000+08
POISSON'S RATIO NU=0.30000-00
COEFFICIENTS OF THERMAL EXPANSION AF=0.84000-05 ATETA=0.84000-05 MASS DENSITY RHO=-0.0000
XAINSf SHEL L STRESS ANALYS... HOT END PRESSURE SHELL MORIMOTO JUN '75

SUBCASE NO 1 FOR FOURTH HARMONIC COS THETA

BOUNDARY CONDITIONS AT STARTING EDGE 1  =  0.0000  3  =  0.0000  7  =  0.0000
BOUNDARY CONDITIONS AT FINAL EDGE 2  =  0.0000  4  =  0.0000  8  =  0.0000

LOADS FOR PART NO 1 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 2 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 3 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 4 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 5 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 6 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
W =  0.0000  U =  0.0000  ITHK =  0.0000
SURFACE AND TEMP LOADS ARE P =  12500+04  PFI =  0.0000  PTHETA =  0.0000  TL =  14000+03  TU =  14000+03

LOADS FOR PART NO 7 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE 0 =  0.0000  M =  0.0000  N =  0.0000
<table>
<thead>
<tr>
<th>PART NO</th>
<th>SUBCASE</th>
<th>LOADS AT END OF PART</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>k_00000  p = 0.12500+04 pfi = -0.0000 ptheta = -0.0000 tl = -1.4000+04 tu = -1.4000+04</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>k_00000  p = 0.12500+04 pfi = -0.0000 ptheta = -0.0000 tl = -1.4000+04 tu = -1.4000+04</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>k_00000  p = 0.12500+04 pfi = -0.0000 ptheta = -0.0000 tl = -1.4000+04 tu = -1.4000+04</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>k_00000  p = 0.12500+04 pfi = -0.0000 ptheta = -0.0000 tl = -1.4000+04 tu = -1.4000+04</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>k_00000  p = 0.12500+04 pfi = -0.0000 ptheta = -0.0000 tl = -1.4000+04 tu = -1.4000+04</td>
</tr>
</tbody>
</table>

**LOADS FOR PART NO 8 SUBCASE NO 1**

**LOADS FOR PART NO 9 SUBCASE NO 1**

**LOADS FOR PART NO 10 SUBCASE NO 1**

**LOADS FOR PART NO 11 SUBCASE NO 1**

**LOADS FOR PART NO 12 SUBCASE NO 1**

**LOADS FOR PART NO 13 SUBCASE NO 1**
RING LOADS AT END OF THIS PART ARE

\[ \theta = 0.000 \quad \phi = 0.000 \quad \theta' = 0.000 \quad \phi' = 0.000 \quad \gamma = 0.000 \]

\[ \theta'' = 0.000 \quad \phi'' = 0.000 \quad \theta'' = 0.000 \quad \phi'' = 0.000 \]

SURFACE AND TEMP LOADS ARE

\[ P = 1.25 \times 10^4 \quad \theta'' = 0.000 \quad \phi'' = 0.000 \quad \gamma = 0.000 \quad TL = 1.200 \times 10^4 \quad TU = 1.200 \times 10^4 \]

LOADS FOR PART NO 14 SUBCASE NO 1

RING LOADS AT END OF THIS PART ARE

\[ \theta = 0.000 \quad \phi = 0.000 \quad \theta' = 0.000 \quad \phi' = 0.000 \quad \gamma = 0.000 \]

\[ \theta'' = 0.000 \quad \phi'' = 0.000 \quad \theta'' = 0.000 \quad \phi'' = 0.000 \]

SURFACE AND TEMP LOADS ARE

\[ P = 1.25 \times 10^4 \quad \theta'' = 0.000 \quad \phi'' = 0.000 \quad \gamma = 0.000 \quad TL = 1.200 \times 10^4 \quad TU = 1.200 \times 10^4 \]
**Kalwins Shell Stress Analysis**

**Deflections and Stress Resultants for Wave Number \(nx = 0\)**

<table>
<thead>
<tr>
<th>W</th>
<th>M</th>
<th>D</th>
<th>UPWH</th>
<th>NPHI</th>
<th>PHI</th>
<th>MPHI</th>
<th>UTHETA</th>
<th>N</th>
<th>NTHETA</th>
<th>MTHETA</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>-53763</td>
<td>0</td>
<td>0.000</td>
<td>-18072</td>
</tr>
<tr>
<td>0.025</td>
<td>25350</td>
<td>0.04</td>
<td>0.000</td>
<td>0.000</td>
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<td>-53763</td>
<td>0</td>
<td>0.000</td>
<td>-18072</td>
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<tr>
<td>0.050</td>
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<td>0.000</td>
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<td>0</td>
<td>0.000</td>
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<tr>
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<td>0</td>
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<tr>
<td>1.025</td>
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<td>0.000</td>
<td>0.000</td>
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<td>0</td>
<td>0.000</td>
<td>-18072</td>
</tr>
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<td>0.000</td>
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<tr>
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<td>0.000</td>
<td>-53763</td>
<td>0</td>
<td>0.000</td>
<td>-18072</td>
</tr>
</tbody>
</table>

**Main Shell Part No. 2**

<table>
<thead>
<tr>
<th>W</th>
<th>M</th>
<th>D</th>
<th>UPWH</th>
<th>NPHI</th>
<th>PHI</th>
<th>MPHI</th>
<th>UTHETA</th>
<th>N</th>
<th>NTHETA</th>
<th>MTHETA</th>
</tr>
</thead>
<tbody>
<tr>
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- Tables represent data for main and material shell parts.
- Specific parts and their corresponding values are listed in the tables.
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**ORIGINAL PAGE 15**

**OF POOR QUALITY**

71-0506-2
### Kalning Shell Stress Analysis

#### Hot End Pressure Shell: Hopimoto Jun 73

**Deflections and Stresses for Wave Number N = 0**

**1250 PSI Plus Temperature Loading**

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**Original Page is of Poor Quality**
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**Loads for Part No. 1 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)

**Loads for Part No. 3 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)

**Loads for Part No. 4 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)

**Loads for Part No. 5 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)

**Loads for Part No. 6 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)

**Loads for Part No. 7 Subcase No. 2**

Ring Loads at End of This Part are:
- \( \phi = -0.0000 \)
- \( \phi = -0.0000 \)
- \( N = -0.0000 \)
- \( U = -0.0000 \)
- \( T = -0.0000 \)
- \( P = -0.0000 \)

Surface and Temp Loads are:
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
- \( P = 28100+04 \)
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<th>Surface and Temp Loads</th>
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**TET Linear Function Generators**:

- No. 1 from 3 points
- Code numbers of variable loads over this part, given by FGENs below: 4, 5

**X Coordinates**: 0.0000, 0.1250, 0.4430

**Y Coordinates**: 0.1400, 0.1400, 0.1200

**X Coordinates**: 0.1250, 0.4430, 0.1200

**Y Coordinates**: 0.1400, 0.1400, 0.1200
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Loads for Part No 14 Subcase No 2

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### KALNINGS SHELL STRESS ANALYSIS... NOT END PRESSURE "SHELL" MORIMOTO JUN 73

### DEFLECTIONS AND STRESS RESULTSANTS FOR WAVE NUMBER NX = 0 2810 PSI PLUS TEMPERATURE LOADING

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### MAIN SHELL PART NO 2

| 0.000 | 0.000 | 0.67949+02 | 0.61299+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |
| 0.000 | 0.000 | 0.93531+02 | 0.89769+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |
| 0.000 | 0.000 | 1.33693+02 | 1.13609+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |
| 0.000 | 0.000 | 0.49324+02 | 0.41249+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |

### MAIN SHELL PART NO 3

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| 0.000 | 0.000 | 0.82984+02 | 0.82984+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |
| 0.000 | 0.000 | 0.82984+02 | 0.82984+02 | -1.66530+04 | -1.1663+01 | -1.0073+02 | 0.0000 | 0.0000 | 1.7293+04 | 0.3021+01 |</p>
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**Note:** The table above contains numerical data and may require specific software or tools to interpret accurately. It appears to be a part of a larger document, possibly related to manufacturing or engineering specifications.
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<th>SHELL 6</th>
<th>SHELL 7</th>
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<tr>
<td>0.663</td>
<td>93130.02</td>
<td>16202.02</td>
<td>12395.02</td>
<td>2440.05</td>
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<td>12395.02</td>
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<td>SHELL WIDTH</td>
<td>SHELL THICKNESS</td>
<td>SHELL TYPE</td>
<td>MILLIMETERS</td>
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<td>19641-01</td>
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<td>21854-01</td>
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<td>22355-01</td>
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<td>MAIN SHELL PART NO</td>
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<td></td>
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<td>-----</td>
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<tr>
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<td>0.0000</td>
<td>23921-02</td>
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<td>31995-02</td>
<td>0.0000</td>
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<tr>
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<td>0.0000</td>
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<tr>
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<td>67655-01</td>
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<td>24839-02</td>
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<td>0.0000</td>
<td>20361-02</td>
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<td>0.1864</td>
<td>67900-01</td>
<td>15332-02</td>
<td>0.0000</td>
<td>19874-02</td>
</tr>
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<td>0.1840</td>
<td>67992-01</td>
<td>15492-02</td>
<td>0.0000</td>
<td>19479-02</td>
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<td>67995-01</td>
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<td>0.2330</td>
<td>67998-01</td>
<td>10881-02</td>
<td>0.0000</td>
<td>20416-02</td>
</tr>
</tbody>
</table>

IBRM = 0 WHICH INDICATES END OF JOB, EXIT CALLED
7.2.4 SHELL AND DISPLACER DEFLECTIONS

THE DEFLECTION OF THE SHELL AND DISPLACER WILL BE CHECKED AGAINST THE CLEARANCES.

SHELL DEFLECTION DUE TO DISTRIBUTED LOAD \( W \) AND END LOAD \( W_e \):

\[
\delta = \frac{Wl^2}{8EI} + \delta_e
\]

\[
\delta_e = 6.79 \times 10^{-4}
\]

\[
W = 0.114 \text{ lb/in.}
\]

\[
I = 0.349 \text{ in}^4
\]

\[
l = 4.0
\]

\[
\frac{1.144 \times 4.4}{8 \times 39.8 \times 10^6 \times 1346} + 6.79 \times 10^{-4}
\]

\[
= 7.14 \times 10^{-6} \text{ in. PER 1 in.}
\]

\[
= 0.00062 \text{ in. PER 8.46 in.}
\]

DISPLACER DEFLECTION:

\[
\delta_d = \frac{WL^3}{48EI}
\]

\[
W = 0.94 \text{ lb/in.}
\]

\[
I = \frac{\pi}{4} (2.272^2 - 2.143^2) = 0.163 \text{ in.}^4
\]

\[
l = 5.0 \text{ in.}
\]

\[
\frac{0.94 \times 5^3}{48 \times 29.8 \times 10^6 \times 1346}
\]

\[
= 0.502 \times 10^{-7} \text{ in. PER 1 in.}
\]

\[
= 0.000043 \text{ in. PER 8.46 in.}
\]

SINCE THE MIN. RADIAL CLEARANCE IS 0.025 in., THE RELATIVE MOTION OF THE SHELL AND DISPLACER IS MUCH LESS THAN THE CLEARANCE.
7.3 FLANGE ASST

7.3.1 NUT PRELOAD

<table>
<thead>
<tr>
<th>PART</th>
<th>L</th>
<th>O.D.</th>
<th>I.D.</th>
<th>A</th>
<th>E</th>
<th>S*</th>
</tr>
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<tr>
<td>1</td>
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<td>3.5</td>
<td>3.05</td>
<td>2.315</td>
<td>2.9 x 10^6</td>
<td>3.773 x 10^-6</td>
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<tr>
<td>2</td>
<td>0.3</td>
<td>3.5</td>
<td>3.0</td>
<td>2.553</td>
<td>2.9 x 10^6</td>
<td>4.053 x 10^-6</td>
</tr>
<tr>
<td>3</td>
<td>0.3</td>
<td>3.5</td>
<td>3.25</td>
<td>1.325</td>
<td>2.9 x 10^6</td>
<td>7.805 x 10^-6</td>
</tr>
<tr>
<td>4</td>
<td>0.5</td>
<td>3.9</td>
<td>3.5</td>
<td>2.325</td>
<td>2.9 x 10^6</td>
<td>7.416 x 10^-6</td>
</tr>
<tr>
<td>5</td>
<td>0.35</td>
<td>3.9</td>
<td>3.6</td>
<td>1.747</td>
<td>2.9 x 10^6</td>
<td>6.850 x 10^-6</td>
</tr>
</tbody>
</table>

\[ \sum S = 1.4246 \times 10^{-6} \]

* \[ S = \frac{L}{AE} \] (lbf/deg)

COMPRESSIVE STACK-UP \[ \sum S_c = 1.558 \times 10^{-8} \text{lbf/deg} \]

TENSILE STACK-UP \[ \sum S_t = 1.4246 \times 10^{-8} \text{lbf/deg} \]

CHANGE IN PRELOAD DUE TO INTERNAL LOAD:

\[ \Delta P = \frac{\sum S_{relax}}{\sum S_n} P_{int} \]

\[ \Delta P = \frac{S_1 + S_2}{\sum S_n} P_{int} = \frac{7.774 \times 10^{-7}}{2.973 \times 10^{-8}} P_{int} \]

\[ \Delta P = +7.41 \text{lbf} \]

AIRESOERCH MANUFACTURING COMPANY
OF CALIFORNIA
7.3.1  **NUT PRELOAD**

**LOAD / TORQUE RELATIONSHIP**

\[
\frac{F_a}{T} = \frac{2}{\left( \frac{1}{VR} - x_t \sin \lambda \right) \left( \tan \lambda + x_f \cos \lambda \right) + \frac{X_h D_m}{\cos \epsilon}}
\]

- \( F_a \): **Axial Load**
- \( T \): **Torque**
- \( D_p \): **Pitch Dia**
- \( \lambda \): **Helix Angle**
- \( x_t \): **The Friction Factor**
- \( X_h \): **Nut Face Friction Factor**
- \( D_m \): **Mean Contact Dia at Nut Face**

\( \epsilon \): For flat surface \( \cos \epsilon \approx 1.0 \)

\( \phi \): **Thread Pressure Angle**

For a 20\( \frac{\text{deg}}{\text{in}} \) pitch and 7\(^\circ\) Pressure Angle with

\[
x_t = x_h = 0.12, \quad \lambda = 2.8648^\circ (0.05 \text{ rad})
\]

\[
R = 1.01758, \quad \sqrt{R} = 1.00875
\]

\[
\frac{F_a}{T} = \frac{2}{\left( \frac{1}{0.991324 - 0.005777} \right) \left( 1.03946 + 0.12 D_m \right)}
\]

\[
\frac{F_a}{T} = \frac{1}{0.98573 D_p + 0.06 D_m}
\]
7.3.1 **NUT PRELOAD**

\[
F_\alpha = \frac{T}{0.8573 D_p + 0.06 D_m}
\]

\[
D_p = 3.580 \text{ in}, \quad D_m = 3.150 \text{ in}
\]

\[
F_\alpha = 2.013 T \quad \text{for } \alpha = 0.12
\]

