



LIGHTWEIGHT EVACUATED MULTILAYER INSULATION SYSTEMS FOR THE SPACE SHUTTLE VEHICLE

FINAL REPORT

CASE FILE
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by

D. L. Barclay, J. E. Bell & D. K. Zimmerman

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16. Abstract This study and experimental program investigated the elements in the evacuated multilayer insulation system and selected the major weight contributors for optimization. Outgassing tests were conducted on candidate vacuum jacket materials. Experiments were conducted to determine the vacuum and structural integrity of selected vacuum jacket configurations. A nondestructive proof test method, applicable to externally pressurized shells was validated on this program.					
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FOREWORD

This report describes a study and experimental program that investigated light-weight evacuated multilayer insulation systems for the Space Shuttle vehicle. The work was performed by The Boeing Company from April 12, 1971 through August 31, 1972, under Contract NAS 3-14369. The work was administered by Mr. J. R. Barber of NASA Lewis Research Center.

Mr. D. K. Zimmerman was program supervisor, Mr. D. L. Barclay was program technical leader, and Mr. J. E. Bell performed the structural analysis and evaluated the test data. Dr. R. E. Jones, originator of the nondestructive proof test method, assisted in the development of test procedures and interpretation of the data. Mr. D. H. Bartlett assisted in program coordination.

Other major participants in the program include:

J. G. Shdo	-	Material Outgassing Tests
M. E. Taylor	-	Material Outgassing Tests
D. E. Giesecking	-	Manufacturing
R. Nelson	-	Manufacturing
J. C. Stevens	-	Nondestructive Proof Tests
E. M. Balog	-	Nondestructive Proof Tests
D. McKenney	-	Vacuum Acquisition Tests
E. J. Zdilar	-	Vacuum Acquisition Tests
K. W. Osborne	-	Documentation

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1.0 SUMMARY

The objective of this program was to develop a high performance evacuated insulation system for the on-orbit propellant (LH_2 and LO_2) tanks of the Space Shuttle Orbiter. The insulation system was to combine maximum performance with minimum weight, be highly reliable, require minimum maintenance and provide a constant level of performance for at least 100 flights.

The work was divided into two major tasks. Design and analytical studies were conducted optimizing structural design, insulation system and cryogen storage method. The experimental program investigated shell materials and vacuum and structural integrity of the shells.

The design and analytical studies consisted of the following:

- 1) Trade studies on shell construction methods to select the least weight configurations.
- 2) Design studies investigated pressure vessel, vacuum jacket, support system, plumbing penetration, insulation layup and manhole access arrangements for a range of L/D 's. (L is the length of the cylindrical section; D is the diameter of the pressure vessel.)
- 3) Thermal analyses studied the effect of external (N_2) and internal (H_2) gas leakage, the effect of non-vented and vented storage systems, and the weight penalty associated with an on-board vacuum pumping system.

The experimental program investigated the following:

- 1) The outgassing characteristics of candidate vacuum shell materials to determine what influence these materials would have on vacuum integrity if exposed to the insulation annulus.
- 2) The vacuum acquisition characteristics of two 45.0 in. (1.14 m) diameter hemispherical sandwich shells. The inner face skin was the vacuum sealing surface for both shells. A spun and chem-milled aluminum alloy inner skin was used on one shell, a bonded aluminum alloy gore arrangement on the other.
- 3) A proof test method for non-destructively determining the buckling strength of externally pressurized shells. Force/stiffness plots were used successfully to predict (within 10% accuracy) the maximum external pressure capability of a 8 ft (2.44 m) and a 45.0 in. (1.14 m) diameter sandwich head.

Program results show that efficient vacuum jacket design is within the scope of present analytical tools, fabrication methods and test procedures. The most efficient design for both the LO₂ and the LH₂ tanks studied was found to be the near spherical shape using the honeycomb sandwich shell construction method. It was also found that for the high L/D designs the semi-rigid construction method was competitive with other methods. The vacuum jacket is the highest weight contributor to an evacuated MLI system. Results show that vacuum jacket weights can be substantially reduced by the optimization and the nondestructive proof test methods used in this program.

Investigation of cryogen storage methods showed that the non-vented storage system is more efficient than the vented system. Based on the outgassing test results, the use of non-metals in the insulation vacuum annulus should be minimized.

2.0 INTRODUCTION

Improved performance opportunities exist for the Space Shuttle Orbiter and future reentry space vehicles when it is verified that the evacuated multilayer insulation (MLI) system is a reliable system capable of withstanding the rigorous cyclic operational requirements of a vehicle such as the Orbiter.

Previous work with MLI has shown it to be fragile, difficult to install and have a wide variability in thermal performance due to installation and handling procedures. Radiation shields of aluminized polyester film have shown excessive degradations of thermal performance when exposed to atmospheric water vapor. For this reason, the evacuated MLI system offers greater promise for fulfilling Orbiter type requirements than its competitor, the purged MLI system. The purged system relies on a non-condensable purge gas e.g., helium to prevent moisture from contaminating the MLI during ground hold and then venting the gas in space to achieve the thermal performance. The complexity of this system leaves much to be desired. The evacuated MLI system on the other hand, assures continuous high performance of the insulation.

The purpose of this program was to develop a lightweight, evacuated MLI system which would maintain vacuum and structural integrity throughout the repeated temperature and pressure cycles of many missions. The study and experimental program discussed in this report addressed itself to this task.

The studies conducted investigated the key elements of the evacuated MLI system, fabricated subscale candidate vacuum shells and conducted vacuum and external pressure tests on these shells. The results from the analytical program were used to define recommended LH₂ and LO₂ tank assemblies for the on-orbit propellant tanks for the Space Shuttle vehicle.

Elements of the evacuated MLI system that were optimized were:

- a) The vacuum jacket; including construction method and the effects of length/diameter ratio.
- b) MLI systems; including the effects of vacuum degradation.

Emphasis was focused on developing a lightweight vacuum jacket design since this was a major component affecting the weight of the system. The vacuum integrity of the shell design is crucial to the thermal performance of the system, which again affects weight. Success of the fabrication process selected to meet the contour and gage requirements will also affect the weight. Structural reliability of lightweight vacuum shells has long been a concern of designers since large scatter in test data has shown need for large conservatism and heavy

structures. As part of the effort to reduce weight in vacuum jackets, a non-destructive proof test method for predicting the buckling pressure of shells was studied in the experimental phase of this program. Successful use of this test method provides a valuable tool for achieving minimum weight shells.

3.0 DESIGN AND ANALYTICAL STUDIES

3.1 Design Criteria

The design and analytical studies of the on-orbit propellant tanks used the criteria outlined below. These criteria were based on the contract work statement and on applicable Space Shuttle design criteria, Reference 1, and Shuttle Phase B contract studies, References 2 and 3.

Pressure Vessel Configuration

LH₂ Tanks - 6 to 15 ft (1.83 to 4.57 m) in diameter with a volume of 2000 cubic ft (56.63 m³).

LO₂ Tanks - 4 to 10 ft (1.22 to 3.05 m) in diameter with a volume of 750 cubic ft (21.24 m³).

Life

One hundred operational flight cycles (launch, orbit, reentry, two weeks ground turnaround) plus one hundred test (temperature) cycles.

Time in Orbit

7 days, 15 days and 30 days.

Thermal Performance

Range of heat flux values - 0.1 to 0.7 Btu/hr ft² (0.32 to 2.21 W/m²).

Loading Conditions

Load Factors

Load factors critical to tank and support structure design are specified in the following table. Plus refers to forward, down or right. All load factors except crash are limit values. The crash load factors are ultimate values. The landing and crash load factors are applied with the tank carrying 30 percent of its maximum propellant weight. All other load factors are applied with a full propellant load.

LIMIT LOAD FACTORS FOR ON-ORBIT PROPELLANT TANKS

	<u>Longitudinal (g's)</u>	<u>Vertical (g's)</u>	<u>Lateral (g's)</u>
Launch	-1.25	0	±1.0
End Boost	-3.0	+0.25	0
Landing	0	+2.5	0
Crash	+9.0	-2.0	+1.5
	-1.5	+4.5	-1.5

Pressure Vessel Supports

Factors of Safety

Yield	1.0
Ultimate	1.5

Vacuum Jacket

Limit Design External Pressure - 14.7 psia (101.4 kN/m²)

Factors of Safety

Yield	1.1
Ultimate	1.4

Pressure Vessel

Material - 2219-T81

Relief Valves - Maximum Pressure Settings

LO ₂	40 psia (275.8 kN/m ²)
LH ₂	30 psia (206.8 kN/m ²)

Factors of Safety

Yield	1.25
Ultimate	1.64

Plumbing Lines

Factor of Safety

Proof	1.5
Ultimate	2.5

Plumbing Lines (Continued)

Sizes

Feed	LO ₂	2.0 in Dia. (5.08 cm)
	LH ₂	2.5 in Dia. (6.35 cm)
	LO ₂	2.5 in Dia. (6.35 cm)
	LH ₂	3.0 in Dia. (7.62 cm)

Maximum Propellant Weight

Maximum propellant weight = (tank internal volume - 4 percent ullage)
x fluid density at maximum relief valve
pressure setting.

Temperature Conditions

Vacuum Shell External Temperature

Ground Hold, Launch, and On-Orbit	+100°F (311°K)
Reentry	+350°F (450°K)

Shuttle Primary Structure Temperature at Tank Support Locations

On-Orbit	+150°F (339°K)
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Minimum Interior Insulation Temperature

-423°F (20.4°K) for LH ₂ Tank
-297°F (90.2°K) for LO ₂ Tank

3.2 Tank Configuration Studies

3.2.1 Evacuated MLI System

The evacuated MLI system for cryogenic containment is described in Figure 3-1. The multilayer insulation (MLI) is located in a vacuum annulus formed by the pressure vessel and the vacuum jacket. The vacuum level in the annulus should be maintained at less than 1×10^{-4} torr (13.3 mN/m²) for the efficient thermal performance of the MLI.

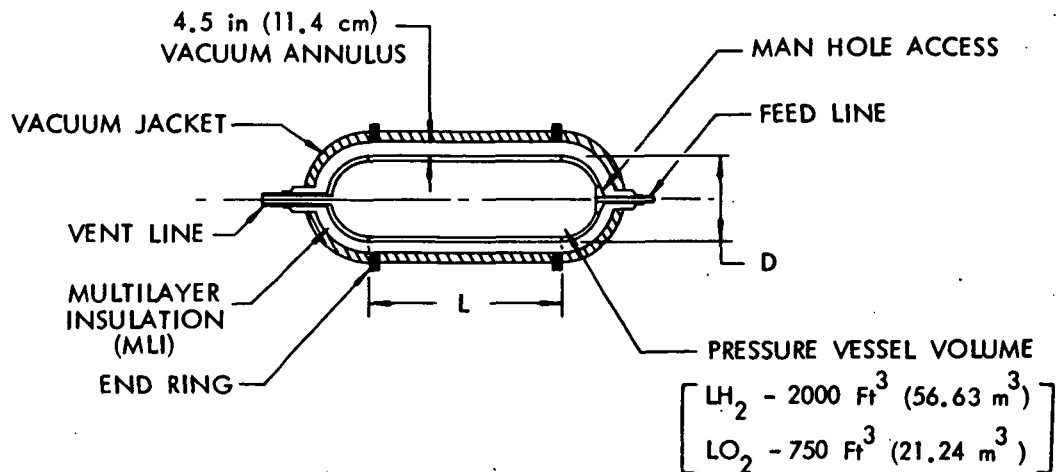


Figure 3-1: EVACUATED MLI SYSTEM

The configuration studies considered tanks with a range of L/D ratio from .09 to 14.3. The 'D' selected was the diameter of the pressure vessel and 'L' was the length of the cylindrical section.

Some of the system items were the subject of independent investigation, apart from the design studies. Results from these investigations were used in the designs. These items are discussed below.

Insulation Annulus

A preliminary investigation was made which selected an insulation annulus of 4.5 in (11.4 cm). This was judged an adequate, but not excessive clearance for installing the MLI and the pressure vessel support system for the range of configurations being considered. This annulus thickness was used throughout the Task I study. This standardized annulus thickness simplified the design and analysis task and provided a common baseline for the weight trades.

MLI Blanket

The thermal analysis studies, Section 3.3, selected double aluminized Mylar (0.15 mil) ($3.8\mu\text{m}$)/dacron B4A net for the MLI blanket configuration. The outer 0.10 in. (0.25 cm) was double aluminized Kapton (0.30 mil) ($7.6\mu\text{m}$)/dacron B4A net. Layer density was 75 layer/in. (29.5 layers/cm).

Pre-assembly of the blanket in panels was considered the most reliable method for maintaining layer density. The panels are layed up on tools shaped to the pressure vessel contour. Layup method assumed for the aluminized Mylar and Kapton film was to drape the film over the tool as it comes from the rolls. The wrinkles along the edges are folded flat. This method of contour control was

recommended by studies on Contract NAS 3-14179, "Multilayer Insulation Panels", since there was good density control of the panel and the original emittance of the aluminized film was maintained. The alternate method of forming the film to contour caused an increase in emittance. The MLI panels are assembled and attached to the pressure vessel with nylon studs and grommets.

The tank assembly configurations discussed below describe the insulation details at the penetrations.

Vacuum Jacket

The major emphasis in this program was to develop a lightweight vacuum jacket since this component made a significant contribution to system weight. Weight trades were performed on different vacuum jacket configurations in the shell trade studies discussed in Section 3.4. The problems associated with the use of non-metallics in the vacuum annulus were investigated in the material outgassing tests described in Section 4.1. The results of these two studies along with an assessment of the different fabrication methods available were used to arrive at the designs considered in the outer shell studies, Section 3.5.

The vacuum jackets shown in the tank configuration studies reflect the above studies. The sandwich construction used had an aluminum inner skin and aluminum Flex-Core. This was the lightest weight construction compatible with eliminating non-metallic materials from the vacuum annulus. The aluminum inner skin was the vacuum sealing skin. Choices for the outer skin included boron/epoxy, fiberglass/polyimide, fiberglass/epoxy and aluminum.

The tank configuration studies show details at the girth ring joints and at the penetrations.

Plumbing Penetrations

Evaluation of different plumbing penetrations and manhole access methods is discussed in detail in the propellant leakage isolation studies in Section 3.3.4. Some details of the penetrations are shown and discussed in the LH₂ tank configuration studies, Section 3.2.2.

3.2.2 LH₂ Tank Configurations

Five LH₂ tank configurations were studied. Geometry and vacuum shell type used for each configuration is shown in Table 3-1. The configurations are discussed below.

Table 3-1: LH₂ TANK CONFIGURATIONS

CONFIGURATION NUMBER	L/D	D		L		VACUUM SHELL TYPE
		In	cm	In	cm	
1	0.09 LOW	180	457.2	16	40.64	SELF SUPPORTING
2						
3						
4	11.1 HIGH	72	182.8	801	2034	SELF SUPPORTING
5						SEMI-RIGID

Low L/D

Figure 3-2 shows the 15 ft. (4.57 m) diameter near spherical tank located in the Orbiter, aft of the main propulsion tanks. It was expected that the vacuum jacket outer surface and the LH₂ tank assembly support structure would experience the same temperature environment. For this reason, insulating the vacuum shell from the support structure was not considered.

Limited definition of the Orbiter structure during the program prevented a meaningful detail study of the tank assembly to vehicle support structure. It appeared, however, that attachment could be made to the fuselage side-frame and/or the fin support structure. The attachment points on the LH₂ tank assembly were located on the vacuum jacket girth ring. This ring is shown in Figure 3-2 aligned vertically in the Orbiter.

Configuration 1

Configuration 1 is shown in Figure 3-3. This configuration investigated the tension/compression strut support system for the pressure vessel. Alternatives for the vent and feedline penetrations and the manhole access were also studied.

