DEVELOPMENT OF AN
SPS/DPS HYDROGEN SHROUDED
CRYOGENIC HELIUM STORAGE
SYSTEM

CONTRACT: NAS9-7337

FINAL REPORT

PREPARED FOR:
NATIONAL AERONAUTICS AND
SPACE ADMINISTRATION -
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HOUSTON, TEXAS 77058

PRESENTED BY:
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FOREWORD

The following report has been prepared by the Bendix Corporation's Instruments & Life Support Division, Davenport, Iowa, under contract No. NAS9-7337. The work described herein was conducted by The Bendix Corporation under the sponsorship of the Propulsion and Power Division of the NASA Manned Spacecraft Center, Houston, Texas. Mr. Donald G. Stafford, EP2, Propulsion and Power Division of the NASA Manned Spacecraft Center was the technical monitor.

Mr. William E. Bald was the project leader for the Bendix Corporation under the administrative supervision of Mr. Paul J. Gardner, Chief Cryogenics Engineer of The Bendix Corporation, Instruments & Life Support Division. Technical consultation during the program was provided by Dr. Blase J. Sollami and Mr. H. R. Lundeen of the Bendix Cryogenics R & D Group. The report summarizes work begun July, 1967, and completed September, 1968.
ABSTRACT

The feasibility of using the concept whereby a primary cryogenic fluid is surrounded by a secondary refrigerant fluid was demonstrated by an earlier NASA contract No. NAS9-4634.

The current program describes the development of an SPS/DPS Hydrogen Shrouded Cryogenic Helium Storage System which is sized to the approximate Lunar Module descent propulsion system requirements.

Extensive testing performed on the completed system demonstrated its ability to meet the anticipated LM mission profile requirements. Methods for considerably extending the standby capabilities of a shrouded dewar are discussed.
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SECTION I

INTRODUCTION

Cryogenically stored supercritical helium gas is presently being used as the pressurant for on-board propellant tanks on the Apollo Lunar module. The present system requires that the helium be transferred into the dewar in a sub-critical liquid state which involves sophisticated ground stand-by equipment to fill the unit. This transfer approach is also sensitive to atmospheric conditions which can affect the final helium state and the amount of usable pressurant stored.

An alternative approach to the above method is to store the helium gas at a higher temperature and pressure using a secondary cryogenic fluid. This concept can be achieved by adopting the Bendix developed integral shroud system whereby the secondary fluid is used as a refrigerant surrounding the primary fluid contained within the inner vessel.

The effectiveness of the above concept for both pre-launch and in-flight standby storage of various cryogens was clearly demonstrated by the results of NASA contract number NAS9-4634 carried out by The Bendix Corporation, Instruments & Life Support Division. Referring in particular to the primary storage of helium enshrouded by various sub-critical refrigerants; figures 10, 13, 16 and 19 of this investigation illustrate the advantages of using hydrogen as the secondary fluid. These figures are reproduced in this report as figure 1, 2, 3 and 4 respectively.

Figures 1 and 2 show that the hydrogen-shrouded-helium system provides the best weight optimized storage system for both pre-launch and in-flight standby with optimum helium storage pressure occurring between 1500 and 3000 PSIA.

A dewar design which incorporates two discrete radiation shields with the inner shield vapor cooled is shown, by figures 3 and 4, to have minimum system weight potential for extended pre-launch and in-flight standby capabilities.

The tankage system described in Section II of this report is an upgrading of the NAS9-4634 tankage to meet the approximate Apollo LM system requirements. Based on the results of figure 1 to 4 the upgraded system is a hydrogen-shrouded-helium storage tank operating at 2000 PSIA.
FIGURE 1.
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF HELIUM BY VARIOUS SHROUD LIQUID REFRIGERANTS

HELIUM STORAGE PRESSURES:
- 3000 psia
- 1500 psia
- 500 psia

SHROUD STORAGE PRESSURE: 14.7 psia

Basis and Explanation in Text

SHROUD SIZED FOR 100 hrs.
FIGURE 2.
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF HELIUM BY VARIOUS SHROUD LIQUID REFRIGERANT

HELUM STORAGE PRESSURES: — 3000 psia
— — 1500 psia
— — — 500 psia

SHROUD STORAGE PRESSURE: 14.7 psia
Basis and Explanation in Test

WET-SHROUD SYSTEM WEIGHT
HELUM WEIGHT

STANDBY TIME, HRS.
FIGURE 3.
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF HELIUM BY LIQUID HYDROGEN
WITH
VARIOUS INSULATION METHODS
HELIUM STORAGE PRESSURE: 1500 PSIA
HYDROGEN STORAGE PRESSURE: 14.7 PSIA
Basis and Explanation in Text

1/2" SI
VACUUM
1" SI
1 SHIELD
2 SHIELDS
1 SHIELD VAPOR COOLED
2 SHIELDS
1 VAPOR COOLED

DRI-SHROUD SYSTEM WEIGHT
HELIUM WEIGHT

VACUUM WITH 50-hr SIZED SHROUD

STANDBY TIME, HRS.
FIGURE 4.
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME FOR IN-FLIGHT COOLING OF HELIUM BY LIQUID HYDROGEN WITH VARIOUS INSULATION METHODS
HELIUM STORAGE PRESSURE: 1500 PSIA HYDROGEN STORAGE PRESSURE: 14.7 PSIA
Basis and Explanation in Text
The basic design employs the Bendix developed discrete radiation shielded concept using two shields with the inner shield being vapor cooled by the vented hydrogen.

Remote monitoring and control of the unit is available by using the Control Panel supplied with the system.

A limited amount of design analysis was performed during the contract period and is included in Section III.

Extensive thermal and dynamic testing was carried out on the completed tankage assembly to verify the basic design principles. This testing together with the graphical results is fully discussed in the text.

Included as appendices to the main text are an updated log of the completed unit and a discussion concerned with the effect of shroud annulus geometry on the heat interception characteristics and standby capabilities of future shrouded dewars.
SECTION II
TANKAGE DESCRIPTION AND FABRICATION

The helium shroud tankage system designed and developed under this contract is shown in Figure 5.

In accordance with the RFQ, Exhibit "A", Statement of Work, under which this contract was awarded, the tankage design utilized a method of construction similar to that used for contract NAS9-4634(1) i.e., the Integral Shroud System Concept.

The dewar was designed to store 50.0 pounds of helium at 2000 psia and 37°F by enshrouding the helium gas with liquid hydrogen at standard atmospheric conditions.

2.1 Dewar Description

A cross sectional view through the dewar is shown in Figure 6 and the physical characteristics of the component parts are presented in Table 1.

Mounted within the inner pressure vessel of Figure 6 is an aluminum support tube around which are wrapped the two inner temperature sensors.

The integral shroud is welded to the inner vessel at the poles to form an annulus around the pressure vessel. Contact between the shroud and inner vessel is also maintained at the six bumper locations where the shroud is dimpled to provide support for the inner vessel/shroud assembly.

Wrapped around the inner vessel outer surface and contained within the shroud annulus is 1/8" thick Teflon felt which is aimed at providing a 'wick action' for the liquid hydrogen in the shroud.

Figure 6 shows that the shroud annulus tapers from the poles to the equator where the shroud gerth weld is located. The reason for this is two fold.

Firstly, tooling costs for manufacturing the shroud hemispheres could be greatly reduced by using existing tooling from the NAS9-4634 contract.
Life Support Division

1. INNER PRESSURE VESSEL
2. TEMPERATURE SENSOR SUPPORT TUBE
3. TEMPERATURE SENSOR LEADS
4. TEFLOM FELT
5. SHROUD
6. INNER VAPOR COOLED RADIATION SHIELD
7. OUTER RADIATION SHIELD
8. OUTER SHELL
9. SHROUD VENT FITTING
10. INNER VENT TUBE
11. ION PUMP
12. ELECTRICAL LEAD FITTING
13. FLUID OUTLET VENT FITTINGS
14. SHIELD HANGER ASSEMBLY
15. SHIELD TO SHIELD SPACER
16. VAPOR COOL TUBING
17. RUPTURE DISC
18. INNER FILL & SUPPLY TUBE
19. SHROUD FILL FITTING
20. RADIAL BUMPERS
21. VAPOR COOL OUTLET FITTING
22. FLUID INLET FILL FITTINGS

FIGURE 6
HELIUM SHROUD TANK DEWAR ASSEMBLY
Secondly, adoption of this existing tooling size to meet the specified shroud volume of approximately (1) cubic foot suggested the tapered shroud concept. In addition, it was found that the tapered shroud eliminated the need for dimpling the shroud at the poles where it is welded to the inner vessel fill and vent fittings. Elimination of these dimples also produces an overall reduction in tankage size of approximately 5% because the shields and outer shell diameters are governed by clearances at the poles. It should be pointed out, however, that the absence of dimples at the shroud poles created certain fabrication problems which will be discussed in Section 2.2.3.

The adoption of a tapered shroud annulus for the present tankage design as opposed to the concentric annulus used in the NAS9-4634 unit introduces interesting new design concepts for future mission standby requirements. These will be discussed later in this report.

Welded into the top of the shroud near its attachment to the inner vent fitting is the shroud vent which has two internal passageways. One port connects to the shroud vent tube and the other to the vapor cooled shield system. Also shown in Figure 6 at the bottom of the shroud is the shroud fill fitting.

Mounted on the inner and shroud fill and vent tubes are (6) Kel-F bumpers, (3) in the plane at 45° above the equator and (3) in the corresponding plane below the equator. These radial bumpers provide an efficient method for isothermally mounting the inner vessel/shroud assembly within the tankage outer shell. The bumpers also provide support for the fill and vent tubes.

The electrical leads from the inner vessel temperature sensors, and from the temperature sensor connected into the shroud vent fitting, are attached to the inner vent tube and pass through the bumpers mounted to this tube.

Within the vacuum annulus between the shroud and outer shell of Figure 6, are mounted (2) discrete radiation shields, the inner shield having a copper vapor cooling coil soldered to its outer surface. The inner shield is supported by thermally insulated hangers from the fill and vent tubes and the outer shield is separated from the inner by Kel-F spacer studs.

The aluminum outer shell of the dewar assembly of Figure 6 provides exits for the inner and shroud fill and vent tubes, electrical temperature sensor leads and for the vapor cooling outlet.
Mounted on the outer shell is a 0.2 liters/second ion pump which preserves the vacuum pressure within the annulus and, at the same time, provides a simple method for measuring this pressure.

A rupture disc is welded into the lower half of the outer shell to prevent excessive pressure buildup within the vacuum annulus caused by small cryogen leakages.
TABLE 1
DEWAR ASSEMBLY PHYSICAL CHARACTERISTICS

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<tr>
<th>Component</th>
<th>Material</th>
<th>Inner Diameter</th>
<th>Wall Thickness</th>
<th>Usable Volume</th>
<th>Maximum Operating Pressure</th>
<th>Proof Pressure</th>
<th>Burst Pressure</th>
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<tr>
<td>Inner Vessel</td>
<td>Inconel 718 (Age Hardened)</td>
<td>25.070 in.</td>
<td>0.190 in.</td>
<td>4.77 cu. ft.</td>
<td>2,575 psia</td>
<td>3,860 psia</td>
<td>5,600 psig</td>
</tr>
<tr>
<td>Shroud</td>
<td>Inconel 718 (Annealed)</td>
<td>28.215 in.</td>
<td>0.035 in.</td>
<td>0.95 cu. ft.</td>
<td>14.7 psia</td>
<td>40.0 psig</td>
<td></td>
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<tr>
<td>Inner Radiation Shield</td>
<td>Aluminum 6061-0</td>
<td>28.450 in.</td>
<td>0.020 in.</td>
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<td></td>
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<tr>
<td>Outer Radiation Shield</td>
<td>Aluminum 6061-0</td>
<td>29.190 in.</td>
<td>0.020 in.</td>
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<td></td>
<td></td>
<td></td>
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<tr>
<td>Fill and Vent Tubes</td>
<td>Stainless Steel MIL-T-8808</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vapor Cool Tubing</td>
<td>Copper (Annealed)</td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>Outer Shell</td>
<td>Aluminum 6061-0</td>
<td>29.740 in.</td>
<td>0.055 in.</td>
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**TABLE I - Continued**

DEWAR ASSEMBLY PHYSICAL CHARACTERISTICS

<table>
<thead>
<tr>
<th>Item</th>
<th>Details</th>
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| Radial Bumpers                      | Material: Kel-F  
Manuf. Diameter: 2.055 in.  
Quantity: 6 |
| Temperature Sensors                 | Manufacturer: Rosemount Engr. Co.  
Type: Platinum Resistance  
Model Number: 137 DE |
| Ion Pump                            | Manufacturer: Varian Assoc.  
Type: 0.2L/S Vacion  
Model Number: 913-0005 |
| Rupture Disc                        | Manufacturer: Black, Sivalls & Bryson  
Type: 77-BEN-025  
Design Burst Pressure: 90 PSID |
| Fluid Transition Fittings           | Manufacturer: Bi-braze Corp.  
Type: Aluminum/St. St. |
| Electrical Leads Transition Tube    | Manufacturer: Whittaker Corp.  
Type: Aluminum/St. St. |
| Dry Dewar Assembly Weight           | Weight: 205 lbs. |
2.2 Dewar Fabrication

2.2.1 Inner Vessel Fabrication

Initial fabrication of the dewar was delayed at the start of this contract because of the difficulty in finding a suitable vendor for fabricating the pressure vessel hemispheres. Various spinning houses and hydro-forming facilities were contacted without success. The major difficulties in forming Inconel 718 in the thickness required for these vessels was the limitations in tooling power available, and the large degree of work-hardening which takes place during the forming process.

After a nation-wide search, Greer Products Inc., Los Angeles, agreed to fabricate (4) hemispheres using their proprietary forming process. The delivered hemispheres were fully annealed at 1850°F for 15 minutes and vidi-gage readings indicated a thickness variation of approximately 0.190 in. at the pole to 0.230 in. at the equator.

The finished machined pressure vessel hemispheres together with the Inconel 718 boss fittings are shown at left prior to welding.

The cross-sectional profile of the boss fittings was fed into the NASA-MSC computer program, number G084, to ascertain the hoop and meridional stress distributions in the pressure vessel regions where changes in cross-section occur. Output from the program indicated a minimum safety factor of 1.65 on yield based on a material yield point of 145,000 lbs./sq. in. and a maximum design pressure of 2580 lbs./sq. in. These values compare favorably with the specified proof safety factor of 1.5 and burst safety factor of 2.0 based on the same design pressure.

Welding of the boss fittings into the hemispheres and the girth welding of the two hemispheres at the equator was carried out at ARDE Inc., New Jersey. Two inner vessels were fabricated, one for the actual dewar and a spare vessel for the burst testing described in Section IV.

The four boss fittings were successfully heli-arc welded into their respective hemispheres and the welds dye-penetrant checked and X-rayed. Success was also achieved with the first girth weld.

During the welding of the second vessel at the girth, however, partial loss of Argon gas purge caused
oxidation of the weld which revealed the weld pattern shown at left when the vessel was cut in half. The two hemispheres were re-machined at the girth to clean-off the weld deposit and the vessel successfully re-welded. This cutting open and re-welding process meant that the second spare pressure vessel was 1/2 inch shorter between the poles than the dewar inner vessel. It was felt that this small ellipsoidal effect would only marginally affect the burst pressure test.

After welding, the two pressure vessels were age-hardened at 1360°F for 10 hours followed by 1175°F for a further 10 hours and then cooled in air. A further hydrochloric/Nitric acid pickling process produced the finished vessels shown below.

The first successfully welded pressure vessel was chosen for the dewar inner vessel and was subsequently proof tested hydraulically to 3860 psig with complete success. Pressurization of the second pressure vessel is described in Section IV.

After proof testing the inner vessel, the closure fittings together with the temperature sensors and support tube were assembled into the vessel and welded. A typical closure fitting prior to welding is shown at left with the short length of stainless steel tubing exiting from the fitting. This tubing was eventually cut to length and sleeve jointed to the inner fill and vent tubes.
2.2.2 Teflon Felt Assembly

Before attachment of the 1/8 inch thick Teflon felt to the outer surface of the inner vessel, a series of tensile test specimens were cut and trial stitched with different diameters andhardnesses of stainless steel wire. These tests resulted in the use of 20 S.W.G. stainless steel wire because of its simplicity in stitching and because, in all cases tested, the stitching was stronger than the parent material. Tensile tests were performed both at room temperature and after immersing the specimens in LN2 for about 10 minutes. Temperature effects were negligible.

A typical tensile test and a typical specimen failure are shown at left.

After completion of the above testing, the Teflon felt was stitched together in segments around the inner vessel in the manner shown at left. The holes in the felt coincide with the shroud dimples and bumper location points.

2.2.3 Shroud Fabrication

The Inconel 718 shroud hemispheres were inturgescently formed at the Bendix facility in Santa Ana, California. Full annealing of the hemispheres was carried out after forming and chem milling. As mentioned before, a tapered shroud annulus was introduced into the dewar design which was achieved by trimming off the shroud hemispheres 1 inch above the true hemisphere equator. A finished machined hemisphere is shown below.
After hemisphere machining, the shroud fill and vent tube assemblies were welded into the ports in their respective hemispheres and each sub-assembly evacuated and leak checked.

A problem was encountered during the assembly of the shroud hemispheres to the inner vessel. Errors in the inner vessel pole to pole dimensions, and in the pole to girth height for each shroud hemisphere, accumulated to cause a poor shroud/inner vessel fit at the closure fittings. This problem was solved by introducing small dimples at the shroud hemisphere poles to provide flexibility and adjustment in the pole dimension during fit-up.