**Hot End Displacer Flange Loads**

- **Max Operating Pressure** = 1250 PSI
- **Proof Pressure**, \(1.68 \times 1250 = 2080 \text{ PSI}\)
- **Burst Pressure**, \(2.25 \times 1250 = 2737.5 \text{ PSI}\)

Consider the seal O.D. as the extent of the pressure area.

\[
F = PA = 1250 \times \frac{\pi}{4} \times 3.25^2
\]

\[
F = 10,370 \text{*}
\]

\[
F_{\text{Proof}} = 17,300 \text{*}
\]

\[
F_{\text{Burst}} = 23,200 \text{*}
\]
7.3.1 NUT PRELOAD

For \( T = 450 \) ft-lbs

\[
F_a = 2.013 \times 12 \times 450.
\]

\[
F_a = 10,900 \text{ lbs}
\]

---

**NOTE:**

3. Axial Stack-Up will separate but deflection will not exceed compression of O-ring or K-seal.

**FIG. 7-4:** FLANGE NUT LOAD VS. EXTERNAL LOAD
7.3.2 STRESS ANALYSIS

1. Nut (Vacuum Support Ring)
2. Hot End Housing Flange
3. Housing Assembly Flange

Shear Area of Internal Thread: At Max Minor of External

\[ S_{RAO} = \frac{3.5713}{2} + 0.08261 \times 0.05 = 1.759 \text{ in} \]

\[ \omega_1 = \frac{3.5713 - 1.759}{2} \tan 7^\circ = 0.00327 \text{ in} \]

\[ \omega_2 = \frac{3.5713 - 1.759}{2} \tan 45^\circ = 0.02665 \text{ in} \]

\[ F = 0.16316 \rho = 0.16316 \times 0.05 = 0.00816 \text{ in} \]

Thickness at \( S \) radius = 0.03808 in

Thickness at \( R = \frac{3.525}{2} = 1.7625 \text{ in} \)

\[ t = 0.00327 \times \frac{1.759}{1.7625} + 0.02665 \times \frac{1.759}{1.7625} + 0.00816 \]

\[ t_{SHAPE} = 0.03802 \text{ in} \]
7.3.2 STRESS ANALYSIS

NUMBER OF ENGAGED THREADS:

\[ N = 0.22 \times 20 = 4.4 \text{ thus} \]

USE 3.0 FULL THREADS

"THE FIRST THREAD WILL CARRY 50% OF THE LOAD"

SHEAR AREA

\[ A = 0.03802 \times 2\pi \times 1.7625 \]

\[ A = 4.21 \text{ in}^2 \]

AT BURST CONDITION

\[ \gamma = \frac{F}{A} = \frac{23,200}{2 \times 4.21} \]

\[ \gamma = 27,600; \text{ PSI O.K.} \]

FOR INCO 718

\[ F_{Su} = (6 \times 180 = 108 \text{ KSI}) \]

\[ M_{J, UL} = \frac{108}{27.6} = 3.81 \]
7.4 SHIELD ASSY, VACUUM, HOT DISPLACED

COPING PRESSURE

\[
E = \frac{1}{2} \times 5.955 + 0.20 = 2.998 \approx 3.00
\]

\[
t = 0.040\text{ m}
\]

\[
L = 5.45 + 0.03 = 5.48
\]

\[
R_{Gg}, 5.49, 126
\]

\[
E = \left(\frac{L}{R}\right)\left(\frac{L - t}{L}\right)^n = 5.48^n \frac{(1 - t)}{3.00 \times 0.04}
\]

\[
\varphi = \frac{c_p \pi^2 D}{2L} = \frac{c_p \pi^2 E t^3}{2L^2 \frac{12}{12} (1 - t)}
\]

\[
c_p = 20. \text{ from Fig. 4, p. 126.}
\]

\[
\varphi_m = \frac{20 \times \pi^2 \times 29.8 \times 10^6 \times 1.04^3}{3.00 \times 5.48 \times 12 \times 0.91}
\]

\[
= 384.35 \quad >> 14.7
\]
7.4 SHIELD ASSY, VACUUM, NOT DISPLACED

Junction of Dome and Cylinder

 hoop stress

\[ f_h = \frac{1.128 \cdot a}{t} \quad \text{for} \quad a/t = 2 \quad \text{RFS 3, page 93} \]

\[ a = 3.00 \]
\[ t = 0.040 \]
\[ f = 14.7 \, \text{psi} \]

\[ f_h = \frac{1.128 \times 3.00 \times 14.7}{0.040} = 1240 \, \text{psi} \]

MATL: 302 Stainless, \( F_{T,7} = 50,000 \, \text{psi} \)

Support Ring

Assume inner edge fixed and supported, outer edge free. R flats \( W \) applied at outer edge and \( P \) acting over

\[ a = 3.00 \quad a/t = 1.65 \quad W = 14.7 \times \pi \times 3.00 = 417,185 \]
\[ b = 1.250 \quad \beta = 0.526 \quad \text{REF. 4.79, 215} \]
\[ t = 0.080 \quad \beta_w = 0.593 \]

\[ f = \frac{P \cdot W - \beta_w a^2}{t} = \frac{224 \times 417}{0.080} - 0.593 \times 14.7 \times 3.00 = 21,900 \, \text{psi} \]

M.J. = \( \frac{30,000}{21,900} \cdot -1 = 0.37 \)
SECTION 8
COLD-END PRESSURE SHELL
STRESS ANALYSIS
8. Cold End Pressure Seal

8.1 Tasks

8.2 Shell Analysis

8.2.1 Pressure Plus Temperature Loading

8.2.2 Vibratory Stresses

8.2.3 Combined Pressure, Temperature, and Vibratory Stresses

8.3 Shell Deflection

8.3.1 Due to Self Induced Vibration Loading

8.3.2 Due to Vibration Loading

8.4 Flange Assy

8.5 Vacuum Jacket Assy
3.1 **Tasks**

The analysis of the cold end pressure shell consists of the following tasks:

- Determining shell stresses due to internal pressure, temperature, and vibratory loading.
- Determining tip deflection due to self-induced vibration.
- Determining tip deflection due to vibration loading.
- Determining net preload required at flange Assy.
- Stress analysis flange Assy.
- Determining collapsing pressure and stresses in vacuum jacket Assy.
8.7 SHELL ANALYSIS

The cold end pressure shell was analyzed using a shell analysis program developed by A. Kalnin. The theoretical basis for the program is contained in Ref. (13).

The shell analysis model is shown on the next page.

The shell was analyzed for 1250 psi max operating pressure and 2800 psi burst pressure, plus a temperature gradient of -343°F (-195°C) at the cold end and 140°F at the flange.

The vibratory stress due to random vibration loading of 35/¢ psi was determined and combined with the max operating pressure stress.

The stresses are shown on pages 8-6 and 8-7. The max M.S. are 0.21 at burst and 0.27 at operating pressure plus vibration.
### S.2.1 Pressure Plus Temperature Stresses

The significant stresses computed by the program are tabulated below. The stresses for the parts not listed are less than 80, KSI.

**At Burst Pressure:**

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>$f_L$ (KSI)</th>
<th>$f_T$ (KSI)</th>
<th>$f_{comb}$ (KSI)</th>
<th>Allowable (KSI)</th>
<th>P.T.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>145.0</td>
<td>16.2</td>
<td>138.0</td>
<td>175.0</td>
<td>0.27</td>
</tr>
<tr>
<td>3</td>
<td>145.0</td>
<td>16.2</td>
<td>138.0</td>
<td>175.0</td>
<td>0.27</td>
</tr>
<tr>
<td>8</td>
<td>125.0</td>
<td>146.1</td>
<td>137.0</td>
<td>175.0</td>
<td>0.28</td>
</tr>
<tr>
<td>9</td>
<td>125.0</td>
<td>146.1</td>
<td>137.0</td>
<td>175.0</td>
<td>0.28</td>
</tr>
<tr>
<td>10</td>
<td>104.0</td>
<td>31.2</td>
<td>92.4</td>
<td>150.0</td>
<td>0.62</td>
</tr>
<tr>
<td>12</td>
<td>118.0</td>
<td>71.2</td>
<td>197.0</td>
<td>240.0</td>
<td>0.21</td>
</tr>
<tr>
<td>13</td>
<td>117.0</td>
<td>72.6</td>
<td>199.0</td>
<td>240.0</td>
<td>0.21</td>
</tr>
</tbody>
</table>

**Notes:**
1. $F_{tu}$ at 140°F
2. $F_{tu}$ at -343°F (65°K)
3. Allowable at weld
   
   \[
   F_a = 0.90 \left[ \frac{F_{tu, annulated, R.T.} \times F_{tu, -340°F}}{F_{tu, at R.T.}} \right]
   \]
   
   \[
   = 0.90 \times 125 \times 240 / 190
   \]
   
   \[
   = 150.0 \text{ KSI}
   \]
4. $f_{comb} = \left( f_L + f_T - f_{tu, -340°F} \right) \times (T=0)$

At max. operating pressure, the stresses at the end of part B are:

$F_L = 55.6 \text{ KSI}$,  
$f_T = 65.2$,  
$p.T. = 61.0 \text{ KSI}$

The max operating stress will be combined with the vibration stress in paragraph 8.1.3.

The total output is included in this section as pages 8-8 through 8-48.
8.2.2 Vibratory Stresses

Bending Stress

From Section 8.3

\[ f_b = 7241 \text{ cfs} \]
\[ L' = 35.0 \text{ in} \]
\[ M = 108.7 \text{ in} \cdot \text{lbf} \quad \text{Sec. 39.2 g/ft} \]
\[ = 97.1 \text{ in} \cdot \text{lbf} \quad \text{Sec. 35.1 g/ft} \]

(See A-A is 6.0 from end,
between 6.0 and 7 of Vib. Analysis)

\[ d_e = 1.025 + 2 \times 0.07 = 1.049 \text{ in.} \]

\[ I = \frac{\pi}{64} \left( 1.049^4 - 1.025^4 \right) = 5.76 \times 10^3 \text{ in.}^4 \]

\[ f = \frac{97.1 \times 5245}{5.76 \times 10^3} = 9700.75 \text{ PSI} \]

8.2.3 Combined Pressure, Temp., and Vibratory Stresses

From Section 8.2.1, pg 8-6, the max operating stress is 61,000 PSI.

\[ f = f_0 + f_v = 61,000 \cdot 9,700.75 \]

\[ = 70,700.75 \text{ PSI} \]

The fatigue allowable for 10^8 cycles is 90,000,750 PSI.

\[ (\text{Ind. 718}) \]

\[ M.S. = \frac{90,000}{70,700} = 0.27 \]
PART NO 1

\[ \begin{align*}
\text{Si} &= 0.0000 \\
\text{Sx} &= 1.5000 \times 10^4 \\
\text{IPa} &= 0 \\
\text{INa} &= 1 \\
\text{SHELL TYPE 2} \\
\end{align*} \]

Cylindrical Shell No. 2

\[ \begin{align*}
K &= 0.0000 \\
R &= 0.5760 \times 10^4 \\
\text{PHI} &= 90.000 \text{ DEGREES} \\
\end{align*} \]

Layer No. 1 from \( Z = 0 \) to \( Z = 1.5000 \times 10^4 \)

Consists of isotropic material, Young's modulus \( E \) is \( 29600 \times 10^5 \) Poisson's ratio \( \nu \) is \( 0.300 \times 10^4 \)

Coefficients of thermal expansion \( \alpha \) is \( 9600 \times 10^5 \) mass density \( \rho \) is \( 0.000 \times 10^4 \)

PART NO 2

\[ \begin{align*}
\text{Si} &= 0.0000 \\
\text{Sx} &= 0.7500 \times 10^4 \\
\text{IPa} &= 0 \\
\text{INa} &= 1 \\
\text{SHELL TYPE 5} \\
\end{align*} \]

Conical Shell No. 6

\[ \begin{align*}
K &= 0.0000 \\
\text{PHI} &= 90.000 \text{ DEGREES} \\
\end{align*} \]

Layer No. 1 from \( Z = 0 \) to \( Z = 1.5000 \times 10^4 \)

Consists of isotropic material, Young's modulus \( E \) is \( 29600 \times 10^5 \) Poisson's ratio \( \nu \) is \( 0.300 \times 10^4 \)

Coefficients of thermal expansion \( \alpha \) is \( 9600 \times 10^5 \) mass density \( \rho \) is \( 0.000 \times 10^4 \)

PART NO 3

\[ \begin{align*}
\text{Si} &= 0.0000 \\
\text{Sx} &= 0.7500 \times 10^4 \\
\text{IPa} &= 0 \\
\text{INa} &= 1 \\
\text{SHELL TYPE 6} \\
\end{align*} \]

Cylindrical Shell No. 2

\[ \begin{align*}
K &= 0.0000 \\
R &= 0.5760 \times 10^4 \\
\text{PHI} &= 90.000 \text{ DEGREES} \\
\end{align*} \]

Layer No. 1 from \( Z = 0 \) to \( Z = 1.5000 \times 10^4 \)

Consists of isotropic material, Young's modulus \( E \) is \( 29600 \times 10^5 \) Poisson's ratio \( \nu \) is \( 0.300 \times 10^4 \)

Coefficients of thermal expansion \( \alpha \) is \( 9600 \times 10^5 \) mass density \( \rho \) is \( 0.000 \times 10^4 \)

PART NO 4

\[ \begin{align*}
\text{Si} &= 0.0000 \\
\text{Sx} &= 0.7500 \times 10^4 \\
\text{IPa} &= 0 \\
\text{INa} &= 1 \\
\text{SHELL TYPE 6} \\
\end{align*} \]

Conical Shell No. 6

\[ \begin{align*}
K &= 0.0000 \\
\text{PHI} &= 90.000 \text{ DEGREES} \\
\end{align*} \]

Layer No. 1 from \( Z = 1.5000 \times 10^4 \) to \( Z = 2.5000 \times 10^4 \)

Consists of isotropic material, Young's modulus \( E \) is \( 29600 \times 10^5 \) Poisson's ratio \( \nu \) is \( 0.300 \times 10^4 \)

Coefficients of thermal expansion \( \alpha \) is \( 9600 \times 10^5 \) mass density \( \rho \) is \( 0.000 \times 10^4 \)