Detail 1 describes the closeout for the vacuum jacket. An aluminum box girth ring joins the two hemispherical heads. The vacuum sealing surface is the inner aluminum skin on these heads. Each head has an aluminum closure ring which is welded to the inner skin and bonded to the outer skin. The closure ring and inner skin are welded together during the first stages of head assembly. The core and outer face skin are then bonded in place. The outer face skin is shown as aluminum, but since this is not the vacuum sealing skin, other materials (i.e., fiberglass/epoxy, boron/polyimide, etc.) can be used if there is an advantage to do so. The head closure rings are welded to the girth ring. The design allows clearance for weld access. Chill bars are necessary to protect

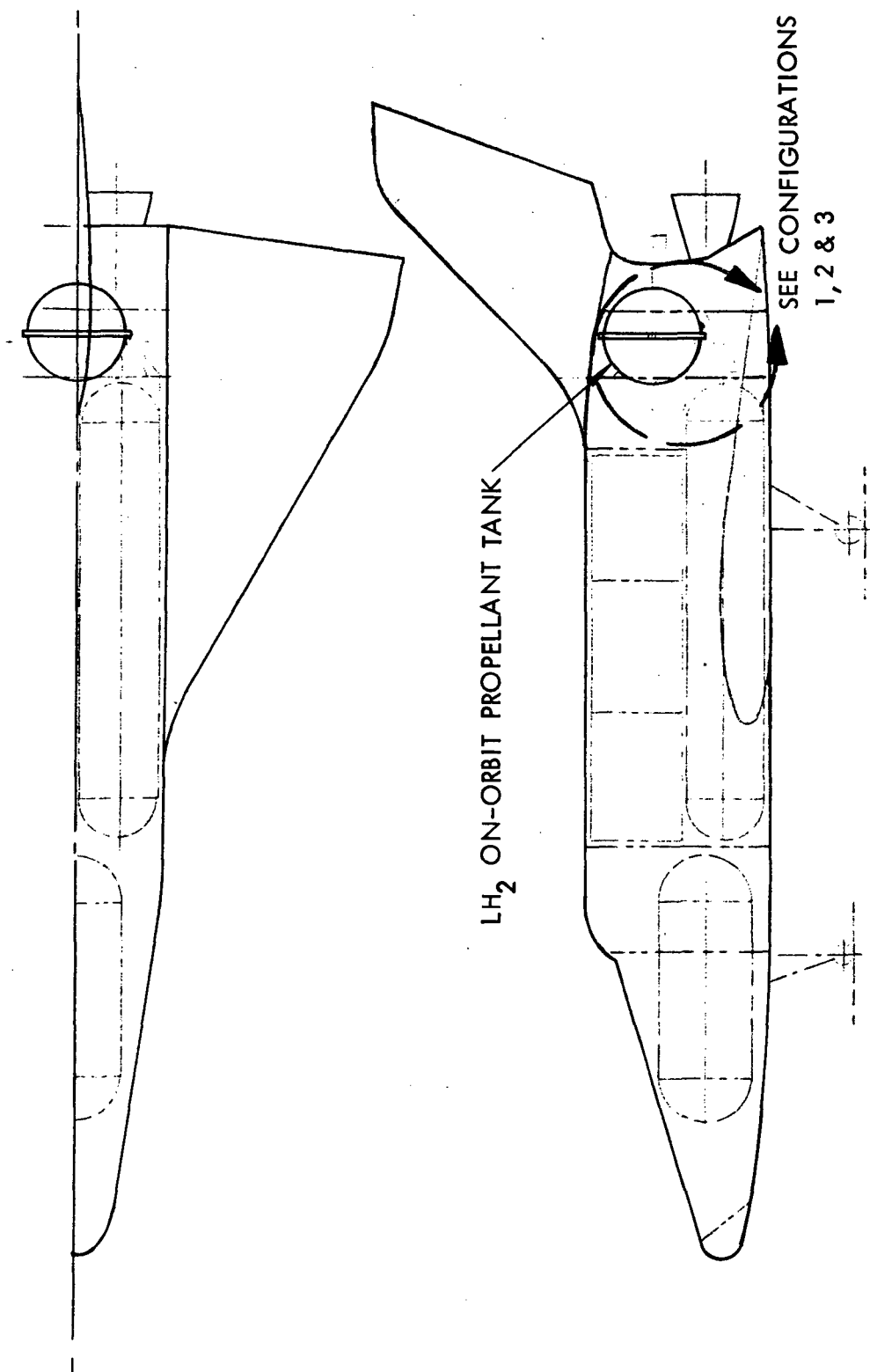


Figure 3-2: TANK LOCATION IN ORBITER

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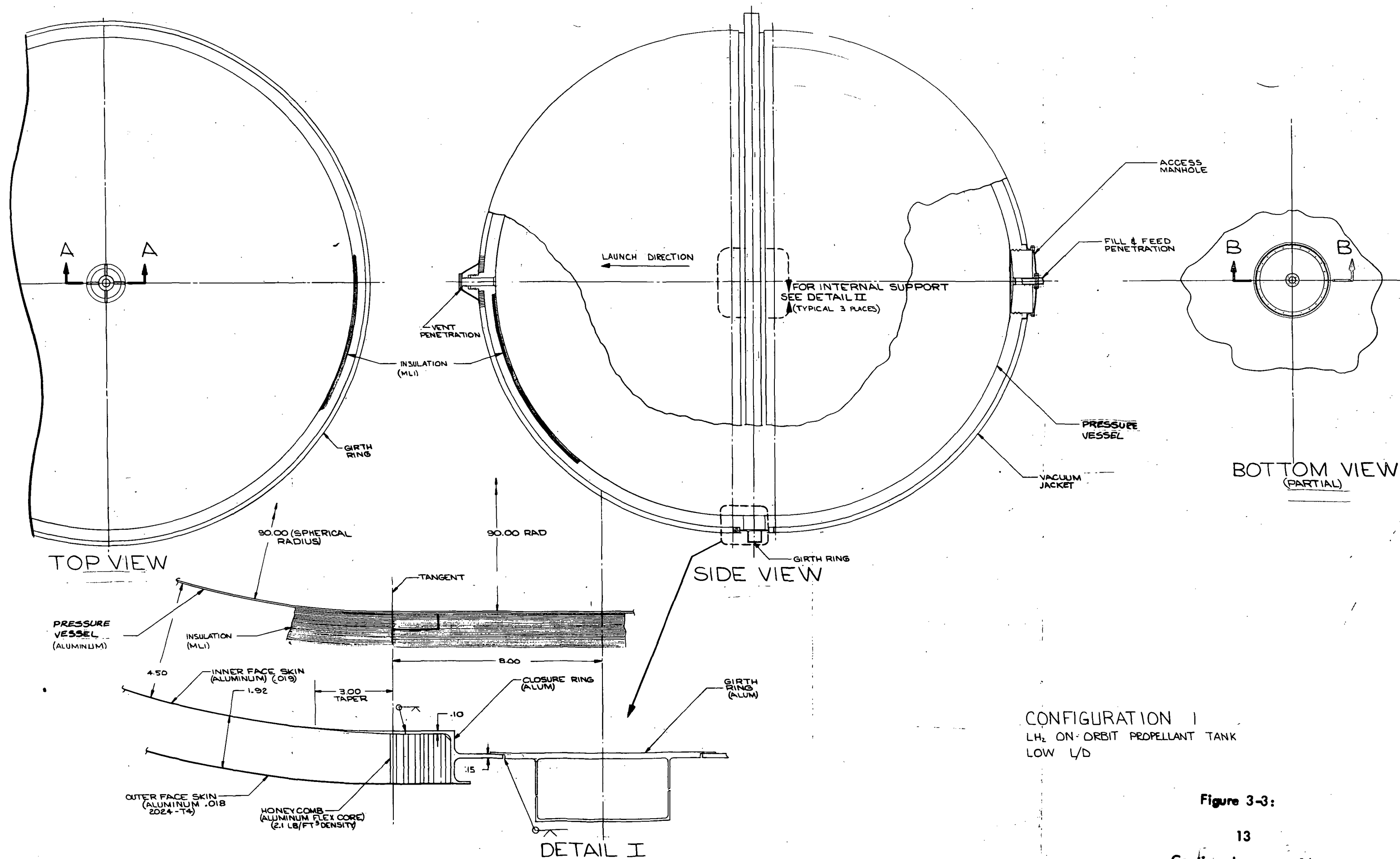
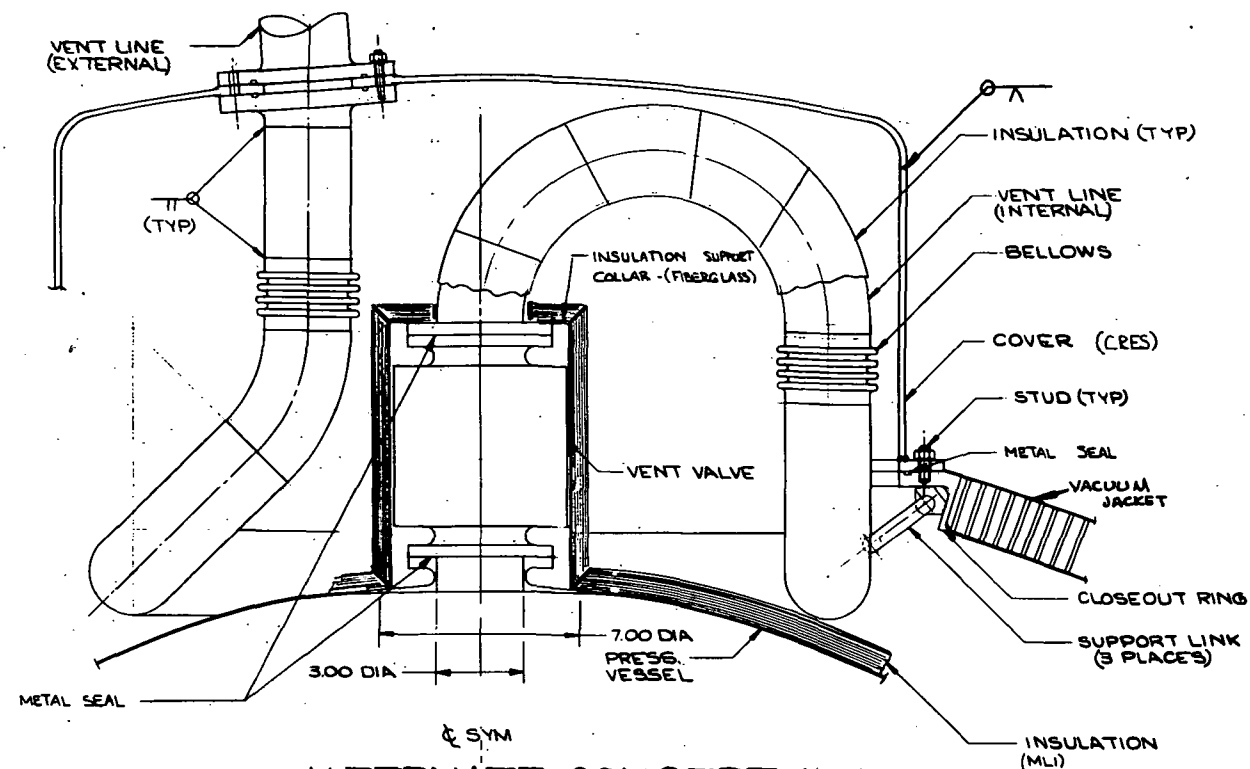
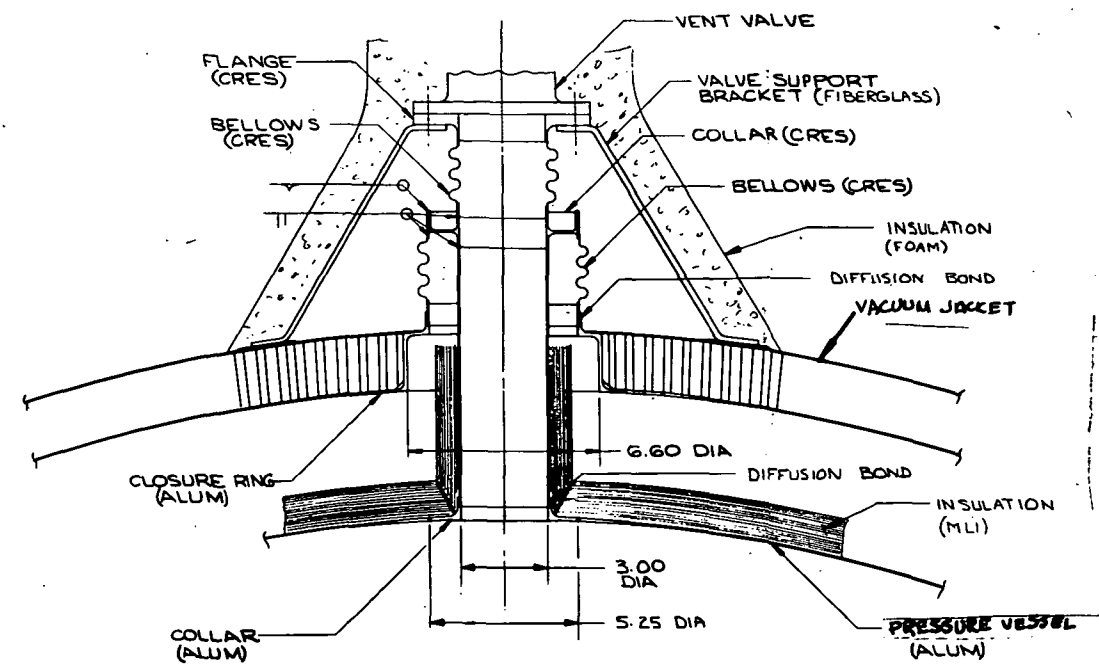


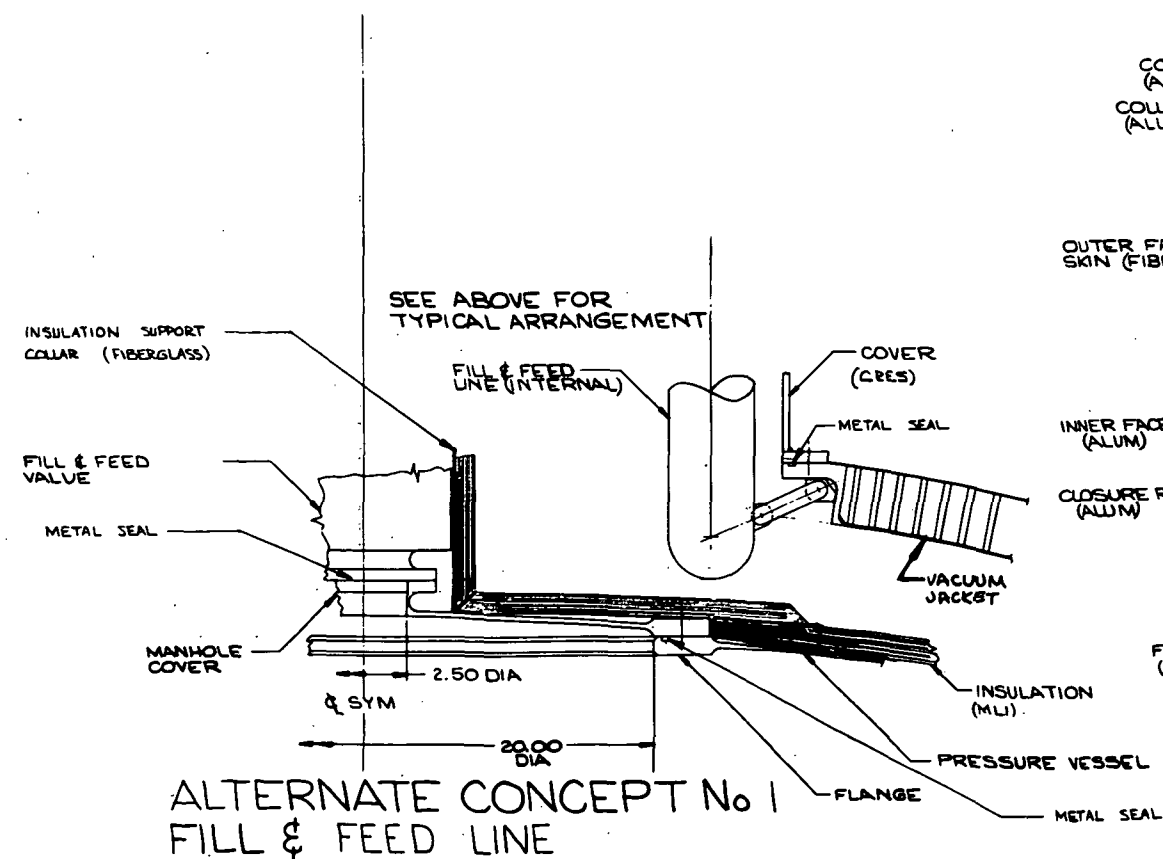
Figure 3-3:



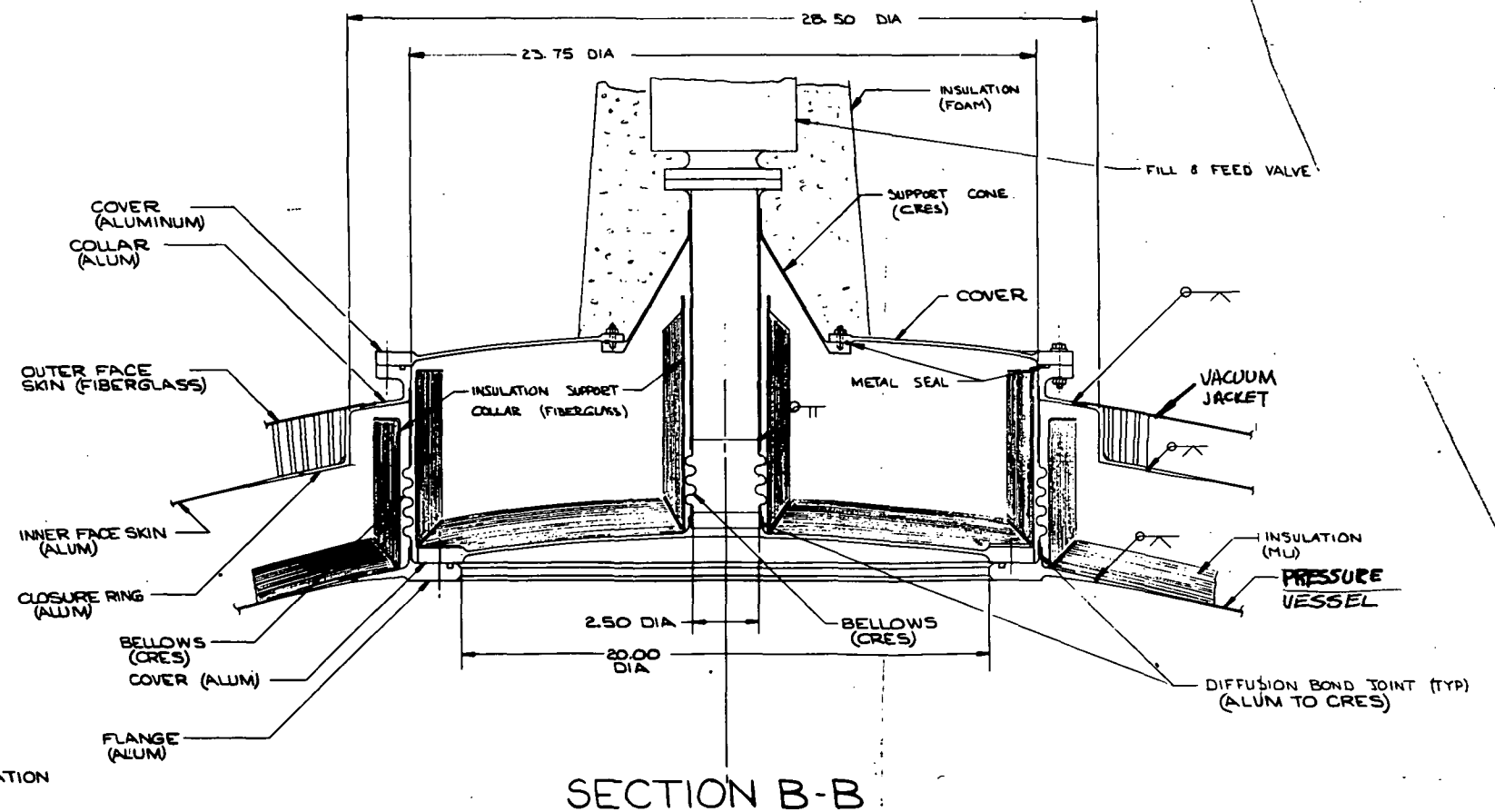
ALTERNATE CONCEPT No. 1
VENT LINE PENETRATION



SECTION A-A



ALTERNATE CONCEPT No. 1
FILL & FEED LINE



SECTION B-B

Figure 3-3:

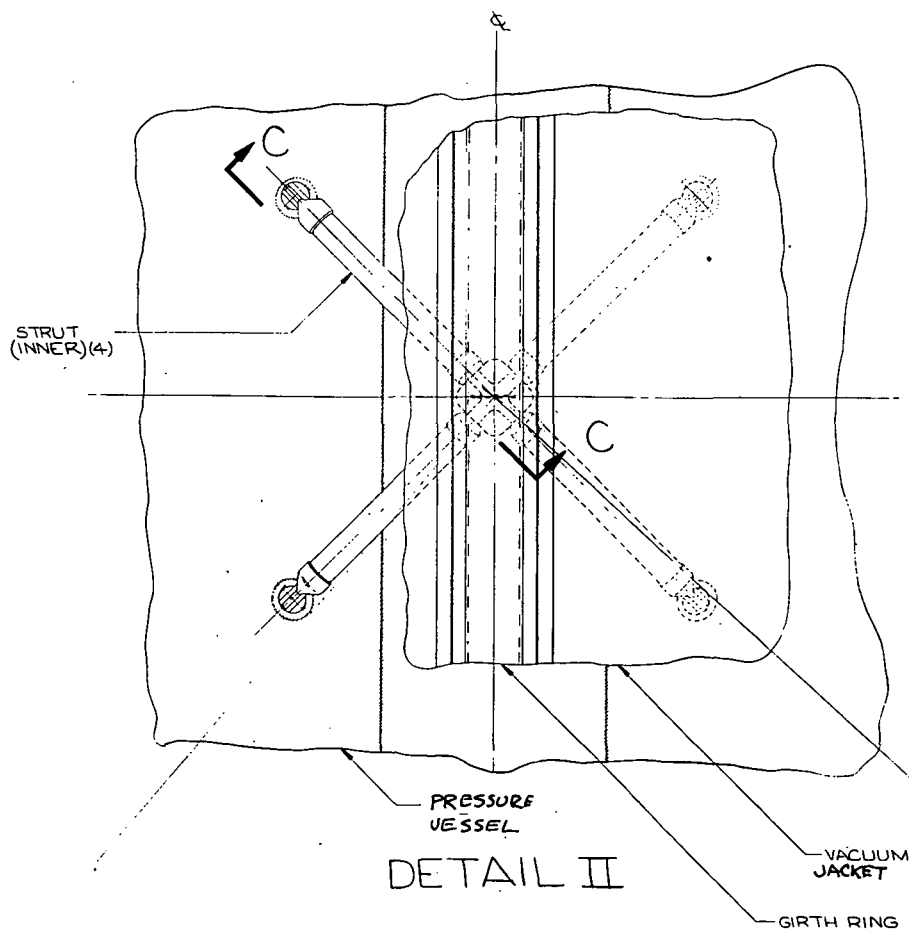
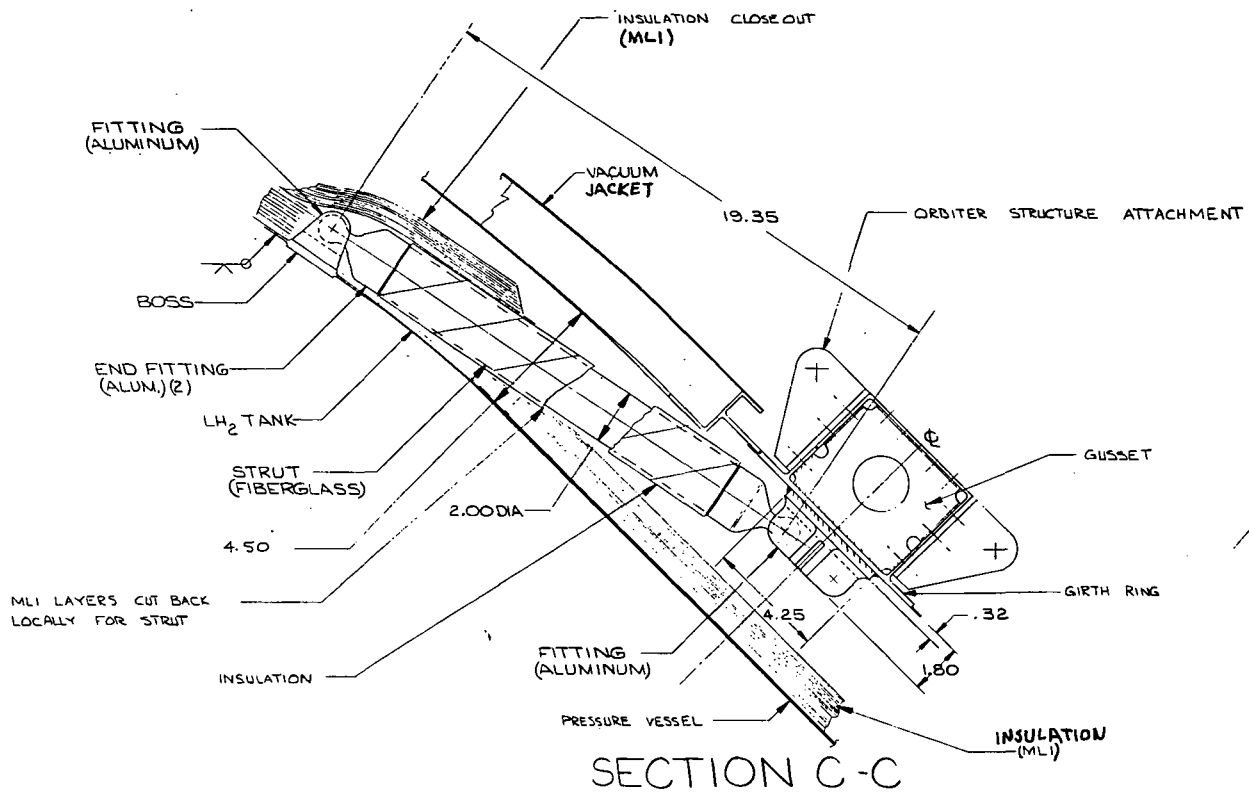


Figure 3-3:

Continued from page 14

adjacent head bond areas during welding. Lips are provided on the girth ring to locate the heads and for weld backup.

The multilayer insulation (MLI) is arranged in gore section panels on the pressure vessel heads. These are joined to the blanket on the 16.0 in. (40.6 cm) cylindrical section by a lap joint.

Detail II shows the fiberglass strut arrangements. Three sets of four tension/compression struts support the pressure vessel from the girth ring. The four struts attached at the Orbiter structure support point on the girth ring and fan out at 45° (0.79 rad) to attach to four tank bosses. The MLI is cut back locally for the struts. Each strut is insulated and an insulation closeout patch covers the disturbed area. Threaded end fittings on the struts provide length adjustment for assembly purposes.

Section A-A shows vent/relief valve mounted external to the vacuum jacket. Bellows are provided for differential thermal shrinkage. This installation protects the vacuum annulus from propellant leakage at the valve. Also, it provides easy accessibility for valve maintenance. However, since the line to the valve is wet, an external insulation such as foam is required. Several wraps of MLI are used on the line between the vacuum jacket closeout and the pressure vessel.

Section B-B describes the feedline and manhole access arrangement with the shut-off valve external to the vacuum jacket. The insulation annulus is protected from propellant leakage at the manhole cover by a closeout bellows between the vacuum jacket and the pressure vessel. Due to the movement of the feedline and manhole closeout bellows a fiberglass collar is required to support the MLI.

The alternate concepts shown for the vent and feedline arrangement place the valves in the vacuum annulus. The long, insulated lines from the valves to the vacuum jacket provide a low heat leak path.

Both of these plumbing and manhole penetration arrangements are discussed in more detail in the propellant leakage isolation study, Section 3.3.4.

Configuration 2

Figure 3-4 shows Configuration 2 which investigated two tension strap arrangements for supporting the pressure vessel. Configuration 1 vent and feedline penetrations are used. The closeout between the vacuum jacket heads and the girth ring is similar to that described in Configuration 1. The girth ring in this configuration is an I section.

Detail 1 describes an aluminum tension band arrangement which is bonded to the pressure vessel head. Aluminum tabs are welded to the bands and provide

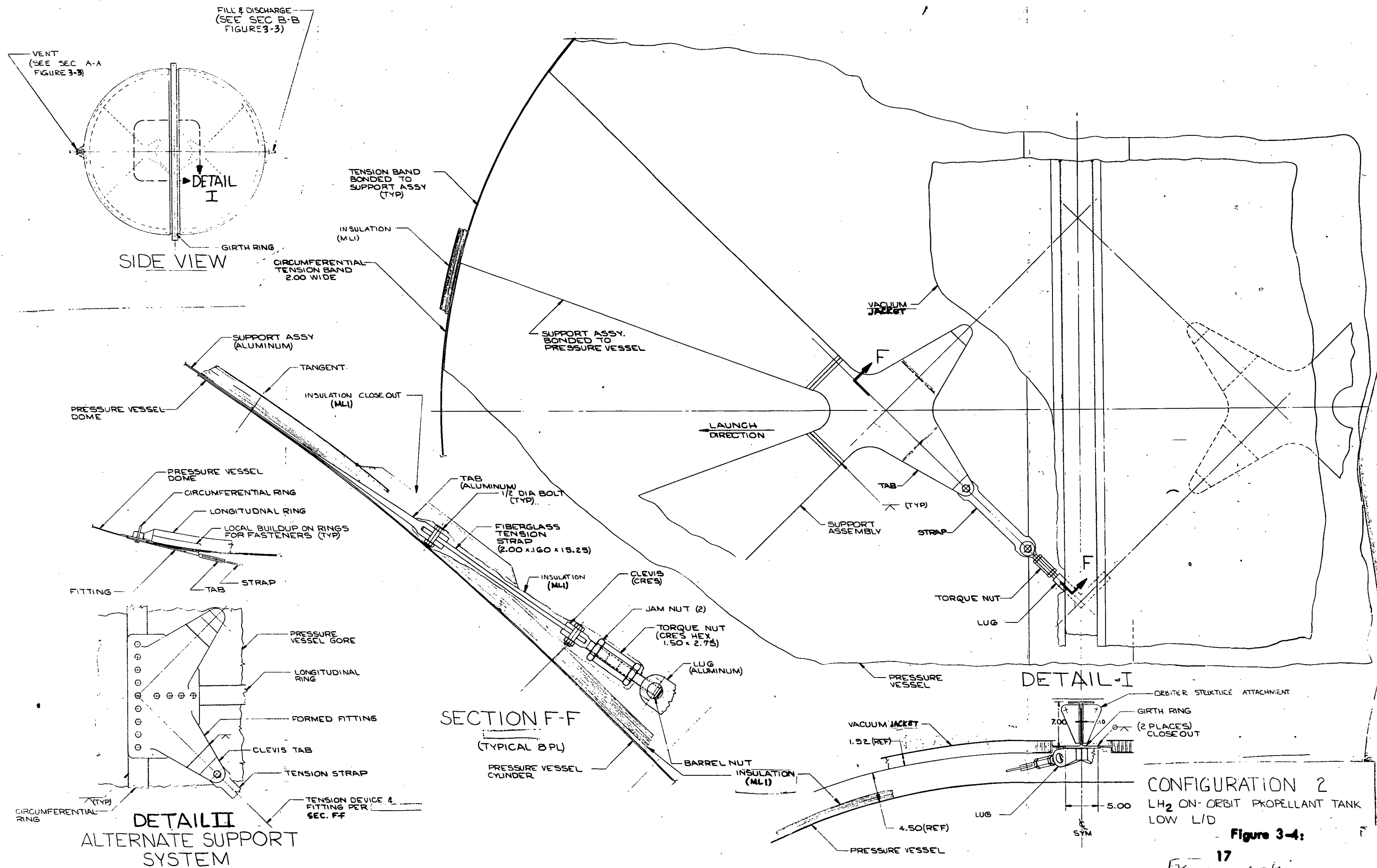


Figure 3-4:

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attachment fittings for the fiberglass tension straps. The straps attach to the girth ring lug through a torque nut. Penetration through the MLI is similar to that described in Configuration 1.

Detail II shows an alternate pressure vessel attachment arrangement for the tension straps. The straps connect to a machined fitting which is bolted to tapped holes in the pressure vessel. Circumferential rings and longitudinal stiffeners, integral parts of the pressure vessel, distribute the loads.

Configuration 3

Configuration 3 shown in Figure 3-5 investigated a crossed tension strap support arrangement for the pressure vessel, the use of valve leakage shrouds, and the routing of the plumbing lines for a short distance around the perimeter within the girth ring envelope.