Shown at left are photographs of the successive assemblies of the shroud to the inner vessel. The first picture shows the upper half shroud in place, the second shows the completed shroud assembly with the fill and vent tubes in situ. The finished inner/shroud assembly, after silver plating, is shown in the third picture at left. Silver plating of the shroud outer surface was vacuum vapor deposition on an epoxy substrate.

2.2.4 Radiation Shield Fabrication

Inturgescent forming at the Bendix facility was again used to fabricate all the aluminum shield hemispheres.

The inner vapor cooled shield hemispheres were copper plated prior to soldering the 1/4 inch copper tubing to their outer surfaces.

After a final fit-up of the inner and outer shields to the inner vessel/shroud assembly, the shield hemispheres were plated on both their inner and outer surfaces using silver on an epoxy substrate.

Shown on the following page is the vapor cooled radiation shield before and after plating and after its final assembly around the shroud.

The final assembly of the outer radiation shield around the vapor cooled shield is also shown on the following page.
2.2.5 Outer Shell Fabrication

The aluminum outer shell hemispheres were inturgescently formed at the Bendix Facility in Santa Ana, California. After forming, the shells were fully annealed and chemically milled to meet the required thickness tolerances.

The finished lower and upper half outer shell hemispheres are, respectively, shown in the column at left. A rupture disc is shown welded into the lower half shell and the three holes where the vapor cool outlet, inner and shroud fill ports weld into the outer shell are also visible.
Shown in the picture of the upper half shell are the ion pump and tip-off tube assembly. The three holes in this case are where the inner and shroud vent ports, and electrical lead wires exit through the outer shell.

An additional outer shell hemisphere was fabricated for the purpose of performing a buckling test to failure on a typical shell hemisphere. This test is fully discussed in Section IV.

After fabrication of the outer shell hemispheres, the inside surfaces were copper plated on an epoxy substrate before assembly around the outer radiation shield. The assembled outer shell before welding is shown below.

At each and everyone of the aforementioned stages of the dewar assembly, systematic helium leak checking of the various welded joints was carried out using the vacuum stand shown below.

Because the outer shell was fabricated in aluminum, and the fill and vent tubing and the electrical lead wire sheaths were necessarily made from 304L stainless steel, some form of bi-metal transition joint was required where these tubes and leads pass through the outer shell.
This problem was solved by using the special aluminum/stainless steel transition fittings shown at the left, each fitting being helium leak checked before assembly into the dewar.

2.2.6 Evacuation and Bakeout

After welding of the outer shell gerth, and the fluid and electrical connection points, the dewar assembly was evacuated and baked out in a temperature controlled oven for a total period of 21 days. During this bakeout period, the temperature at locations within the oven and the vacuum pressure inside the dewar vacuum annulus were continuously monitored. Systematic degassing of the diffusion pump system was carried out to extinguish any gas molecules which might congregate due to temperature out-gassing of the dewar assembly.

A stabilized bakeout temperature of 275°F was used which maintained the dewar plated components at a temperature below their epoxy substrate curing temperature of 300°F. At this bakeout temperature, a minimum vacuum annulus pressure of $2.5 \times 10^{-7}$ torr was achieved, this pressure falling to $1.7 \times 10^{-7}$ torr when the dewar cooled down to room temperature of 73°F.

A further reduction in vacuum annulus pressure occurred when the shroud was filled with LN2, the cryo-pumping effect producing a final vacuum pressure of $2.5 \times 10^{-8}$ torr as measured by the ion pump ammeter reading.

2.3 System Description

The complete shroud tankage system is represented schematically in Figure 7 and the pertinent characteristics of the external components are listed in Table II. The electrical wiring circuit for monitoring temperatures and pressures and for controlling the solenoid-operated valves is shown in Figure 8.

The dewar assembly described in Section 2.2 is mounted in a cradle-type mount carriage which is designed to support inertia or shock loadings in any given direction. Figure 9 shows the mount carriage assembly and the locations of the (6) shock mounts which are the attachment points between the helium tankage assembly and the vehicle or propellant cascade system. An analysis of the cradle-type mount structure subjected to inertia loading along the vertical axis is presented in Section III. This was carried out to check the peak G loading effects on the mount structure during the vibration testing phase of this contract.
FIGURE 7.
HELIUM SHROUD TANKAGE SYSTEM SCHEMATIC
BARRY SHOCK MOUNTS
TYPE NC-2040-T6
3 PAIRS SPACED
120° APART

34.75 DIA.

29.740 DIA.

FIGURE 9
MOUNT CARRIAGE ASSEMBLY

Top
Referring to the system of Figure 7, manual Teflon-seated ball valves are provided for the inner and shroud fill lines and also for the shroud direct or vapor cooled vent lines. Double-latching, remotely operated solenoid valves are connected to the inner high pressure helium system for venting the inner pressure or for supplying high pressure GHe to the propellant cascade system. The double latching mechanism incorporated in these valves allows the valve to remain 'normally open' or 'normally closed' for an indefinite period without using D.C. power to energize the solenoids.

A 20 micron filter has been incorporated between the inner fill valve and pressure vessel to prevent the ingress of dirt particles into the system causing valve malfunctioning. The necessity for this precaution is discussed in Section IV.

Bendix designed relief valves are connected into both the inner and shroud vent lines to prevent system pressure rising above the maximum operating levels.

Also connected into the inner and shroud vent lines are pressure transducers which enable the system pressures to be remotely monitored.

Remote control of the Helium Tankage system and remote monitoring of the system temperatures and pressures is afforded by the control panel shown in Figure 10 and represented schematically by Figure 8.

The electrical system operates from a 28VDC supply, the full scale output from the temperature sensors and pressure transducers being 5VDC as indicated by the voltmeter on the control panel.

The two double-latching solenoid valves operate directly from the 28VDC supply, the red and green lights connected in parallel with their respective solenoids indicating the valve position. A GREEN light indicates that the valve is OPEN and a RED light indicates the CLOSED position. Opening and closing of the valves is actuated by the ON-OFF-ON switches mounted on the control panel of Figure 10.

A six-position rotary selector switch is provided on the control panel for switching to the PRT circuit, the inner or shroud pressure sensor circuits, and the meter voltage supply.

An additional six-position rotary switch enables the
four individual temperature sensor circuits to be selected.

Calibration of the individual temperature sensors and pressure transducers in relation to the control panel voltmeter reading will be discussed in Section IV under System Testing.

2.4 System Operation

Operation of the tankage system shown in figure 7 is straightforward and self-explanatory after the unit has been filled with LH2 and GHe. A detailed description of the procedure to be followed during system fill is included as part of the testing program of Section IV.

After the unit is filled it will be assumed that all the manual and solenoid operated valves are in the CLOSED position. In this condition environmental heat leak into the cryogen will gradually build-up pressure in the shroud until the shroud relief valve opens and hydrogen venting takes place through the direct vent line. Opening the shroud manual vent valve will relieve the shroud pressure and allow direct shroud venting to take place at atmospheric pressure conditions.

To provide longer system standby requirements shroud venting through the vapor cooled shield is available. Venting through the vapor cooled circuit is achieved by closing the shroud manual vent valve and opening the vapor cool valve.

Supply of high pressure GHe from the inner vessel is obtained by selecting the solenoid operated supply valve to the OPEN position. During long in-flight standby periods the inner vessel pressure increases due to the environmental heat transfer and is eventually relieved by the inner P.R.V. Emergency venting or pressure relief of the inner vessel pressure is available by opening the inner solenoid operated vent valve.
# Physical Characteristics of the External Components

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturer</th>
<th>Model Number/Type Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manual Fill &amp; Vent Valves</td>
<td>Jamesbury</td>
<td>St. St. Ball Valve 1/4&quot; HPO - 36MT</td>
</tr>
<tr>
<td>Solenoid Operated Valves</td>
<td>Retek Corp.</td>
<td>2-way Double Latching</td>
</tr>
<tr>
<td>Operating Pressure Range</td>
<td>0 - 2,500 psig</td>
<td>-420°F to +250°F</td>
</tr>
<tr>
<td>Operating Temperature Range</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inner Pressure Relief Valve</td>
<td>Bendix</td>
<td>1620420-1 2550 psia</td>
</tr>
<tr>
<td>Cracking Pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shroud Pressure Relief Valve</td>
<td>Bendix</td>
<td>FR121A 40 psig</td>
</tr>
<tr>
<td>Cracking Pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inner Pressure Sensor</td>
<td>Statham Instruments Inc.</td>
<td>PA419TC-5M 0 - 5000 psia</td>
</tr>
<tr>
<td>Type</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating Range</td>
<td>Statham Instruments Inc.</td>
<td>PA419TC-100 0 - 100 psia</td>
</tr>
<tr>
<td>Shroud Pressure Sensor</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>James, Pond &amp; Clark, Inc.</td>
<td>4342F - 20CU 3000 psig 20 Micron</td>
</tr>
<tr>
<td>Inner Vessel Filter</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Max. Operating Pressure</td>
<td>Barry Controls</td>
<td>NC-2040-T6 30 to 50 lbs.</td>
</tr>
<tr>
<td>Filtration</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mount Carriage</td>
<td>Magnesium Alloy</td>
<td>10.2 lbs.</td>
</tr>
<tr>
<td>Material</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Weight</td>
<td></td>
<td></td>
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<tr>
<td>Shock Mounts</td>
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<tr>
<td>Manufacturer</td>
<td>Barry Controls</td>
<td>NC-2040-T6 30 to 50 lbs.</td>
</tr>
<tr>
<td>-------------------------------</td>
<td>---------------------</td>
<td></td>
</tr>
<tr>
<td>Manufacturer</td>
<td>Platinum Resistance</td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>137 EC</td>
<td></td>
</tr>
<tr>
<td>Model</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| Dry Dewar Assembly Weight     | 205 lbs.            |
| (From Table I)                |                     |
| Weight of External Components | 45 lbs.             |
| Total Dry System Weight       | 250 lbs.            |
SECTION III

SYSTEM ANALYSIS

The analyses included in this section were performed during the period of this contract. Some of the topics discussed are incomplete because of the limited time available but in these instances directives for future extensions to these analyses are included.

3.1 Mount Carriage Analysis

At the initiation of the present contract levels of anticipated acceleration, shock, and vibration loadings were not specified for on-board helium pressurization tankage systems of this type. An analysis of the cradle-type mount carriage shown in figure 9 was, however, initiated in anticipation of future static and dynamic load requirements for a structure of this type.

The following analysis considers the particular case of an instantaneous inertia loading acting along the polar axis of the tank or X axis shown in figure 9. The analysis will also consider the general case of a mount carriage having any arbitrary number of support arms, it being assumed that the number of arms are equal to, and coincide with, the number of radial bumpers within the Dewar.

Considering each half of the cradle-type mount carriage, any individual arm can be represented by the equivalent spring system shown in figure 11.

Let $F =$ maximum inertia force acting in direction of vehicle launch.

$n =$ number of radial bumpers in each bumper plane.

$\alpha =$ bumper plane angle above and below horizontal plane.

$K_s =$ outer shell stiffness.

$K_R =$ support ring in-plane stiffness.

$q_R =$ angular stiffness of support ring.

$M_e =$ external couple produced by eccentricity of radial load relative to support ring center of twist.
\[ W_a = \text{horizontal load acting on each arm.} \]
\[ W_R = \text{horizontal load acting on support ring at each bumper load point.} \]
\[ W_s = \text{horizontal load acting on outer shell at each bumper load point.} \]
\[ M_R = \text{couple acting on support ring at each bumper load point.} \]
\[ M_a = \text{couple acting on each arm.} \]
\[ \delta = \text{linear displacement.} \]
\[ \theta = \text{angular displacement.} \]

For equilibrium of forces in the y direction
\[
W_R + W_s + W_a = F/n \cot \alpha \quad (3.1)
\]

For equilibrium of moments
\[
M_a + M_R = M_e \quad (3.2)
\]

The compatibility conditions at the juncture between the arm and support ring are
\[
\delta_s = \delta_R = (\delta_a) \quad (3.3)
\]
\[
\theta_a = \theta_R
\]

Equations (3.1) to (3.3) give the five necessary conditions for finding the unknowns \( W_R, W_s, W_a, M_a \) and \( M_R \).

3.1.1 Analysis of the Cradle-type Arm

The forces and moments acting on each arm are as shown at left.

The strain energy of the system can be written
\[
U = \int_0^\alpha \frac{M^2 R d\phi}{2EI_a}
\]
i.e.
\[
U = \frac{R}{2EI_a} \left[ W_a R (\sin \alpha - \sin \phi) + F/n R (\cos \phi - \cos \alpha) - M_a^2 \right] d\phi
\]
Life support
Division

Now

\[
(\delta_a)_y = \frac{3W_a}{3W} = \frac{R}{2EI_a} \int_0^\alpha 2[W_a R(\sin \alpha - \sin \phi) + \\
F/n R(\cos \phi - \cos \alpha) - M_a] R(\sin \alpha - \sin \phi) \, d\phi
\]

\[
(\delta_a)_y = \frac{R^2}{EI_a} \int_0^\alpha [W_a R(\sin^2 \alpha + \sin^2 \phi - 2 \sin \alpha \sin \phi) + \\
F/n R(\sin \alpha \cos \phi - \sin \alpha \cos \phi - \sin \phi \cos \phi + \cos \alpha \sin \phi) - \\
M_a(\sin \alpha - \sin \phi)] \, d\phi
\]

\[
(\delta_a)_y = \frac{R^2}{EI_a} [W_a R \left\{\phi \sin^2 \alpha + \phi/2 - \frac{1}{4} \sin 2\phi + \\
2 \sin \alpha \cos \phi\right\} + \frac{F}{n} R \left\{\sin \alpha \sin \phi - \phi \sin \alpha \cos \alpha - \\
\frac{1}{2} \sin^2 \phi - \cos \alpha \cos \phi\right\} - M_a \left\{\phi \sin \alpha + \cos \phi\right\}]^\alpha_0
\]

\[
(\delta_a)_y = \frac{R^2}{EI_a} [W_a R \left\{(\phi \sin^2 \alpha + \frac{\alpha}{2} - \frac{1}{4} \sin 2\alpha + 2 \sin \alpha \cos \alpha - \right. \\
2 \sin \alpha\right\} + \frac{F}{n} \cdot R \sin^2 \alpha - \alpha \sin \alpha \cos \alpha - \frac{1}{2} \sin^2 \alpha - \cos^2 \alpha + \right. \\
\left. \cos \alpha\right\} - M_a \left\{\alpha \sin \alpha + \cos \alpha - 1\right\}]
\]

\[
(\delta_a)_y = \frac{R^2}{EI_a} [W_a R \left\{\frac{\alpha}{2} - 2 \sin \alpha + \sin^2 \alpha + \frac{3}{4} \sin 2\alpha\right. \\
\left. + \frac{F}{n} \cdot R \left\{\frac{1}{2} \sin^2 \alpha + \phi \sin 2\alpha - \frac{3}{2} \sin^2 \alpha - \cos^2 \alpha + \cos \alpha\right\} - \\
M_a \left\{\alpha \sin \alpha + \cos \alpha - 1\right\}\right\}
\]

Also

\[
(\delta_a)_x = \frac{3W}{\phi(F/n)} = \frac{R}{2EI_a} \int_0^\alpha 2[W_a R(\sin \alpha - \sin \phi) + \\
F/n \cdot R \left\{(\cos \phi - \cos \alpha) - M_a\right\} R(\cos \phi - \cos \alpha) \, d\phi
\]

32
\[
\begin{align*}
\theta_a &= \frac{\partial U}{\partial M_a} = - \frac{R}{E_1 a} \int_0^\alpha \left[ W_a R (\sin \alpha - \sin \phi) + \frac{F}{n} \cdot R (\cos \phi - \cos \alpha) - M_a \right] d\phi = - \frac{R}{E_1 a} \left[ W_a R (\phi \sin \alpha + \cos \phi) + \frac{F}{n} \cdot R (\sin \phi - \phi \cos \alpha) - M_a \right]_0^\alpha \\
\theta_a &= \frac{R}{E_1 a} \left[ W_a R (\alpha \sin \alpha + \cos \alpha - 1) + \frac{F}{n} \cdot R (\sin \alpha - \cos \alpha) - M_a \right]_0^\alpha
\end{align*}
\]
Applying the condition \( \theta_a = \theta_R = \frac{M_R}{q_R} \) to equation (3.6) hence, 

\[
M_R = \frac{R \left[ RW_a (\alpha \sin \alpha + \cos \alpha - 1) + R \cdot \frac{F}{n} (\sin \alpha - \alpha \cos \alpha) - M_e \right]}{E l / a q_R - \alpha R}
\] \hspace{1cm} (3.7)

From the condition \( \delta_s = \delta_R \) we find 

\[
W_s / K_s = W_R / K_R
\]

and using the value of \( W_R \) given by equation (3.1) 

\[
W_s = \frac{K_s}{K_R} \left( \frac{F}{n} \cot \alpha - W_s - W_a \right)
\]

i.e., 

\[
W_s = \frac{K_s / K_R (\frac{F}{n} \cot \alpha - W_a)}{1 + K_s / K_R}
\] \hspace{1cm} (3.8)

The remaining condition to be satisfied is 

\[
\delta_R = (\delta_a)_y
\]

i.e., 

\[
W_R / K_R = (\delta_a)_y
\]

or using equation (3.1) 

\[
\frac{F/n \cot \alpha - W_a - W_s}{K_R} = (\delta_a)_y
\]

and substituting the value of \( W_s \) given by equation (3.8) in the above expression 

\[
(\delta_a)_y = \frac{F/n \cot \alpha - W_a}{K_R + K_s}
\] \hspace{1cm} (3.9)
using equation (3.4) together with (3.2) in the above
equation (3.9) and solving for $W_a$

$$W_a = \frac{[\alpha_2 R^3 + \frac{\alpha_3^4 R^4}{E I a/q_R} - aR + \frac{E I a}{K_R + K_S} \frac{F}{n} M e [\alpha_3 R^2 + \frac{\alpha_3^3 R^3 a}{E I a/q_R - a_R}]}{[\alpha_1 R^3 + \frac{\alpha_2^2 R^4}{E I a/q_R} - aR + \frac{E I a}{K_R + K_S}]} \quad (3.10)$$

where:

$$\alpha_1 = \frac{\alpha}{2} - 2 \sin \alpha + \frac{3}{4} \sin 2\alpha$$

$$\alpha_2 = \frac{\alpha}{2} \sin 2\alpha + \cos^2 \alpha - \frac{1}{2} \sin^2 \alpha - \cos \alpha$$

$$\alpha_3 = \alpha \sin \alpha + \cos \alpha - 1$$

$$\alpha_4 = \alpha \cos \alpha - \sin \alpha$$

3.1.2 Equivalent Spring Stiffnesses

To solve the foregoing equations it is now necessary to
define the equivalent spring stiffnesses $K_S$, $K_R$ and $q_R$.