PART NO 5

\[ \begin{align*}
\text{Si} &= 0.0000 \\
\text{Sx} &= 0.0000 \\
\text{IPa} &= 0 \\
\text{INa} &= 1 \\
\text{SHELL TYPE 6} \\
\end{align*} \]

Cylindrical Shell No. 2

\[ \begin{align*}
K &= 0.0000 \\
R &= 1.1000 \times 10^4 \\
\text{PHI} &= 90.000 \text{ DEGREES} \\
\end{align*} \]

Layer No. 1 from \( Z = 1.5000 \times 10^4 \) to \( Z = 2.5000 \times 10^4 \)

Consists of isotropic material, Young's modulus \( E \) is \( 29600 \times 10^5 \) Poisson's ratio \( \nu \) is \( 0.300 \times 10^4 \)

Coefficients of thermal expansion \( \alpha \) is \( 9600 \times 10^5 \) mass density \( \rho \) is \( 0.000 \times 10^4 \)
PART NO 6

ST= .00000  SX= .35000-00  IPAR= 3  ING= 1  SHELL TYPE 6 NTP= 0 LAYERS MLY= 1

CONICAL SHELL NO 6  XZ= .00000  PHI= 180.000 DEGREES  AZ= -11000+01

LAYER NO 1 FROM Z= -.15000-00 TO Z= .15000-00
CONSISTS OF ISOTROPIC MATERIAL  YOUNGS MODULUS E= .29000+08  POISSONS RATIO NU= .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI= .76000+05  ATHETA= .76000+05  MASS DENSITY RHO= .00000

PART NO 7

ST= .00000  SX= .12700-00  IPAR= 7  ING= 1  SHELL TYPE 6 NTP= 0 LAYERS MLY= 1

CONICAL SHELL NO 6  XZ= .00000  PHI= 180.000 DEGREES  AZ= .76000+00

LAYER NO 1 FROM Z= -.15000-00 TO Z= .15000-00
CONSISTS OF ISOTROPIC MATERIAL  YOUNGS MODULUS E= .29000+08  POISSONS RATIO NU= .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI= .76000+05  ATHETA= .76000+05  MASS DENSITY RHO= .00000

PART NO 8

ST= .00000  SX= .70697-00  IPAR= 7  ING= 2  SHELL TYPE 6 NTP= 0 LAYERS MLY= 1

CONICAL SHELL NO 6  XZ= .00000  PHI= 90.050 DEGREES  AZ= .44060+01

VARIABLE Z=COORDINATES OF THIS PART FOLLOW

Z1  LINEAR FUNCTION GENERATOR NO. 1 FROM 2 POINTS

Y COORDINATES =.6000-01  =.6000-02
X COORDINATES =.6000  =.6000-00

Z2  LINEAR FUNCTION GENERATOR NO. 2 FROM 2 POINTS

Y COORDINATES =.6000-01  =.6000-02
X COORDINATES =.6000  =.6000-00

LAYER NO 1 FROM Z= -.6000-01 TO Z= -.6000-02
CONSISTS OF ISOTROPIC MATERIAL  YOUNGS MODULUS E= .29000+08  POISSONS RATIO NU= .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI= .76000+05  ATHETA= .76000+05  MASS DENSITY RHO= .00000

PART NO 9

ST= .00000  SX= .50850+01  IPAR= 25  ING= 1  SHELL TYPE 2  NTP= 0  LAYERS MLY= 1

CYLINDRICAL SHELL NO 2  XZ= .00000  RE= .51800-00  PHI= 90.000 DEGREES

LAYER NO 1 FROM Z= -.6000-00 TO Z= .6000-00
CONSISTS OF ISOTROPIC MATERIAL  YOUNGS MODULUS E= .29000+08  POISSONS RATIO NU= .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI= .76000+05  ATHETA= .76000+05  MASS DENSITY RHO= .00000
PART NO 10

$1 = 0.0000$ $sx = 125000.00$ $ipar = 4$ $ing = 1$ SHELL TYPE 2 $ntr = 0$ LAYERS MLY = 1

CYLINDRICAL SHELL NO 2 $kw = 0.0000$ $rn = 51800.00$ $phi = 90.000$ DEGREES

LAYER NO 1 FROM Z = -745000.02 TO Z = 14000.01
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS $E = 298000.00$ POISSON'S RATIO $\nu = 0.30000.00$

COEFFICIENTS OF THERMAL EXPANSION $A = 76000.05$ $\alpha = 76000.05$ MASS DENSITY $\rho = 0.00000$

PART NO 11

$1 = 0.0000$ $sx = 23040.00$ $ipar = 8$ $ing = 1$ SHELL TYPE 6 $ntr = 0$ LAYERS MLY = 1

CONICAL SHELL NO 6 $kw = 0.0000$ $phi = 180.000$ DEGREES $\alpha = 51800.00$

LAYER NO 1 FROM Z = -745000.01 TO Z = 78000.01
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS $E = 298000.00$ POISSON'S RATIO $\nu = 0.30000.00$

COEFFICIENTS OF THERMAL EXPANSION $A = 76000.05$ $\alpha = 76000.05$ MASS DENSITY $\rho = 0.00000$

PART NO 12

$1 = 0.0000$ $sx = 220000.00$ $ipar = 6$ $ing = 1$ SHELL TYPE 2 $ntr = 0$ LAYERS MLY = 1

CYLINDRICAL SHELL NO 2 $kw = 0.0000$ $rn = 28700.00$ $phi = 90.000$ DEGREES

LAYER NO 1 FROM Z = -125040.01 TO Z = 125040.01
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS $E = 298000.00$ POISSON'S RATIO $\nu = 0.30000.00$

COEFFICIENTS OF THERMAL EXPANSION $A = 76000.05$ $\alpha = 76000.05$ MASS DENSITY $\rho = 0.00000$

PART NO 13

$1 = 0.0000$ $sx = 237000.00$ $ipar = 5$ $ing = 1$ SHELL TYPE 6 $ntr = 0$ LAYERS MLY = 1

CONICAL SHELL NO 6 $kw = 0.0000$ $phi = 180.000$ DEGREES $\alpha = 28700.00$

LAYER NO 1 FROM Z = -1175000.01 TO Z = 1175000.01
CONSISTS OF ISOTROPIC MATERIAL, YOUNG'S MODULUS $E = 298000.00$ POISSON'S RATIO $\nu = 0.30000.00$

COEFFICIENTS OF THERMAL EXPANSION $A = 76000.05$ $\alpha = 76000.05$ MASS DENSITY $\rho = 0.00000$
### Kalwin Shell Stress Analysis: Cold End Pressure Shell

**4 Jan 74**

#### Subcase No. 1 for Fourier Harmonic Cos 0 Theta

**Boundary Conditions at Starting Edge**
- 1 = 0.0000
- 3 = 0.0000
- 5 = 0.0000
- 7 = 0.0000

**Boundary Conditions at Final Edge**
- 2 = 0.0000
- 6 = 0.0000

**Boundary Condition at Branch Edge No. 2**
- 1 = 0.0000
- 3 = 0.0000
- 6 = 0.0000

#### Loads for Part No. 1 Subcase No. 1

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 2 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 0.0000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 3 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 4 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 5 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 6 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000

**Loads for Part No. 7 Subcase No. 1**

**Ring Loads at End of This Part Are**
- q1 = 0.0000
- q2 = 0.0000
- q3 = 0.0000
- q4 = 0.0000
- q5 = 0.0000
- q6 = 0.0000

**Surface and Temp Loads Are**
- P = 1250000
- P1 = 0.0000
- P2 = 0.0000
- P3 = 0.0000
RIFT LOADS AT END OF THIS PART ARE
\[ \theta = 0.0000 \quad NPHI = -0.0000 \quad NP = -0.0000 \]
\[ WK = -0.0000 \quad UK = -0.0000 \quad UTHK = -0.0000 \]

SURFACE AND TEMP LOADS ARE
\[ P = 12500 + 0.0 \quad PFI = -0.0000 \quad PTHETA = 0.0000 \]
\[ TL = 14000 + 03 \quad TU = 14000 + 03 \]

LOADS FOR PART NO 6 SUBCASE NO 1
RIFT LOADS AT END OF THIS PART ARE
\[ \theta = 0.0000 \quad NPHI = -0.0000 \quad NP = -0.0000 \]
\[ WK = -0.0000 \quad UK = -0.0000 \quad UTHK = -0.0000 \]

SURFACE AND TEMP LOADS ARE
\[ P = 12500 + 04 \quad PFI = -0.0000 \quad PTHETA = 0.0000 \]
\[ TL = 14000 + 03 \quad TU = 14000 + 03 \]

CODE NUMBERS OF VARIABLE LOADS OVER THIS PART ARE GIVEN BY FSNS BELOW:
\[ 4 \quad 5 \]

TI LINEAR FUNCTION GENERATOR NO. 3 FROM 2 POINTS
Y COORDINATES: 14000 + 03, 64000 + 02
X COORDINATES: 0.0000, 0.7070 + 00

TO LINEAR FUNCTION GENERATOR NO. 4 FROM 2 POINTS
Y COORDINATES: 14000 + 03, 64000 + 02
X COORDINATES: 0.0000, 0.7070 + 00

LOADS FOR PART NO 9 SUBCASE NO 1
RIFT LOADS AT END OF THIS PART ARE
\[ \theta = 0.0000 \quad NPHI = -0.0000 \quad NP = -0.0000 \]
\[ WK = -0.0000 \quad UK = -0.0000 \quad UTHK = -0.0000 \]

SURFACE AND TEMP LOADS ARE
\[ P = 12500 + 00 \quad PFI = -0.0000 \quad PTHETA = 0.0000 \]
\[ TL = 14000 + 03 \quad TU = 14000 + 03 \]

CODE NUMBERS OF VARIABLE LOADS OVER THIS PART ARE GIVEN BY FSNS BELOW:
\[ 4 \quad 5 \]

TL LINEAR FUNCTION GENERATOR NO. 5 FROM 2 POINTS
Y COORDINATES: 64000 + 02, 34300 + 03
X COORDINATES: 0.0000, 0.5075 + 01

TU LINEAR FUNCTION GENERATOR NO. 6 FROM 2 POINTS
Y COORDINATES: 64000 + 02, 34300 + 03
X COORDINATES: 0.0000, 0.5075 + 01

LOADS FOR PART NO 10 SUBCASE NO 1
RIFT LOADS AT END OF THIS PART ARE
\[ \theta = 0.0000 \quad NPHI = -0.0000 \quad NP = -0.0000 \]
\[ WK = -0.0000 \quad UK = -0.0000 \quad UTHK = -0.0000 \]

SURFACE AND TEMP LOADS ARE
\[ P = 0.0000 \quad PFI = -0.0000 \quad PTHETA = 0.0000 \]
\[ TL = 34300 + 05 \quad TU = 34300 + 05 \]

LOADS FOR PART NO 11 SUBCASE NO 1
<table>
<thead>
<tr>
<th>Part No. 12 Subcase No. 1</th>
<th>Part No. 13 Subcase No. 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ring Loads at End of This Part Are</td>
<td>Ring Loads at End of This Part Are</td>
</tr>
<tr>
<td>( \sigma = -0.0000 )</td>
<td>( \sigma = -0.0000 )</td>
</tr>
<tr>
<td>( \mu = -0.0000 )</td>
<td>( \mu = -0.0000 )</td>
</tr>
<tr>
<td>( \nu = -0.0000 )</td>
<td>( \nu = -0.0000 )</td>
</tr>
<tr>
<td>Surface and Temp Loads Are</td>
<td>Surface and Temp Loads Are</td>
</tr>
<tr>
<td>( P = 12500 + 0 )</td>
<td>( P = 12500 + 0 )</td>
</tr>
<tr>
<td>( F = -0.0000 )</td>
<td>( F = -0.0000 )</td>
</tr>
<tr>
<td>( \theta = -0.0000 )</td>
<td>( \theta = -0.0000 )</td>
</tr>
<tr>
<td>( L = 3.4300 + 0 )</td>
<td>( L = 3.4300 + 0 )</td>
</tr>
<tr>
<td>( T = 3.4300 + 0 )</td>
<td>( T = 3.4300 + 0 )</td>
</tr>
</tbody>
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### DEFLECTIONS AND STRESS RESULTANTS FOR HOLE NUMBER X = 0

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<tr>
<th>K</th>
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<th>UPHI</th>
<th>NPHI</th>
<th>BPHI</th>
<th>UPHI</th>
<th>NPHI</th>
<th>UEFFTA</th>
<th>N</th>
<th>NTHETA</th>
<th>NTHETA</th>
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<tbody>
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<td>0.09006</td>
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<td>0.76455+04</td>
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<td>0.250</td>
<td>0.33505+03</td>
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<td>0.25512+03</td>
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### MAIN SHELL PART NO. 2

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<th>UPHI</th>
<th>NPHI</th>
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<th>N</th>
<th>NTHETA</th>
<th>NTHETA</th>
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</thead>
<tbody>
<tr>
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<td>0.2136+02</td>
<td>0.35757+04</td>
<td>-6.298-06</td>
<td>0.5021-01</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.191+05</td>
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</tr>
<tr>
<td>0.193</td>
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<tr>
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</table>

### MAIN SHELL PART NO. 3

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<th>N</th>
<th>NTHETA</th>
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**Loads for Part No. 1 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 2 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 3 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 4 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 5 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 6 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000

**Loads for Part No. 7 Subcase No. 2**

- Ring Loads at End of This Part are:
  - WK = -0.0000
  - UH = -0.0000
  - UHK = 0.0000

- Surface and Temp Loads are:
  - P = 23000.00
  - PT = 0.0000
RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +28000.04, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +4000.04, & T_u &= +4000.04 \\
\end{align*} \]

Code numbers of variable loads over this part, given by PGENS below, are 4 5

Linear function generator no. 3 from 2 points

\[ \begin{align*}
Y \text{ coordinates} &= +1400.03, +8400.02 \\
X \text{ coordinates} &= +0.0000, +7070.00 \\
\end{align*} \]

To linear function generator no. 4 from 2 points

\[ \begin{align*}
Y \text{ coordinates} &= +1400.03, +8400.02 \\
X \text{ coordinates} &= +0.0000, +7070.00 \\
\end{align*} \]

RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +28000.04, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +84000.04, & T_u &= +34300.04 \\
\end{align*} \]

Code numbers of variable loads over this part, given by PGENS below, are 4 5

Linear function generator no. 5 from 2 points

\[ \begin{align*}
Y \text{ coordinates} &= +8400.02, +34300.03 \\
X \text{ coordinates} &= +0.0000, +5075.01 \\
\end{align*} \]

To linear function generator no. 6 from 2 points

\[ \begin{align*}
Y \text{ coordinates} &= +8400.02, +34300.03 \\
X \text{ coordinates} &= +0.0000, +5075.01 \\
\end{align*} \]

RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +0000.00, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +34300.03, & T_u &= +34300.03 \\
\end{align*} \]

RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +0000.00, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +34300.03, & T_u &= +34300.03 \\
\end{align*} \]

RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +0000.00, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +34300.03, & T_u &= +34300.03 \\
\end{align*} \]

RIN loads at end of this part are

\[ \begin{align*}
G &= \pm0.0000, & N\phi &= \pm0.0000, & N\phi &= \pm0.0000, & N &= \pm0.0000 \\
W_k &= \pm0.0000, & U_k &= \pm0.0000, & U\theta_k &= \pm0.0000 \\
\end{align*} \]

Surface and temp loads are

\[ \begin{align*}
P &= +0000.00, & P\phi_i &= \pm0.0000, & P\theta &= \pm0.0000, & T_\ell &= +34300.03, & T_u &= +34300.03 \\
\end{align*} \]
RING LOADS AT END OF THIS PART ARE  Q = -00000  NPHI = -00000  MPHI = -00000  N = -00000
WK = -00000  UK = -00000  UTHK = -00000

SURFACE AND TEMP LOADS ARE P = 25000+04  PFI = -00000  PTHETA = -00000  TL = -34300+03  TU = -34300+03

LOADS FOR PART NO 12 SUBCASE NO 2
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LOADS FOR PART NO 13 SUBCASE NO 2
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SURFACE AND TEMP LOADS ARE P = 25000+04  PFI = -00000  PTHETA = -00000  TL = -34300+03  TU = -34300+03
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- Branch Shell Part No. 2 includes additional deflections and stress resultants for wave number N = 0.
- Main Shell Part No. 3 includes additional deflections and stress resultants for wave number N = 0.
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- Surveyor:
- Date:
- Location:

**M&R SHELL PART NO 10**

- Description:
- Calibration:
- Owner:
- Surveyor:
- Date:
- Location:

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8.3 SHELL DEFLECTION

THE DEFLECTION AT THE TIP OF THE COLD END SHELL DUE TO SELF-INDUCED VIBRATION WAS DETERMINED BY USING COMPUTER PROGRAM V025.