The crossed titanium tension straps are arranged at 30 degree (0.52 rad) intervals. Attachment studs are positioned on the pressure vessel cylinder along the circumferential centerline. Tank support brackets, linking the two vacuum jacket girth rings, provide the outer attachment point for the straps. A 0.60 in. (1.52 cm) slot machined into the support bracket allows for a ± 0.10 in. (0.25 cm) difference in strap length. This allows for tension link lateral movement to compensate for unequal strap length. The pre-tensioning is accomplished by a double nut locking tension link device which is torqued to the tension requirement. The tank support brackets also provide load paths for the vacuum jacket heads through the girth rings. Intermediate girth ring brackets can be added as required by this loading condition.

The vent and the feed valves are mounted in the vacuum annulus. Leakage shrouds shown in Details I and II guard against vacuum degradation from valve leakage.

The dry vent and feedline runs from the valves are contained within the girth ring assembly chamber. These lines are attached to the tank support brackets. The lines are insulated along their length and form the support brackets to minimize conductive heat leak into the cryogen.

Results

The advantages and disadvantages of the low L/D configurations are itemized in Tables 3-2 and 3-3.

Results of this study favor a single vacuum jacket girth ring. Tension straps supporting the pressure vessel from this girth ring, and oriented to resist the load environment, simplified MLI, pressure vessel, support system and vacuum jacket assembly procedures and simplified pressure vessel design. The preferred location for the plumbing line penetration was at the tank apex.

Table 3-2: ADVANTAGES OF LH₂ ON-ORBIT PROPELLANT
TANK LOW L/D CONFIGURATIONS

ADVANTAGES	CONFIGURATION		
	1	2	3
Assembly of Pressure Vessel Support System and Girth Ring Provides a Convenient Arrangement for Supporting Pressure Vessel During MLI Blanket Installation	✓	✓	✓
Plumbing and Manhole Access Penetrations Discussed in Propellant Leakage Isolation Section 3.3-4.	✓	✓	✓
Single Girth Ring Minimizes Vacuum Close-Out Welds	✓	✓	
Box Section Girth Ring has Excellent Torsional Rigidity	✓		✓
Simple Welded Pressure Vessel Design Using Gores, Polar Caps, Cylindrical Section and Attachment Fitting	✓	Detail I Only	✓

Table 3-3: DISADVANTAGES OF LH₂ ON-ORBIT PROPELLANT
TANK LOW L/D CONFIGURATIONS

DISADVANTAGES	CONFIGURATION		
	1	2	3
Plumbing and Manhole Access Penetrations Discussed in Propellant Leakage Isolation Section 3.3-4.	✓	✓	✓
Difficulty in Matching Head Closure Rings With Girth Ring for Welding	✓	✓	✓
Bending Moment on Pressure Vessel Wall at Support Strut Attachment Bases Due to Offset of Applied Load and Reaction	✓		
Reaction to Side Load Soft		Detail I Only	
Pressure Vessel Design Complex		Detail II Only	
Accessibility to Plumbing Lines and Valves Difficult			✓

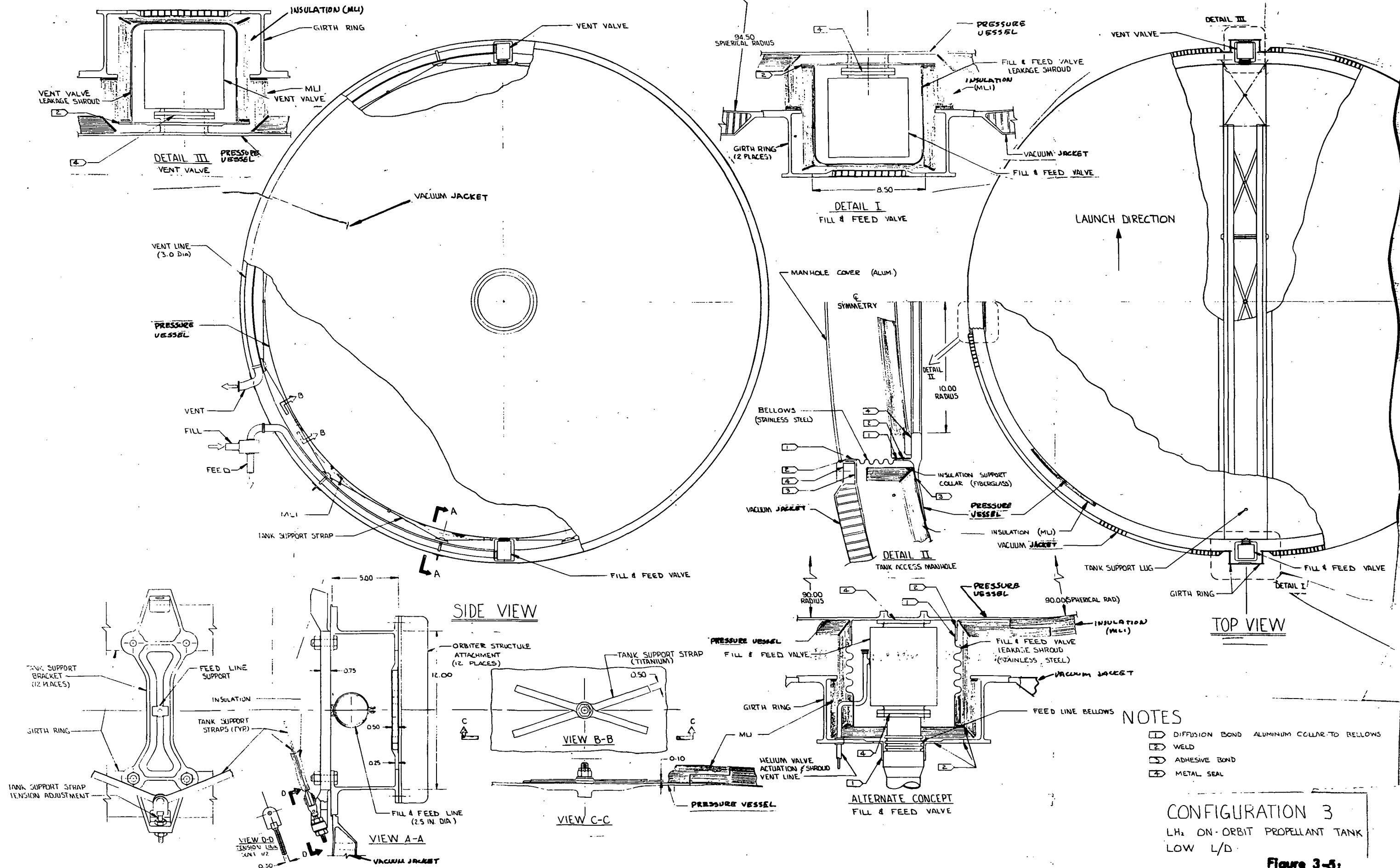


Figure 3-5:

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High L/D

Configuration 4

Configuration 4 is a LH_2 tank assembly with a $L/D = 11.1$. This design, shown in Figure 3-6, uses a honeycomb sandwich for the vacuum jacket hemispherical heads and cylindrical section. The support arrangement and the vacuum jacket assembly features described in this design are also applicable to tanks with a lower L/D and to tanks using other self-supporting vacuum jacket construction methods.

The vacuum jacket comprises the two hemispherical heads and the cylindrical section. The cylinder is fabricated in several sections as determined by autoclave size restrictions. Vacuum tight welds join the cylindrical sections, the two girth rings and the two heads.

The pressure vessel is supported by 8 fiberglass/epoxy tension straps at each head, attached to the girth rings. These straps support the pressure vessel against loads in the direction of its longitudinal axis. Eight fiberglass/epoxy bumpers which penetrate the insulation are mounted at each end of the pressure vessel cylinder to react loads normal to the longitudinal centerline. A disadvantage with this system is the higher heat transfer rate through the MLI at the bumpers. At room temperature conditions there is a 0.10 in. (0.25 cm) clearance between the bumpers and the vacuum jacket girth ring. Tank shrinkage at cool-down will increase this clearance to approximately 0.24 in. (0.61 cm). The length of the tank will decrease by approximately 1.5 in. (3.81 cm) at cool-down, thereby relaxing the tension straps. In effect then, the loaded pressure vessel "floats" within the vacuum jacket. The bumpers and tension strap prevent excessive "float" which would compress the MLI. In this regard, however, special attention in MLI layup at the tension straps is necessary to minimize damage to the insulation from the strap movement.

The tank assembly is attached to the orbiter primary structure by 8 brackets at each of the girth rings. Although not shown, the attachment arrangement would allow for misalignment, and insure that orbiter structure loads are not transferred through the vacuum jacket.

Feed and vent line penetration, and the manhole access arrangement, are typical arrangements suitable for all tank configurations studied.

Configuration 5

The shell trade studies in Section 3.4 concluded that the semi-rigid vacuum shell showed a weight advantage over the self-supporting shells for an $L/D = 11.1$ and 4.31. Figure 3-7 is a preliminary design of a semi-rigid shell arrangement for an L/D ratio of 11.1.

All the design features of Configuration 4 that were applicable to the semi-rigid design were included in Configuration 5. The sandwich heads, girth rings, pressure vessel support system, vent and feed line penetration and manhole access arrangement were used.

The comparison between the two designs then rested with the differences in manufacturing, storage and installation problems associated with the semi-rigid and the self-supporting shells.

The self-supporting shell arrangement by definition is an integral unit, which can be vacuum pumped, stored and installed in the vehicle using normal procedures and precautions. The semi-rigid shell design on the other hand requires special support tooling before vacuum pumping can commence, and while the tank assembly is in storage. The installation in the orbiter must locate and fix the jacket heads and ensure that primary structure movement does not distort the vacuum jacket/pressure vessel relationship so that the support straps become ineffective for transmitting load, or that the girth ring is forced against the bumpers.

3.2.3 LO_2 Tank Study

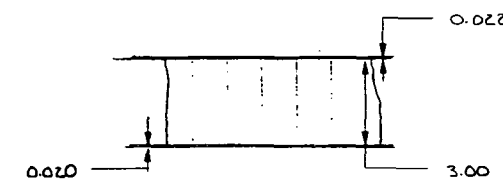
The design features developed in Configurations 1 through 5 for the LH_2 tanks were also applicable to the LO_2 tanks. In general, a LO_2 tank can be configured using the LH_2 tank details modified to account for the differences in volume and L/D. Pressure vessel, vacuum jacket, plumbing penetrations, manhole access, and MLI arrangement details can be similar for both cryogens.

The pressure vessel support system for the two cryogens may be different however, because liquid oxygen is sixteen times heavier than liquid hydrogen. For the low L/D LO_2 tank, a titanium tension strap was selected in place of fiberglass/epoxy. This provided a smaller cross-sectional area for the strap, thereby minimizing the disturbance to the MLI layup. In the high L/D LO_2 tank, an intermediate set of fiberglass bumpers was needed to support the pressure vessel during a landing condition with 30% fuel remaining. Additional supports, from the vacuum jacket to the Orbiter primary structure were also needed at the bumper locations.

The preferred outer face skin material for the LO_2 tank vacuum jacket was aluminum alloy. This was because of possible LO_2 spills on the vacuum jacket during loading. Aluminum alloys are relatively stable to LOX under ballistic impact whereas organic materials generally are not.

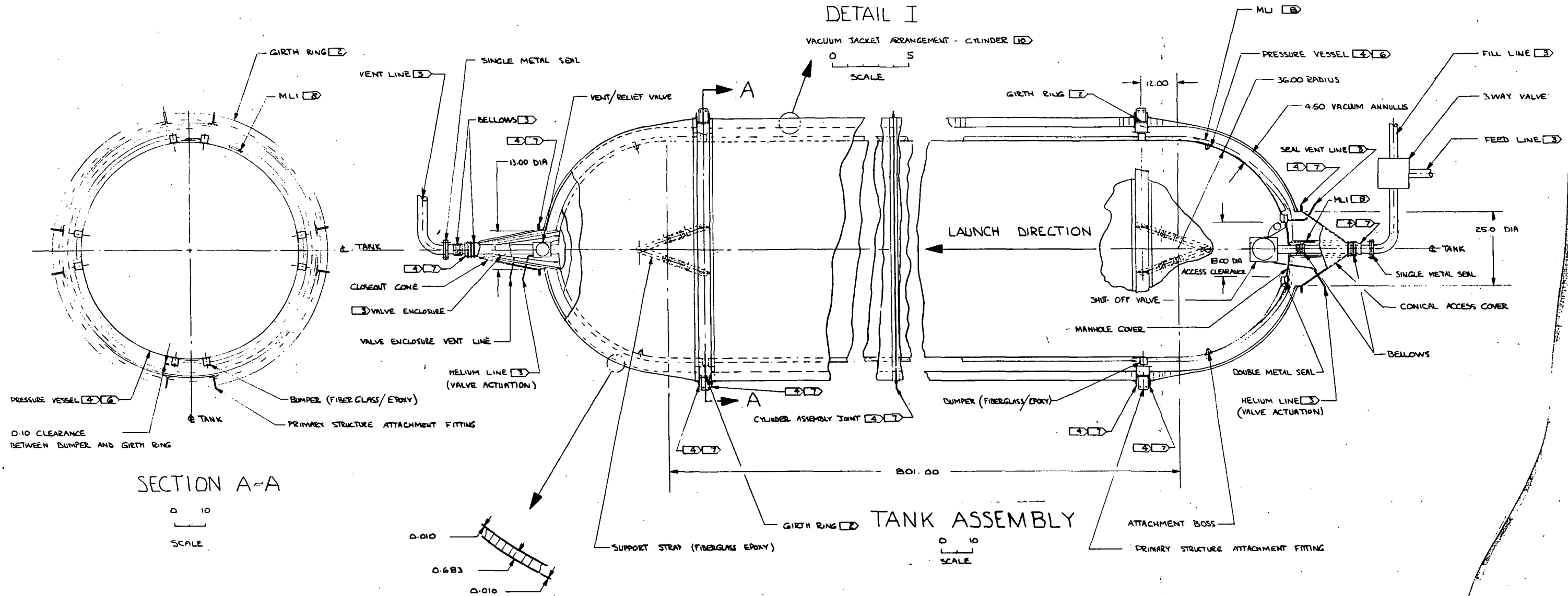
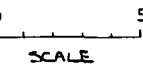
3.3 Thermal Studies

The thermal analyses were performed to determine insulation requirements and to identify methods of achieving a vacuum in the vacuum annulus and maintaining it against leakage. The primary effect of degraded vacuum in an evacuated



DETAIL I

VACUUM JACKET ARRANGEMENT - CYLINDER

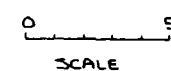


SECTION A-A



DETAIL II

VACUUM JACKET ARRANGEMENT - HEMISPHERE

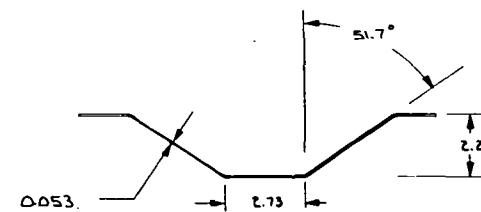


NOTES

- 1 ALUMINUM ALLOY 2219-T81
- 2 ALUMINUM ALLOY ROLL FORGED RING 2219-T852
- 3 STAINLESS STEEL
- 4 WELD
- 5 DIFFUSION BOND
- 6 WELDS IN 'AS WELDED' CONDITION
- 7 VACUUM SEAL
- 8 MULTILAYER INSULATION (MLI)
 - a) ALUMINIZED MYLAR (0.15 MIL)/DACRON NET (BAA) - 75 LAYERS/INCH
 - b) OUTER 0.10 INCH LAYER - ALUMINIZED KAPTON (0.30 MIL)/DACRON NET (BAA) - 75 LAYERS/INCH
- 9 ASSEMBLE WITH NYLON STUDS AND GROMMETS
- 10 SANDWICH CONSTRUCTION
 - a) FACE SKINS - ALUMINUM ALLOY 2024 T3
 - b) CORE - 505G ALUMINUM FLEX-CORE (2.1 LB/FT³)