3.1.2.1 Outer Shell Stiffness $K_S$

Since no solutions are available for a spherical shell
subjected to a point loading it will be assumed that
the proportion of load taken by the outer shell $W_a$ is
uniformly distributed around the bumper plane circle
as shown at left.

where

$$Q = \frac{nW_s}{2\pi a \cos \alpha} \quad (3.12)$$

For shells 1 and 2 the displacement and rotation at
the edges can be obtained using the formulas derived in
Timoshanko "Theory of Plates and Shells" page 470.

i.e.,

$$\theta_1 = \frac{4\lambda^3 M}{E a} - \frac{2\lambda^2 \sin \psi}{E t} \cdot Q_1$$

$$\delta_1 = \frac{2\alpha \sin^2 \psi}{E t} \cdot Q_1 - \frac{2\lambda^2 \sin \psi}{Et} \cdot M$$

$$\theta_2 = \frac{4\lambda^3 M}{E a} - 2\lambda^2 \sin (\pi - \psi) \cdot Q_2 \quad (3.13)$$
\[
\delta_2 = \frac{2a\lambda \sin^2(\pi - \psi)}{Et} \cdot Q_2 - \frac{2\lambda^2 \sin(\pi - \psi)}{Et} \cdot M
\] (3.13)

Using the shell compatibility conditions \( \vartheta_1 = \vartheta_2 \) and \( \delta_1 = \delta_2 \) together with equations (3.13) it is found that

\[
Q_1 = Q_2 = Q/2
\]

\[
M = \frac{Qa \sin \psi}{4\lambda}
\] (3.14)

The shell deflection at the load point is then given by

\[
\delta_s = \frac{2a\lambda \sin^2 \psi}{Et} \cdot \frac{Q}{2} - \frac{2\lambda^2 \sin \psi}{Et} \cdot \frac{Qa \sin \psi}{4\lambda}
\]

\[
\delta_s = \frac{Qa \sin^2 \psi}{2Et}
\] (3.15)

Using equation (3.12) the above equation becomes

\[
\delta_s = \frac{a\lambda \sin^2(\pi/2 + \alpha)}{2Et} \cdot \frac{nW_s}{2\pi a \cos \alpha}
\]

and by definition,

\[
K_s = \frac{W_s}{\delta_s} = \frac{4\pi E \cdot t}{\delta_s \cdot n \lambda \cos \alpha}
\] (3.16)

where \( \lambda^4 = 3(1 - \nu^2)(\frac{b}{c})^2 \) (3.17)

3.1.2.2 Support Ring Stiffness \( K_R \)

From Table VIII, page 174 of "Formulas for Stress and Strain" by Roark the deflection of the ring at each bumper load point is

\[
\delta_R = \frac{W_R b^3}{2E I_R} \left[ (\frac{\pi}{2n} + \frac{1}{2} \sin \frac{\pi n}{n} \cos \frac{\pi}{n}) \csc^2 \frac{\pi}{n} - \frac{n}{\pi} \right]
\]

and by definition,

\[
K_R = \frac{W_R}{\delta_R} = \frac{2E I_R}{b^3 \left[ (\frac{\pi}{2n} + \frac{1}{2} \sin \frac{\pi n}{n} \cos \frac{\pi}{n}) \csc^2 \frac{\pi}{n} - \frac{n}{\pi} \right]}
\] (3.18)
3.1.2.3 Angular Stiffness of Support Ring $q_R$

The support ring stiffness $q_R$ can be found by determining the angle of twist of the support ring at the bumpers due to the couples $M_R$ of $n$ in number. This approach, however, would yield stiffness values lower than the true values because the restraint on support ring twisting caused by the outer shell has been neglected in this analysis.

To take some account of the shell restraint on twisting the total twisting moment $nM_R$ will be assumed to be uniformly distributed around the support ring circumference.

hence, \[
\theta_R = \frac{b^2}{E_l} \cdot \frac{nM_R}{2\pi b} \]

and by definition,

\[
q_R = \frac{\theta_R}{b} = \frac{nM_R}{2\pi E_l R} \tag{3.19}
\]

3.1.3 Maximum Outer Shell Displacement Condition

A possible limitation on the mount structure design is that the stiffness of the support system in the $y$ direction must be such as to prevent overstressing of the outer shell due to meridional bending.

If the allowable bending stress in the outer shell is $\sigma_b = 6M/t^2$ then $M_{\text{max}} = t^2 \sigma_b / 6$

and using (3.14) and (3.15) $Q_{\text{max}} = \frac{4\lambda}{asin\psi} \cdot \frac{t^2 \sigma_b}{6}$

\[
(\delta_s)_{\text{max}} = \frac{a\lambda\sin^2\psi}{2Et} \cdot \frac{4\lambda t^2 \sigma_b}{6asin\psi} \]

\[
(\delta_s)_{\text{max}} = \frac{\lambda^2 t \sigma_b \cos\alpha}{3E_s} \tag{3.20}
\]

3.1.4 Application to Mount Carriage Design

The foregoing mount carriage analysis was applied to the existing mount structure design to check its suitability to withstand the LM vibration spectrum requirements outlined in Section IV. This approach was necessary because specific mount carriage loading requirements were not specified until after the design phase of the contract was completed.
The analysis showed that the peak G levels, which could be attained by the LM Launch and Boost Random inputs, could severely overstress the mount carriage arms. This was the main reason for proposing the modified random inputs enunciated in Section IV. In this way the peak G level along any of the three principal axes of the shroud tankage was reduced from 25.57g to 9.37g.

Using a value for $F = 9.37g = 2440$ LBS equations (3.1) to (3.20) showed a maximum bending stress of 27,000 LBS/sq. ins in the mount carriage arms at the section approximately 22 1/2° from the tank center plane. This maximum stress value gives a minimum safety factor of 1.25 based on the material yield point which was considered satisfactory in this case because of the random nature of the peak loading.

3.1.5 Recommended Extensions to Mount Carriage Analysis

Extensions to the foregoing analysis are necessary before predictions can be made as to the performance of a tankage system subjected to a full on-board mission load profile.

The analysis requires expanding to consider instantaneous inertia loadings in the Y and Z directions and to study the degree of error involved in using thin ring theory to analyse the support arms.

A computerized version of the extended analysis can then be applied to specific cradle-type mount carriage problems to produce a weight optimized design.

3.2 Dynamical Characteristics of The Tankage System

The radial bumper suspension system can be represented by the following spring system with the center of the cryogen mass coinciding with the origin of the co-ordinate axes. This spring system assumes that suitable pre-loading has been carried out which prevents separation during dynamical g loadings. If pre-loading is impossible, then the springs in one of the bumper planes have zero stiffness.
For any arbitrary position of the cryogen mass $m$ defined by the co-ordinates $x$, $y$, $z$, the unbalanced tension in each spring will be:

$$S_i = -K_t (\alpha_i x + \beta_i y + \gamma_i z)$$  \hspace{1cm} (3.21)

where $\alpha_i$, $\beta_i$, $\gamma_i$ are the direction cosines of the axis of any $i$th spring having stiffness $K_t$.

The equations of motion can then be written:

$$m\ddot{x} = \sum_{i=1}^{n} S_i \alpha_i$$

$$m\ddot{y} = \sum_{i=1}^{n} S_i \beta_i$$  \hspace{1cm} (3.22)

$$m\ddot{z} = \sum_{i=1}^{n} S_i \gamma_i$$

The number of springs $n$ are equal to the total number of bumpers in the system, it being assumed that the spring positions are symmetrical about the $zoy$ plane.
Substituting (3.21) in (3.22) and re-arranging gives:

\[ \begin{align*}
mx + c_{11}x + c_{12}y + c_{13}z &= 0 \\
my + c_{12}x + c_{22}y + c_{23}z &= 0 \quad (3.23) \\
mz + c_{13}x + c_{23}y + c_{33}z &= 0
\end{align*} \]

where:

\[ \begin{align*}
c_{11} &= K_t \sum_{i=1}^{n} \alpha_i^2 \\
c_{12} &= K_t \sum_{i=1}^{n} \alpha_i \beta_i \\
c_{22} &= K_t \sum_{i=1}^{n} \beta_i^2 \\
c_{13} &= K_t \sum_{i=1}^{n} \alpha_i \gamma_i \\
c_{33} &= K_t \sum_{i=1}^{n} \gamma_i^2 \\
c_{23} &= K_t \sum_{i=1}^{n} \gamma_i \beta_i \\
\end{align*} \quad (3.24) \]

Taking the harmonic solutions of equations (3.23) in the form:

\[ \begin{align*}
x &= \lambda_1 \sin (pt + \delta) \\
y &= \lambda_2 \sin (pt + \delta) \\
z &= \lambda_3 \sin (pt + \delta) \quad (3.25)
\end{align*} \]

Substituting (3.25) in (3.23) gives:

\[ \begin{align*}
(c_{11} - mp^2)\lambda_1 + c_{12}\lambda_2 + c_{13}\lambda_3 &= 0 \\
c_{12}\lambda_1 + (c_{22} - mp^2)\lambda_2 + c_{23}\lambda_3 &= 0 \quad (3.26) \\
c_{13}\lambda_1 + c_{23}\lambda_2 + (c_{33} - mp^2)\lambda_3 &= 0
\end{align*} \]
Solutions for equations (3.26) different from zero require that:

\[
\begin{vmatrix}
(c_{11}-mp^2), c_{12}, c_{13} \\
c_{12}, (c_{22}-mp^2), c_{23} \\
c_{13}, c_{23}, (c_{33}-mp^2)
\end{vmatrix} = 0 \quad (3.27)
\]

Equation (3.27) is the Frequency Equation which yields a cubic in \( p^2 \); the three roots \( p_1, p_2, \) and \( p_3 \) being the required phase rates or frequencies of the system.

### 3.2.1 Spring Stiffness \( K_m \)

The vibrating mass \( m \) considered in the foregoing equations consists of the inner vessel/shroud assembly together with their contained cryogens.

The elastic support on which this mass rests is made up of a combination of the elasticity in the bumpers, pressure vessel, outer shell and mount carriage.

Let \( K_m \) = Stiffness of mount carriage when subjected to a radial loading.

\( K_B \) = Bumper stiffness at cryogenic temperature.

\( K_{pv} \) = Pressure vessel wall stiffness at cryogenic temperature.

\( K_S \) = Outer shell wall stiffness.

The compatibility and equilibrium conditions for the application of any radial load \( W \) to the elastic support are as follows:

\[
\delta = \delta_B + \delta_{pv} + \delta_{os}
\]

\[
\delta_{os} = \delta_m
\]

\[
W = W_{pv} = W_B = W_m + W_{os}
\]
The equivalent spring system representing the elastic support is shown at the left, and the resultant spring stiffness $K_t$ is then given by:

$$\frac{1}{K_t} = \frac{1}{K_B} + \frac{1}{K_{PV}} + \frac{1}{K_s + K_m}$$

(3.28)

### 3.2.2 Effect of Shock Mounts on Natural Frequencies

The foregoing theory for calculating the sinusoidal natural frequencies of the radial bumper suspension system assumes that the helium shroud tankage system is rigidly attached to the vehicle. In cases where flexible shock mounts are introduced into the suspension system in a manner similar to that shown in Figure 9, the frequencies calculated from equation (3.27) require modification.

A simple approach to this aspect of the dynamical problem is to reduce the overall system to the two-degree-of-freedom system shown at left for each of the principal modes of the bumper suspension system.

In this system, the mass $m_1$ represents the combined mass of outer shell, mount carriage, and external system components attached to the mount carriage. The spring stiffness $K_i$ represents the combined stiffness of the shock mounts or isolators in the direction of principal mode considered.

The equivalent spring stiffness $K_e$ represents the dynamical stiffness of the bumper suspension system considered on page 39 again in the direction of axis considered. If the natural frequency of the radial bumper suspension system in any one of the three co-ordinate directions is represented by $f_n$ then the dynamical stiffness $K_e$ can be found from:

$$K_e = m(2\pi f_n)^2$$

(3.29)

and the two natural frequencies in the chosen direction can be found from the roots of the following frequency equation:

$$mm_1p^4 - (K_{e}m + K_{e}m_{1} + K_{e}m_{1}m_{1})p^2 + K_{e}K_{e} = 0$$

(3.30)
The two frequencies calculated from the above equation (3.30) are the frequencies when the inner vessel/shroud/cryogen mass $m$ vibrates in phase, and $180^\circ$ out of phase, with the mass $m_1$.

3.2.3 Effect of Mount Carriage Pre-load on Dynamical Characteristics

The dynamical characteristics of the radial bumper suspension system discussed so far have assumed a purely linear system which requires a certain degree of pre-loading of the dewar assembly to satisfy these linearity requirements. The degree of pre-load necessary to meet these requirements will now be discussed.

During the initial build-up of a dewar leading to the girth welding of the outer shell hemispheres, it can be assumed that contact between the outer shell and pressure vessel via the bumpers occurs on all bumpers in both bumper planes.

When the dewar is subsequently filled with cryogen separation of the suspension system can occur leading to non-linear vibration problems involving impact loading conditions. This separation can be prevented by pre-loading the mount structure prior to cryogen fill; the degree of pre-loading being based on zero separation in a $1g$ environment.

Separation can then only occur at high acceleration loads when vibration inputs can be ignored.

The following analysis will be aimed at designing the amount of pre-load required for a given system to prevent separation in a $1g$ environment. This is the optimum condition which will limit vibration levels and at the same time, reduce thermal conduction through the bumpers to a minimum.

Let $\alpha_p = \text{linear expansion coefficient for PV material}$

$\alpha_B = \text{linear expansion coefficient for bumper material}$

$x_t = \text{diametral contraction of support system due to cryogen fill.}$

$x_p = \text{increase in PV diameter due to cryogen pressure}$

$\Delta t = \text{temperature difference between ambient and cryogen temperature}$
\[ K = \text{suspension System stiffness at ambient} \]
\[ K_t = \text{suspension system stiffness at cryogenic temperatures} \]
\[ W = \text{mount carriage pre-load} \]
\[ W_c = \text{cryogen weight in a lg environment} \]
\[ n = \text{total number of bumpers} \]
\[ D = \text{P.V. diameter} \]
\[ h = \text{P.V. wall thickness} \]
\[ d = \text{bumper diameter} \]
\[ \sigma_{yp} = \text{P.V. material yield point} \]
\[ E = \text{Young's modulus for P.V. material} \]
Assuming a mean change in bumper temperature of \( \frac{1}{2} \Delta t \), the thermal contraction can be written:

\[
x_t = \Delta t (\alpha_p D + \alpha_b d)
\]  

(3.31)

Also,

\[
x_p = \frac{PD^2}{4he} (1-\nu)
\]  

(3.32)

If the allowable P.V. stress = \( \frac{2}{3} \sigma_p \) based on a proof safety fact of 1.5 then:

\[
P = \frac{4h}{D} \frac{2}{3} \sigma_p = \frac{8h}{3D} \sigma_p
\]

which on substitution in (3.32) gives:

\[
x_p = \frac{2D}{3} (1-\nu) \frac{\sigma_P}{E}
\]  

(3.33)

For Inconel 718, (3.33) reduces to \( x_p = 2.25 \times 10^{-3} D \)

Now, initial pre-load in each spring = \( 2\sqrt{2} \) W/n and initial compression of each spring \( \delta_1 = \frac{2\sqrt{2}W}{nK} \)

Final compression of each spring

\[
\delta_2 = \delta_1 - \frac{1}{2}(x_t - x_p) + \frac{\sqrt{2}W}{nKt}
\]  

(3.34)

The above values for \( \delta_1 \) and \( \delta_2 \) assume a bumper plane disposed 45° above and below the center plane.

The condition for separation is \( \delta_2 = 0 \) which when applied to equation (3.34) gives:

\[
\frac{2\sqrt{2}W}{nK} - \frac{1}{2}(x_t - x_p) - \frac{\sqrt{2}W}{nKt} = 0
\]

or pre-load

\[
W = \frac{nK}{2\sqrt{2}} \left( \frac{1}{2}(x_t - x_p) + \frac{\sqrt{2}W}{nKt} \right)
\]
$W = \frac{nK}{2\sqrt{2}} \left[ \frac{1}{2} \left\{ \Delta t (\alpha_D + \alpha_B) - \frac{2D}{3}(1-v) \frac{\sigma_P}{E} \right\} + \frac{\sqrt{2}W_C}{nK_t} \right] (3.35)$

More generally, for increased acceleration loads defined by Ng's equation (3.35) for the pre-load in this case is:

$$W = \frac{nK}{2\sqrt{2}} \left[ \frac{1}{2} \left\{ \Delta t (\alpha_D + \alpha_B) - \frac{2D}{3}(1-v) \frac{\sigma_P}{E} \right\} + \frac{\sqrt{2}NW_C}{nK_t} \right] (3.36)$$

The suspension stiffness $K$, during actual pre-loading of the assembly, is different from the suspension stiffness defined by equation (3.28) when the system is vibrating. This is because the manner or origin of loading has changed.