THE MODEL USED IN THE ANALYSIS IS SHOWN ON PAGE 8-41. THIS MODEL WAS ALSO USED TO FIND THE FIRST MODE FREQUENCY AND TIP DEFLECTION DUE TO RANDOM VIBRATION.

THE RESULTS WERE:

- Tip deflection due to self-induced vib = 8.46 x 10^-6 in. (REF. PG. 8-46)
- First mode frequency = \( f = 241 \text{ cps.} \) (REF. PG. 8-49)
- Tip deflection due to random vib = 0.00086 in. (REF. PG. 8-52)

THE TIP DEFLECTION DUE TO SELF-INDUCED VIBRATION (8.46 x 10^-6 in.) IS MUCH LESS THAN THE SPECIFIED MAXIMUM OF 0.000200 in.

THE FIRST MODE FREQUENCY, WITH AN ASSUMED AMPLIFICATION FACTOR OF 19, GIVES A RANDOM VIBRATION LOAD OF 35.0 G's.

APPLYING 35.0 G's TO THE SHELL RESULTED IN 0.00086 INCHES DEFLECTION AT THE TIP, WHICH IS GREATER THAN THE 0.005 INCH RADIAL CLEARANCE. THEREFORE, UNDER RANDOM VIBRATION LOADING, CONTACT WILL OCCUR BETWEEN THE SHELL AND DISPLACER. SUCH CONTACT, HOWEVER, IS NOT DETRIMENTAL BECAUSE:

1) THE DEFLECTIONS ARE ELASTIC AND MOMENTARY. THE STRESS ASSOCIATED WITH THE MAX DEFLECTION AND MAX OPERATING PRESSURE ARE MUCH LOWER THAN THE ELASTIC LIMIT.

2) THE PROBABLE CONSEQUENCE OF SUCH CONTACT WOULD BE A MOMENTARY STALLING OF THE ENGINE, AFTER WHICH THE ENGINE WOULD CONTINUE TO FUNCTION.
8.3.1 DEFLECTION DUE TO SELF-INDUCED VIBRATION

Fig 8-3 Cold End Shell Deflection Model
8.3.1 DEFLECTION DUE TO SELF INDUCED VIBRATION.

\[ M_1 = 0.078 \times 386 = 0.000 20267 \text{ lb-sec}^2/\text{in.} \]

\[ M_2 = M_1 = M_3 = M_4 = 1.00 \times 386 = 0.000 4876 \text{ lb-sec}^2/\text{in.} \]

\[ M_7 = 1.00 \times \frac{3175}{5312} = 0.000 15239 \text{ lb-sec}^2/\text{in.} \]

\[ M_8 = \frac{224}{386} \times (1.240 - 1.072)(2.00) = 0.000 52388 \text{ lb-sec}^2/\text{in.} \]

\[ x_8_{cc} = \frac{63125 + 100}{7.8125} = 8312.5 \]

\[ M_9 = 0.74(1.240 - 1.072)(3.25) = 0.000 79482 \text{ lb-sec}^2/\text{in.} \]

\[ x_9_{cc} = 8312.5 + 1.25 = 8316.25 \]

\[ M_{10} = \frac{224(1.240 - 1.072)(3.25)}{386} = 0.000 92728 \text{ lb-sec}^2/\text{in.} \]

\[ x_{10_{cc}} = 8316.25 + 1.25 = 8317.5 \]

\[ M_{11} = \frac{(95 + 2.25)}{386} = 0.000 22539 \text{ lb-sec}^2/\text{in.} \] (UNUSUALLY LARGE DISPLACEMENT)

\[ x_{11_{cc}} = 12125 + 2.25 = 12127.25 \]

\[ M_{12} = \frac{224(2.380 - 2.250)(2.00)}{386} = 0.000 71658 \text{ lb-sec}^2/\text{in.} \]

\[ x_{12_{cc}} = 12127.25 + 1.00 = 12128.25 \]
8.3.1 DEFORMATION DUE TO SELF INDUCED VIBRATION

SUPPORT STIFFNESS

MOMENT RESISTANCE AT FIXED SUPPORT

\[ \alpha = 1.6 \]
\[ t = 0.8 \]
\[ b/a = 0.375 \]
\[ e = \gamma M \]
\[ E t^3 \]
\[ \gamma = 0.30 \quad (\text{Revised, p. 24}) \]
\[ \frac{M}{E} = \gamma E t^3 = 0.30 \times 29.8 \times 10^6 \times 30^3 = 24,200,000 \text{ in}-\text{lb/in}, \]

IN-PLANE STIFFNESS WILL BE HIGH.
APPROXIMATE BY USING TWO PLATES - ONE IN TENSION, ONE IN COMPRESSION.

\[ L = a-b = 1.00'' \quad A = 0.333 \times 1.26 = 0.420 \text{ in}^2 \]
\[ w = 6 = 1.20 \]
\[ t = 0.30 \]
\[ k = \frac{24E}{L} = \frac{1360 \times 29.8 \times 10^6 \times 2}{1.00} \]
\[ = 10.7 \times 10^6 \text{ lb/in}, \text{ in} \]
\[ = 21.4 \times 10^6 \text{ lb/ft} \]
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**V.H. CYCLOGENIC ENGINE - COLD END SHELL DYNAMIC RESPONSE AT 490 RPM**

**NIKE MUNIMOTO JAN 74**

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SEARCHING COMPLETED FROM 5.6549E+03 TO 1.1938E+04 RAD/SEC
8.3.2 DEFORMATION DUE TO VIBRATION LOADING

RANDOM VIBRATION LOAD

\[ f = 241 \text{ cfs} \]
\[ T = 10.0 \text{ (assumed)} \]
\[ PSD = 0.026 \text{ g}^2 / \text{cfs} \text{ (Ref. Section 3)} \]

ONE SIGMA RESPONSE

\[ \ddot{X}_1 = \left( \frac{\pi}{2} \times 0.036 \times 10.0 \times 241 \right)^{0.5} \]
\[ = 11.7 \text{ g's} \]

THREE SIGMA RESPONSE

\[ \ddot{X}_3 = 35.0 \text{ g's} \]

SINEOISED VIBRATION

THE SINEOISED VIBRATION INPUT AT 241 CFS IS 2.3 G'S (REF. SECTION 3). ASSUMING AN AMPLIFICATION FACTOR OF 10, THE SINEOISED VIBRATION LOAD BECOMES 23 G'S.

THE RANDOM VIBRATION LOADING IS MORE SEVERE.
B.3.2 DEFLECTION DUE TO VIBRATION LOADING

The deflection due to vibration loading was determined using program V0245 with the inertia loads applied as static loads at each mass station.

\[ P_c = \text{load applied at the mass stations} = W_i \times L = M_i \times g \times L \]

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Note: The above were based on a 39.2 g acceleration. Since the load is 35.0 g's, the deflection from V0245 will be corrected by multiplying by \( \frac{35.0}{39.2} \).

The deflection from V0245 is 0.00964 in. (at 39.2 g's).

At 35.0 g's, \( y = 0.00964 \times \frac{35.0}{39.2} = 0.0086 \) in.
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| 3      | 5.659E-03 | 0.0     | 1.651E-03 | 3.457E-00 | 0.0   | 0.0   |
| 4      | 3.210E-03 | 0.0     | 1.651E-03 | 3.457E-00 | 0.0   | 0.0   |
| 5      | 1.309E-03 | 0.0     | 1.651E-03 | 3.457E-00 | 0.0   | 0.0   |

| 1      | 9.697E-03 | 0.0     | 1.861E-03 | 3.457E-00 | 0.0   | 0.0   |
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| 4      | 3.210E-03 | 0.0     | 1.651E-03 | 3.457E-00 | 0.0   | 0.0   |
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**SEARCHING COMPLETED FROM 0.0 TO 2.0985E 00 RAD/SEC**
**8.4 Cold End Flange Assy**

**NUT PRELOAD**

**JOINT FLEXIBILITY**

---

**ORIGINAL PAGE IS OF POOR QUALITY**

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\[
\Delta P_{\text{Nut}} = \frac{5 + 5}{5} = \frac{1.22 \times 10^{-8}}{2.99 \times 10^{-8}} = 0.410
\]

\[
P_{\text{Nut}} = 4.10 \times P_{\text{Applied}}
\]

\[
P_{\text{Nut}} = P_{\text{Preload}} + 0.410 \times P_{\text{Applied}}
\]
8.4 COLD END FLANGE ASSY

NUT PRELOAD

\[ F_A = \frac{D_p}{T} \left( \frac{2}{(\frac{1}{T} - f_p \sin \lambda)} \left( \frac{T \tan \lambda}{R} + f_p \cos \lambda \right) + \frac{D_m}{\cos \lambda} \right) \]

\( D_p = \) PITCH DIA. = 2.00

\( \lambda = \) HELIX ANGLE = .05720, = 2.8696°

\( f_p = \) THREAD PRESSURE ANGLE = 7°

\( R = 1 + \tan \lambda + \tan^2 \phi = 1.01758 \quad \therefore R^2 = 1.03875 \)

\( f_p = \) THREAD FRAC. FACTOR = .12

\( D_m = \) MEAN CONTACT DIA., AT NUT FACE = 1.605 in.

\( \xi = (180 - COUNTERTURN ANGLE) / 2 = 0 \)

\( f_n = \) NUT FACE FRAC. FACTOR = .12

\( F_A = 3.127 \quad T \quad T = \) in lb
8.4 COLD END FLANGE ASSY

NUT PRELOAD

APPLIED LOADS

CONSIDER SEAL O.D. AS PRESSURIZED AREA

SEAL: AP 26 - 1688 - 7 - SP
OUTSIDE DIA. = 1.491

\[ P_A = \frac{\pi \times 1.471^2 \times 1250.2}{2} = 2010.05 \text{ at operating pressure} \]
\[ = 2.746 \times 2080 = 5700.185 \text{ at proof pressure} \]
\[ = 2.746 \times 2800 = 7730.2 \text{ at burst pressure} \]

FOR \( T = 65 \text{ FT-LBS} = 780.1 \text{ IN-LBS} \) (MIN)

\[ P_{\text{preload}} = 3.727 T = 2910.02 \text{ LBS} \]

AT PROOF PRESSURE

\[ P_{\text{nut}} = 4820.1 \text{ LBS} \]

AS SHOWN ON THE DIAGRAM (FOLLOWING PAGE), THE JOINT WILL NOT SEPARATE AT PROOF PRESSURE.

USE: \( T = 65 \text{ to 70 FT-LBS} \)
8.4 Cold End Flange Assy

T = 65. FT- LBS, (MIN)

P_{\text{bolt}} = P_{\text{preload}} + 0.410 P_{\text{applied}}

P_{\text{preload}} = 3.727 T (110-LBS)

- NUT LOAD - 1000 LBS.
- APPLIED LOAD - 1000 LBS.
8.4 COLD END FLANGE ASSY

THREAD SHEAR

**Shear Area of External Thread at Minor Dia. of Internal Thd.**

Assume \( g = 0.0042 \) (Ref NBS HDPK H28)

\[ D_o = \text{Major Dia. of Ext. Thd} = 1.996 \]

\[ D_m = \text{Minor Dia. of Int. Thd} = D_{basic} - 1.2\phi \]

\[ = 2.000 - 1.2 \times 0.05 = 1.940 \text{ in.} \]

**Thickness at 1/16 Radius**

\[ t = \frac{(D_o - D_m) \tan 7^\circ + F_1 + (D_o - D_m) \tan 45^\circ}{2} \]

\[ F = 0.16314 \]

\[ = \frac{(1.996 - 1.940) \tan 7^\circ + 0.16314 \times 0.05 + 0.028 \tan 45^\circ}{2} \]

\[ = 0.00344 + 0.00316 + 0.028 = 0.0396 \text{ in.} \]

**Shear Area per Thread**

\[ A_\phi = 0.0396 \times \pi \times 1.940 = 0.241 \text{ in}^2 \]

**No. of Engaged Thds (Min.)**

\[ N = 20 \times 20 = 4 \]

**Use** \[ N = 3 \], \( 1 \frac{3}{4} \) in. and carries 50% of load

**Shear Stress at Burst**

\[ \sigma_S = \frac{50 \times 6230}{0.241} = 13,000, \text{ PSI} \]

\[ \text{M.I.S} = \frac{180,000 \times 60 - 1}{13,000} \]

\[ \text{HIGH} \]
9.5 cm O.D. VACUUM JACKET ASSEMBLY

**COLLAPSING PRESSURE**

Assuming external pressure of 150 psi acts on end and radially on cylindrical portion.