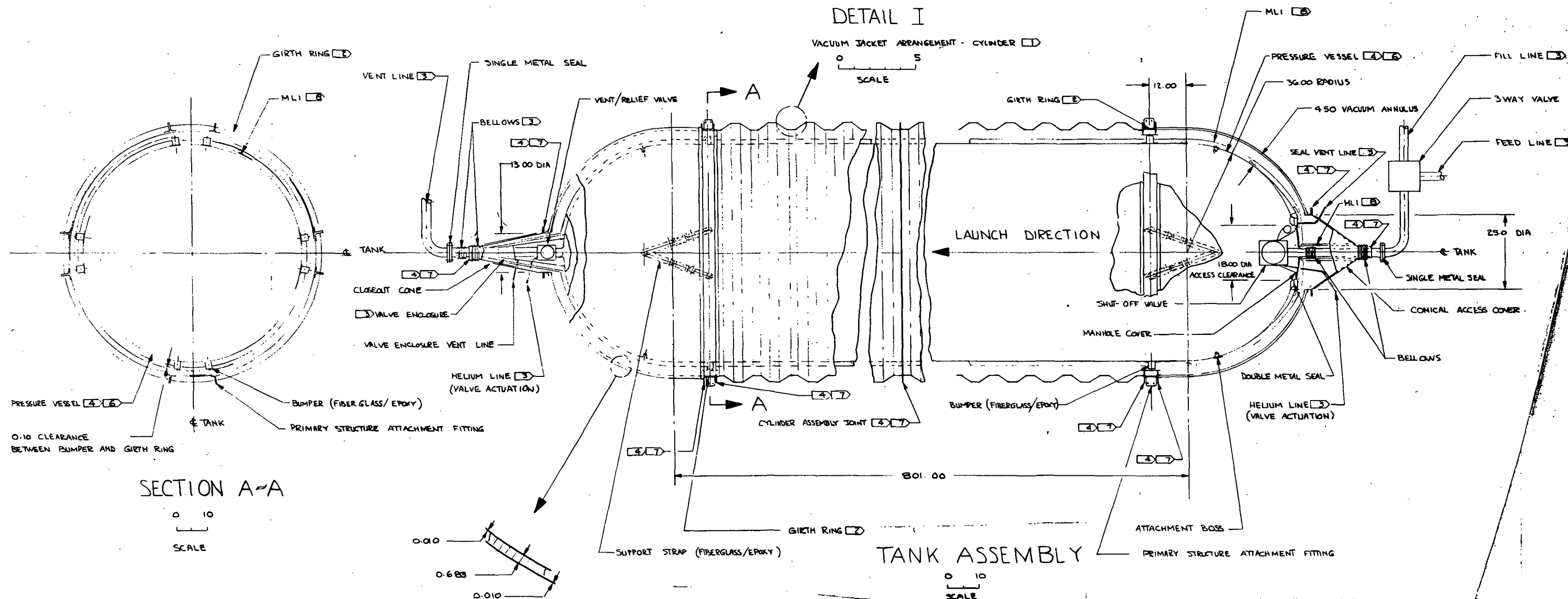
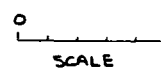
CONFIGURATION 4
LH₂ ON-ORBIT PROPELLANT TANK
HIGH L/D

Figure 3-6:



DETAIL I

VACUUM JACKET ARRANGEMENT - CYLINDER



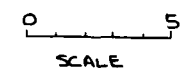
TANK ASSEMBLY

NOTES

- 1 ALUMINUM ALLOY 2219-T81
- 2 ALUMINUM ALLOY ROLL FORGED RING 2219-T82
- 3 STAINLESS STEEL
- 4 WELD
- 5 DIFFUSION BOND
- 6 WELDS IN 'AS WELDED' CONDITION
- 7 VACUUM SEAL
- 8 MULTILAYER INSULATION (MLI)
 - a) ALUMINIZED MYLAR (0.15 MIL)/DACRON NET (BAA) - 75 LAYERS/INCH
 - b) OUTER 0.10 INCH LAYER - ALUMINIZED KAPTON (0.30 MIL)/DACRON NET (BAA) - 75 LAYERS/INCH
- 9 ASSEMBLY WITH STUDS AND GROMMETS
- 10 SANDWICH CONSTRUCTION
 - a) FACE SKINS - ALUMINUM ALLOY 2024 T3
 - b) CORE - SOSG ALUMINUM FLEX-CORE (2.1 LB/FT³)

DETAIL II

VACUUM JACKET ARRANGEMENT - HEMISPHERE



CONFIGURATION 5

LH₂ ON-ORBIT PROPELLANT TANK
HIGH V/D SEMI-RIGID

Figure 3-7:

multilayer insulation system is increased heat flow through the insulation. If allowed for in the design, this results in increased system weight, and if not, in unscheduled venting during the mission.

The study approach was to select several insulations and screen to one, based on performance with and without vacuum leaks, and then to evaluate several vacuum acquisition techniques by calculation of vacuum level versus time and its effect on system weight.

3.3.1 Insulation Selection

Several MLI systems were screened to determine those best suited for use on the 15 ft. (4.57 m) diameter LH₂ tank configuration. Factors considered were (1) MLI weight, (2) outgassing characteristics, (3) thermal performance prediction accuracy, and (4) installation complexity. It was assumed that there were sufficient Kapton layers on the outside of the blanket to reduce working temperatures to the point where the materials studied could be used.

Typical support and fluid line configurations were assumed and the heat flow for these subtracted from the total allowance (0.1 to 0.7 Btu/ft²-hr) (0.32 to 2.21 W/m²). Eight fiberglass tank supports, approximately 10 in. (25.4 cm) long, were selected. The 2.5 in. (6.35 cm) and 3.0 in. (7.62 cm) diameter fill and vent lines had a 0.035 in. (0.089 cm) wall thickness and were routed 1/4 of the distance around the hemispherical tank head before exiting the vacuum annulus.

Table 3-4 shows the MLI systems studied, the thermal conductivity and relative insulation weights. At this point dacron net spacers were assumed to have a thermal performance similar to the other net spacers. As will be shown later, the conductivity and relative weight of aluminized Mylar-Tissuglas shown here appears to be much too high.

Thermogravimetric analyses performed on past Boeing IR&D programs have shown that room temperature weight loss for these materials is:

<u>Material</u>	<u>Weight Loss (% of Original)</u>
Aluminized Mylar	0.072
Dacron Net	0.132
Nylon Net	3.16
Silk Net	7.00
Tissuglas	Not Tested

Table 3-4 : RELATIVE INSULATION WEIGHTS OF MLI SYSTEMS

$$k = k_r (T_1^2 + T_2^2) (T_1 + T_2) + k_c (T_1 + T_2), \text{ BTU/FT-HR-}^\circ\text{R (W/m}^\circ\text{K)}$$

where T_1 and T_2 are the boundary temperatures

$$k_r = \frac{\sigma}{12n \left(\frac{1}{\epsilon} - 1\right)} \quad (\text{unless taken directly from reference 4})$$

σ = Stefan-Boltzmann Constant
 n = layers/inch
 ϵ = .025

k_c = constant selected to fit test data
 $T_1 = 560^\circ\text{R (312}^\circ\text{K)}$
 $T_2 = 40^\circ\text{R (22}^\circ\text{K)}$

INSULATION	n	ρ		.15 Mil (3.8 μ m)	THROUGH LAYERS						RELATIVE WEIGHT
		lb/ft ³	kg/m ³		k_r		k_c		Ref.		
					∇	∇	∇	∇			
NRC-2	70	0.94	15.04	9.6×10^{-14}	166.1	0.89×10^{-8}	15.4	4	1.00		
Aluminized Mylar-nylon net	70	3.25	52.00	2.5×10^{-14}	43.26	0.53×10^{-8}	9.17	5	1.18		
Aluminized Mylar-2 silk net	23	0.98	15.68	7.8×10^{-14}	135.0	2.20×10^{-8}	38.07	6	1.24		
Aluminized Mylar-Tissuglas	82	2.11	33.80	3.7×10^{-14}	64.05	3.10×10^{-8}	53.65	4	3.45		

∇ BTU/FT-HR- $^\circ$ R * Mylar Thickness ∇ FW/m $^\circ$ K ∇ nW/m $^\circ$ K

The NRC-2 MLI system appears to be the best choice based on relative weight and outgassing. However, experience has shown that NRC-2 is difficult to apply to a specific layer density and is especially sensitive to compression loading. In thicker blankets, gravity will influence the applied thickness in different locations on the tank. With NRC-2 the heat flow will be more difficult to predict accurately and the labor involved in application of the MLI will be greater than for other systems.

Experience has shown that net spacers add resilience and strength to a MLI system. Application to a specific layer ratio is more easily accomplished, thus the accuracy of thermal performance predictions is better. However, the nylon and silk nets do not appear to be a good choice for this application because of their initial high moisture content and affinity for water vapor. It would be possible to achieve optimum performance with these materials through preconditioning, however, loss of vacuum during ground turnaround would necessitate repetition of the preconditioning procedure. It appears that a dacron net spacer would be an excellent choice, based on the small amount of thermal conductivity data available, and the low outgassing. No data was available on the outgassing of Tissuglas, so its suitability for this application is not verified. It is considered a candidate because its fibrous construction should give much lower gas conduction than the nets at low pressure.

The choices for radiation shields include .25 mil ($6.35 \mu\text{m}$) and .15 mil ($3.8 \mu\text{m}$) aluminized Mylar, 0.31 mil ($7.6 \mu\text{m}$) aluminized Kapton and 0.30 mil ($7.6 \mu\text{m}$) goldized Kapton. Considering weight and cost, the 0.15 mil ($3.8 \mu\text{m}$) aluminized Mylar is the best choice. However, where the temperature capability of Mylar is exceeded, the Kapton must be used. In the present application, the better environmental resistance of the expensive gold-coating is not needed.

Three MLI systems were chosen based on the above considerations:

- (1) 0.30 mil ($7.6 \mu\text{m}$) Double Aluminized Kapton/B4A Dacron Net
- (2) 0.30 mil ($7.6 \mu\text{m}$) Double Aluminized Kapton/Tissuglas
- (3) 0.3 mil ($7.6 \mu\text{m}$) Double Aluminized Kapton/B4A Dacron Net
over 0.15 mil ($3.8 \mu\text{m}$) Double Aluminized Mylar/B4A Dacron Net

The first two use aluminized Kapton throughout assuming a conservative requirement for 350°F (450°K) capability through the thickness. The Tissuglas spacer in (2) provides a comparison of sensitivity to gas pressure. The third selection provides a minimum weight system accounting for actual penetration of elevated temperature during reentry. Some calculations were also done with 0.30 mil ($7.6 \mu\text{m}$) goldized Kapton/B4A dacron net to show the effect on weight of the type of metallizing.

The amount of Kapton layers in the third system was determined by a transient thermal analysis of the vacuum shell and insulation, simulating the reentry condition. Typical results of the analysis are shown in Figure 3-8. This shows the very small temperature drop through the honeycomb sandwich vacuum jacket, which is typical of shells using aluminum honeycomb core. The outer surface of the insulation also follows this temperature profile closely. Peak temperatures versus depth for several cases are cross-plotted in Figure 3-9. This shows that the elevated temperature penetration from the external surface is not too sensitive to annulus gas pressure or blanket thickness. A thickness of 0.1 in. (0.25 cm) was selected for the outer layers using Kapton in MLI system (3).

3.3.2 System Optimization

Tank With Constant Pressure

Optimum thicknesses were determined by hand thermal analysis and graphical optimization. For simplicity, the assumption was made initially that the LH_2 tank would be vented at constant pressure through the mission. This approach was used for the first analyses to determine baseline insulation requirements and was subsequently used in determining the effect of the vacuum acquisition method on system weight. However, further analysis indicated that least system weight for both the LH_2 and LO_2 tanks was obtained with a non-vented system.

A simplified thermal analysis was made to determine penetration heat leaks for the LH_2 tank for use in the system analysis. The total estimated heat leak for the plumbing and tank supports was 42 Btu/hr (12 W) and for the insulation fasteners was $0.0088/L \text{ Btu/ft}^2\text{-hr}$ (0.028 W/m^2) where L is thickness of insulation in in. (m).

The insulation heat transfer was calculated using the equation form from Reference 7, with the addition of a gas conduction term. The gas conduction was based on test data, using Dexiglas test data for the Tissuglas and nylon net test data for the dacron net. The conductivity equation coefficients for aluminized Kapton/Tissuglas came directly from Reference 7. The only data available for B4A dacron net spacers was in Reference 8, but with goldized Kapton radiation shields. This data was used to construct an equation for aluminized radiation shields with dacron net, using the approaches of Reference 7. The theoretical radiation contribution of the goldized shields was subtracted to obtain the solid conduction, which was fit to a curve to obtain the required coefficients. The radiation contribution was replaced by the equivalent values for aluminized radiation shields from Reference 7. Based on the data of Reference 8, a layer density of 75 layers/inch (29.5 layers/cm) was chosen for the dacron net insulation as a density equivalent to the minimum layer densities in Reference 7.

The thermal conductivity equations used in the analyses were as follows:

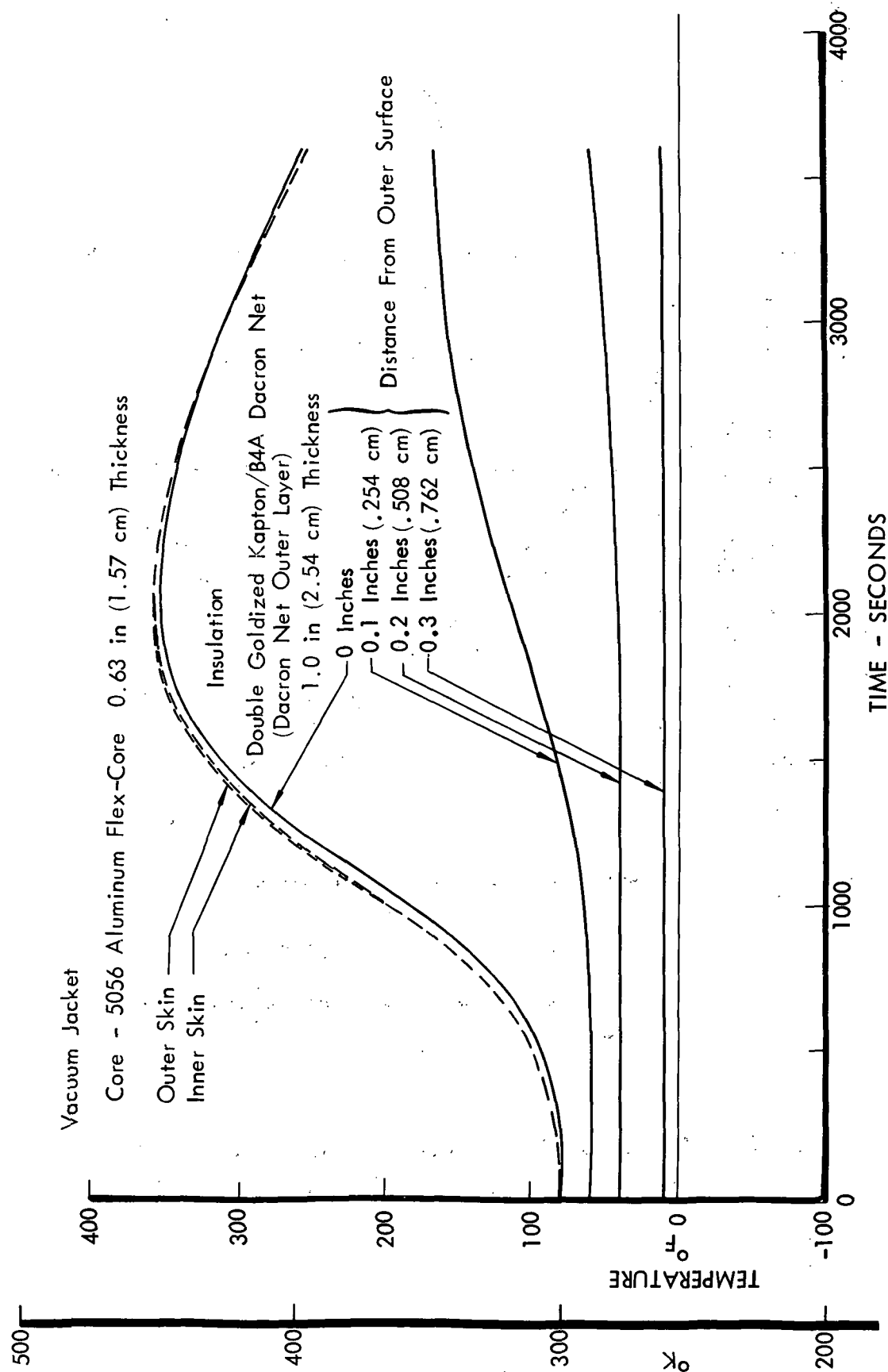


Figure 3-8: VACUUM JACKET AND INSULATION REENTRY TEMPERATURES

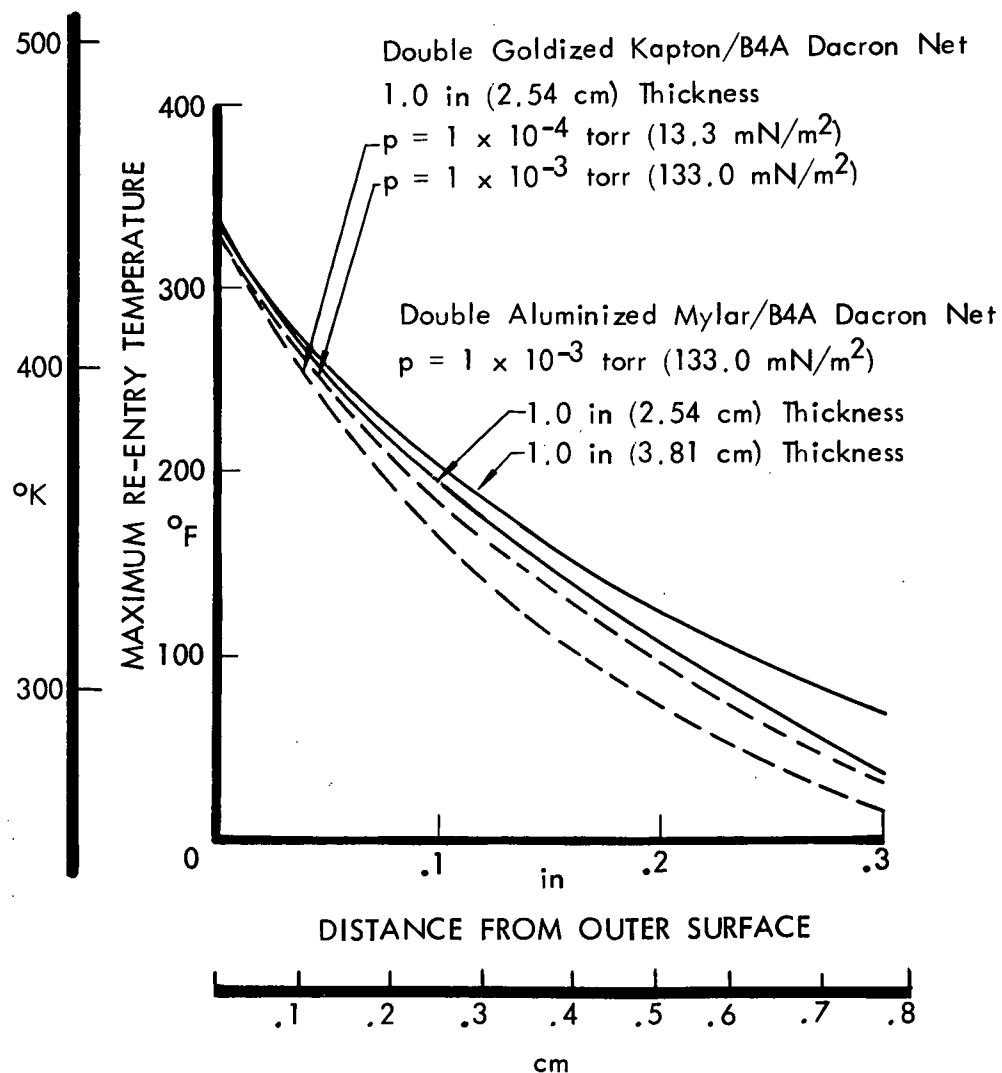


Figure 3-9: EFFECT OF GAS PRESSURE, METALLIZING, AND THICKNESS ON REENTRY TEMPERATURES IN MLI

Aluminized Kapton (or Mylar)/B4A dacron net

$$\begin{aligned}
 k &= 1.2 \left[.206 \times 10^{-6} \left(\frac{T_1 + T_2}{2} \right) + 4.4 \times 10^{-15} \left(\frac{T_1^{4.67} - T_2^{4.67}}{T_1 - T_2} \right) \right] \\
 &+ 14.05 \left(\frac{P_1}{T_1^{0.5}} \right) \frac{\text{Btu-in}}{\text{ft}^2 \text{-hr-}^\circ\text{R}} ; \quad \begin{array}{l} P \text{ in Torr} \\ T \text{ in } ^\circ\text{R} \end{array} \\
 &= .00173 \left[.371 \times 10^{-6} \left(\frac{T_1 + T_2}{2} \right) + 3.805 \times 10^{-14} \left(\frac{T_1^{4.67} - T_2^{4.67}}{T_1 - T_2} \right) \right] \\
 &+ 1.14 \times 10^{-4} \left(\frac{P_1}{T_1^{0.5}} \right) \frac{\text{watt-cm}}{\text{cm}^2 \text{-}^\circ\text{K}} ; \quad \begin{array}{l} P \text{ in N/m}^2 \\ T \text{ in } ^\circ\text{K} \end{array}
 \end{aligned}$$

Aluminized Kapton/Tissuglas

$$\begin{aligned}
 k &= 1.2 \left[1.429 \times 10^{-7} \left(\frac{T_1 + T_2}{2} \right) + 3.912 \times 10^{-15} \left(\frac{T_1^{4.67} - T_2^{4.67}}{T_1 - T_2} \right) \right. \\
 &\left. + 1.58 \left(\frac{P_1}{T_1^{0.5}} \right) \right] \frac{\text{Btu-in}}{\text{ft}^2 \text{-hr-}^\circ\text{R}} ; \quad \begin{array}{l} P \text{ in Torr} \\ T \text{ in } ^\circ\text{R} \end{array} \\
 &= .00173 \left[2.57 \times 10^{-7} \left(\frac{T_1 + T_2}{2} \right) + 3.38 \times 10^{-4} \left(\frac{T_1^{4.67} - T_2^{4.67}}{T_1 - T_2} \right) \right. \\
 &\left. + .89 \times 10^{-2} \left(\frac{P_1}{T_1^{0.5}} \right) \right] \frac{\text{watt-cm}}{\text{cm}^2 \text{-}^\circ\text{K}} ; \quad \begin{array}{l} P \text{ in N/m}^2 \\ T \text{ in } ^\circ\text{K} \end{array}
 \end{aligned}$$

The temperature dependence of the gas conduction term is based on the kinetic theory of gases at low pressure.

Optimization results for the vented LH_2 tank for a 30 day mission are shown in Figure 3-10. With negligible residual gas pressure in the annulus, the aluminized Kapton-Mylar/dacron net combination provides the least system weight at an optimum thickness of 1.38 in. (3.51 cm).

A somewhat more realistic situation is achieved by accounting for some gas pressure in the annulus. Figure 3-11 shows the optimum insulation and boiloff weights, and insulation thickness for an annulus gas pressure of 1×10^{-4} (13.3 mN/m²) torr. The Kapton-Mylar/dacron net insulation thickness increases from 1.38 in (3.51 cm) to 1.57 in (3.99 cm) for the 30 day mission when the gas pressure is included.

Tank With Pressure Rise

An investigation was made to determine what system weight improvement could be made by taking advantage of the large heat capacity available in the cryogen. The results of a system weight trade, assuming launch at a LH_2 tank pressure of 16 psia (110.3 kN/m²), are shown in Figure 3-12. The analysis is based on thermal equilibrium in the hydrogen. For the solid lines (no burn) the minimum system weight occurs at about 30 psia (206.8 kN/m²) and the tank remains non-vented throughout the mission. The results indicate that replacing boil-off with pressure rise as a sink for the heat leak results in impressive weight reduction. The comparable weight of the 35 psia (241.3 kN/m²) boil-off system (obtained by adding tank weight to Figure 3-11) is about 1070 lb (485 kg) or about 500 lb (226.8 kg) more than the optimum system from Figure 3-12.

The dashed lines in Figure 3-12 show the optimization results with an engine burn halfway through the mission. Insulation and boil-off weight are reduced because of the pressure drop during the burns. The analysis used did not include effects due to the pressurization during the burn. Determination of a critical design condition for optimizing the insulation would require definition of mission timelines and analysis of possible variations. The few variations analyzed indicate that the "no burn" assumption is a reasonable (and perhaps conservative) model for the present analyses.

Optimum weights and insulation thicknesses for the three insulations as a function of mission duration are shown in Figure 3-13. The breaks in the curves are due to the tank heads and cylinder minimum gage constraint of .030 in (7.62 cm).

The LO_2 tank also optimizes as a nonvented system, which could be expected because of its large heat capacity. The nonvented system weight and insulation thickness versus pressure are shown in Figure 3-14.

2000 ft³ (56.63 m³) LH₂ Tank

30 Day Mission

Constant Tank Vent Pressure = 35 psia (241 kN/m²)

Negligible Vacuum Annulus Gas Pressure

Penetration Heat Leaks = 42 Btu/hr (12 W)

Fastener Heat Leak = 0.0088/L Btu/ft²-hr (0.028 W/m²) -

Double Aluminized Kapton

Double Aluminized Mylar/B4A Dacron Net

N = 75 Layers/in (29.5 Layers/cm)

Double Aluminized Kapton/B4A Dacron Net

N = 75 Layers/in (29.5 Layers/cm)

Double Aluminized Kapton/Tissuglas

N = 125 Layers/in (49.21 Layers/cm)