At ambient temperature conditions, during the pre-load phase, the compatibility and equilibrium conditions are:

- $W_B = W_{pv}$
- $\delta_B = \delta_{os} - \delta_{pv}$
- $W = W_m$
- $\delta = \delta_{os} + \delta_m$

The above conditions are satisfied by the spring system shown at left where, in deference to Section 3.2.1, $K_B$ and $K_{pv}$ are the bumper and pressure vessel wall stiffnesses at ambient temperature.

The combined system stiffness $K$ is found from:

$$\frac{1}{K} = \frac{1}{K_m} + \frac{1}{K_s} + \frac{1}{1/K_B + 1/K_{pv}} (3.37)$$

3.2.4 Recommend Extensions to Dynamical Analysis

Application of the foregoing equations to the subject Helium Shroud Tankage are discussed in relation to the vibration testing of the unit described in Section IV.
Extensions to the foregoing dynamical analysis are necessary particularly in the areas which can better guarantee a linear vibrating system. This requires design studies aimed at alternative methods of pre-loading the tankage assembly to prevent separation of the inner vessel/shroud assembly from the bumpers and outer shell due to cryogenic shrinkage. While improving the dynamical characteristics of the unit, alternate pre-loading methods should not impair the thermal characteristics of the dewar.

Parallel investigations should be carried out alongside the purely linear analysis to ascertain some of the non-linear characteristics of the radial bumper suspension system having various degrees of separation. This approach would be aimed at determining the amount of shock loading and displacement which can occur within the dewar without degrading the thermal and thermo-dynamic requirements of the storage tank.

### 3.3 Dewar Thermal Analysis

The following thermal analysis of the evacuated dewar is aimed at correlating the measured heat leaks discussed in Section IV with the theoretically predicted values.

Heat transfer from the surrounding atmosphere into the contained cryogen can take place by the modes of conduction, radiation or natural convection. Because of the low vacuum annulus pressure of $2.5 \times 10^{-8}$ torr achieved by the methods described in Section 2.2.6, the natural convection transfer mode is negligible and has, therefore, been ignored in the following analysis.

#### 3.3.1 Radiation

For the general case of radiative heat transfer between $n$ concentric spheres, the heat transfer rate can be expressed as:

$$Q_R = \sigma E_o A_1 (T_n^4 - T_1^4) \quad \text{(3.38)}$$

where:

- $\sigma$ = Stefan-Boltzmann constant
- $A_1$ = area of enclosed surface or shroud
- $T_n$ = absolute temperature of the outer shell
The overall emissivity factor $E_0$ is given by:

$$\frac{1}{E_0} = \frac{n-1}{\sum_{i=1}^{n-1} A_i} \left[ \frac{1}{A_i e_i} + \frac{1}{A_{i+1} e_{i+1}} \left( \frac{1}{e_{i+1}} - 1 \right) \right] \quad (3.39)$$

The main problem involved in calculating the radiative heat transfer for any given dewar using equations (3.38) and (3.39) is assigning accurate values to the individual surface emissivities $e_1, e_2 \ldots e_n$.

Each emissivity value is a function of surface temperature which, for all practical purposes, is a linear relationship as illustrated by Figure 12 for the emissivity of silver surfaces.

A computer program exists which uses an iterative process for calculating the radiation shield temperatures which, in turn, allows successively more accurate estimations to be made of the shield emissivity values using the known relationships of Figure 12.

3.3.2 Conduction

Conductive heat transfer between the outer shell and the stored cryogen occurs principally through the radial bumpers and the fill and vent tubes. This mode of heat transfer can be generally expressed as:

$$Q_c = \frac{K A \Delta T}{L} \quad (3.40)$$

where:

- $K = $ mean thermal conductivity
- $A = $ cross-sectional area
- $L = $ length of conductive path

and:

- $\Delta T = $ temperature difference

Application of equation (3.40) to the case of the internal fluid tubes is straightforward but the problem of the conductive heat flow through the bumpers is more involved.

The bumper conductivity is dependent on the area of contact between the bumpers and outer shell or
EMISSIVITY OF SILVER SURFACES AS A FUNCTION OF SURFACE TEMPERATURE

**Figure 12.**
inner vessel/shroud assembly. It also involves a continuous variation in cross-sectional area $A$ and a non-linear temperature gradient throughout the bumper material.

Because of these difficulties an empirical factor is used in the computer program referred to in Section 3.3.1 to cater for the conductive heat transfer through the radial bumpers. This factor is based on prior experience with testing cryogenic storage dewars incorporating the discrete radiation shield design concept.

### 3.3.3 Vapor Cooling

As described in Section 11, a vapor cooling system is attached to the inner radiation shield to reduce the environmental heat leak to the cryogen and improve the dewar's standby capabilities. This is achieved by the evaporated fluid from the shroud passing through the vapor cooling coil and intercepting the environmental heat leak $Q_1$ at a rate $Q_2 = m \Delta h$ where $m$ is the vapor cooled mass flow and $\Delta h$ is its change in specific enthalpy.

The heat flow equation which expresses the vapor cooling effectiveness is:

$$Q_2 = Q_1 - m \Delta h$$  \hspace{1cm} (3.41)

where $Q_2$ is the resulting heat leak to the cryogen.

Equation (3.41) is included in the thermal computer program by accordingly adjusting the heat balances during the iterative procedure.

### 3.3.4 Computer Predicted Heat Leaks

Outputs from the computer program predicted the following vented heat leak values for the shroud tankage using the NASA best fit emissivity curve of Figure 12.

<table>
<thead>
<tr>
<th>INNER VESSEL &amp; TOTAL VENTED HEAT LEAK [BTU/HR]</th>
<th>REMARKS</th>
</tr>
</thead>
<tbody>
<tr>
<td>LN$_2$</td>
<td>14.40</td>
</tr>
<tr>
<td>LN$_2$</td>
<td>10.13</td>
</tr>
<tr>
<td>LH$_2$</td>
<td>11.64</td>
</tr>
<tr>
<td>LH$_2$</td>
<td>5.95</td>
</tr>
</tbody>
</table>

The above values should be compared with the experimental values listed in Table IV, page 73.

Correlation between the theoretical and experimental heat leak values suggested that the actual surface emissivities were closer to the NASA best fit curve than the existing Bendix design curve.
SECTION IV
SYSTEM TESTING

The system testing described in this section was designed to verify the structural, thermal and thermo-dynamic capabilities of the helium shroud tankage described in Section II. In most instances a comparison between the results of each test and the applicable theoretical analysis is included.

4.1 Hemisphere Buckling Tests

During the process of manufacturing the Dewar the shroud hemispheres must undergo helium leak checks before they are assembled into the unit. This necessarily involves evacuation of the inside of these hemispheres which subjects them to an external pressure load which governs the material strength of the shroud.

The outer shell is also subjected to external atmospheric pressure difference during the normal operation of the tankage on the ground. To verify the structural integrity and design safety factors of these shells buckling tests to failure were carried out on sample shroud and outer shell hemispheres. These hemispheres were inturgescently formed from the same batch material and underwent the same degree of process annealing as the shells used for the helium tankage fabrication. Each shell was trimmed off at the equator to form a pure hemisphere.

The inturgescent forming machine used to form the actual hemispheres at the Bendix Santa Ana facility was adapted for the buckling tests. This machine is shown in the figure at left.

The hemisphere to be tested was mounted inside a larger diameter cavity and the annulus between the shell and cavity pressurized with hydraulic fluid. Prior to filling the cavity with fluid a vacuum of approximately 6 psig was drawn on the space inside the hemisphere to prevent it floating while the cavity was filled with fluid.

Actual testing was carried out by gradually increasing the hydraulic fluid pressure until the shell collapsed and noting the difference between the fluid pressure in the annulus and the vacuum pressure inside the hemisphere.

The Inconel 718 shroud hemisphere and aluminum 6061 outer shell hemisphere after buckling are shown in the respective photographs at the left. The shroud hemisphere
bucked at a pressure differential of 75 psi and the outer shell hemisphere at 68 psi.

Figure 13 illustrates the relationship between the results of these two buckling tests and the existing theoretical and empirical buckling design curves for monocoque spherical shells.

Based on the early theoretical work of Zoelly (2) and others, the instability of monocoque spherical shells subjected to external pressure can be represented in the form

\[
P_{cr} = \frac{K_p}{(R/t)^2}
\]

where:

- \( P_{cr} \) = external buckling or critical pressure
- \( E \) = modulus of elasticity for the shell material
- \( R \) = mean radius of the shell
- \( t \) = mean thickness of the shell.

and \( K_p \) = empirical or theoretical correction factor.

The curves of figure 13 are representations of equation (4.1) using the following values for \( K_p \):

<table>
<thead>
<tr>
<th>Curve</th>
<th>( K_p )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zoelly</td>
<td>1.212</td>
</tr>
<tr>
<td>Von Karman</td>
<td>0.365</td>
</tr>
<tr>
<td>NASA</td>
<td>0.350</td>
</tr>
<tr>
<td>Bendix</td>
<td>0.260</td>
</tr>
</tbody>
</table>

It has been well known for some time that the value of \( K_p = 1.212 \) derived by Zoelly using linear shell theory was too idealistic for practical shell problems. To surmount this problem Von Karman (3) proposed a non-linear theory using the principle of minimum potential energy of the deformed shell to predict the critical buckling pressure and arrived at a theoretical factor of \( K_p = 0.365 \).

The factor derived in the NASA shell analysis manual (4) of \( K_p = 0.350 \) was based on a best fit curve for experimentally available data on spherical caps using a 90% probability level.

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THEORETICAL AND EMPIRICAL CURVES
FOR BUCKLING OF MONOCOQUE SPHERICAL SHELLS

FIGURE 13

PREVIOUS BENDIX EXPERIMENTAL VALUES

EXPERIMENTAL VALUES OBTAINED FROM
NAS 9 - 7337
BUCKLING TESTS

CRITICAL PRESSURE DIVIDED BY YOUNG'S MODULUS

$(\frac{P_{cr}}{E}) \times 10^{-6}$

RATIO OF RADIUS TO THICKNESS

$R/t$

200 400 600 800 1000 1200 1400

ZOELLY'S EQUATION

VON KARMAN'S MINIMUM PRESSURE CURVE

SHROUD

NASA DESIGN CURVE

BENDIX DESIGN CURVE

OUTER SHELL
A value of $K_p = 0.260$ was derived by Bendix on the basis of their own experiments with hemispherical shell buckling. These experimental points obtained prior to the present testing are plotted on Figure 13.

The results of the two buckling tests carried out within the frame of the present contract suggest that for values of $R/t < 500$ the Von Karman curve more closely approximates the experimental critical points. It also appears reasonable to assume that for values of $R/t < 200$ where the hemispherical or spherical shells are becoming more rigid the experimental critical points should approach the Zoelly curve predicted by linear theory.

### 4.2 Pressure Vessel Burst Test

A hydrostatic burst test was performed on the spare inner pressure vessel which was fabricated by the methods discussed in Section 2.2.1. This test was aimed at verifying the minimum burst safety factor of 2.0 specified in Exhibit 'A', Statement of Work.

The vessel was filled completely with water and the pressure gradually increased by observing the pressure gage connected to the supply line. Failure of the vessel occurred at an observed pressure of 5,600 P.S.I.G. which indicates a burst safety factor of 2.17 at room temperature based on a maximum shroud system operating pressure of 2,580 P.S.I.G. This safety factor assumes that failure occurs in accordance with the maximum principal stress theory of failure.

Photographs at the left show the inner pressure vessel before and after the above described burst test.
4.3 Temperature and Pressure Sensor Calibration

Prior to system testing of the completed helium tankage unit the temperature and pressure sensors were calibrated to obtain conversion charts for each sensor. These charts were subsequently used to convert temperatures and pressures throughout the various phases of the test program.

4.3.1 Temperature Sensor Calibration

Temperature sensors designated PRT1, PRT2 and PRT3 on the wiring diagram of figure 8 are supplied with their own signal conditioning unit and are therefore directly calibrated in relation to the voltmeter reading on the control panel. Calibration figures for these units were supplied by the manufacturer and are plotted in figure 14. A check on this curve was made by immersing these sensors at LN$_2$ and LH$_2$ temperatures and good correlation resulted.

Temperature sensor PRT4 which measures the vapor cool outlet temperature was introduced into the system at an intermediate part of the contract and, therefore has no direct read out capabilities. For the purposes of the present test program this sensor was connected to a separate Wheatstone Bridge unit to monitor its resistance change in relation to the supplier's calibration points represented by figure 15. This curve was again checked against the sensor resistance at LN$_2$ and LH$_2$ temperatures.

Figures 14 and 15 are the calibration curves used to convert voltmeter and resistance changes into actual temperature readings.

4.3.2 Pressure Transducer Calibration

To calibrate the pressure transducer connected to the shroud vent system the shroud was gradually pressurized with GH$_e$ in 5 psi increments until the shroud P.R.V. opened. Actual helium pressure was measured with a calibrated pressure gage connected to the system, each pressure gage reading corresponding to a voltmeter reading on the control panel.

The resulting calibration curve for the shroud pressure transducer is shown in figure 16.

Calibration of the pressure transducer connected to the inner high pressure system was carried out during a system hydrostatic pressure check-out using Freon as the
NAS 9 - 7337 CALIBRATION CURVE
FOR TEMPERATURE SENSORS PRT1, PRT2, PRT3

FIGURE 14

VOLTAGE OUTPUT (VDC)

5.0
4.0
3.0
2.0
1.0
0

TEMPERATURE °F

-100
-200
-300
-400
-500

A5226 - 35
NAS 9 - 7337  CALIBRATION CURVE
FOR TEMPERATURE SENSOR PRT4

FIGURE 15

SENSOR RESISTANCE (Ω)

TEMPERATURE °F

100 0 -100 -200 -300 -400

100 80 60 40 20

0

A5226 - 36
NAS 9 - 7337 HELIUM SHROUD TANKAGE
CALIBRATION CURVE FOR SHROUD PRESS. SENSOR P2
PRV OPENING = 39.5 psig

FIGURE 16

VOLTOMETER READING (VDC)

ROOM TEMPERATURE

GAGE PRESSURE (LBS./SQ. INS.)
NAS 9 - 7337 HELIUM SHROUD TANKAGE
CALIBRATION CURVE FOR INNER PRESS. SENSOR P1
PRV OPENING = 2600 psig

FIGURE 17

GAGE PRESSURE (LBS. / SQ. INS.)

VOLTMETER READING (VDC)
pressurant. The inner pressure was gradually raised in 200 psi increments until the inner P.R.V. opened, pressure gage readings corresponding to the control panel voltmeter reading again being the method of calibration.

The calibration curve for the inner pressure transducer is shown in figure 17.

4.4 Dewar Fill Procedure

Before performing any cryogenic testing on the helium shroud unit the following purge and fill procedures should be followed to prevent moisture entering the system resulting in partial or total blockage of the fluid lines. These procedures should be read in conjunction with figure 7 of Section 11.

4.4.1 High Pressure System Purge

To insure efficient and reliable operation of the solenoid valves connected to the inner vessel high pressure system, condensation of moisture on these valve seats should be avoided. Ice formation on the valve seats can be minimized using the following procedure.

4.4.1.1 Energize the inner vent and main supply solenoid valves and insure that they are in the OPEN position.

4.4.1.2 Blank-off the main supply line on the outlet side of the main supply valve.

4.4.1.3 Connect a source of dry helium gas directly to the inlet side of the inner manual fill valve and allow the system to purge for five minutes.

4.4.1.4 Close the inner manual fill valve.

4.4.1.5 Connect a vacuum pump to the outlet side of the inner vent solenoid valve and pull a partial vacuum on the inner vessel system.

4.4.1.6 Open the inner manual fill valve and disconnect the vacuum pump, thus allowing a free flow of helium gas (approximately 15 liters/minute) through the system for about 20 minutes.

4.4.1.7 Energize the main supply solenoid valve to the CLOSED position and remove the blanking cap.

4.4.1.8 Energize the inner vent solenoid valve to the CLOSED position and close the inner manual fill valve.
4.4.2 Shroud System Purge

To expel air and moisture from the shroud system and prevent blockage of the fluid tubes due to ice formation, the following procedure should be applied.

4.4.2.1 Set the shroud manual fill and V.C. valves to the OPEN position and the shroud manual vent to the CLOSED position.

4.4.2.2 Connect a source of dry helium gas directly to the inlet side of the shroud manual fill valve and allow a free flow of helium gas (approximately 15 liter/minute) through the vapor cooling system for about five minutes.

4.4.2.3 Set the shroud manual vent valve to the OPEN position and the V.C. valve to the CLOSED position and continue to purge with helium gas for a further five minutes.

4.4.2.4 Close the shroud manual fill and vent valves.

4.4.3 Shroud Fill

The following procedure for filling the shroud with LH$_2$ should be carried out AFTER the system purges described in sections 4.4.1 and 4.4.2 have been completed. It should be emphasised that failure to insure an adequate inner and shroud system purge before filling with cryogen can lead to ice blockage in the fluid tubes which could prevent operation of the P.R.V's.

4.4.3.1 Connect a LN$_2$ dewar to the inlet side of the shroud manual fill valve and set this valve to the OPEN position.

4.4.3.2 OPEN the shroud vent valve and continue the fill operation until liquid escapes through the vent valve. CLOSE the shroud fill valve at this juncture.