From Ref. 5

\[
T_{col} = \frac{C_P D \pi}{4L^2} \quad \text{where} \quad D = \frac{E t^3}{12(1 - \nu^2)}
\]

\[
T_{col} = \frac{C_P E t^3 \pi}{12(1 - \nu^2) R L^2}
\]

\[
\frac{t}{t_{min}} = \frac{1}{1.012}
\]

\[
D_i = 2.237/2.217
\]

\[
E_m = \frac{1}{2} (2.237 + 0.012) = 1.124
\]

\[
L = 5,300,000
\]

**MATERIAL: B-204 CRES**

\[
E = 29,000,000 \quad G = 12.5 \times 10^6 \quad \nu = 0.16
\]

\[
\frac{t}{t_{min}} = \left[ \frac{5,600}{1.124} \right] \left[ 5,600 + 0.012 \right] = 2,300
\]

\[
C_P = 5.01 \quad \text{from Fig. 4, pg 126, Ref. 5}
\]
8.5 Cold End Vacuum Jacket Assy

Collapsing Pressure (cont'd.)

\[
\frac{50 \times 29 \times 10^6 \times 0.123 \times \pi}{12 \times (1.16) \times 1.124 \times 5.6 \times 10^{-6}} = 60.4 \text{ psi}
\]

\[
\frac{15}{5,3} = \frac{60.0}{15} = 3.00
\]
8.5 Cold End Vacuum Jacket

Stress at Cylinder/Ring Junction

Assume ring is rigid.

\[ M = 0.304 \text{ psi} \]

\[ R = 1.124 \]

\[ t = 0.012 \text{ (min.)} \]

\[ \sigma = \frac{6M}{t} = \frac{1.824}{0.012} = 152 \text{,000 psi} \]

\[ f_c = \frac{7R}{2t} = \frac{15 \times 1.124}{2 \times 0.012} = 704 \text{ psi} \]

\[ f_t + f_c = 3,800 \text{ psi} \]

Low collapsing pressure - roughen outer using Ref. 4

\[ l_1 = 4.76 \times R \left( \frac{R}{6t} \right)^{1/3} = 4.76 \times 1.125 \left( \frac{1.125}{0.012} \right)^{1/3} = 53.5 \]

\[ l = 6 \]

Use short tube formula \((l < l_1)\)

\[ \delta' = \frac{0.087 \times t^3}{L R} \left[ \frac{1}{1-m^2} \right] \left( \frac{t}{R} \right) \]

\[ = \frac{25.984 \times 10^6 \times 0.012}{6.711 \times 1.125 \left( \frac{0.012}{1.125} \right)} = 57.3 \text{ psi} \]

\[ M.S. = \frac{57.3}{15} - 1 = 2.82 \]
SECTION 9
HOT-END DISPLACER
STRESS ANALYSIS
9. Hot End Displacer

9.1 Task

9.2 Buckling Analysis

9.3 Shell Analysis

9.4 Bearing Journal Deflection
9.1 TASK

THE ANALYSIS OF THE HOT END DISPLACEMENT CONSISTS OF THE FOLLOWING TASKS:

- DETERMINE THE EXTERNAL COLLAPSING PRESSURE
- DETERMINE STRESSES DUE TO EXTERNAL PRESSURE AND TEMPERATURE GRADIENTS.
- DETERMINE BEARING JOURNAL DEFORMATION AND COMPARE TO CLEARANCE.
9.2 Buckling Analysis

The method described in REF. 5, page 125, involves the utilization of a family of curves relating \( F \) and \( C_p \) for various values of \( F \). These parameters are defined as:

\[
F = \frac{12 (1 - \nu^2)}{L^2 t^3} \left[ I_f + A_f (\frac{r - \bar{z}}{t})^2 \right] + \frac{L_e}{L_f} \left( 1 + 12 \frac{\bar{z}}{t} \right) - 1
\]

Where: 
- \( A_f \) = Area of Ring 
- \( I_f \) = Moment of Inertia of Ring 
- \( e \) = Distance from Shell Mid Surface to Centroid of Ring Cross Section. 
- \( \bar{z} \) = Distance from Shell Mid Surface to Centroid of Combined Section of Ring Frame and Effective Width of Sheet. 
- \( L_f \) = Ring Spacing. 
- \( L_e \) = Total Effective Width of Sheet Assumed Acting with the Ring.

\[
C_p = \frac{D R L \nu}{D \pi^2}
\]

\[
D = \frac{E t^3}{12 (1 - \nu^2)}
\]

\[
\frac{L}{R} = \left( \frac{L}{R} \right) \left( \frac{1}{L} \right) \left( 1 - \nu^2 \right) \frac{1}{R}
\]

The coefficient \( C_p \) is obtained from curves relating \( C_p, \bar{z}, \) and \( F \) on page 9-41.
9.2 Buckling Analysis

\[ L_f = 1.4375 \]
\[ L = 3.75 \]
\[ R = 1.125 - 0.020 = 1.105 \text{ in} \]
\[ A_f = 0.080 \times 1.105 = 0.088 \text{ in}^2 \]
\[ I_f = \frac{0.080 \times 1.105^3}{12} = 0.667 \times 10^{-5} \text{ in}^4 \]
\[ A_e = 0.040 \times 1.4375 = 0.0575 \text{ in}^2 \]
\[ \varepsilon = \frac{0.040 \times 1.4375 + 0.090 \times 0.008}{0.0575 + 0.008} - 0.020 \]
\[ e = \varepsilon (0.090 + 1.00) = 0.075 \]
\[ = 0.00355 \text{ in} \]

\[ F = \frac{12 \times 91}{1.4375 \times 0.040} \left( \frac{0.667 \times 10^{-5} + 0.008 (0.070 - 0.00855)}{0.040} \right) \]
\[ + \left( 1 + 12 \times \frac{0.00855}{0.040} \right) - 1 \]
\[ = 4.92 \]
\[ \varepsilon = \frac{3.75 \times 91}{1.105 \times 0.040} = 3.04 \]
7.2 Buckling Analysis

The curves relating \( C_P \) and \( Z \) for various values of \( F \) are shown below. These curves were obtained from Ref (5).

For \( \pi = 3.94 \) and \( F = 5 \), \( C_P = 60. \)

\[
\pi = \frac{C_P \cdot \pi^3 \cdot \pi}{12 \cdot (1 - \mu^2) \cdot R \cdot L}
\]

\[
= 60 \times 2.44 \times 10^6 \times 0.40^3 \times \pi
\]

\[
= \frac{12 \times 9.1 \times 1.15 \times 3.75}{2}
\]

\[
= 5450, \text{ PSI}
\]
9.2 Buckling Analysis

At Max Operating Pressure,

\[
M.J. = \frac{5450}{1250} \cdot (-1) = 3.36
\]

At Proof Pressure,

\[
M.J. = \frac{5450}{2060} \cdot (-1) = 1.65
\]

At Burst Pressure,

\[
M.J. = \frac{5450}{2800} \cdot (-1) = 0.95
\]
9.3 SHELL ANALYSIS

The hot end displacer was analyzed using a Shell Analysis program developed by A. Kalnin. The theoretical basis for this Shell Analysis program is contained in Ref. (13).

The Shell Analysis model is shown on the next page.

The displacer was analyzed for 2,800 psi burst pressure with a temperature gradient along the displacer centerline. The max stresses were:

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>MAX. STRESS (ksi)</th>
<th>ALLOWABLE STRESS (ksi)</th>
<th>MARGIN OF SAFETY</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>127</td>
<td>180</td>
<td>0.40</td>
</tr>
<tr>
<td>11</td>
<td>124</td>
<td>140</td>
<td>0.13</td>
</tr>
<tr>
<td>12</td>
<td>123</td>
<td>140</td>
<td>0.14</td>
</tr>
<tr>
<td>14</td>
<td>60.5</td>
<td>140</td>
<td>1.31</td>
</tr>
<tr>
<td>27</td>
<td>35.2</td>
<td>35.3</td>
<td>0.80</td>
</tr>
</tbody>
</table>

The Shell program input data are defined on Pages 9-8 through 9-20. The computer output is shown on Page 9-22 through 9-45.

The stress for Part No. 14 was computed manually on Page 9-46. The computer value is erroneously high because the local reinforcement was not inputted into the computer.

Notes:
1. FTU at 1200°F
2. Based on 1250 psi operating pressure
3. Creep allowable at 1200°F; 0.7%; 20,000 hrs.
HOT DISPLACER

SHELL ANALYSIS MODEL
### 9.3 SHELL ANALYSIS

**Input Data for SHELL Program**

**Temperature Gradient and Elastic Properties**

<table>
<thead>
<tr>
<th>Distance Along C (in)</th>
<th>Temp. (°F)</th>
<th>E (10^6 psi)</th>
<th>µ</th>
<th>α (10^-6/°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-</td>
<td>29.3</td>
<td>0.300</td>
<td>7.4</td>
</tr>
<tr>
<td>0.798</td>
<td>-</td>
<td>29.3</td>
<td>0.300</td>
<td>7.4</td>
</tr>
<tr>
<td>1.56</td>
<td>140.0</td>
<td>29.3</td>
<td>0.300</td>
<td>7.4</td>
</tr>
<tr>
<td>2.226</td>
<td>37.8</td>
<td>28.8</td>
<td>0.300</td>
<td>7.4</td>
</tr>
<tr>
<td>3.841</td>
<td>74.4</td>
<td>27.0</td>
<td>0.300</td>
<td>7.8</td>
</tr>
<tr>
<td>9</td>
<td>3.979</td>
<td>80.7</td>
<td>27.0</td>
<td>0.300</td>
</tr>
<tr>
<td>11/4</td>
<td>5.576</td>
<td>72.0</td>
<td>24.4</td>
<td>0.300</td>
</tr>
<tr>
<td>12</td>
<td>5.578</td>
<td>72.0</td>
<td>24.4</td>
<td>0.300</td>
</tr>
<tr>
<td>6.255</td>
<td>120.0</td>
<td>24.4</td>
<td>0.300</td>
<td>8.4</td>
</tr>
<tr>
<td>7.625</td>
<td>120.0</td>
<td>24.4</td>
<td>0.300</td>
<td>8.4</td>
</tr>
</tbody>
</table>
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 1 CYLINDER (LSL=2)

\[ E = 0.0 \quad J = 0.798 \quad R = 0.398 \quad \phi = 20^\circ \quad \theta = -10^\circ \pm \ldots \]

\[ T = 140^\circ F \quad E = 29.8 \times 10^6 \text{psi} \quad \alpha = 7.6 \times 10^{-6} \text{in./in./}^\circ F \]

PART NO. 2 ELLIPTIC (REFERENCE SURFACE) \[ y \]

\[ A = \frac{1}{2} (2.7235 + 2.14 + 15) = 1.09 \text{ in.} \]

\[ B = \frac{1}{2} A = 0.5455 \text{ in.} \]

\[ \frac{\alpha}{\beta} \cdot \frac{\beta}{\alpha} = 1. \]

\[ x = \ldots \]

AT \[ y = 0.398 \text{ in.} \]

\[ X = b \left( 1 - \frac{\gamma}{\alpha} \right) \]

\[ = 0.5455 \left( 1 - \frac{0.398}{1.091} \right) = 0.5079 \]

\[ \Theta = \tan^{-1} \left[ \frac{\alpha}{\beta} \left( \frac{x}{(\alpha^2 + \beta^2)^{1/2}} \right) \right] \]

\[ = \tan^{-1} \left( 0.5455 \cdot 0.5079 \right) \]

\[ = 27.7^\circ = 54.5^\circ = 1.277 \text{ rad} \]

AT \[ y = 0.750 \text{ in.} \]

\[ X = 0.5455 \left( 1 - \frac{0.750}{1.091} \right) = 0.3946 \]

\[ \Theta = \tan^{-1} \left( \frac{0.5455 \cdot 0.3946}{(0.5455^2 + 0.3946^2)^{1/2}} \right) \]

\[ = \tan^{-1} (2.1129) \]

\[ = 66^\circ - 46.3^\circ = 69.6^\circ = 1.192 \text{ rad} \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 2

THICKNESS

INSIDE DIAMETER, \( A = 1.049 \), \( B = 0.5245 \)

\[ x = b \left( 1 - \frac{y}{a} \right)^{\frac{1}{2}} = 0.5245 \left( 1 - \frac{0.378}{1.049} \right)^{\frac{1}{2}} = 0.3671 \]

\[ \Theta = \tan^{-1} \left[ \frac{2 \times 0.3671}{1.5245 - 0.3671} \right] = \tan^{-1} \left[ \frac{2 \times 0.3671}{1.5245 - 0.3671} \right] = \tan^{-1} (1.9558) = 62° - 55', 14.59 = 62.919° \]

AT \( Y = 0.750 \) (GAGE POINT)

\[ x = 0.5245 \left( 1 - \frac{0.750}{1.049} \right)^{\frac{1}{2}} = 0.2957 \]

\[ \Theta = \tan^{-1} \left[ \frac{2 \times 0.2957}{1.5245 - 0.2957} \right] = \tan^{-1} \left[ \frac{2 \times 0.2957}{1.5245 - 0.2957} \right] = \tan^{-1} (4.0772) = 78° - 24', 77 = 78.417° \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 2

THICKNESS

OUTSIDE ELLIPSE

\[ A = 1.112 \quad \frac{B}{2} = 0.5584 \]

AT \( y = 0.399 \):

\[ X = b \left( 1 - \frac{y^2}{a^2} \right) = 0.5584 \left( 1 - \frac{0.399^2}{1.112^2} \right) = 0.5241 \]

\[ \theta = \tan^{-1} \left[ \frac{a \times X}{b \left( b^2 - X^2 \right)^{1/2}} \right] = \tan^{-1} \left[ \frac{0.5584 \times 0.5241}{(0.5584^2 - 0.5241^2)^{1/2}} \right] \]

\[ = \tan^{-1} (5.2179) = 79.929' = 79.151^\circ \]