○ Optimum

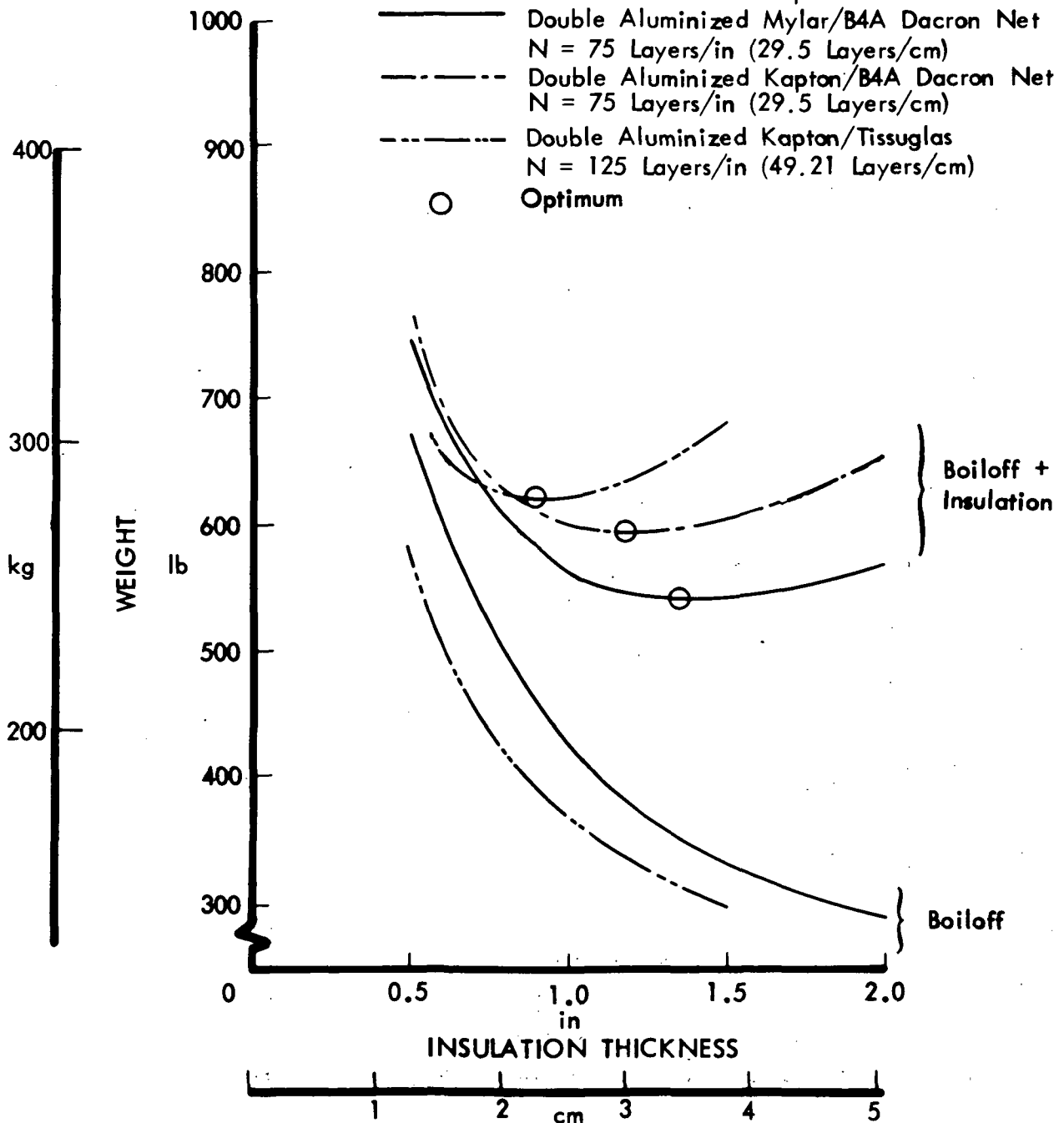


Figure 3-10: MLI OPTIMIZATION RESULTS

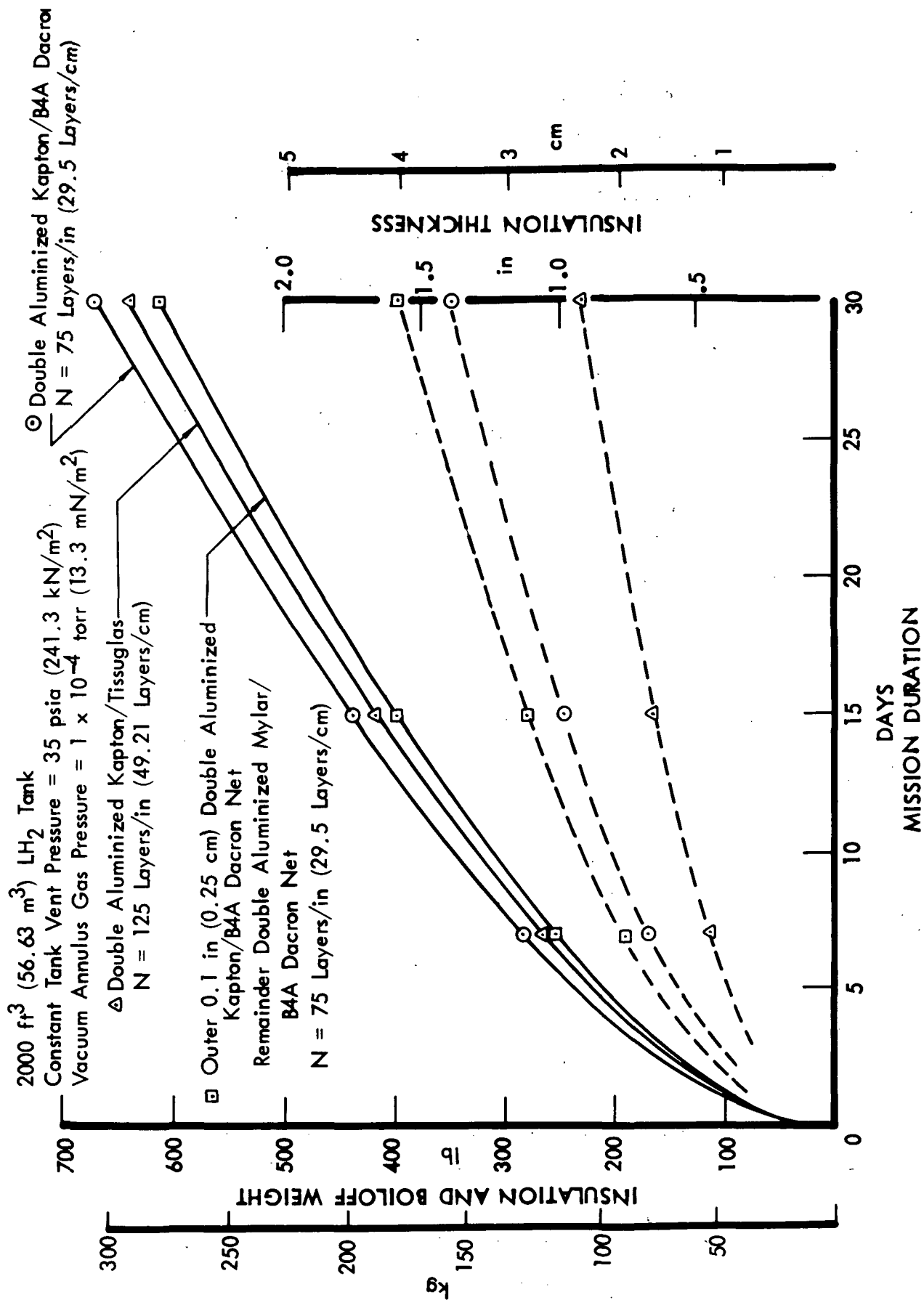


Figure 3-11: OPTIMUM INSULATION AND BOIL-OFF WEIGHT AND INSULATION THICKNESS

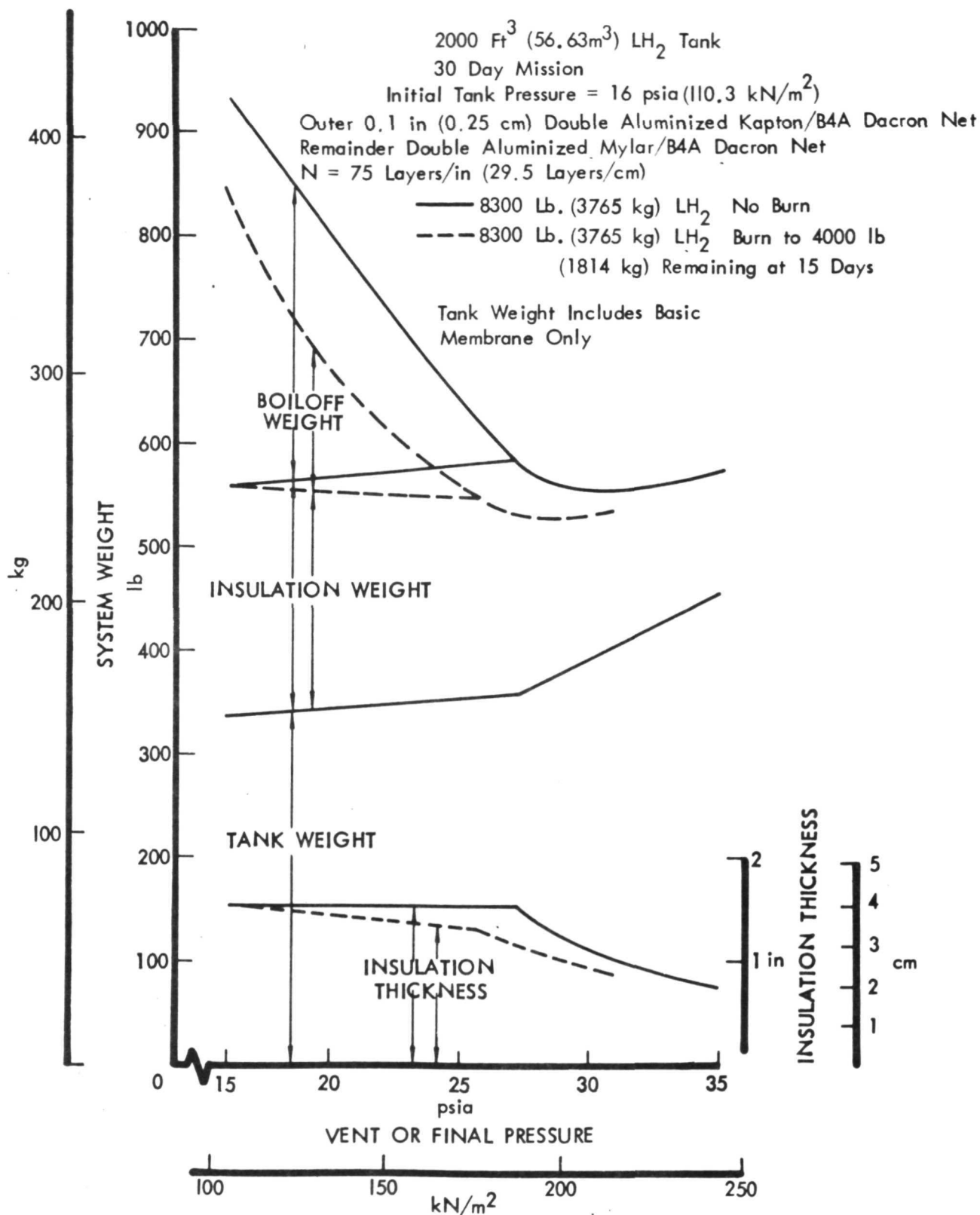


Figure 3-12: OPTIMUM SYSTEM WEIGHT VERSUS PRESSURE FOR LH_2 TANK

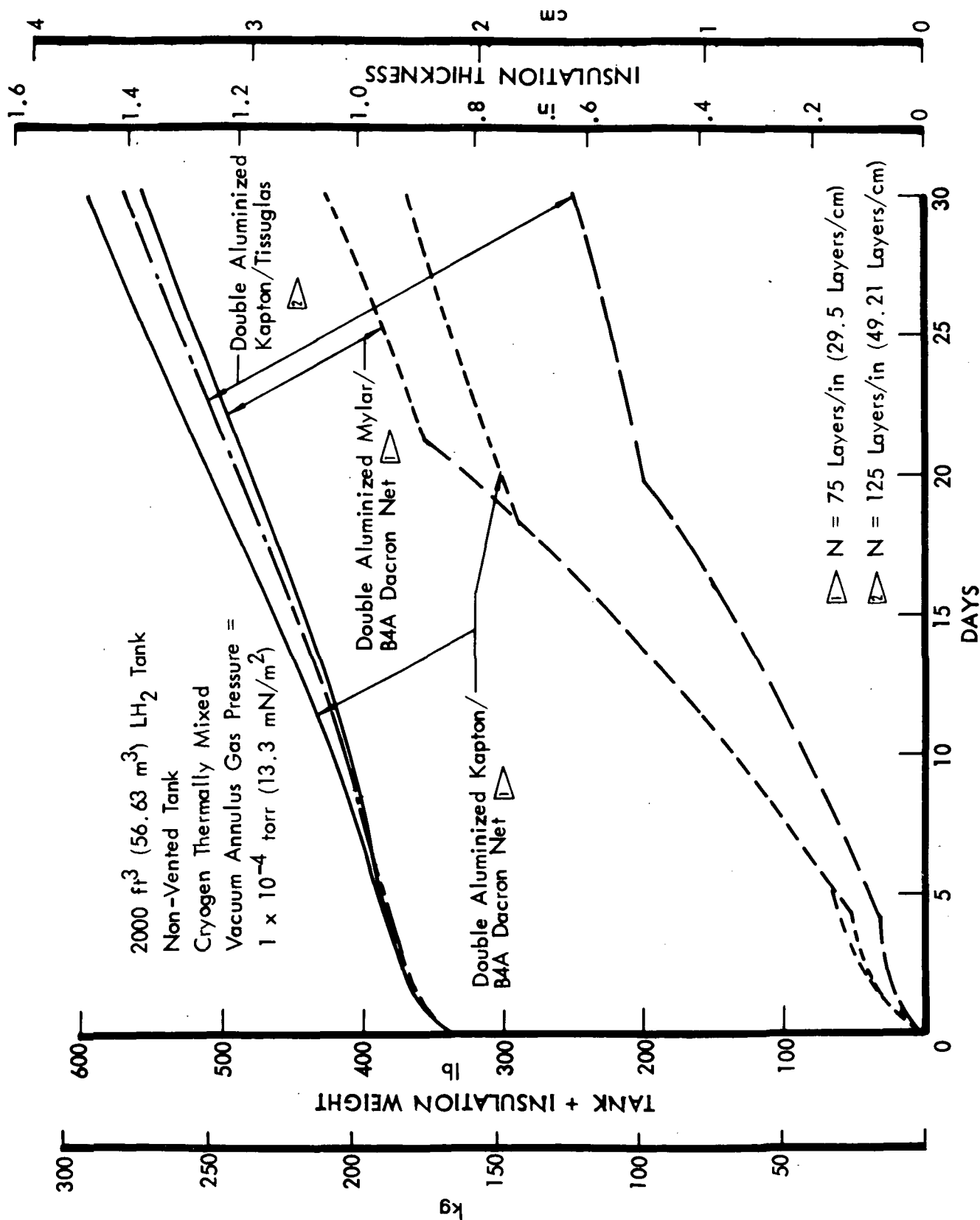


Figure 3-13: OPTIMUM INSULATION AND TANK WEIGHT AND INSULATION THICKNESS

750 ft³ (21.24 m³) LO₂ Tank

30 Day Mission

Initial Tank Pressure = 16 psia (110.3 kN/m²)

Outer 0.1 in (0.25 cm) Double Aluminized Kapton/B4A Dacron Net.

Remainder Double Aluminized Mylar/B4A Dacron Net

N = 75 Layers/in (29.5 Layers/cm)

Tank Weight Includes Basic Membrane Only

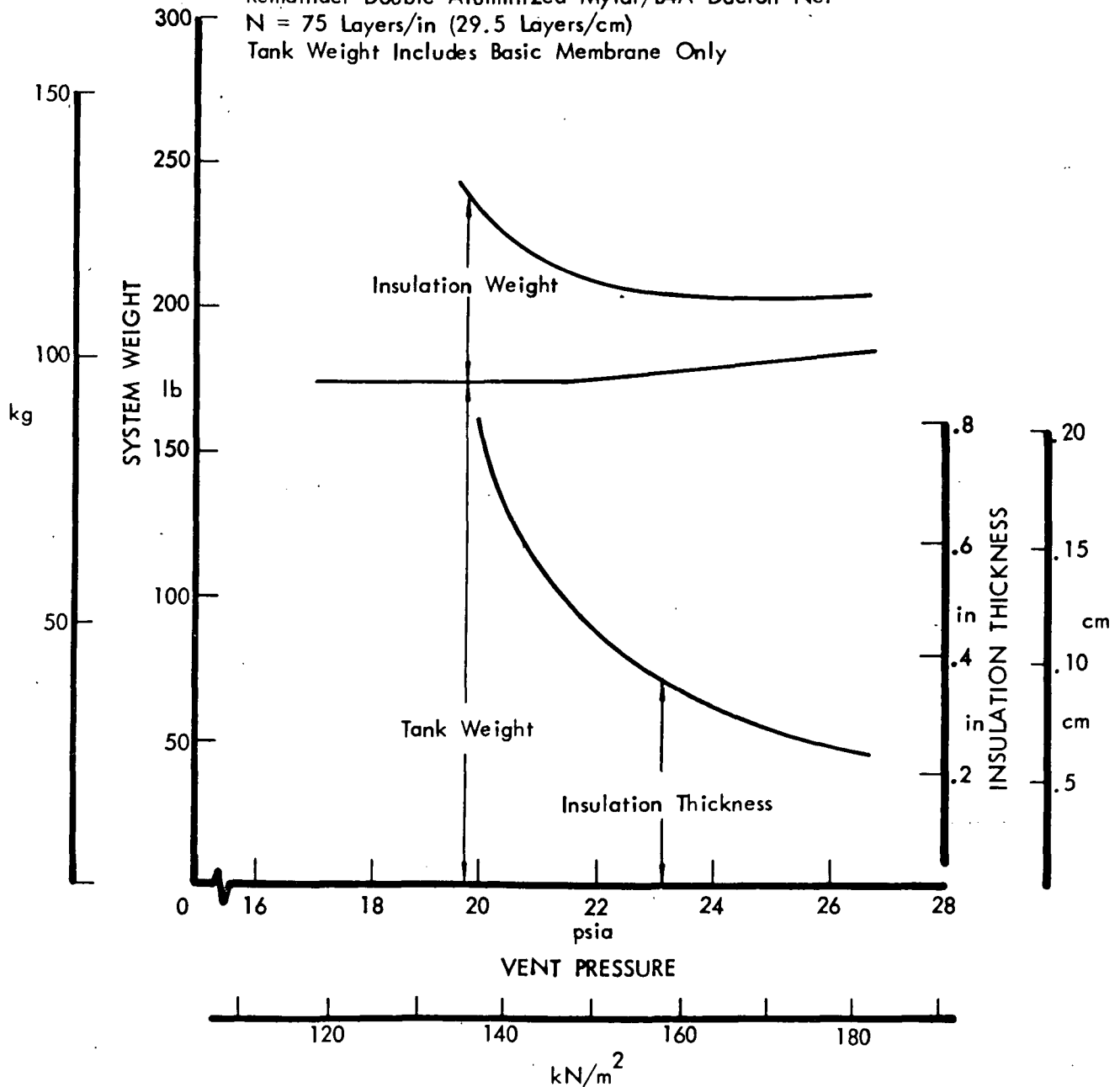


Figure 3-14: OPTIMUM SYSTEM WEIGHT VS PRESSURE FOR LO₂ TANK

3.3.3 Vacuum Acquisition Studies

Vacuum Acquisition Procedures

Figure 3-15 describes the vacuum acquisition equipment and the pumpdown procedures. Options, with and without space venting, are shown. Briefly, the insulation annulus is leak checked and evacuated after tank final assembly. Pumping continues with heavy duty equipment until the vacuum pressure and degradation rate requirements are met. A boil-off test is conducted to determine thermal performance of the system.

During tank storage Vaclon* pumps are activated and a vacuum pressure gage is periodically monitored. This equipment is mounted in a ground service line which is removed prior to installation of the tank into the orbiter. Vacuum pumping continues as required during ground hold and ground turn-around. In the space venting option, the vent valve and the tank shut-off valve are opened in orbit. Prior to reentry, these valves are closed.

Effect of Hydrogen Gas Leakage - LH₂ Tank

This study determined the sensitivity of the selected MLI systems to hydrogen gas leakage. Figure 3-16 shows the effects of hydrogen leak rate on boil-off. The annulus pressures calculated by the program are shown in Figure 3-17. These **curves** are linear, but distorted due to the log scale. The aluminized Kapton/Tissuglas insulation is seen (Figure 3-16) to be less sensitive to increasing gas pressure due to leaks than the dacron net system. This is due to the lower gas conductivity of the Tissuglas as shown by the equations of Section 3.3.2. Because gas flow rates are small, the relative evacuation capability of gas flow resistance, does not influence the results. It was assumed that adequate design provision could be used to prevent very high hydrogen leaks, and that dacron net would be satisfactory. Dacron net insulation was chosen for use in the remaining vacuum acquisition analyses, since it was also acceptable in terms of outgassing.

Effect of Nitrogen Gas Leakage - LH₂ Tank

Several trades were made using a condensible gas flow subroutine for the thermal analysis computer program. The gas flow subroutines, described in Appendix A, are used to determine gas pressure through the insulation thickness from which the gas conductivity can be determined.