4.4.3.3 When the system has stabilized with LN$_2$, in the shroud, re-purge the inner vessel with gaseous helium which has been pre-cooled by passing through a LN$_2$ heat exchanger connected to the inner manual fill valve. For this purge the inner fill and vent valves should be OPEN and the main supply valve CLOSED.

4.4.3.4 CLOSE the inner fill and vent valves.

4.4.3.5 Connect the LN$_2$ heat exchanger to the outlet side of the shroud manual vent valve and connect the helium gas source to the heat exchanger.
4.4.3.6 OPEN the shroud fill valve and empty the shroud with gaseous helium which has been pre-cooled by passing through the LN₂ heat exchanger.

4.4.3.7 CLOSE the shroud fill valve and momentarily OPEN the V.C. valve to expel any LN₂ trapped in the vapor cool system.

4.4.3.8 CLOSE the shroud vent valve and disconnect the LN₂ heat exchanger and gaseous helium source.

4.4.3.9 Connect the LH₂ source to the inlet side of the shroud manual fill valve and remote vent stacks to the outlet sides of the shroud vent and V.C. valves. These stacks should carry the vented hydrogen gas away from the immediate vicinity of the test area.

4.4.3.10 OPEN the shroud manual fill and vent valves and fill the shroud with LH₂, replenishing as necessary until the system reaches equilibrium temperature.

4.4.3.11 CLOSE the shroud fill valve.

4.4.3.12 Re-connect the LN₂ heat exchanger to the inlet side of the inner manual fill valve and connect the gaseous helium source to the heat exchanger.

4.4.3.13 OPEN the inner fill valve and pressurize the inner vessel up to its operating pressure of 2000 psia. During this helium pressurization the LH₂ in the shroud should be replenished by opening the shroud fill valve.

4.4.3.14 CLOSE the inner fill valve and the shroud vent valve and disconnect the helium and hydrogen sources.
4.5 Vented Heat Leak Tests

An initial study of the thermal insulating characteristics of the helium shroud tankage was carried out by performing vented heat leak tests using LN\(_2\) and LH\(_2\) as the stored cryogens. A vented heat leak test is defined as a test which ascertains the cryogen boil-off rate at atmospheric pressure due to environmental heat absorption.

A total number of seven vented heat leak tests were carried out within the scope of the testing program, the type of test performed and the cryogens used being summarized in Table III.

To perform each test the tankage assembly was placed on a weigh-scale and a Wet Test Meter (W.T.M.) connected to either the shroud vent or shroud vapor cooled vent line depending on whether the test was with or without vapor cooling. Figure 18 shows the Wet Test Meter system connected to the vapor cooled vent valve for a typical vented heat leak test with vapor cooling.

Vented gas from the shroud bubbles through a water container where it is totally saturated before flowing through the W.T.M. Connected to the downstream side of the W.T.M. is a pressure regulator which can be adjusted to maintain the shroud vent pressure sensibly constant at a standard atmospheric pressure of 760 millimeters of mercury. This pressure is indicated by the mercury barometer connected into the shroud vent system as shown in Figure 18.

Accumulated volumetric flow through the W.T.M. was corrected for temperature and pressure in accordance with the following expression.

\[
V_s = \frac{P_m - P_w}{P_m} = V_m \tag{4.2}
\]

where:

\[
V_m = \text{accumulated W.T.M. reading}
\]
\[
P_m = \text{W.T.M. pressure from barometer}
\]
\[
P_w = \text{water vapor pressure at temperature } T_m
\]

Because the vented shroud gas was maintained close to standard atmospheric conditions by the back pressure regulator, standard gas tables for density at various temperatures were used to correct \(V_s\) from expression (4.2) to mass flow.
TABLE III
NAS9-7337 HELIUM SHROUD TANKAGE
VENTED HEAT LEAK TESTS

<table>
<thead>
<tr>
<th>TEST NO.</th>
<th>INNER VESSEL CRYOGEN</th>
<th>SHROUD CRYOGEN</th>
<th>INNER FILL VALVE (MANUAL)</th>
<th>SUPPLY VALVE (SOLENOID OPERATED)</th>
<th>SHROUD FILL VALVE (MANUAL)</th>
<th>INNER VENT VALVE (SOLENOID OPERATED)</th>
<th>SHROUD VENT VALVE (MANUAL)</th>
<th>REMARKS</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>LN₂</td>
<td>LN₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>OPEN</td>
<td>CLOSED No Vapor Cooling</td>
</tr>
<tr>
<td>2</td>
<td>LN₂</td>
<td>LN₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>CLOSED</td>
<td>OPEN Shield Vapor Cooled</td>
</tr>
<tr>
<td>3</td>
<td>LH₂</td>
<td>LH₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>OPEN</td>
<td>CLOSED No Vapor Cooling</td>
</tr>
<tr>
<td>4</td>
<td>LH₂</td>
<td>LH₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>CLOSED</td>
<td>OPEN Shield Vapor Cooled</td>
</tr>
<tr>
<td>5</td>
<td>LH₂</td>
<td>EMPTY</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>OPEN</td>
<td>CLOSED No Vapor Cooling</td>
</tr>
<tr>
<td>6</td>
<td>LN₂</td>
<td>LN₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>CLOSED</td>
<td>After Vibration Search. No Vapor Cooling</td>
</tr>
<tr>
<td>7</td>
<td>LN₂</td>
<td>LN₂</td>
<td>CLOSED</td>
<td>CLOSED</td>
<td>closed</td>
<td>OPEN</td>
<td>OPEN</td>
<td>After Vibration Search. Shield Vapor Cooled</td>
</tr>
</tbody>
</table>

Test Nos. 6 and 7 are repeats of tests 1 and 2, respectively, to verify that the vibration tests described below have not impaired the thermal performance of the Dewar.
FIGURE 18 — THERMAL TEST EQUIPMENT SCHEMATIC
The total combined vented loss from the shroud and inner vessel was measured using the automatically recorded weigh scale reading, the rate of change of the weigh scale reading being proportional to the total heat leak. Knowing the vented loss from the shroud and the total loss then the heat leak into the inner vessel is obviously the difference between these two values.

4.5.1 LN₂ Vented Heat Leak Tests

Test numbers 1 and 2 of Table III were performed after the shroud and inner vessels were completely filled with LN₂.

Both instantaneous and average W.T.M. readings were recorded during the tests, the non-vapor cooled test being continued for a sufficient period to ascertain the overall heat leak characteristics from a full to empty shroud.

Figures 19 and 21 show, respectively, the rate of decrease of the weigh scale reading for the non-vapor cooled and vapor cooled tests. A slight change in slope was observed in Figure 19 after about 60 hours of testing which indicated a small reduction in total heat leak but no discernable change was noted for the vapor cooled total heat leak of Figure 21.

The variation of shroud heat interception with time, together with the environmental temperature readings, are plotted in Figures 20 and 22 for the LN₂ non-vapor cooled and vapor cooled tests respectively. These curves show that a considerable period elapsed after the tests were initiated before the shroud heat interception reached its maximum equilibrium state. This was attributed to stratification taking place within the inner vessel and between the inner and shroud cryogens during this stabilization period.

Instantaneous values plotted in Figures 20 and 22 tended to have significant errors compared to the more accurate average values taken over a period because of errors involved in reading the W.T.M. As a result of this, the shroud heat interception curve has been drawn through the average points of Figures 20 and 22.

Probably the most significant aspect of these vented heat leak tests was the rapid decrease in shroud heat interception illustrated by Figure 20 as the shroud liquid level fell. Another significant factor
NAS 9 - 7337 HELIUM SHROUD TANKAGE
TOTAL VENTED LOSS OF LN₂ FROM SHROUD & INNER VESSEL
NON-VAPOR COOLED

WEIGHT SCALE READING (LBS)

STANDBY TIME (HOURS)

FIGURE 19

\( \frac{\Delta W}{\Delta t} = \frac{dQ}{dt} \)

IN RANGE \( 0 \leq t \leq 59.5 \)

\( \frac{dQ}{dt} = 16.85 \text{ BTU/HR} \)

IN RANGE \( 71.8 \leq t \leq 118 \)

\( \frac{dQ}{dt} = 16.2 \text{ BTU/HR} \)

AFTER VIBRATION TESTING
LN\textsubscript{2} NON-VAPOR COOLED VENTED HEAT LEAK TEST

\begin{figure}
\centering
\includegraphics[width=\textwidth]{figure20}
\caption{SHROUD HEAT INTERCEPTION (BTU/HR) vs. STANDBY TIME (HOURS)}
\end{figure}
NAS 9 - 7337 HELIUM SHROUD TANKAGE
TOTAL VENTED LOSS OF LN₂ FROM SHROUD & INNER VESSEL
VAPOR COOLED

WEIGH SCALE READING (LBS)

STANDBY TIME (HOURS)

AFTER VIBRATION TESTING

\[ \frac{dQ}{dt} = 13 \text{ BTU/HR} \]

FIGURE 21
INNER & SHROUD LN₂ HEAT LEAK CHARACTERISTICS

HEAT INTERCEPTION (BTU/HR)

STANDBY TIME (HOURS)

Figure 23

A5226-57
was the "levelling off" in the shroud heat interception characteristic when the vapor cooling system was in use.

From the experimental data of Figures 19 to 22, the heat leak values listed in Table IV have been extracted for the equilibrium state when the shroud and inner vessels are completely filled with LN₂. This table also compares the equilibrium heat leaks per unit area of shroud surface with the corresponding values obtained by NASA-MSC from their tests on the NAS9-4634 shrouded unit.

Figure 23 summarizes the heat leak characteristics of the shroud and inner vessels for both the vapor cooled and non-vapor cooled conditions using LN₂ as the stored fluid.

4.5.2 LH₂ Vented Heat Leak Tests

Test numbers 3 and 4 of Table III, and test number 5 which was a continuation of test number 4 since vapor cooling ceases after the shroud is empty, were performed in Bendix's remote test facility where the shroud and inner vessels were completely filled with LH₂ at ambient pressure.

An overall view of the shroud unit mounted on the remote test cell weigh-scale and a close-up of the vacuum jacketed valves used to transfer the LH₂ into the shroud and inner vessels are shown respectively at the left. These jacketed valves were used for the LH₂ vented heat leak tests to conserve cryogen.

The monitoring of test measurements and the presentation of the results is similar to the LN₂ tests described in Section 4.5.1 with one exception. Temperature measurements of the vented hydrogen at the inlet and outlet to the vapor cooling system were recorded throughout the LH₂ vented heat leak tests and all subsequent testing. These readings were unattainable during the LN₂ testing because system assembly and calibration was completed after these tests to comply with the tight testing program schedule.

Figures 24 to 27 illustrate the rate of decrease of the weigh scale reading and the variation of shroud heat interception and environmental temperature with time for both the vapor cooled and non-vapor cooled test conditions.

The vapor cooled phase of the LH₂ heat leak tests were, contrary to the LN₂ testing, performed before the
### TABLE IV

**VENTED HEAT LEAK DATA**

<table>
<thead>
<tr>
<th>DEWAR</th>
<th>INNER VESSEL FLUID</th>
<th>SHROUD FLUID</th>
<th>MEAN AMBIENT TEMP. °F</th>
<th>EQUILIBRIUM HEAT LEAK [BTU/HR]</th>
<th>TOTAL HEAT LEAK/UNIT AREA OF SHROUD SURFACE [BTU/(FT²·HR)]</th>
<th>REMARKS</th>
</tr>
</thead>
<tbody>
<tr>
<td>NAS9-4634</td>
<td>---</td>
<td>LN₂</td>
<td>70</td>
<td>---</td>
<td>9.79</td>
<td>9.79</td>
</tr>
<tr>
<td>NAS9-7337</td>
<td>LN₂</td>
<td>LN₂</td>
<td>80</td>
<td>6.25</td>
<td>10.0</td>
<td>16.25</td>
</tr>
<tr>
<td>NAS9-4634</td>
<td>---</td>
<td>LN₂</td>
<td>70</td>
<td>---</td>
<td>8.87</td>
<td>8.87</td>
</tr>
<tr>
<td>NAS9-7337</td>
<td>LN₂</td>
<td>LN₂</td>
<td>80</td>
<td>5.0</td>
<td>8.0</td>
<td>13.0</td>
</tr>
<tr>
<td>NAS9-4634</td>
<td>---</td>
<td>LH₂</td>
<td>70</td>
<td>---</td>
<td>8.6</td>
<td>8.6</td>
</tr>
<tr>
<td>NAS9-7337</td>
<td>LH₂</td>
<td>LH₂</td>
<td>--</td>
<td>3.3</td>
<td>9.3</td>
<td>12.6</td>
</tr>
<tr>
<td>NAS9-4634</td>
<td>---</td>
<td>LH₂</td>
<td>70</td>
<td>---</td>
<td>5.11</td>
<td>5.11</td>
</tr>
<tr>
<td>NAS9-7337</td>
<td>LH₂</td>
<td>LH₂</td>
<td>70</td>
<td>1.4</td>
<td>7.3</td>
<td>8.7</td>
</tr>
</tbody>
</table>

NAS9-4634 Shroud Surface Area = 7.9 FT²
NAS9-7337 Shroud Surface Area = 16.28 FT²
NAS 9-7337  HELIUM SHROUD TANKAGE
TOTAL VENTED LOSS OF LH2 FROM SHROUD & INNER VESSEL
NON-VAPOR COOL
LH₂ NON-VAPOR COOLED VENTED HEAT LEAK TEST

SHROUD HEAT INTERCEPTION

INSTANTANEOUS W.T.M. READING

AVÉRAGE W.T.M. READING

ENVIRONMENTAL TEMP Tₑ

SHROUD HEAT INTERCEPTION (BTU/HR)

ENVIRONMENTAL TEMP Tₑ (°F)

STANDBY TIME (HOURS)

FIGURE 25

A5226-65
NAS 9-7337 HELIUM SHROUD TANKAGE
TOTAL VENTED LOSS OF LH$_2$ FROM SHROUD & INNER VESSEL
VAPOR COOLED

\[ \frac{dQ}{dt} = 9.58 \text{ BTU/HR.} \]

\[ \frac{dQ}{dt} = 8.67 \text{ BTU/HR} \]

\[ \frac{dQ}{dt} = 10.35 \text{ BTU/HR}. \]

FIGURE 26.
LH₂ VAPOR COOLED VENTED HEAT LEAK TEST

ENVIRONMENTAL TEMP (°F)

STANDBY TIME (HOURS)

SHROUD HEAT INTERCEPTION

SHROUD HEAT INTERCEPTION (BTU/HR)

INSTANTANEOUS W.T.M. READING

AVERAGE W.T.M. READING

FIGURE 27
INNER & SHROUD LH₂ HEAT LEAK CHARACTERISTICS

HEAT INTERCEPTION (BTU/HR)

STANDBY TIME (HOURS)

FIGURE 28
non-vapor cooled phase. Figure 26 illustrates three distinct total heat leak rates which signify the following conditions:

(i) An initial stabilizing period of between 30 and 40 hours during which time stratification within and between the shroud and inner vessel takes place.

(ii) A period of stable equilibrium heat leak extending over approximately 60 hours during which time there is a steady fall in the shroud liquid level.

(iii) A final period of increasing total heat absorption after the shroud has emptied and vapor cooling becomes ineffective.

A full spectrum of reducing shroud heat interception is signified by Figure 27 for the vapor cooled test from a completely full shroud until it was empty. The area under this curve indicates a total heat capacity of 709 BTU's which would suggest a shroud volume of 0.85 cubic ft. and an initial 91% full condition at the start of the test.

Figures 24 and 25 show the results of the LH2 non-vapor cooled heat leak tests. The erratic behavior of these thermal characteristics, particularly during the first 60 hours of testing, was attributed to the warming of the inner vapor cooled radiation shield immediately after the vapor cooled testing phase.

The equilibrium heat leak values suggested by Figures 24 through 27 are again tabulated in Table IV and compared with the results from the NAS9-4634 unit.

Test 5 of Table III was a continuation of the vapor cooled test after the shroud had emptied. This test confirmed a gradual increasing environmental heat input until the non-vapor cooled heat leak had reached about 13 BTU/HR.

A summary of the LH2 vented heat leak characteristics is contained in Figure 28 for both the vapor cooled and non-vapor cooled tests.

Analysis of the vapor cooled inlet and outlet temperatures for tests 3, 4 and 5 proved disappointing and suggested that the positioning of the probe for PRT #4 caused inaccurate outlet temperature measurements resulting in errors in the vapor cool shield heat
interception Q3 discussed in Section 3.3.3. These results were therefore excluded from the presentation.

Throughout the LN$_2$ and LH$_2$ vented heat leak tests described above continuous monitoring of the environmental temperature was carried out. These temperatures are plotted in Figures 20, 22, 25 and 27.

As might be expected, the inner vessel temperatures were sensibly those of the liquid/gas phase change at atmospheric pressure for nitrogen and hydrogen. Pressure transducer measurements for the inner and shroud systems were constant throughout the atmospheric conditions.

Comparing the total equilibrium heat leak values of Table IV with the theoretical computer values listed on page 50, suggests that the theoretical conduction factors are low. This is particularly evident for the LN$_2$ tests and is attributed to differences in bumper contact caused by the thermal contraction discussed in Section 3.2.3.

4.6 Pressure Build-Up Tests

After completion of the vented heat leak tests described in the foregoing Section 4.5, the vacuum jacketed valves shown on page 72 were disconnected from the inner and shroud fill ports and the permanent tankage system plumbing re-connected. The piping between the inner and shroud fill valves was lagged as shown at left to prevent excessive boil-off and external ice formation during cryogen fill.

4.6.1 Tankage System Fill

To perform the tankage system pressure build-up tests described below, the unit was filled with LH$_2$ and GHe using the fill procedure described in Section 4.4.

A schematic showing the equipment used to achieve this fill procedure is presented in Figure 29.