AT \( y = 0.750 \) (GAGE POINT):

\[ X = 0.5584 \left( 1 - \frac{0.750}{1.112} \right) = 0.41227 \]

\[ \theta = \tan^{-1} \left[ \frac{X + 0.41227}{(0.5584^2 - 0.41227^2)^{1/2}} \right] = \tan^{-1} (2.1893) \]

\[ = (50.0 - 27.135') = 25.451^\circ \]
9.3 SHELL ANALYSIS

INPUT DATA

\[ E - \text{COORDINATE} \ vs \ \phi \]

\[ \begin{align*}
\phi = 0.19346 \cos (70.7^\circ) & \quad \text{AT} \quad \phi = 0.19346 \cos (70.7^\circ) \\
\phi = 0.44204 \cos (0.750) & \quad \text{AT} \quad \phi = 0.44204 \cos (0.750) \\
\phi = \pi/2 = 1.57080 \cos (0) & \quad \text{AT} \quad \phi = \pi/2 = 1.57080 \cos (0)
\end{align*} \]

\[ \begin{align*}
\tau_1 &= (X_k - X_m) \cos (70.7^\circ) = -0.0222 & \tau_2 &= (X_k - X_m) \cos (70.7^\circ) = 0.0122 \\
\tau_1 &= (Y_k - Y_m) \cos (70.7^\circ) = 0.0354 \text{ in.} & \tau_2 &= (Y_k - Y_m) \cos (70.7^\circ) = 0.0412 \text{ in.} \\
\tau_1 &= (Z_k - Z_m) \cos (70.7^\circ) = 0.0063 \text{ in.} & \tau_2 &= (Z_k - Z_m) \cos (70.7^\circ) = 0.002 \text{ in.}
\end{align*} \]

\[ T = 140^\circ F \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 3 CYLINDER (I55 = 8)

\[ S_1 = 0, \quad S_x = 2.56 - 1.62 + \frac{1}{2} \times 0.040 = 0.360 \text{ in.} \]

\[ R = \frac{1}{4} (2.2735 + 2.1415) = 1.09, \quad \phi = 90^\circ \]

\[ t_{mm} = 0.040 \text{ in.}, \quad \frac{1}{2} t_{g} = 0.0 \]

\[ Z = -1.020 \text{ to } 0.020 \]

\[ T_I = 140^\circ F, \quad T_X = 378^\circ F \]

\[ \mu = 0.500, \quad \alpha = 7.6 \times 10^{-6} \]

\[ E_1 = 29.8, \quad EX = 28.8 \]

PART NO. 10 CONE (I55 = 6)

\[ S_1 = 0, \quad S_x = 1.091 - \frac{1}{2} \times 1.844 = 0.119, \quad \phi = 180^\circ \]

\[ T_I = 287^\circ F \]

\[ E = 27.0 \times 10^{-6} \text{ psi}, \quad \mu = 0.300, \quad \omega = 7.8 \times 10^{-6} \]

\[ A_x = -1.091 \]

\[ Z = -1.020 \text{ to } 0.020 \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 9 CYLINDER (E55 = 8)

\[ \sigma_x = 0, \quad x = 1.575, \quad R = 1.091, \quad \phi = 90^\circ \]
\[ z = -0.02 \quad \tau = 0,02 \]
\[ T = 441.9^\circ F, \quad T_x = 887.9^\circ F \]
\[ E = 28.8 \times 10^6 \quad \alpha_T = 27.0 \quad \mu = 0.300 \]
\[ \lambda = 7.1 \times 10^{-6} \quad \lambda_T = 7.3 \times 10^{-6} \]

PART NO. 4 CONE (E55 = 6)

\[ \sigma_x = 0, \quad x = 1.091 - \left( \frac{1}{4} \times 1.3014 - 0.036 \right) = 0.170 \quad m, \]
\[ \phi = 90^\circ, \quad \alpha = -1.091 \]
\[ z = -0.025 \quad \tau = 0.025 \]
\[ T = 378.0^\circ F, \quad E = 28.8, \quad \mu = 0.300, \quad \alpha = 7.6 \]

PART NO. 5 CYLINDER (E55 = 2)

\[ \sigma_x = 0, \quad x = 0.175, \quad R = 0.921, \quad \phi = 90^\circ \]
\[ z = -0.015 \quad \tau = 0.015 \]
\[ T = 378^\circ F \]
\[ E = 28.8 \times 10^6 \quad \mu = 0.300 \]
\[ \alpha = 7.6 \times 10^{-6} \quad \text{in/\textbf{m}/^\circ \text{F}} \]
9.3 SHHELL ANALYSIS

INPUT DATA

PART NO. 6 CYLINDER

\( S_{Z} = 0, \quad S_{X} = 0.155, \quad R = 1.091, \quad \phi = 90^\circ \)

\( Z = 0.020 \) to \( 0.020 \)

\( T_{I} = 378^\circ F, \quad T_{F} = 441^\circ F, \quad E = 28.8, \quad \alpha = 30, \quad \alpha = 27.6 \)

PART NO. 7 CONE

\( S_{Z} = 0, \quad S_{X} = 1.091 - \frac{1}{4}x10\times19 + 0.025 = 0.177 \)

\( A = 1.091, \quad \phi = 180^\circ \)

\( Z = 0.020 \) to \( 0.020 \)

\( T = 441^\circ F, \quad E = 28.8 \times 10^6, \quad \alpha = 7.6 \times 10^6 \)

PART NO. 8 CYLINDER

\( S_{Z} = 0, \quad S_{X} = 0.150, \quad R = 0.974, \quad \phi = 270^\circ \)

\( Z = 0.023 \) to \( 0.023 \)

\( T_{I} = 441^\circ F, \quad E = 28.8 \times 10^6, \quad \alpha = 7.6 \times 10^6 \)

PART NO. 11 CYLINDER

\( S_{Z} = 0, \quad S_{X} = 1.450, \quad R = 1.091, \quad \phi = 90^\circ, \quad \phi = 90^\circ \)

\( Z = 0.020 \) to \( 0.020 \)

\( T_{I} = 807^\circ F, \quad T_{F} = 1200^\circ F \)

\( E_{I} = 27.0, \quad E_{F} = 24.4, \quad \alpha_{I} = 7.8, \quad \alpha_{F} = 8.4, \quad \mu = 0.30 \)
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 12 COIL (I = 6)

\[ \sigma_t = 0.0, \quad A = - (2.182^2 + 1.091)\beta, = -2.43955 \]

\[ 3x = (130^o + 0.65)^{\beta}, = 1453^o \]

\[ \theta = \tan^{-1} (0.65/130) = 26.525^o \]

\[ \phi = 90 + \theta = 116.525^o \]

\[ z_1 = -0.020, \quad z_2 = 0.050 \]

\[ T = 1200^o F, \quad E = 24.4, \quad \nu = 0.3, \quad \alpha = 8.4 \times 10^{-6} \]

PART NO. 13 CYLINDER (I = 2)

\[ \sigma_t = 0, \quad 3x = 0.3, \quad R = 1.026, \quad \phi = 90^o \]

\[ t = 0.0595 \text{ in}, \quad z = -0.030, \quad T = 1200^o F \]

\[ L = 2.37 \times 0.166 = 0.302 \text{ in.} \]

\[ E = 24.4 \times 10^6, \quad \alpha = 8.4 \times 10^{-6} \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 14 ELLIPSOID

REFERENCE (SURFACE) ELLIPSOID  \( A = 1.026 \), \( B = 0.518 \)

AT \( y = 0.504 \)

\[
\begin{align*}
X &= 0.513 \left( 1 - \frac{0.504}{1.026} \right) = 0.44684 \\
\theta_1 &= \tan^{-1} \left[ \frac{2 \times 0.44684}{(1.132 - 0.44684)} \right] = \tan^{-1} (3.5463) \\
&= 74^\circ - 15', 150'' = 74^\circ 0', 252^\circ = 1.27595 \text{ rad} \\
\phi_1 &= \frac{\pi}{2} + \theta_1 = 2.19675 \text{ rad}.
\end{align*}
\]

AT \( y = 0.750 \) (GAGE POINT)

\[
\begin{align*}
X &= 0.513 \left( 1 - \frac{0.750}{1.026} \right) = 0.35006 \\
\theta_2 &= \tan^{-1} \left[ \frac{2 \times 0.35006}{(1.132 - 0.35006)} \right] = \tan^{-1} (11.8670) \\
&= 61^\circ - 49', 53'' = 61.82564^\circ = 1.07906 \text{ rad} \\
\phi_2 &= \frac{\pi}{2} + \theta_2 = 2.164586 \text{ rad}.
\end{align*}
\]
7.3 SHELL ANALYSIS

INPUT DATA

PART NO. 14

INSIDE CIRCLE: \( A = 1.962 \), \( B = 0.498 \)

\[
\Delta y = 0.504
\]

\[
x = 0.498 \left( 1 - \frac{0.504}{0.9962} \right) = 0.42965
\]

\[
\theta = \tan^{-1} \left[ \frac{2 \times 0.42965}{(0.498 - 0.42965) \times 0.9962} \right] = 73.1377
\]

\[
= 73.0 - 40.083' = 73.000'
\]

\[
\Delta y = 0.750
\]

\[
x = 0.52803 \text{ in.}
\]

\[
\theta = \tan^{-1} (1.7509) = 60.16038' = 60.263'
\]

OUTSIDE CIRCLE: \( A = 1.112 \), \( B = 0.5584 \)

\[
\Delta y = 0.750
\]

\[
x = 0.5584 \left( 1 - \frac{0.504}{1.112} \right) = 0.47775
\]

\[
\theta = \tan^{-1} \left[ \frac{2 \times 0.47775}{(0.5584 - 0.47775) \times 1.112} \right] = 75.441456
\]

\[
= 75.44 - 14.567' = 75.736'
\]

\[
\Delta y = 0.750
\]

\[
x = 0.41227
\]

\[
\theta = 55.451'
\]

SCEC CALC, FROM POINT NO. 12

ARIESearch MANUFACTURING COMPANY OF CALIFORNIA
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 14

\[ z \text{ COORDINATES VS. } \phi \]

\[ AT \phi = \frac{\pi}{4} = 1.5708 \text{ Radians} \quad (x=0, y=0) \]

\[ z_1 = -(1.026 - 0.969) (100) = -0.0292 \]

\[ z_c = (1.112 - 1.026) = 0.0860 \]

\[ t = z_1 + z_c = 0.115 \text{ IN.} \]

\[ AT \phi = 2.64986 \quad (y = 0.750) \]

\[ z_1 = -(0.35006 - 0.32803) (0.88152) = -0.0194 \]

\[ z_c = (0.41227 - 0.35006) (0.88152) = 0.0548 \]

\[ t = 0.0742 \text{ IN.} \]

\[ AT \phi = 2.86475 \quad (y = 0.504) \]

\[ z_1 = -(0.44684 - 0.42965) (0.96246) = -0.0165 \]

\[ z_c = (0.49775 - 0.44684) (0.96246) = 0.0490 \]

\[ t = z_1 + z_c = 0.0655 \text{ IN.} \]

\[ T = 1200.4^\circ F \quad E = 2.4 \times 10^6 \quad D = 8.4 \times 10^9 \]
9.3 SHELL ANALYSIS

INPUT DATA

PART NO. 15 CYLINDER (L=55 =2)

\( \delta \delta = 0, \quad \delta x = 1.30 \)

\( R = \frac{1}{4} (0.624 + 0.374) = 0.504, \quad \phi = 90^\circ \)

\( t = 0.123 \text{ in.} \)

\( z = -0.415, \quad T = 1015 \)

\( T = 1200^\circ F \)

PART NO. 16 CONE (L=55 = 6)

\( \delta \delta = 0, \quad \delta x = 0.404, \quad A = -0.504, \quad \phi = 100^\circ \)

\[ \begin{array}{cccc}
\delta & t & z_1 & z_2 \\
0 & 0.200 & -0.100 & 0.100 \\
0.404 & 0.120 & -0.100 & 0.100 \\
\end{array} \]

\( T = 1200^\circ F \)
9.3 SHELL ANALYSIS

The shell program showed the following Max stresses due to 2800 psi burst pressure plus temperature loading.

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>MAX. STRESS (KSI)</th>
<th>ALLOW. STRESS (KSI)</th>
<th>M. S.</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>12.9</td>
<td>120.1</td>
<td>0.40</td>
</tr>
<tr>
<td>11</td>
<td>12.4</td>
<td>140.*</td>
<td>0.13</td>
</tr>
<tr>
<td>12</td>
<td>12.3</td>
<td>140.*</td>
<td>0.14</td>
</tr>
<tr>
<td>14</td>
<td>11.9</td>
<td>140.*</td>
<td>0.18</td>
</tr>
</tbody>
</table>

*At 1200°F,

The computer output was included as pages 9-22 through 9-45.

The stress at the junction of part 14 and 15 was computed by the program is higher than actual. This is because the local reinforcement was not included into the program. The stress in that area will be computed manuverly.
KALWIN SHELL STRESS ANALYSIS........ NOT END DISPLACEMENT MEGUMOTO MAR 73

STATIC ANALYSIS . PARTS=.16 BRANCHES=3 NUMBER OF SUBCASES=1

ANGLES OF ROTATION OF BOUNDARY CONDITIONS ARE ALFL= -.00000 ALFR= -.00000 -.00000

ANALYSIS. PART NO 1

PART NO 1

Cylindrical Shell No 2 K= -.00000 R= .79900=0 PH= 90.000 DEGREES

Layer NO 1 FROM Z= -.10900+00 TO Z= .10900+00

CONSISTS OF ISOTROPIC MATERIAL. YOUNG'S MODULUS E= .29800+08 POISSON'S RATIO NU= .50000-00

COEFFICIENTS OF THERMAL EXPANSION AFI= .76000=05 ATHETA= .76000=05 MASS DENSITY RHO= .00000

PART NO 2

ELIPSOIDAL SHELL NO 5 K= -.00000 R= .10910+00 PHT= 90.000 DEGREES

Layer NO 1 FROM Z= -.40000-01 TO Z= .40000-01

CONSISTS OF ISOTROPIC MATERIAL. YOUNG'S MODULUS E= .29800+08 POISSON'S RATIO NU= .50000-00

COEFFICIENTS OF THERMAL EXPANSION AFI= .76000=05 ATHETA= .76000=05 MASS DENSITY RHO= .00000

PART NO 3

VARIABLE Z-COORDINATES OF THIS PART FOLLOW

Y COORDINATES .2220+01 -.2660+01 .4200+01

X COORDINATES .1935+00 .4420+00 .1571+01

Z0 LINEAR FUNCTION GENERATOR NO. 2 FROM 3 POINTS

Y COORDINATES .1320+01 .1440+01 .3200+01

X COORDINATES .1935+00 .4420+00 .1571+01

Layer NO 1 FROM Z= .29800-01 TO Z= .29800-01

CONSISTS OF ISOTROPIC MATERIAL. YOUNG'S MODULUS E= .29800+08 POISSON'S RATIO NU= .50000-00

COEFFICIENTS OF THERMAL EXPANSION AFI= .76000+05 ATHETA= .76000+05 MASS DENSITY RHO= .00000

VARIABLE ELASTIC PROPERTIES OF LAYER NO. 1 FOLLOW

EVAR LINEAR FUNCTION GENERATOR NO. 3 FROM 2 POINTS

Y COORDINATES .2980+08 .2880+08

X COORDINATES .0000 .9600+00

MU LINEAR FUNCTION GENERATOR NO. 4 FROM 2 POINTS
ALPHA LINEAR FUNCTION GENERATOR NO. 5 FROM 2 POINTS

Y COORDINATES .7600-05 .7600-05
X COORDINATES -.0000 .9600-00

LAYER NO. 1 FROM Z = -.20000-01 TO Z = -.20000-01
CONSISTS OF ISOTROPIC MATERIAL; YOUNG'S MODULUS E = .28800-08; POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .76000-05; ATHETA = .76000-05; MASS DENSITY RHO = -.00000

PART NO. 4

CONICAL SHELL NO. 6 KM = -.00000 PHI = 100.00 DEGREES A = -.10910+01
LAYER NO. 1 FROM Z = -.25000-01 TO Z = -.25000-01
CONSISTS OF ISOTROPIC MATERIAL; YOUNG'S MODULUS E = .28800-08; POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .76000-05; ATHETA = .76000-05; MASS DENSITY RHO = -.00000

PART NO. 5

CYLINDRICAL SHELL NO. 2 KM = -.00000 RE = -.21000-00 PHI = 90.00 DEGREES
LAYER NO. 1 FROM Z = -.15000-01 TO Z = -.15000-01
CONSISTS OF ISOTROPIC MATERIAL; YOUNG'S MODULUS E = .28800-08; POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .76000-05; ATHETA = .76000-05; MASS DENSITY RHO = -.00000

PART NO. 6

CYLINDRICAL SHELL NO. 2 KM = -.00000 RE = -.10910+01 PHI = 90.00 DEGREES
LAYER NO. 1 FROM Z = -.20000-01 TO Z = -.20000-01
CONSISTS OF ISOTROPIC MATERIAL; YOUNG'S MODULUS E = .28800-08; POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .76000-05; ATHETA = .76000-05; MASS DENSITY RHO = -.00000

PART NO. 7

CONICAL SHELL NO. 6 KM = -.00000 PHI = 100.00 DEGREES A = -.10910+01
LAYER NO. 1 FROM Z = -.20000-01 TO Z = -.20000-01
CONSISTS OF ISOTROPIC MATERIAL; YOUNG'S MODULUS E = .28800-08; POISSON'S RATIO NU = .30000-00
COEFFICIENTS OF THERMAL EXPANSION AFI = .76000-05; ATHETA = .76000-05; MASS DENSITY RHO = -.00000

PART NO. 8
### Layer 1
- **From Z:** 0.23000.01 to **Z:** 0.23000.01
- **Material:** Isotropic
- **Young's Modulus:** 2.28000.08
- **Poisson's Ratio:** 0.30000.00
- **Thermal Expansion Coefficients:**
  - \( \alpha = 0.78000.05 \)
  - \( \alpha = 0.78000.05 \)
- **Mass Density:** 0.00000.00

### Layer 2
- **From Z:** 0.23000.01 to **Z:** 0.23000.01
- **Material:** Isotropic
- **Young's Modulus:** 2.28000.08
- **Poisson's Ratio:** 0.30000.00
- **Thermal Expansion Coefficients:**
  - \( \alpha = 0.78000.05 \)
  - \( \alpha = 0.78000.05 \)
- **Mass Density:** 0.00000.00

### General Layer
- **Type:** General
- **K:** 0.00000.00
- **R:** 0.97400.00
- **Phi:** 270.000.00

### Part No. 9
- **Si:** 0.00000.00
- **SX:** 0.15750.01
- **IPAR:** 10
- **ING:** 2
- **Shell Type:** 2
- **NT:** 0
- **Layers:** MLY: 1

### Part No. 10
- **Si:** 0.00000.00
- **SX:** 0.11900.00
- **IPAR:** 2
- **ING:** 2
- **Shell Type:** 6
- **NT:** 2
- **Layers:** MLY: 1

### Part No. 11
- **Si:** 0.00000.00
- **SX:** 0.10500.01
- **IPAR:** 10
- **ING:** 3
- **Shell Type:** 8
- **NT:** 0
- **Layers:** MLY: 1

### General Shell No. 8
- **K:** 0.00000.00
- **R:** 0.10910.01
- **Phi:** 90.000.00
### Variable Elastic Properties of Layer No. 1 Follow

#### Linear Function Generator No. 9 from 2 Points

<table>
<thead>
<tr>
<th>X Coordinates</th>
<th>Y Coordinates</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0000</td>
<td>2.7000+08</td>
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<tr>
<td>1.450+01</td>
<td>2.440+08</td>
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#### Linear Function Generator No. 10 from 2 Points

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<thead>
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<th>X Coordinates</th>
<th>Y Coordinates</th>
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<td>0.0000</td>
<td>3.000+00</td>
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<tr>
<td>1.450+01</td>
<td>3.000+00</td>
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#### Linear Function Generator No. 11 from 2 Points

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<tr>
<th>X Coordinates</th>
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<tr>
<td>0.0000</td>
<td>7.800+05</td>
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<tr>
<td>1.450+01</td>
<td>8.400+05</td>
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#### Layer No. 1 from Z = -2.0000+01 to Z = 2.0000+01

- Consists of isotropic material, Young's modulus $E = 2.7000+08$, Poisson's ratio $\nu = 3.0000+00$
- Coefficients of thermal expansion $\alpha_T = 7.8000+05$, $\alpha_{\theta} = 7.8000+05$, Mass density $\rho = -.00000$

---

### Part No. 12

- $S_1 = -.00000$, $S_\theta = 1.453+00$, $I_P = 5$, $I_N = 1$, shell type 6, $N_T = 0$, layers $N_L = 1$
- Conical shell, $R = 0.0000$, $\phi = 116.565$ degrees
- Linear Function Generator No. 12 from 2 Points

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<tr>
<td>1.653+00</td>
<td>-5.0000+01</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>X Coordinates</th>
<th>Y Coordinates</th>
</tr>
</thead>
<tbody>
<tr>
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<td>-1.0000+00</td>
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<tr>
<td>1.653+00</td>
<td>-.0000</td>
</tr>
</tbody>
</table>

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### Part No. 13

- $S_1 = -.00000$, $S_\theta = 3.0000+00$, $I_P = 6$, $I_N = 3$, shell type 2, $N_T = 0$, layers $N_L = 1$
- Cylindrical shell, $R = 0.0000$, $\phi = 90.000$ degrees
- Linear Function Generator No. 13 from 2 Points

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<thead>
<tr>
<th>X Coordinates</th>
<th>Y Coordinates</th>
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<td>0.0000</td>
<td>-3.0000+01</td>
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<tr>
<td>1.0260+01</td>
<td>90.000</td>
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</table>

- Layer No. 1 from Z = -3.0000+01 to Z = 3.0000+01
- Consists of isotropic material, Young's modulus $E = 2.4400+08$, Poisson's ratio $\nu = 3.0000+00$
- Coefficients of thermal expansion $\alpha_T = 8.4000+05$, $\alpha_{\theta} = 8.4000+05$, Mass density $\rho = -.00000$
PART NO 14

S# 15708+01 3X 26668+01 IPAR= 10 ING= 3 SHELL TYPE 5 NTP= 0 LAYERS MLY= 1

ELLIPSOIDAL SHELL NO 5  K= 50000  A= 10260+01  B= 51500+00  DIRECTn= 1.

VARIABLE Z-COORDINATES OF THIS PART FOLLOW

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<tr>
<th>ZI</th>
<th>LINEAR FUNCTION GENERATOR NO. 14 FROM 3 POINTS</th>
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<td>---</td>
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<tr>
<td>26668+01</td>
<td>1571+01</td>
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<tr>
<th>Z0</th>
<th>LINEAR FUNCTION GENERATOR NO. 15 FROM 3 POINTS</th>
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<td>---</td>
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<tr>
<td>2667+01</td>
<td>1571+01</td>
</tr>
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</table>

LAYER NO 1 FROM Z= 40000+01 TO Z= 40000+01

CONSISTS OF ISOTROPIC MATERIAL YOUNG'S MODULUS E = 24400+08 POISSONS RATIO NU = 0.30000+00

COEFFICIENTS OF THERMAL EXPANSION AF= 0.0000+00 ATHETA= 0.84000+05 MASS DENSITY HMD= 0.0000+00

PART NO 15

S# -0000 3X 13000+01 IPAR= 10 ING= 1 SHELL TYPE 2 NTP= 0 LAYERS MLY= 1

CYLINDRICAL SHELL NO. 2  K= -0000  R= 59400+60 PHN= 90.000 DEGREES

LAYER NO 1 FROM Z= 61500+01 TO Z= 61500+01

CONSISTS OF ISOTROPIC MATERIAL YOUNG'S MODULUS E = 24400+08 POISSONS RATIO NU = 0.30000+00

COEFFICIENTS OF THERMAL EXPANSION AF= 0.0000+00 ATHETA= 0.84000+05 MASS DENSITY HMD= 0.0000+00

PART NO 16

S# -0000 3X 40900+00 IPAR= 3 ING= 2 SHELL TYPE 6 NTP= 0 LAYERS MLY= 1

CONICAL SHELL NO 5  K= -0000 PHN= 180.000 DEGREES A= 50400+00

VARIABLE Z-COORDINATES OF THIS PART FOLLOW

<table>
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<th>ZI</th>
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<tr>
<td>40900+00</td>
<td>1000+00</td>
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</table>

<table>
<thead>
<tr>
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<td>---</td>
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<tr>
<td>40400+00</td>
<td>1000+00</td>
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</tbody>
</table>
LAYER NO 1 FROM Z = -.10000+00 TO Z = .10000+00
CONSIST OF ISOTROPIC MATERIAL YOUNG'S MODULUS E = 244000+08
POISSON'S RATIO N = .30000+00
YOUNG'S MODULUS  = .10000+00
POISSON'S RATIO N = .30000+00
COEFFICIENTS OF THERMAL EXPANSION \( Af \) = .64000-05
\( \Theta = .64000-05 \)
MASS DENSITY \( \rho \) = -.00000
## TNINS SHELL STRESS ANALYSIS

### Hot End Displacement

**Author:** Morimoto

**Date:** Mar 73

---

**Subcase No. 1 for Fourier Harmonic COS Θeta**

<table>
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<tr>
<th>Boundary Conditions at Starting Edge</th>
<th>1</th>
<th>0.0000</th>
<th>3</th>
<th>0.0000</th>
<th>5</th>
<th>0.0000</th>
<th>7</th>
<th>0.0000</th>
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</thead>
<tbody>
<tr>
<td>Boundary Conditions at Final Edge</td>
<td>2</td>
<td>0.0000</td>
<td>4</td>
<td>0.0000</td>
<td>6</td>
<td>0.0000</td>
<td>8</td>
<td>0.0000</td>
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</tbody>
</table>

**Loads for Part No. 1 Subcase No. 1**

- Ring loads at end of this part are:
  - \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000
  - \( \psi \) = 0.0000, \( \psi \) = 0.0000, \( \psi \) = 0.0000
- Surface and temp loads are:
  - \( P = 2800 \pm 64 \) \( \psi \) = 0.0000, \( \psi \) = 0.0000, \( \psi \) = 0.0000

**Loads for Part No. 2 Subcase No. 1**

- Ring loads at end of this part are:
  - \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000
- Surface and temp loads are:
  - \( P = 2800 \pm 64 \) \( \psi \) = 0.0000, \( \psi \) = 0.0000, \( \psi \) = 0.0000

**Loads for Part No. 3 Subcase No. 1**

- Ring loads at end of this part are:
  - \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000
- Surface and temp loads are:
  - \( P = 2800 \pm 64 \) \( \psi \) = 0.0000, \( \psi \) = 0.0000, \( \psi \) = 0.0000

**Linear Function Generator**

- 10 points from 2 points

**Y Coordinates**
- 1.4000 + 03, 3.780 + 03

**X Coordinates**
- 0.0000, 9.600 + 00

**Y Coordinates**
- 1.4000 + 03, 3.780 + 03

**X Coordinates**
- 0.0000, 9.600 + 00

**Loads for Part No. 4 Subcase No. 1**

- Ring loads at end of this part are:
  - \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000, \( \phi \) = 0.0000
- Surface and temp loads are:
  - \( P = 3780 \pm 60 \) \( \psi \) = 0.0000, \( \psi \) = 0.0000, \( \psi \) = 0.0000

---

**Note:** LOADS FOR PART NO. 4 SUBCASE NO. 1

**Page:** 5-29

**Original Page Quality:** Poor

---
LOADS FOR PART NO 5 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE: QZ = 0.0000, QPHI = 0.0000, MPU = 0.0000, N = 0.0000
WK = 0.0000, UK = 0.0000, UTHK = 0.0000
SURFACE AND TEMP LOADS ARE: P = -0.0000, PF = 0.0000, PTHETA = 0.0000, TL = 37800+03, TU = 37800+03

LOADS FOR PART NO 6 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE: QZ = 0.0000, QPHI = 0.0000, MPU = 0.0000, N = 0.0000
WK = 0.0000, UK = 0.0000, UTHK = 0.0000
SURFACE AND TEMP LOADS ARE: P = -28000+03, PF = 0.0000, PTHETA = 0.0000, TL = 37800+03, TU = 44100+03

CODE NUMBERS OF VARIABLE LOADS OVER THIS PART, GIVEN BY FGNS BELOW, ARE 4 5

Y COORDINATES: 0.3780+03, 0.4410+03
X COORDINATES: 0.0000, 1.5500

LOADS FOR PART NO 7 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE: QZ = 0.0000, QPHI = 0.0000, MPU = 0.0000, N = 0.0000
WK = 0.0000, UK = 0.0000, UTHK = 0.0000
SURFACE AND TEMP LOADS ARE: P = -0.0000, PF = 0.0000, PTHETA = 0.0000, TL = 44100+03, TU = 44100+03

LOADS FOR PART NO 8 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE: QZ = 0.0000, QPHI = 0.0000, MPU = 0.0000, N = 0.0000
WK = 0.0000, UK = 0.0000, UTHK = 0.0000
SURFACE AND TEMP LOADS ARE: P = -0.0000, PF = 0.0000, PTHETA = 0.0000, TL = 44100+03, TU = 44100+03

LOADS FOR PART NO 9 SUBCASE NO 1
RING LOADS AT END OF THIS PART ARE: QZ = 0.0000, QPHI = 0.0000, MPU = 0.0000, N = 0.0000
WK = 0.0000, UK = 0.0000, UTHK = 0.0000
SURFACE AND TEMP LOADS ARE: P = -28000+04, PF = 0.0000, PTHETA = 0.0000, TL = 44100+03, TU = 80700+03

CODE NUMBERS OF VARIABLE LOADS OVER THIS PART, GIVEN BY FGNS BELOW, ARE 4 5

Y COORDINATES: 0.4410+03, 0.8070+03
X COORDINATES: 0.0000, 1.5500
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<thead>
<tr>
<th>LINEAR FUNCTION GENERATOR NO.</th>
<th>FROM 2 POINTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>X COORDINATES</td>
<td>-0.0000</td>
</tr>
<tr>
<td>Y COORDINATES</td>
<td>-0.0000</td>
</tr>
<tr>
<td>X COORDINATES</td>
<td>1.450001</td>
</tr>
</tbody>
</table>