Helium gas supply to the inner vessel flowed directly from the gas storage bottles, or via the air driven gas pump shown at left, depending on the pressure differential between the supply and dewar. Pre-cooling of the GHe entering the dewar was carried out using a LN$_2$ heat exchanger in which the liquid level was maintained sufficient to submerge the heat exchanger coil.

Throughout the inner vessel fill, a steady flow of LH$_2$ to the shroud was maintained to replenish evaporated hydrogen caused by the incoming warmer helium. A simple method of ascertaining when the shroud is full
HYDROGEN/HELIUM DEWAR FILL SCHEMATIC

FIGURE 29.
was discovered during the fill procedure by watching the control panel voltage reading for PRT #3. When the shroud liquid level reaches this PRT probe, a sudden decrease in voltage output occurs, the converse occurring when the liquid level in the shroud falls off.

Venting of the hydrogen gas through the remote vent stack was accelerated during fill by maintaining a nitrogen purge to the stack throughout.

The view at the left shows the 1000 liter LH₂ and LN₂ supply dewars used during the fill procedure. Vacuum jacketed piping shown at left was used to connect the LH₂ dewar to the shroud manual fill valve.

4.6.2 Non-Vapor Cooled Pressure Build-Up Test

After completion of the dewar LH₂ and GHₑ fill procedures, the W.T.M. equipment described in Section 4.5 and illustrated in Figure 18 was connected to the shroud manual vent valve. In this manner the rate of hydrogen evaporation from the shroud could be measured in the same way as during the vented heat leak tests.

Helium gas temperatures and pressures inside the inner vessel were continually monitored throughout these tests using the temperature and pressure sensor readouts on the control panel voltmeter.

On completion of system fill, and after the hydrogen and helium fill lines had been disconnected, a decrease in the inner vessel pressure was observed over a period of 9 hours during which time the pressure fell from 2000 psig to 1850 psig and then stabilized at this pressure. This pressure decay was attributed to stratification taking place within the inner vessel.

Because of the limited LH₂ supply it was decided to perform the non-vapor cooled pressure build-up test from 1850 psig up to the inner PRV blow-off pressure.

The results of the non-vapor cooled pressure build-up test are presented in Figure 30. These curves show that the helium pressure rises at an increasing rate as the shroud liquid level decreases.

The area under the shroud heat interception curve is equivalent to a total heat input of 586 BTU which suggests that the total volume of LH₂ evaporated was 0.7 cubic feet. This infers that 0.24 ft.³ of LH₂ was evaporated during the 9 hour stratification period when the inner pressure fell from 2000 to 1850 psig.

When the inner vessel GHₑ pressure reached 2600 psig the inner PRV opened but failed to re-seat resulting...
NAS 9-7337 HELIUM SHROUD TANKAGE
INNER VESSEL GHe PRESSURE-BUILD-UP TEST
NON-VAPOR COOLED

FIGURE 30

INNERS VESSEL PRESSURE (PSIG)

ENVIRONMENTAL TEMP

INNER GHe PRESSURE

SHROUD HEAT INTERCEPTION

INSTANTANEOUS W.T.M. READING

AVERAGE W.T.M. READING

STANDBY TIME (HOURS)

ENVIRONMENTAL TEMP (FT/L)
in a total loss of the stored gas. The relief valve was dismantled and examination of the seat revealed the dirt particles shown at left. Slight leakage problems associated with the solenoid-operated supply and vent valves were also attributed to dirt particles in the inner vessel system.

As a result of the dirt ingress to the high pressure system, it was decided to repair the inner PRV and replace it on the system for the remainder of the testing program but prevent its opening unless an emergency should arise. The solenoid valves were scheduled for examination and repair after completion of the pressure build-up and system flow tests.

To prevent future valve failures caused by dirt particles, a 20 micron filter was introduced into the inner fill line as shown in Figure 7. An industrial filter was also incorporated between the gas pump and dewar during filling as shown in Figure 29.

4.6.3 Vapor Cooled Pressure Build-Up Test

To perform the vapor cooled pressure build-up test the dewar was re-filled with hydrogen and helium in the same manner as described in Section 4.6.1. When the helium pressure reached 2000 psig with a full LH₂ shroud, a 5 hour stabilization period was required before an equilibrium state was achieved. During this period, the GHe pressure decayed and was replenished by additional helium supply while maintaining LH₂ flow to the shroud. The overall fill time required to achieve a stable inner vessel pressure of 2000 psig at -423°F was 15 hours.

Figure 31 shows the results of the vapor cooled pressure build-up test from the point of stable equilibrium at 2000 psig until the inner vessel pressure reached 2500 psig when the test was terminated by performing the system flow tests described later. An initial increase in the shroud heat interception during the first 25 hours of the test can be explained by the slight warming of the vapor cooled shield after cessation of LH₂ flow through the vapor cool system.

The most significant aspect of this test was the constant inner vessel pressure maintained during the first 57 hours of testing while the shroud liquid level was high. This phenomena indicates that during this period all environmental heat leak into the dewar
NAS 9 - 7337 HELIUM SHROUD TANKAGE
INNER VESSEL GHe PRESSURE-BUILD-UP TEST
VAPOUR COOLED

INNER VESSEL PRESSURE (PSIG)

SHROUD HEAT INTERCEPTION

ENVIRONMENTAL TEMP T_e

INSTANTANEOUS W.T.M. READING

AVERAGE W.T.M. READING

PRESSURE

STANDBY TIME (HOURS)

FIGURE 31

A5226-69
is absorbed by the shroud fluid. It also confirms earlier predictions that a shrouded system can provide indefinite standby capabilities for the primary fluid if the shroud is replenished before the inner vessel PRV opens.

Referring to both Figures 30 and 31 for the non-vapor cooled and vapor cooled tests, respectively, it will be observed that the total equilibrium heat leak values are greater than the values of Table IV obtained from the LH2 vented heat leak tests. In the case of the pressure build-up tests, the total environmental heat leak is the same as the rate of evaporation from the shroud while the inner vessel pressure is constant.

NASA-MSC (5) reported a similar phenomena during testing of the NAS9-4634 shrouded unit although their comparison was between the total heat leak of a Hydrogen Shrouded/empty inner vessel unit and a hydrogen shrouded/pressurized helium inner vessel unit.

A summary of the total equilibrium heat leaks into the dewar for the LH2 temperature conditions applicable to the present contract are given below.

<table>
<thead>
<tr>
<th>TYPE of TEST</th>
<th>TOTAL EQUILIBRIUM HEAT LEAK [BTU/HR]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LH2 VENTED HEAT LEAK TEST</td>
</tr>
<tr>
<td>Non-Vapor Cooled</td>
<td>12.6</td>
</tr>
<tr>
<td>Vapor Cooled</td>
<td>8.7</td>
</tr>
</tbody>
</table>

A possible explanation for the above increased heat leak values during the pressure build-up tests is the increase in radial bumper contact area due to the following two factors.

(i) The inner vessel/shroud assembly weight increases by an amount equal to the difference in weight between a LH2 filled inner vessel and a GHe filled inner vessel at 2000 psia. This weight increase amounts to 29 lbs, which would be shared among the lower three bumpers.

(ii) An increase in the pressure vessel diameter caused by the GHe pressure can cause increases
in bumper contact area depending on the degree of pre-load defined by Section 3.2.3. An indication of this effect can be ascertained by applying expression (3.35) to the pressurized and unpressurized LH\textsubscript{2} temperature conditions.

Contact will be lost with the upper bumpers during the LH\textsubscript{2} vented heat leak tests if the pre-load is less than 7,500 lbs, whereas a pre-load of 4,700 lbs. is required to maintain contact during the pressure build-up tests. This means that if contact with the upper bumpers was just lost during the LH\textsubscript{2} vented heat leak tests, an additional 2,800 lbs. could be shared among all the bumpers (i.e., 467 lbs. per bumper) during the LH\textsubscript{2} shrouded pressure build-up tests.

The analysis discussed in Section 3.2.3 was completed after the assembly of the dewar and consequently the degree of pre-load applied to the unit is not known.

4.7 System Flow Test

When the vapor cooled pressure build-up test reached a GH\textsubscript{3} pressure of 2,500 psig, a system flow test was performed to ascertain the mass flow capabilities of the unit.

The solenoid-operated supply valve shown in Figure 7 was opened for approximate 10 second intervals and then closed again. Stop-watch readings of the actual flow time and weigh scale and inner vessel pressure readings were noted after each flow period.

Differences in the weigh scale readings at the start and finish of each flow period were used to calculate the mean mass flow rate over the interval. These flow rates together with the GH\textsubscript{3} pressure decay and temperature curves are presented in Figure 32. The instantaneous mass flow rate immediately after valve opening is the maximum flow capability of the system and was linearly interpolated from the readings as 0.8 LBS/SEC.

Throughout the system flow test readings were taken of the GH\textsubscript{3} temperature in the inner vessel using the voltmeter readouts from PRT #1 and 2. Transient temperature measurements were not possible during
NAS 9 - 7337 HELIUM SHROUD TANKAGE
GHe SYSTEM FLOW CHARACTERISTIC & PRESSURE DECAY PROFILE

GHe MASS FLOW (LBS/SEC)

TIME AFTER VALVE OPENING (SECS)

FIGURE 32

A5226-70
the initial phase of the flow test because of equipment limitations but the trend toward helium liquefaction due to isenthalpic expansion is apparent from the temperature curve of Figure 32.

4.8 Vibration Testing

On completion of the thermal testing the shroud tankage was shipped to the Martin Marietta Corporation's test facilities in Denver, Colorado to carry out specified vibration testing.

It was originally agreed between Bendix and NASA-MSC, Houston to perform the tests with two different cryogen storage conditions, namely:

(i) A LN$_2$ filled shroud and empty inner vessel
(ii) A LH$_2$ filled shroud with 50 lbs. of GHe in the inner vessel at 2000 psia.

However, the dangers associated with the hydrogen embrittlement of Inconel 718 material which were prevalent at the time of testing dictated the cancellation of condition (ii) above.

Sinusoidal and random input vibrations described below were therefore performed with a LN$_2$ filled shroud and empty inner vessel (i.e. empty in the sense that it had negligible mass although it contained a small purge of helium gas) which closely simulates the total mass of an actual hydrogen/helium system. By this means the dynamic response of the dewar assembly could be closely simulated without the risk of permanent damage to the unit.

Levels of vibration input excitation along the three principal axes of the tank shown in Figure 9 were chosen and agreed between Bendix and NASA-MSC based on the actual LM input requirements and the mount carriage design limitations discussed in Section 3.1.4. These agreed levels are illustrated in Figures 33 and 34 for the sinusoidal and random spectrums respectively. The anticipated resonance regions shown on Figures 33 and 34 were calculated using the theoretical approach discussed in Section 3.2 taking into account the effect of the shock mounts and assuming insufficient mount carriage pre-loading to prevent separation.

4.8.1 Description of Test Equipment

Two magnesium vibration fixtures were utilized for the
FIGURE 33 — SINUSOIDAL INPUT

INPUT AMPLITUDE [INCHES]

0.1
0.03
0.01
0.001

FREQUENCY [c. p. s.]

1 10 30 100

LEM INPUT REQUIREMENT

PROPOSED INPUT FOR RESONANCE SEARCH

ANTICIPATED RESONANCE REGIONS

LAUNCH & BOOST

A5226-2
FIGURE 34 — RANDOM INPUT

ANTICIPATED RESONANCE REGIONS

FREQUENCY [c.p.s.]
vibration testing. The vertical X axis fixture of Figure 35 consisted of three pairs of vertical pillars each pillar coinciding with one of the shroud tank shock mounts. The flat plate type of fixture shown in Figure 36 with a clearance hole bored through its center was used for testing along the Y and Z axes. These test fixtures afford excitation through the center of gravity of the tankage assembly in all axes.

A Ling model A-249 shaker in conjunction with a Ling model PP175/240 amplifier was used to excite the system, control being exhibited by one of the accelerometers attached to the fixture at the mounting point. Vibration levels were controlled with a spectral dynamics model 105-A sine servo in the case of the sinusoidal tests and a Ling model ASDE-80 automatic equalizer for the random vibration tests.

Both input and output vibration levels were measured using accelerometers positioned as illustrated in Figure 35, 36, 37 and 38.

During all the vibration tests performed on the unit the ion pump was disconnected and the ion pump magnet removed from its support bracket.

4.8.2 Input Excitation Levels

Table V summarizes the actual vibration input levels achieved during the testing.

Testing commenced with sinusoidal excitation along the X axis, performing a series of sweeps of progressively increasing amplitude at a constant sweep rate of 3 octaves per minute. At a double amplitude of 0.04 inches audible movement of the inner vessel/shroud assembly within the dewar was causing concern particularly at the higher frequency end of the range. A final sinusoidal sweep along the X axis at the proposed level of 0.06 inches double amplitude was terminated at 22 Hz to avoid damage to the unit.

Random input along the X axis was performed in a continuous series of 3 sweeps progressing in RMS G level from 1 through 3 G for a total time of 7 1/2 minutes or approximately 2 1/2 minutes at each level. A power spectral density plot of the actual random input is shown in Figure 39. These input levels demonstrated the ability of the shroud tankage assembly to meet the proposed random input requirements completely.
FIGURE 35 X Axis Vibration Test Fixture
FIGURE 37 Close-up of shroud unit mounted in Y and Z axis test fixture
FIGURE 38
ACCELEROMETER LOCATIONS
### TABLE V
VIBRATION INPUT LEVELS

<table>
<thead>
<tr>
<th>Date</th>
<th>Run</th>
<th>Type</th>
<th>Axis</th>
<th>Level</th>
</tr>
</thead>
<tbody>
<tr>
<td>8/27/68</td>
<td>1</td>
<td>5-100 Hz Sine</td>
<td>X</td>
<td>0.006&quot; Displacement, 1.5G's</td>
</tr>
<tr>
<td>8/27/68</td>
<td>2</td>
<td>5-100 Hz Sine</td>
<td>X</td>
<td>0.01&quot; Displacement, 1.5G's</td>
</tr>
<tr>
<td>8/27/68</td>
<td>3</td>
<td>5-100 Hz Sine</td>
<td>X</td>
<td>0.02&quot; Displacement, 1.5G's</td>
</tr>
<tr>
<td>8/27/68</td>
<td>4</td>
<td>5-100 Hz Sine</td>
<td>X</td>
<td>0.04&quot; Displacement, 1.5G's*</td>
</tr>
<tr>
<td>8/27/68</td>
<td>5</td>
<td>Random</td>
<td>X</td>
<td>1 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/27/68</td>
<td>6</td>
<td>Random</td>
<td>X</td>
<td>2 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/27/68</td>
<td>7</td>
<td>Random</td>
<td>X</td>
<td>3 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/27/68</td>
<td>8</td>
<td>5-22 Hz Sine</td>
<td>X</td>
<td>0.06&quot; Displacement*</td>
</tr>
<tr>
<td>8/28/68</td>
<td>9</td>
<td>5-100 Hz Sine</td>
<td>Z</td>
<td>0.01&quot; Displacement 1.5G's</td>
</tr>
<tr>
<td>8/28/68</td>
<td>10</td>
<td>5-100 Hz Sine</td>
<td>Z</td>
<td>0.02&quot; Displacement, 1.5G's</td>
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<tr>
<td>8/28/68</td>
<td>11</td>
<td>5-100 Hz Sine</td>
<td>Z</td>
<td>0.04&quot; Displacement, 1.5G's*</td>
</tr>
<tr>
<td>8/28/68</td>
<td>12</td>
<td>Random</td>
<td>Z</td>
<td>1 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/28/68</td>
<td>13</td>
<td>Random</td>
<td>Z</td>
<td>2 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/28/68</td>
<td>14</td>
<td>Random</td>
<td>Z</td>
<td>3 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/28/68</td>
<td>15</td>
<td>5-100 Hz Sine</td>
<td>Y</td>
<td>0.01&quot; Displacement, 1.5G's</td>
</tr>
<tr>
<td>8/28/68</td>
<td>16</td>
<td>5-100 Hz Sine</td>
<td>Y</td>
<td>0.02&quot; Displacement, 1.5G's</td>
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<tr>
<td>8/28/68</td>
<td>17</td>
<td>5-100 Hz Sine</td>
<td>Y</td>
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</tr>
<tr>
<td>8/28/68</td>
<td>18</td>
<td>Random</td>
<td>Y</td>
<td>1 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/28/68</td>
<td>19</td>
<td>Random</td>
<td>Y</td>
<td>2 Grms 2.5 Minutes</td>
</tr>
<tr>
<td>8/28/68</td>
<td>20</td>
<td>Random</td>
<td>Y</td>
<td>3 Grms 2.5 Minutes</td>
</tr>
</tbody>
</table>

* Transmissibility Plots Made
FIGURE 39  Power spectral density plot of actual random input for X, Y and Z axes.
As shown by Table V, the next series of vibration tests were performed by exciting the unit along the Z axis. In this case, the sinusoidal input was terminated after the 0.04 inch double amplitude sweep because the audible response of the unit was causing concern at the lower end of the frequency range in addition to the higher frequency problems encountered with the X axis.

Again the random input requirements were successfully achieved along the Z axis using the same continuous sweep technique as was used for the X axis testing.

The vibration testing was completed by exciting the unit along the Y axis of Figure 9. These tests resulted in a similar unit response to that experienced with the Z axis. At 0.04 inch double amplitude sinusoidal input, the shroud unit response at frequencies above 10 Hz was at the stage where further excitation might have caused permanent damage to the unit.

Random vibration input requirements for testing along the Y axis were fully achieved.

The results of the vibration input levels in relation to the proposed input requirements can best be summarized by referring to Figures 40, 41 and 34. For the levels achieved it was found that the resonance regions shown in Figures 33 and 34 could be ignored principally because of the sweep rate used for the testing.