**LOADS FOR PART NO 10 SUBCASE NO 1**

RING LOADS AT END OF THIS PART ARE:  
- W = -0.0000  
- U = -0.0000  
- UTH = 0.0000  

SURFACE AND TEMP LOADS:  
- P = -0.0000  
- PHI = -0.0000  
- T = -12000+04

**LOADS FOR PART NO 11 SUBCASE NO 1**

RING LOADS AT END OF THIS PART ARE:  
- W = -0.0000  
- U = -0.0000  
- UTH = 0.0000  

SURFACE AND TEMP LOADS:  
- P = -28000+04  
- PHI = -0.0000  
- T = 12000+04

**LOADS FOR PART NO 12 SUBCASE NO 1**

RING LOADS AT END OF THIS PART ARE:  
- W = -0.0000  
- U = -0.0000  
- UTH = 0.0000  

SURFACE AND TEMP LOADS:  
- P = -28000+04  
- PHI = -0.0000  
- T = 12000+04

**LOADS FOR PART NO 13 SUBCASE NO 1**

RING LOADS AT END OF THIS PART ARE:  
- W = -0.0000  
- U = -0.0000  
- UTH = 0.0000  

SURFACE AND TEMP LOADS:  
- P = -28000+04  
- PHI = -0.0000  
- T = 12000+04

**LOADS FOR PART NO 14 SUBCASE NO 1**

RING LOADS AT END OF THIS PART ARE:  
- W = -0.0000  
- U = -0.0000  
- UTH = 0.0000  

SURFACE AND TEMP LOADS:  
- P = -28000+04  
- PHI = -0.0000  
- T = 12000+04
SURFACE AND TEMP LOADS ARE 
P = \cdot 26000 + 04 
PF = \cdot 00000 
PTh = A = \cdot 00000 
TL = \cdot 12000 + 04 
TU = \cdot 12000 + 04 

LOADS FOR PART NO 15 SUBCASE NO 1

RING LOADS AT END OF THIS PART ARE 
R = \cdot 00000 
MPH = \cdot 00000 
NPHI = \cdot 00000 
N = \cdot 00000 

SURFACE AND TEMP LOADS ARE 
P = \cdot 26000 + 04 
PF = \cdot 00000 
PTh = A = \cdot 00000 
TL = \cdot 12000 + 04 
TU = \cdot 12000 + 04 

LOADS FOR PART NO 16 SUBCASE NO 1

RING LOADS AT END OF THIS PART ARE 
R = \cdot 00000 
MPH = \cdot 00000 
NPHI = \cdot 00000 
N = \cdot 00000 

SURFACE AND TEMP LOADS ARE 
P = \cdot 26000 + 04 
PF = \cdot 00000 
PTh = A = \cdot 00000 
TL = \cdot 12000 + 04 
TU = \cdot 12000 + 04
### MAIN SHELL PART NO. 1

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<tr>
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<th>Q</th>
<th>UPHI</th>
<th>NPHI</th>
<th>BPNI</th>
<th>NPHI</th>
<th>UTHETA</th>
<th>N</th>
<th>NTHETA</th>
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<tbody>
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<td>1.0000</td>
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<td>1.0000</td>
<td>-4362+03</td>
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### MAIN SHELL PART NO. 2

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<th>UTHETA</th>
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<tbody>
<tr>
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<td>-52162+03</td>
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**KAINING SHELL STRF37 ANALYSIS........... HOT END DISPLACER (ORIMOTO) MAR 73**

DEFLECTIONS AND STRESS RESULTANTS FOR WAVE NUMBER NX = 0 2000 PSI BURST PRESSURE PLUS TEMPERATURE LOADING

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**ORIGIHAL PAGE IS OF POOR QUALITY**

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**Page 933**

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**Page 98-996-3**
<table>
<thead>
<tr>
<th>SHELL PART NO 15</th>
<th>SHELL PART NO 16</th>
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**Note:** The table represents deflections and stresses for wave number \( n = 0 \). The conditions include 2000 PSI burst pressure plus temperature load.
| Page 74-896.2 | Original Page IS of Poor Quality |
9.3 SHELL ANALYSIS

DOME/BEARING JOURNAL JUNCTION, OUTBD END

CHECK LOCAL REINFORCED AREA

\[ a_o = 1.175 \quad a = 1.08 \]
\[ t = 0.090 \]

For \( t = \frac{h}{a} \):
\[ R = 2a = 2.16 \]
\[ P = 0.297 - 0.0625 = 0.234 \]
\[ t_c = 0.125 \]

\[ \rho = \frac{R_o}{R_i} \left( \frac{R_i}{t_i} \right)^k = \frac{0.234}{(2.160 \times 0.090)^{0.5}} = 0.553 \]
\[ \frac{t_c}{t_i} = 1.39 \quad 0.90 \]

\[ S_C F = 1.8 \quad \text{(Ref 8, Fig 3.6)} \]

\[ f_{max} = 1.8 \times 1250 \times 2.160 = 27,000 \text{ psi - OPERATING} \]
\[ f_{max} = 27,000 \times 0.090 = 0.30,500 \text{ psi - BURST} \]

\[ F_{com} = 35,000 \text{ psi} \quad (1200 \text{ F} \times 20,000, \text{wire and 2\%}) \]
\[ F_{tu} = 140,000 \text{ psi} \quad (1200 \text{ F}) \]

\[ M_{J} = \frac{35,000}{27,000} - 1 = \frac{0.30}{1} \]
9.4 Bearing Journal Deflection

Vibratory Load = 87.05 (Ref. page 7-11)

Load, \( P = \frac{1}{2} \) displacement \( \times \) load factor

\( = 0.47 \times 87 = 40.9 \text{ lb} \)

Assume load acts at \( \frac{1}{3} L \) from free end.

\[ \delta = \frac{PL^3}{3EI} \]

\[ L = 1.0 \text{ in.} \]

\[ I = \frac{\pi}{4} (0.25^4 - 0.19^4) = 0.466 \times 10^{-3} \text{ in.}^4 \]

\[ \delta = \frac{40.9 \times 1.0^3}{3 \times 30 \times 10^6 \times 0.466 \times 10^{-3}} \]

\[ = 0.000070 \text{ in.} \]

Since \( \delta < 0.0002 \) clearance, this is OK.
SECTION 10
COLD-END DISPLACER
STRESS ANALYSIS
10. COLD END DISPLACER

10.1 TASKS
10.2 BUCKLING ANALYSIS
10.3 SHELL ANALYSIS
10.4 DISPLACEMENT AT END
10.1 TASKS

THE ANALYSIS OF THE COLD END DISPLACER CONSISTS OF THE FOLLOWING TASKS:

- DETERMINE EXTERNAL COLLAPSING PRESSURE
- DETERMINE STRESSES AT DOME/ CYLINDER JUNCTION
- DETERMINE DISPLACER DEFORMATION AT THE END DUE TO RANDOM VIBRATION LOADING.
10.2 Buckling Analysis

The method explained on page 9-3 will be used.

\[ D_0 = 0.3925 \text{ (min.)} \quad D_f = 0.3720 \text{ (max.)} \]
\[ t = 0.010 \text{ (min.)} \quad R_M = 0.1911 \]

\[ b = \frac{0.024 - 0.010}{12} \text{ (min.)} \]
\[ d = \frac{0.040}{12} \text{ (min.)} \]

\[ L_f = \text{Ring Span} = 0.49 + 0.024 = 0.514 \text{ in.} \]
\[ L = 5.034 \text{ in.} \]
\[ A_f = 0.024 \times 0.040 = 0.96 \times 10^{-3} \text{ in}^2 \]
\[ I_f = \frac{0.024 \times 0.040^3}{12} = 1.28 \times 10^{-6} \text{ in}^4 \]
\[ e = \frac{t}{2} \left(0.010 + 0.024\right) = 0.025 \text{ in.} \]
\[ \bar{z} = \frac{0.025 \times 0.96 \times 10^{-3}}{0.96 \times 10^{-3} + 0.025 \times 0.010} = 0.00174 \text{ in.} \]
\[ F = 12 \times (1 - 0.30^2) \left[ \frac{1.28 \times 10^{-6} + 0.96 \times 10^{-3} (0.025 - 0.00174^2)}{0.514 \times 0.010^3} \right]^2 \]
\[ + \left(1 + \frac{12 \times 0.00174^2}{0.010^2} \right) - 1 \]
\[ = 14.1 \]
10.2 Buckling Analysis

\[ \gamma = \left(5.034 \right) \left(5.034 \right) \left(1 - 0.05^{2} \right)^{0.5} = 12,600. \]

\[ c_p = 800. \text{ From Fig. 7, p. 128, Ref. 5} \]

\[ p = \frac{800 \times 29.8 \times 10^6 \times 0.103 \times \pi}{12 \left(1 - 0.05^2 \right) \times 1911 \times 5.034 \times 0.5} = 4,450 \text{ psi} \]

\[ M_5 = \frac{4450}{2060} - 1 = \frac{2.56}{1.14} \text{ open} \]

\[ = \frac{4450}{2060} - 1 = \frac{1.14}{1.14} \text{ fail} \]

\[ = \frac{4450}{2060} - 1 = \frac{0.57}{2809} \text{ burst} \]
10.3 SHELL ANALYSIS

DOME/ CYLINDER JUNCTION

The method of Ref. 17 (pg 14.7.2-10) will be used to estimate the stress at the junctions.

\[
\sigma^a = \frac{K E}{2} \frac{R}{t}
\]

\[
F = 3.93 \quad R = 1.96 \text{ in.}
\]

\[
D = \frac{A_t}{A_t} = 39.3 \quad t/\delta = 5.8
\]

\[
\sigma_c = \frac{K E}{2} \frac{R}{t}
\]

\[
K = 2.5 \quad \text{(from Ref. 17, Fig. 14.7.2-13)}
\]

\[
\sigma_c = 2.5 \times 2800 \times 1.96 = 137,000 \text{ PSI}
\]

\[
f_c = \frac{7 R}{2} = 27,500 \text{ PSI}
\]

\[
f_c + \sigma_c = 164,000 \text{ PSI}
\]

At -343°F (65°K), Fr. 718 is 240,000 PSI.

\[
M.S. = \frac{240,000}{164} = 1.46
\]

At room temp., Fr. = 180,000 PSI

\[
M.S. = \frac{180}{164} = 0.11
\]
10.4 DEFLECTION AT DISPLACEMENT END

\[ D_0 = 0.393 \quad L = 0.010 \]

\[ l = 5.875 \quad \text{ (from end to BRG) } \]

\[ \omega = \frac{\pi}{2} \left( 0.393 - 0.373^2 \right) \times 2.97 = 0.00357 \text{ rad/in} \]

\[ I = \frac{\pi}{4} \left( 0.393^4 - 0.373^4 \right) = 2.71 \times 10^{-4} \text{ in.}^4 \]

\[ \gamma = \frac{\omega I L}{EI} = \frac{0.00357 \times 5.875^4}{8 \times 29.8 \times 10^6 \times 2.71 \times 10^{-4}} \]

\[ = 80.9 \times 10^{-6} \text{ in. at } 1 \text{ g}. \]

\[ f = \frac{a_0}{2\pi} \left( \frac{E I}{a_0 L} \right)^{1/2} \]

\[ = 3.52 \left( \frac{29.8 \times 10^6 \times 2.71 \times 10^{-4} \times 386}{0.00357 \times 5.875^4} \right)^{1/2} \]

\[ = 433, \text{ cfs} \]

Random Vibration Load at 433 cfs is 53.5 g's (3σ response).

(See next page)

Deflection due to Random Vibration Load

\[ \gamma = 0.00432 \text{ in.} \]
10.4 DEFLECTION AT DISPLACEMENT END

VIBRATION LOAD - RANDOM VIBRATION SPECTRUM

\[ X = \left( \frac{\pi^2}{2} \times PSD \times T \times f_n \right)^{\frac{1}{2}} \]

\[ f_n = 433.1 \text{ c/s} \]

\[ PSD = 0.045 \text{ g}^2/\text{c/s} \]

Assuming \( T = 10.0 \)

\[ X = \left( \frac{\pi^2}{2} \times 10.05 \times 10.1 \times 433.1 \right)^{\frac{1}{2}} \]

\[ = 17.5 \text{ g/s} \]

\[ \ddot{X} = 52.5 \text{ g/s} \]
APPENDIX A
WEIGHT AND C.G. DATA
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**DATE**

Rev: 6-15-73

**CUSTOMER**

NASA

**APPLICATION**

Spacecraft

**TOTAL WEIGHT - DRY**

LBS.

**TOTAL WEIGHT - WET**

LBS.

**PROJ. ENGR.**

Nickskiiewicz

**TITLE**

Valve Assembly Regenerative Engine 48 Watt

**NEXT ASST.**

Page 1 of 10

**CHANGES**

DWG. NO. 56-7570-1

**DATE**

6-23-73

**CUSTOMER**

NASA

**APPLICATION**

Space

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Page 1 of 10

**CHANGES**

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**SUMMARY**

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**CERTIFIED**

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**CUSTOMER**

**APPLICATION**

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**TOTAL WEIGHT - WET**

**PROJ. ENGR.**

**TITLE**

**NEXT ASSY.**

**PAGE**

**ChANGES**

**Dwg. No.**

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**TOTAL WEIGHT-WET**

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**APPLICATION:** Space  
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**TITLE:** Von Linmer Refrigeration Engine 1/4 Watt  
**GREAT.** Nick Loewen  
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**WEIGHT CALCULATION SHEET**

**DATE** 5-23-77  **CUSTOMER** NASA  **APPLICATION** Space  **TOTAL WEIGHT - DRY** LBS.

**PROJ. ENGR.**  **TITLE** Vulkemier Refrigeration Engine 1/4 Watt

**GALC. BY** Nicklausen  **NEXT ASY.**  **PAGE** 6 of 10  **CHANGES**

**DVG. NO.** 8-5/26-2
## WEIGHT CALCULATION SHEET

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**Total Weight - Wet:** 25.587 lbs

**Customer:** NASA

**Application:** Volumentier Refrigeration Engine 1/4 Watt

**Date:** 5 - 23 - 73

**Project Engr.:**

**Title:** Volumentier Refrigeration Engine 1/4 Watt

**Calculations by:** Nick Loewen

**Page 9 of 10**
APPENDIX B
REFERENCES
APPENDIX B

REFERENCES


5. "Instability of Thin Elastic Shells" by Y. C. Fung and E. E. Sechler, California Institute of Technology, Pasadena, California.