Figure 40 shows that only a small area in the 22 - 25 Hz frequency band was unattainable for excitation along X axis.

The area between the 0.02" and 0.03" amplitude regions (or 0.04" and 0.06" double amplitude regions) shown in Figure 41 signifies the area requiring further investigation for vibratory motion along the Y and Z axes.

Actual random vibration inputs exceeded the proposed levels by virtue of the shaded resonance regions shown in Figure 34.

The results of the vibration input levels in relation to the proposed input requirements can best be summarized by referring to Figures 40, 41 and 34. For the levels achieved it was found that the resonance regions shown in Figures 33 and 34 could be ignored.
ACTUAL SINUSOIDAL INPUT VIBRATION SPECTRUM
FOR X AXIS EXCITATION

INPUT AMPLITUDE [INCHES]

0.1
0.03
0.02
0.01
0.001

FREQUENCY [c. p. s.]
1 10 25 35 100

AGREED LEVELS
NOT ACHieved
EXCEEDS AGREED LEVELS

FIGURE 40

A5226-71
ACTUAL SINUSOIDAL INPUT VIBRATION SPECTRUM
FOR Y AND Z AXES EXCITATION

FIGURE 41.
principally because of the sweep rate used for the testing.

Figure 40 shows that only a small area in the 22-25 Hz frequency band was unattainable for excitation along the X axis.

The area between the 0.02" and 0.03" amplitude regions (or 0.04" and 0.06" double amplitude regions) shown in Figure 41 signifies the area requiring further investigation for vibratory motion along the Y and Z axes.

Actual random vibration inputs exceeded the proposed levels by virtue of the shaded resonance regions shown in Figure 34.

4.8.3 Sinusoidal Output Response Levels

Accelerometer #1, positioned on the fixtures as shown in Figures 36 and 38, was used as the datum input readout for measuring the gain response of the tankage assembly. The accelerometer readings for the tests donated by an asterisk in Table V were recorded on magnetic tape and the data analyzed in the form of transmissibility plots by comparing each accelerometer output with the output from accelerometer #1.

In all the cases analyzed, the output response was greatest for the accelerometers attached to the mount carriage adjacent to the lower radial bumper load points. Typical response plots for these maximum gain regions are shown in Figures 42, 43, 44 and 45.

Figure 42 shows two distinct resonance regions, the lower resonance occurring within the 15-30 C/S frequency band and the upper less pronounced resonance at about 90 C/S. These resonance points agreed reasonably well with the theoretical values of 27.5 C/S and 112 C/S predicted by the theory of Section 3.2 for excitation along the X axis.

Figures 44 and 45 for excitation along the Z and Y axes, respectively, show that the response of the unit is less pronounced at the higher end of the frequency range than for the X axis. The transmissibility plots indicate resonant frequencies of 12 C/S for displacements along the Z axis and 12 1/2 C/S for Y axis excitation. A second higher resonance point probably occurs at a frequency greater than the sinusoidal input upper limit of 100 C/S.
All the transmissibility plots presented in the text exhibit the same non-linear response superimposed on the sinusoidal wave particularly at the resonance points. This is indicative of impact loadings occurring within the dewar caused by the thermal separation discussed in Section 3.2.3.

After completion of the foregoing vibration testing the unit was returned to Instruments & Life Support Division of The Bendix Corporation where vented heat leak tests numbers 6 and 7 of Table III were performed.

As shown by Figures 19 and 21, the slopes of the total vented loss lines are sensibly the same indicating the same heat leak rate before and after vibration testing.
Note: Multiply the frequency by 0.1 for actual value.

FIGURE 42 Transmissibility Plot – 0.04° DA along X Axis.
TRANSMISSIBILITY PLOT

Note: Multiply the frequency by 0.1 for actual value.

FIGURE 43  Transmissibility Plot – 0.06" DA along X Axis.
FIGURE 44 Transmissibility Plot – 0.04" DA along Z Axis.

Note: Multiply the frequency by 0.1 for actual value.
Note: Multiply the frequency by 0.1 for actual value.

FIGURE 45 Transmissibility Plot – 0.04" DA along Y Axis.
SECTION V

CONCLUSIONS & RECOMMENDATIONS

5.1 Conclusions

The work performed under this contract and described in the foregoing sections confirms the recommendations put forward in the NAS9-4634 contract. A hydrogen shrouded helium storage dewar can be designed and fabricated to store any quantity of GHₑ at densities greater than the present lunar module SHₑ pressurization system. This increased density allows the storage of the same pressurant quantity in a smaller tankage envelope.

An indefinite pre-launch standby capability is available with a shroud tank without loss of primary stored fluid provided the shroud liquid is replenished at certain intervals. The unit constructed under the present contract to approximate LM descent propulsion system requirements would require LH₂ replenishment approximately every 100 hours of ground standby without loss of GHₑ. This ground standby capability can be increased by a factor of 3 by adopting one of the shroud geometries discussed in Appendix II.

During initial fill of the unit, a stabilization period of about 5 hours is required after reaching system pressure to ensure a uniform LH₂ temperature distribution throughout the GHₑ mass stored within the inner vessel.

A total ground fill time of 15 hours is required for the unit although this period may be greatly reduced when using a large capacity high pressure helium gas source.

An equilibrium vented heat leak of 8.7 BTU/HR was achieved when the inner and shroud vessels were filled with LH₂ at ambient pressure. This heat leak figure increased to 12.4 BTU/HR when the inner vessel was filled with GHₑ at 2000 psig.

The vapor cooling system attached to the inner radiation shield produced an environmental heat leak reduction of between 20% and 30% over direct venting for storage at LH₂ temperatures.

System flow tests carried out on the helium shroud tankage show a maximum flow capability of 0.8 LBS/SEC.
and a mean flow of 0.04 LBS/SEC over the pressure range 2500 - 0 psig.

Modified LM spectrum vibration inputs showed that the helium tankage described in this report satisfied the requirements completely for random input excitation. Certain areas of the sinusoidal input spectrum require additional analysis and design study to upgrade the bumper suspension system to meet anticipated lunar flight conditions.

5.2 Recommendations

The development work carried out on the NAS9-4634 and current programs have clearly shown the feasibility of an integral shrouded cryogenic storage system. Application of this type of unit to future space storage dewars should be seriously considered for life support and reactant supply systems in addition to propellant pressurization systems. In this respect, the shrouding of ignitable cryogens by inert gases during pre-launch standby is worth consideration.

It is recommended that the hydrogen shrouded helium pressurization system described in this report be upgraded to meet the present lunar module pre-launch and on-board propellant tankage standby requirements. This upgrading should concentrate particularly on providing a weight optimized unit which satisfies the anticipated LM vibration spectrum requirements.

To achieve this upgrading, the adoption of one of the two alternative shroud geometries, shown by Appendix II to considerably extend standby capabilities, is recommended.

Adoption of the concentric shroud annulus should be carried out in conjunction with the use of a cryoformed stainless steel inner vessel which would provide a 25% reduction of the pressure vessel weight. This annulus geometry would increase the overall tankage envelope from 29 3/4 inches to 31 inches resulting in an approximate overall tankage weight reduction of 20% and a mission standby capability of 125 hours.

The glass-fiber reinforced (GFR) oblate spheroidal inner vessel with metal liner can provide an inner pressure vessel weight reduction of between 25 and 50% depending on the liner material. The possible disadvantages of vacuum outgassing and plating associated with the composite surrounding this type of vessel do not apply to the shrouded dewar concept.
Adopting aluminum for the GFR pressure vessel liner would allow the use of this material for the shroud which would save a further 30% in shroud weight.

The shroud annulus geometry associated with a GFR inner vessel would again increase the overall tankage envelope from 29 3/4 inches to 31 inches resulting in an overall tankage weight reduction of 30% and a mission standby capability of approximately 340 hours.

The above increased tankage envelope size compares favorably with the existing LM supercritical helium pressurant tanks which have an envelope of 33 inches.

It is further recommended that improved structural and dynamic performance of the shrouded unit should be pursued by extending the analysis contained in Section III of the main report. This would be specifically aimed at the effect of various pre-loading techniques on eliminating the non-linear shock loads experienced by the dewar during vibration.
REFERENCES


APPENDIX I

HEL IUM SHROUD TANKAGE LOG
<table>
<thead>
<tr>
<th>DATE</th>
<th>COMPONENT</th>
<th>LOG</th>
<th>SIGNATURE</th>
</tr>
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<tbody>
<tr>
<td>11/20/67</td>
<td>Shroud</td>
<td>Hemispheres adopted for Tankage System are:</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N's 1 &amp; 2 - Inconel 718 material</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 1 - 3 Processing anneals</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 2 - 3 Processing anneals</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Evacuated and Leak checked before machining.</td>
<td></td>
</tr>
<tr>
<td>12/7/67</td>
<td>Shield #1</td>
<td>Hemispheres adopted for Tankage System are:</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N's 4 &amp; 5 - Aluminum 6061-0 material</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 4 - 4 processing anneals at 650°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 5 - 5 processing anneals at 650°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td>12/22/67</td>
<td>Shield #2</td>
<td>Hemispheres adopted for Tankage System are:</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N's 2 &amp; 8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 2 - material 6061-0 Aluminum, 5 processing anneals at 650°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 8 - material 3003-0 Aluminum, 2 processing anneals at 775°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td>12/22/67</td>
<td>Outer Shell</td>
<td>Hemispheres adopted for Tankage System are:</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N's 4 &amp; 5 - Alum. 6061-0 material</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 4 - 4 processing anneals at 650°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 5 - 4 processing anneals at 650°F for 10 minutes.</td>
<td></td>
</tr>
<tr>
<td>1/5/68</td>
<td>Pressure Vessel</td>
<td>Hemispheres adopted for Tankage System are:</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N's 1 &amp; 3 - Inconel 718 material</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>S/N 1 &amp; 3 - 2 processing anneals at 1950°F for 1/2 hour.</td>
<td></td>
</tr>
<tr>
<td>1/5/68</td>
<td>Pressure Vessel</td>
<td>S/N 1 &amp; 3 - Hemispheres annealed at 1850°F for 15 minutes and processed in accordance with Bendix MCI-168, P.O. SP-C-1252 and S.F. 87638.</td>
<td></td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td>SIGNATURE</td>
</tr>
<tr>
<td>----------</td>
<td>-------------------</td>
<td>-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>2/1/68</td>
<td>Pressure Vessel</td>
<td>S/N 1 &amp; 3 - Hemispheres Alkaline Electro cleaned for 1 - 5 minutes after machining prior to welding. Boss fittings also electro cleaned.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/5/68</td>
<td>Pressure Vessel</td>
<td>S/N's 1 &amp; 3 - Hemispheres heli-arc welded together with boss fittings to form spherical pressure vessel. Radiographic and dye-penetrant checked.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/15/68</td>
<td>Shield #1</td>
<td>Hemispheres etch cleaned, de-oxidized, placed in Zinc dip and then electro-plated with copper for 15 minutes.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/16/68</td>
<td>Pressure Vessel</td>
<td>Vessel pressurized to 1000 psig while in annealed condition to extend pole dimensions.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/16/68</td>
<td>Pressure Vessel</td>
<td>Vessel Alkaline Electro-cleaned for 6 minutes and coated with &quot;Turco&quot; product prior to Age Hardening</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/17/68</td>
<td>Pressure Vessel</td>
<td>Vessel Age Hardened at 1360°F for 10 hours + 1175°F for 10 hours and then air cooled. Vessel purged with Argon gas throughout heat treat.</td>
<td>W.B.Ba:Bd</td>
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<tr>
<td>2/20/68</td>
<td>Pressure Vessel</td>
<td>Vessel Acid Pickled after age hardening.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/21/68</td>
<td>Shield #1</td>
<td>Copper vapor cooling tubing soldered to hemispheres using SN 50 Alloy, Resin Five Flux, 66-core Flux - Poly Pole.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/22/68</td>
<td>Pressure Vessel</td>
<td>Vessel pressurized to 2,800 psig, at which pressure, one of the fitting &quot;O&quot; rings blew. Proof pressure test was then discontinued.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>2/29/68</td>
<td>Pressure Vessel</td>
<td>Vessel pressurized to 2,500 psig and then released to check fittings. Repressurized to 2,300 psig when &quot;O&quot; ring leakage occurred. Proof pressure test was then discontinued.</td>
<td>W.B.Ba:Bd</td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td></td>
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<tr>
<td>--------</td>
<td>----------------------------</td>
<td>-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>3/1/68</td>
<td>Temperature Probe Assy.</td>
<td>Temperature Sensor leads brazed into P.V. vent fitting and then removed for modification. One of the temperature sensor leads broken on removal. Repaired sensor to be fitted into shroud vent fitting and new sensor replaced in P.V. fitting.</td>
<td></td>
</tr>
<tr>
<td>3/12/68</td>
<td>Pressure Vessel (Proof Test)</td>
<td>Vessel pressurized to Proof Pressure of 3,860 psi and then released. Vessel again pressurized to 3,860 psi in steady increments measuring diametral change throughout, Stress/Strain curve was linear throughout test. Proof test performed with fill and vent fittings and temperature probe assy. welded into vessel.</td>
<td></td>
</tr>
<tr>
<td>3/12/68</td>
<td>Shroud</td>
<td>Lower shroud evacuated to leak check fill fitting. Shroud hemisphere tended to buckle in region of dimples. Evacuation discontinued at this stage and leak checks on shroud fill and vent fittings carried out by locally isolating fittings.</td>
<td></td>
</tr>
<tr>
<td>3/13/68</td>
<td>Pressure Vessel</td>
<td>Vessel evacuated before and after Proof testing to Helium leak check the welds.</td>
<td></td>
</tr>
<tr>
<td>3/18/68</td>
<td>Pressure Vessel</td>
<td>Small indentation observed on radiused bend of inner fill tube of closure fitting.</td>
<td></td>
</tr>
<tr>
<td>3/27/68</td>
<td>Pressure Vessel/Shroud Assy.</td>
<td>Shroud evacuated to leak check shroud girth welds. Fault in vacuum pump system caused test to be postponed.</td>
<td></td>
</tr>
<tr>
<td>3/27/68</td>
<td>Temp. Sensor Leads for Inner Vessel</td>
<td>Loss of ground connection during heliarc welding of shrouds caused arcing between cart and leads. Electrically checked leads as O.K. St. St. sleeve brazed over damaged region.</td>
<td></td>
</tr>
<tr>
<td>3/27/68</td>
<td>Shroud</td>
<td>Prior to welding, shroud hemispheres were alkaline electro-cleaned for a total of 9 minutes each.</td>
<td></td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td>SIGNATURE</td>
</tr>
<tr>
<td>--------</td>
<td>----------------------------------</td>
<td>----------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>4/4/68</td>
<td>Shroud/P.V. Assembly</td>
<td>Shroud Evacuated Three (3) times during helium leak check after welding to pressure vessel.</td>
<td></td>
</tr>
<tr>
<td>4/5/68</td>
<td>Outer Shell</td>
<td>S/N 4 - Lower shell with burst disc welded in.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 5 - Upper shell with Ion Pump transition tube welded in.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Both shells evacuated to leak check welds.</td>
<td></td>
</tr>
<tr>
<td>4/5/68</td>
<td>Outer Shell</td>
<td>Outer shell hemispheres etch cleaned prior to copper plating.</td>
<td></td>
</tr>
<tr>
<td>4/5/68</td>
<td>Shield #1</td>
<td>Hemispheres etch cleaned before plating.</td>
<td></td>
</tr>
<tr>
<td>4/5/68</td>
<td>Shield #2</td>
<td>Hemispheres etch cleaned before plating.</td>
<td></td>
</tr>
<tr>
<td>4/13/68</td>
<td>P.V./Shroud Assembly</td>
<td>Shroud outside surface silver vapor plated at 300°F in accordance with ES 647.</td>
<td></td>
</tr>
<tr>
<td>4/13/68</td>
<td>Shield #1</td>
<td>Inside and outside surfaces silver vapor plated at 300°F in accordance with ES 647.</td>
<td></td>
</tr>
<tr>
<td>4/13/68</td>
<td>Shield #2</td>
<td>Inside and outside surfaces silver vapor plated at 400°F in accordance with ES 647.</td>
<td></td>
</tr>
<tr>
<td>4/13/68</td>
<td>Outer Shell</td>
<td>Shell hemispheres copper plated on inside surface at 400°F in accordance with ES 647.</td>
<td></td>
</tr>
<tr>
<td>4/17/68</td>
<td>P.V./Shroud/Shield #1 Assy.</td>
<td>Shroud evacuated to leak check vapor cooling system.</td>
<td></td>
</tr>
<tr>
<td>4/31/68</td>
<td>Burst Disc</td>
<td>During helium leak checking of dewar Assy, leakage detected on blow-out patch of Burst Disc. Leakage sealed with Epoxy.</td>
<td></td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td></td>
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<tr>
<td>----------</td>
<td>----------------------</td>
<td>----------------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>5/21/68</td>
<td>Dewar Assy</td>
<td>Dewar evacuated and Baked Out at 275°F for 21 days and successfully tipped-off. Shroud filled with LN₂ to further reduce vacuum in annulus. Pressure Vessel fill and vent tubes iced up at this stage.</td>
<td></td>
</tr>
<tr>
<td>6/7/68</td>
<td>to 6/20/68</td>
<td>Dewar Assy.</td>
<td></td>
</tr>
<tr>
<td>6/7/68</td>
<td>to 6/20/68</td>
<td>Shroud &amp; Inner Vessels filled with LN₂ at ambient pressure and non-vapor cooled and vapor cooled vented heat leak tests performed. Vessels emptied with gaseous nitrogen purge after testing.</td>
<td></td>
</tr>
<tr>
<td>7/3/68</td>
<td>to 7/17/68</td>
<td>Dewar Assy.</td>
<td></td>
</tr>
<tr>
<td>7/19/68</td>
<td>Shroud</td>
<td>Shroud &amp; Inner Vessels filled with LH₂ at ambient pressure and non-vapor cooled and vapor cooled vented heat leak tests performed. Vessels emptied with gaseous Helium purge after testing.</td>
<td></td>
</tr>
<tr>
<td>7/20/68</td>
<td>Inner Vent</td>
<td>Faulty latch mechanism adjusted by rotating solenoid clockwise by approximately 5°.</td>
<td></td>
</tr>
<tr>
<td>7/20/68</td>
<td>Solenoid Valve</td>
<td>Inner vessel filled with LN₂ at ambient pressure overnight.</td>
<td></td>
</tr>
<tr>
<td>7/21/68</td>
<td>Dewar Assy</td>
<td>Inner vessel emptied of LN₂ with gaseous helium purge via LN₂ heat exchanger. Inner vessel pressure raised to 200 psig. Shroud filled with LH₂ during which inner vessel pressure fell to 50 psig. Time to fill shroud approx. 30 minutes.</td>
<td></td>
</tr>
<tr>
<td>7/21/68</td>
<td>Dewar Assy</td>
<td>5 lbs. of gaseous He pumped into Inner Vessel to ascertain flow requirements during fill. Shroud topped-off with LH₂ during this fill process. Maximum pressure reached in Inner 250 psig.</td>
<td></td>
</tr>
<tr>
<td>7/21/68</td>
<td>Dewar Assy</td>
<td>LH₂ in shroud emptied with GHe and Inner Vessel He vented off and vessel warmed back to ambient overnight.</td>
<td></td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td></td>
</tr>
<tr>
<td>--------------</td>
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<td>-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>7/23/68</td>
<td>Dewar Assy.</td>
<td>Inner Vessel System pressurized to 2,600 psig with Freon and inner pressure sensor calibrated. Solenoid supply valve leaking at 600 psig.</td>
<td></td>
</tr>
<tr>
<td>7/25/68 to</td>
<td>Dewar Assy.</td>
<td>Inner Vessel pre-cooled with LN\textsubscript{2}. Shroud filled with LH\textsubscript{2} at ambient pressure and 30 lbs. (GH_e) pumped into Inner Vessel up to maximum pressure of 2,050 psig. (He) leakage developed in Inner PRV and solenoid valves. Pressure Build-up tests discontinued at this stage and Inner and Shroud Vessels emptied and warmed back to ambient.</td>
<td></td>
</tr>
<tr>
<td>7/26/68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7/31/68 to</td>
<td>Dewar Assy.</td>
<td>Inner Vessel pre-cooled with LN\textsubscript{2}. Shroud filled with LH\textsubscript{2} at ambient pressure and (GH_e) pumped into Inner Vessel. At completion of fill (GH_e) press. - 2,000 psig and (GH_e) temp. = -423°F. After shut-down (GH_e) pressure dropped and stabilized at 1875 psig.</td>
<td></td>
</tr>
<tr>
<td>8/1/68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8/4/68</td>
<td>Dewar Assy.</td>
<td>At 10:00 a.m. LH\textsubscript{2} completely vented from shroud and (GH_e) pressure in Inner Vessel built-up to 2,630 psig. Slight leakage from pipe connections to inner pressure sensor which apparently caused voltmeter reading to deflect to maximum reading for this sensor. Pressure gage connected to system to read 2,630 psig value. At 7:00 p.m. inner PRV opened to relieve pressure but failed to close due to dirt particles and inner vessel pressure fell to 150 psig. Pressure build-up test discontinued.</td>
<td></td>
</tr>
<tr>
<td>8/9/68 to</td>
<td>Dewar Assy.</td>
<td>Inner Vessel pre-cooled with LN\textsubscript{2}. Shroud filled with LH\textsubscript{2} and (GH_e) pumped into Inner Vessel. After pressure reached 2,000 psig system took 5 hours to stabilize at 2,000 psig and -423°F without drop-off in (GH_e) pressure. Vapor cooled pressure-build up test completed when inner pressure reached 2,500 psig. Max. flow test carried out by opening inner vent solenoid valve until inner pressure was zero. Inner and Shroud vessels warmed back to ambient.</td>
<td></td>
</tr>
<tr>
<td>8/14/68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>DATE</td>
<td>COMPONENT</td>
<td>LOG</td>
<td>SIGNATURE</td>
</tr>
<tr>
<td>-----------</td>
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<td>-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>8/26/68</td>
<td>Dewar Assy.</td>
<td>Shroud filled with LN$_2$ and inner vessel purged with 10 psig GHe. Sinusoidal and random input vibration levels applied to the helium tankage assembly along the X, Y, and Z axes. Inner and Shroud Vessels warmed back to ambient before shipping back to Davenport. Slight damage caused to one of the mount carriage legs due to airfreighting from Davenport to Denver.</td>
<td>W.B. Rely</td>
</tr>
<tr>
<td>8/29/68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9/6/68</td>
<td>Dewar Assy.</td>
<td>Shroud and Inner Vessels filled with LN$_2$ at ambient pressure and non-vapor cooled and vapor cooled vented heat leak tests performed. Inner and Shroud Vessels warmed back to ambient.</td>
<td>D.B. Rely</td>
</tr>
<tr>
<td>9/11/68</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9/14/68</td>
<td>Pressure Vessel</td>
<td>Inner Vessel LOX cleaned to MSFC Spec. 164 using UCON Solvent 113-LRI.</td>
<td>W.B. Rely</td>
</tr>
<tr>
<td>9/14/68</td>
<td>Tankage Assy.</td>
<td>Outer shell painted with white speed-enamel Q.D. and air dried. Mount carriage painted with white speed-enamel Q.D. and baked at 225°F for 10 minutes.</td>
<td>W.B. Rely</td>
</tr>
<tr>
<td>9/18/68</td>
<td>Tankage Assy.</td>
<td>Inner Vessel System pressurized with GHe at 2,000 psig at ambient temperature to check for leakages after final assembly.</td>
<td>W.B. Rely</td>
</tr>
</tbody>
</table>
APPENDIX II

SHROUD HEAT INTERCEPTION CHARACTERISTICS - A GENERAL DISCUSSION OF THE EFFECTS OF SHROUD ANNULUS GEOMETRY

A significant factor affecting the standby capabilities of future shrouded dewars is the heat interception characteristics of the shroud fluid. This characteristic has been illustrated by the experimental curves of figures 23, 28, 30 and 31 contained in the main text of this report.

As discussed in Section 11, a tapered shroud geometry was included as part of the dewar design as opposed to the concentric shrouded arrangement used for the NAS9-4634 unit.

Using the oblate spheroidal inner vessel discussed in the recommendations section of this report introduces a third possible shroud annulus geometry.

The following discussion will therefore treat, in a qualitative rather than a quantitative manner, the effects of the above three geometrical annulii on the shroud heat interception characteristics. These three geometries are shown in figure 46.

The experimental results obtained from the thermal test program of section IV show that during the vented heat leak tests the total environmental heat leak is divided between the shroud and inner vessels in continuously varying degrees. This sharing of the heat absorption, however, does not occur when the inner vessel is filled with GHe at pressures above atmospheric.

The discussion will, therefore be limited to the actual pressurant storage system having LH₂ in the shroud and GHe contained within the inner vessel at pressure.

Results of the pressure build-up tests for a hydrogen/helium system show that the liquid level in the shroud falls considerably before a measurable pressure rise occurs in the inner vessel. The area under the shroud heat interception curve of figure 31 indicates an approximate 75% evaporation of shroud LH₂ before the GHe pressure increases.

The dependence of GHe pressure rise on shroud liquid level suggests that it must be a function of the Prandtl and Grashof numbers for the helium or, in other words, the helium's ability to convect the environmental heat back to the shroud fluid. It must also
be a function of the inner vessel surface area exposed to the shroud fluid gas phase.

A qualitative comparison among the three types of shroud annulus geometry shown in figure 46 can therefore be made by considering the respective times required to expose a pre-determined amount of inner vessel surface area.

If the shroud heat interception characteristic curve can be predicted theoretically then the inner vessel pressure rise curve is readily evaluated. The discussion will therefore concentrate on the shroud curve.

Assuming a vapor cooled discrete radiation shielded dewar, the shroud characteristic must have three distinct phases.

(i) A period of constant heat interception $\dot{Q}_{\text{max}}$ during which time the inner vessel pressure rise is negligible. This condition assumes that complete thermal stabilization exists within the dewar before standby commences thus eliminating the heat interception increase shown by the curve of figure 31.

(ii) A period of reducing shroud heat interception $\dot{Q}$ as the LH$_2$ level falls and the inner vessel pressure increases at an increasing rate.

(iii) A final period of zero shroud heat absorption during which time the effect of vapor cooling disappears and the total environmental heat leak increases producing a still more rapid increase in pressure build-up.

Consider the general case of the tapered shroud annulus shown at left which represents the actual shape adopted for the tankage described in the main text.

The time taken from $t = 0$ when the shroud is 100% full until time $t = t_1$, when the liquid level reaches the top of the inner vessel is easily derived as:

$$t_1 = \frac{\pi \rho \Delta h (R_S - R_i - x)}{\dot{Q}_{\text{max}}} \left[ \frac{1}{3} (R_S - R_i - x) (2R_S + R_i + x) - \frac{1}{4} D^2 \right]$$

(1)
where $\rho = \text{LH}_2$ density
$\Delta h = \text{enthalpy change at liquid/gas inter-phase}$
$D = \text{inner vessel boss fitting diameter.}$
$R_S = \text{shroud radius}$
$R_i = \text{inner vessel radius}$
$x = \text{displacement between shroud and inner centers of curvature.}$

Consider now the evaporation of shroud fluid from time $t = t_1$ to any other arbitrary level $u > x$.

At any instant of time $t$ the shroud level will be $u + du$ and after a small interval $dt$ the level will fall a distance $du$ caused by the absorption of an amount of heat $\dot{Q} dt$.

Applying the basic heat equation to the situation shown at left gives the expression

$$\dot{Q} dt = \rho \Delta h \pi \left[ (R_S^2 - u^2) - (R_i^2 - (u - x)^2) \right] du$$

Integrating the above expression between the appropriate limits and introducing the negative sign because of falling level

$$\dot{Q} dt = -\rho \Delta h \pi \int \left[ \frac{u}{R_i + x} \right] [R_S^2 - u^2] - (R_i^2 - (u - x)^2)] du \quad (ii)$$

If the instantaneous heat interception $\dot{Q}$ is assumed constant over the period, which is a reasonable assumption until at least $u = x$ based on the experimental evidence, then putting $\dot{Q} = \dot{Q}_{\text{max}}$ in equation (ii) and solving gives $t = t_1 + \frac{\rho \Delta h \pi R_i}{\dot{Q}_{\text{max}}} \left[ (R_S^2 - R_i^2 + x^2)(R_i + x - u) - ((R_i + x)^2 - u^2)x \right] \quad (iii)$

which is the time required for the shroud cryogen level to fall to any arbitrary level $u$ above the datum line.

Putting $u = x$ in expression (iii) gives the time required for the shroud to become 50% full.

i.e.,

$$t_2 = t_1 + \frac{\rho \Delta h \pi R_i}{\dot{Q}_{\text{max}}} \left[ R_S^2 - R_i^2 - x (R_i + x) \right] \quad (iv)$$
Expression (i) and (iv) can also be applied to the case of a concentric shroud annulus, such as the one used for the NAS9-4634 contract, by simply putting $x = 0$ in these equations.

For comparative purposes within the confines of this discussion, the third possible shroud annulus geometry using an oblate spheroidal inner vessel will be treated by putting $x = -x$ in equations (i) and (iv) derived above. Exact expressions using an elliptical equation for the inner vessel shape can be derived using the same technique as outlined above.

Consider now the evaporation of shroud fluid from time $t = t_2$ to the point where the inner vessel $\text{GHe}$ pressure starts to increase at time $t = t_p$. During this period the heat interception is still constant at $\dot{Q}_{\text{max}}$ by virtue of the assumptions on page .

The integral equation representing the liquid level change shown at left is

$$
\int_{t_2}^{t_p} \dot{Q}_{\text{max}} \, dt = \rho \Delta h \pi \int_{x}^{c} [(R_s^2 - u^2) - (R_i^2 - (u - x)^2)] \, du
$$

which on solving gives

$$
t_p = t_2 + \frac{\rho \Delta h \pi}{\dot{Q}_{\text{max}}} [(R_s^2 - R_i^2 + x^2) (c-x) - x (c^2 + x^2)] \quad (v)
$$

Equation (v) can be used to calculate the standby time required for the hydrogen shrouded helium system to sense any pressure increase inside the inner vessel provided the shroud liquid level $c$ at this stage is known. As mentioned before this level is a function of the inner vessel surface area exposed to shroud gas phase and requires further theoretical and experimental study to resolve the problem.

The next phase of the shroud evaporation cycle is from time $t = t_p$ until time $t = t_3$ when the inner vessel surface is completely exposed. During this period, the shroud heat interception is no longer constant but is a variable function of time.
Let us assume for comparative purposes that this function is given by the following second degree polynomial

\[ \dot{Q} = \alpha + \beta t + \gamma t^2 \]  

(vi)

where \( \alpha, \beta \) and \( \gamma \) are constants to be found from the boundary conditions.

The conditions which the shroud heat interception characteristic must satisfy over the range \( 0 \leq t \leq (t_3 - t_p) \) are as follows:

\[ \dot{Q} = \dot{Q}_{\text{max}} \quad \text{when } t = 0 \]

\[ \dot{Q} = \dot{Q}_3 \quad \text{when } t = t_3 - t_p \]  

(vii)

and

\[ \int_0^{t_3 - t_p} \dot{Q} \, dt = \rho \Delta h_{\text{m}} \int \left( R_s u^2 - \{R_i R_i - (u-x)^2\} \right) \, du \]  

Inserting the above conditions in equation (vi) gives for the constants

\[ \alpha = \dot{Q}_{\text{max}} \]

\[ \beta = \frac{6 \rho \Delta h_{\text{m}} f}{(t_3 - t_p)^2} - \frac{2 \dot{Q}_3 + 4 \dot{Q}_{\text{max}}}{t_3 - t_p} \]  

(viii)

\[ \gamma = \frac{3(\dot{Q}_{\text{max}} + \dot{Q}_3)}{(t_3 - t_p)^2} - \frac{6 \rho \Delta h_{\text{m}} f}{(t_3 - t_p)^3} \]

and the shroud heat interception characteristic becomes:

\[ \dot{Q} = \dot{Q}_{\text{max}} + \left[ \frac{6 \rho \Delta h_{\text{m}} f}{t_3 - t_p} - 2 \dot{Q}_3 - 4 \dot{Q}_{\text{max}} \right] \frac{t}{t_3 - t_p} + \left[ 3(\dot{Q}_{\text{max}} + \dot{Q}_3) - \frac{6 \rho \Delta h_{\text{m}} f}{t_3 - t_p} \right] \left( \frac{t}{t_3 - t_p} \right)^2 \]  

(ix)

The remainder of the shroud evaporation cycle from time \( t = t_3 \) to \( t = t_4 \) when the shroud is entirely gas phase can be approximated by assuming a linearly decaying interception characteristic. This assumption leads to the heat equation.
\[
\frac{1}{2}(t_4-t_3)\dot{Q}_3 = \pi \rho \Delta h (R_S - R_i - x) \left[ \frac{1}{3}(R_S - R_i - x)(2R_S + R_i + x) - \frac{1}{4}D^2 \right]
\]

or

\[
t_4 = t_3 + \frac{2\pi \rho \Delta h (R_S - R_i - x)}{\dot{Q}_3} \left[ \frac{1}{3}(R_S - R_i - x)(2R_S + R_i + x) - \frac{1}{4}D^2 \right]
\]

The foregoing equations as applied to the three geometrical shroud annuli of figure 46 produced the curves of figures 47 and 48.

The theoretical shroud heat interception characteristics shown in figure 47 were based on the following assumptions.

(i) The shroud heat interception was assumed constant and equal to the total environmental heat leak from the 100% full condition until the LH2 level reached a value equivalent to the 1/4 full condition for the NAS9-7337 shroud design. This occurred at a value of \( c = 5.67 \) inches or a distance of 4.67 inches below the dewar center plane. At this shroud level, the same area of inner vessel surface is exposed to the hydrogen gas phase for all three geometries which is assumed as the condition when pressure build-up within the inner vessel commences.

(ii) The total environmental heat leak was assumed to be the same for all three geometries.

(iii) The value \( \dot{Q}_3 \) was assumed the same for all three geometries based on the assumption that the heat absorbed by the GHe is dependent on exposed inner vessel surface area.

(iv) The inner vessel volume was capable of storing 50 lbs. of GHe at 2000 PSIA and 37°F.

It is evident from the curves of figures 47 and 48 that considerable improvements in the standby capabilities of the NAS9-7337 shroud tankage are possible by adopting optimized shroud annulus geometry.

Further research in this area is necessary to examine these improved standby capabilities in relation to overall tankage weight optimization.
EFFECT OF SHROUD GEOMETRY ON SHROUD HEAT INTERCEPTION CHARACTERISTICS

![Graph showing effect of shroud geometry on shroud heat interception characteristics.](image-url)
EFFECT OF SHROUD GEOMETRY ON GHe PRESSURE BUILD-UP AND TANKAGE STANDBY CAPABILITIES

GHe INNER VESSEL PRESSURE (PSIA)

STANDBY TIME (HOURS)

NAS 9-7337 SHROUD

OBLATE SPHEROIDAL GFR INNER VESSEL WITH SPHERICAL SHROUD

CONCENTRIC SHROUD

FIGURE 48.

A5226-77