The analyses show that, for constant vacuum leak and pumping rates, the presence of cryopumping establishes a steady state pressure distribution in the insulation within the first few hours. These steady state pressures are considerably lower than those previously obtained with a hydrogen gas leak. Figure 3-18 shows gas pressure in the outer and inner portions of the two part insulation blanket as a function of gas leak rate. The gas assumed is nitrogen. Previous

* Product of Varian Associates

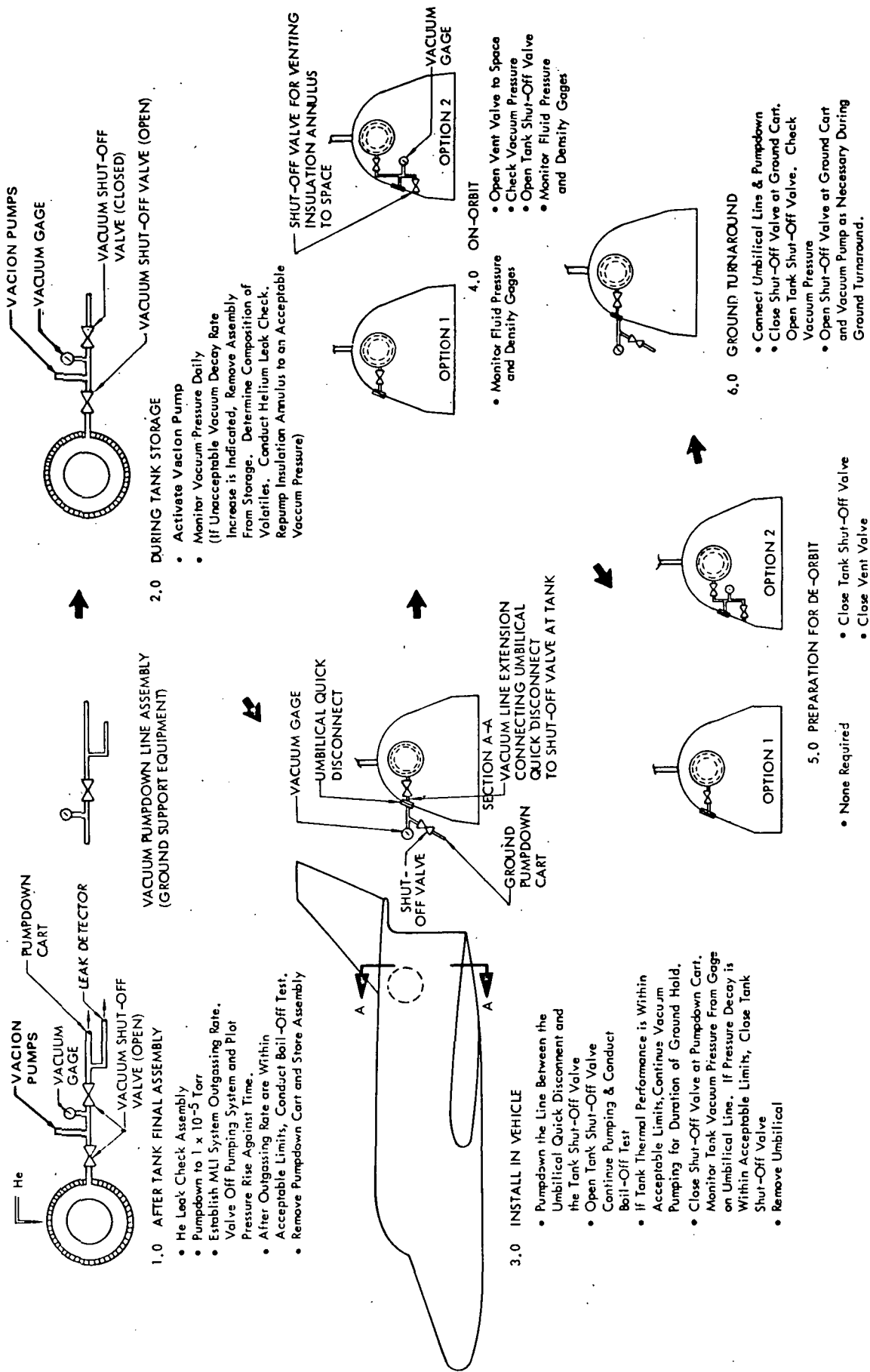


Figure 3-15: VACUUM ACQUISITION SYSTEMS FOR LO_2 AND LH_2 TANKS

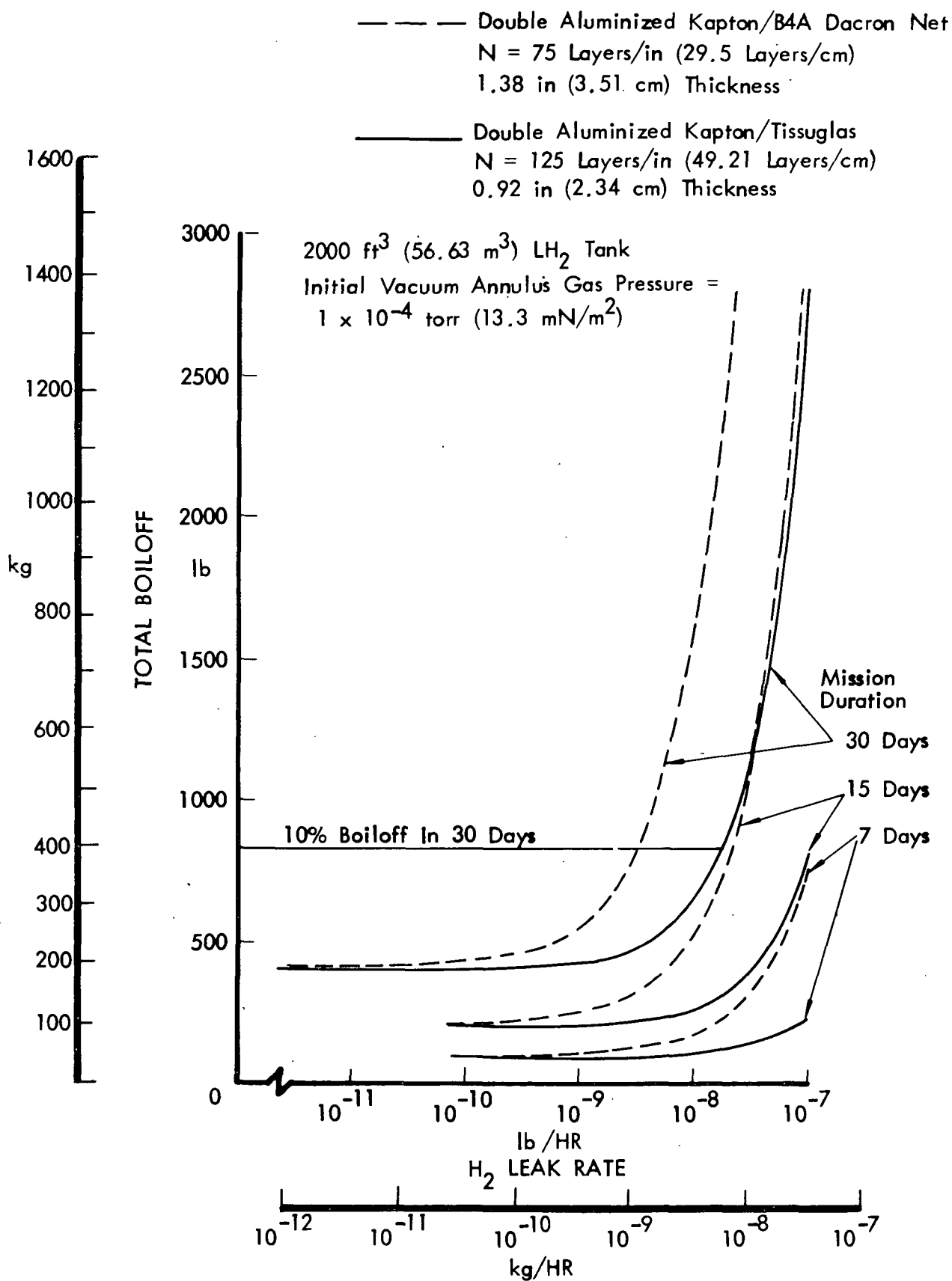


Figure 3-16: EFFECT OF HYDROGEN GAS LEAKAGE ON LH₂ TANK BOIL-OFF

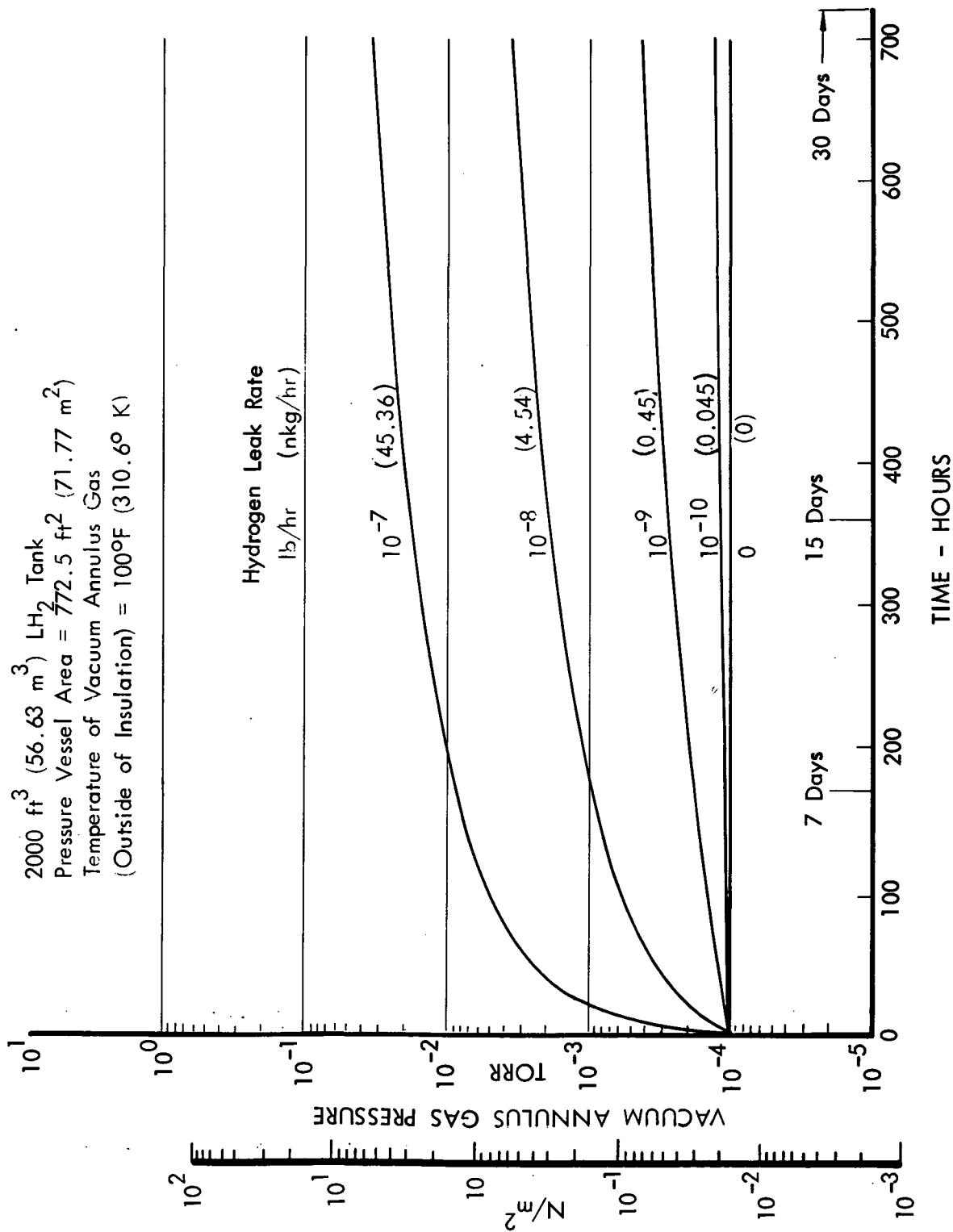


Figure 3.17: VACUUM ANNULUS GAS PRESSURES FOR CONSTANT H₂ GAS LEAK RATES

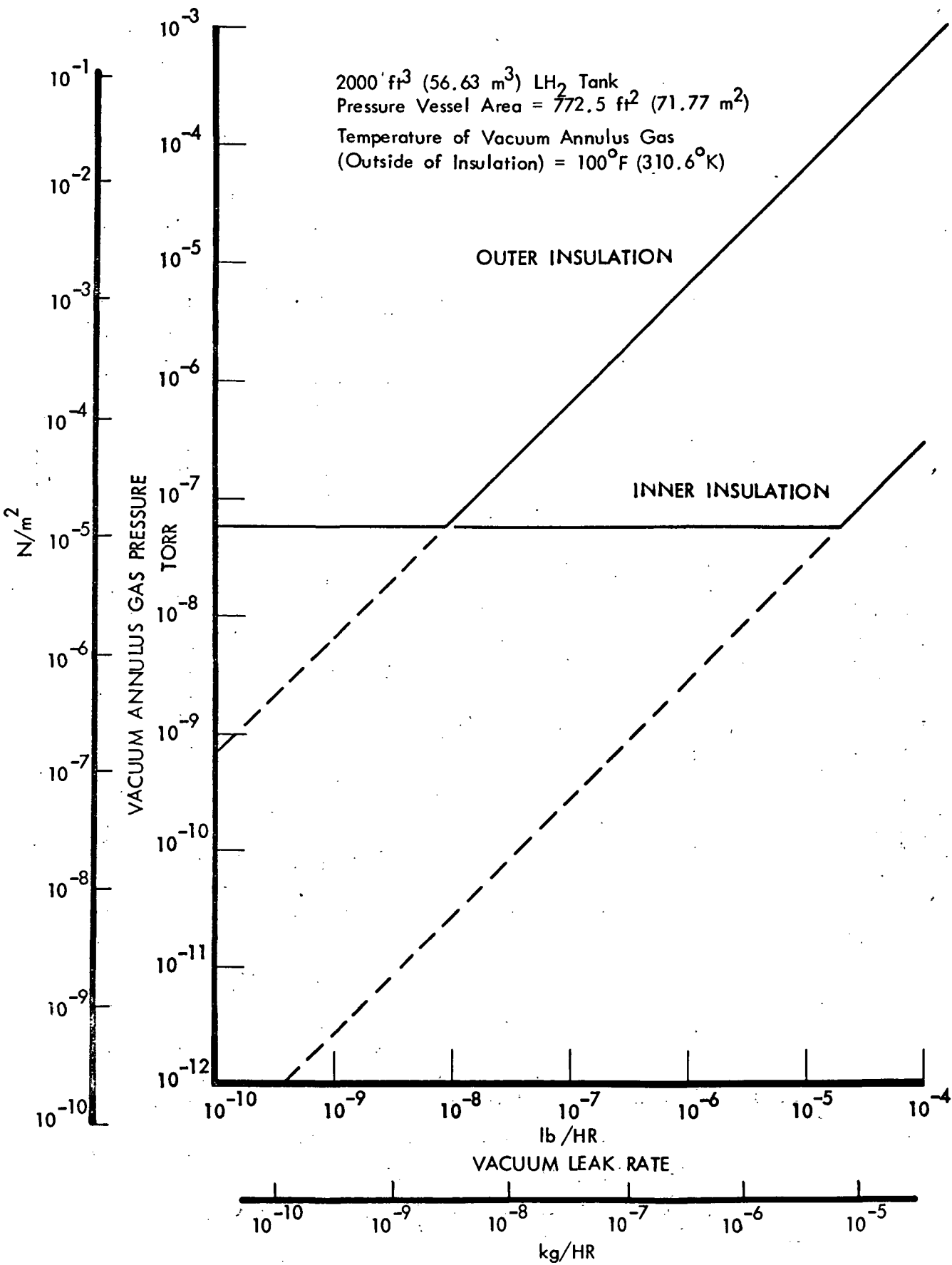


Figure 3-18: EFFECT OF N₂ GAS LEAK RATE ON VACUUM ANNULUS GAS PRESSURE AFTER 30 DAYS

results showed a pressure of 3×10^{-2} torr (4 N/m^2) after 30 days for a 1×10^{-7} lb/hr (12.5 pkg/sec) hydrogen leak. With condensing nitrogen at the same leak rate, the pressure is 6.5×10^{-7} torr ($86.5 \mu \text{ N/m}^2$) in the outer insulation layers.

The pressure curves are shown dashed below 5.7×10^{-8} torr ($7.6 \mu \text{ N/m}^2$) in Figure 3-18 because this is the minimum pressure expected with LH_2 cryopumping of N_2 gas. As discussed in Appendix A the analytical model did not include this effect due to computer stability problems.

Figure 3-19 shows the amount of LH_2 boil-off as a function of the nitrogen leak rate. From these curves, it appears that a condensible gas leak of 1×10^{-6} lb/hr (1.3 pkg/sec), or 2.9×10^{-4} std cc helium/sec, could be tolerated with essentially no increase in boil-off. This is about 4 orders of magnitude higher than the tolerable gaseous hydrogen leak rate.

The analyses indicate that sealing of the vacuum jacket (the source of condensible gas leaks) is much less critical than sealing of the pressure vessel (the source of hydrogen leaks). However, the gas pressure in the insulation undoubtedly will be higher than predicted by the condensible gas analysis due to the presence of non-condensable gases and dissociation of molecules containing hydrogen, such as water. Prediction of actual gas concentrations would be very difficult and probably not accurate enough to justify the effort.

Figures 3-20 and 3-21 show the effect of pumping during flight on the gas pressures and total boil-off for nitrogen gas leak rate of 1×10^{-4} lb/hour (12.5 nkg/sec). Origin of this gas might be air trapped in the core unable to vent to space through a sealed outer face skin. The upper curve of Figure 3-21 is reproduced in Figure 3-22 along with the additional weight of a Vaclon pump and magnet required to get the pumping speed on the abscissa. It is evident from total weight on Figure 3-22 that the on-board pump system has little advantage, saving about 12 lb (5.44 kg) at the most for the assumed leak rate. The weight of the controller for the pump has not been included, and it is expected that the complete system would not save any weight. A further disadvantage with the ion pumps is the relatively high gas pressure at the pump (the upper curve of Figure 3-18) which would result in very short pump life.

Effect of Gas Leakage - LO_2 Tank

Leak tightness of the vacuum jacket is critical on the LO_2 tank. Nitrogen gas does not cryopump to the -297°F (90.2°K) pressure vessel surface. Vacuum integrity therefore relies on obtaining low leakage rates through the vacuum jacket.

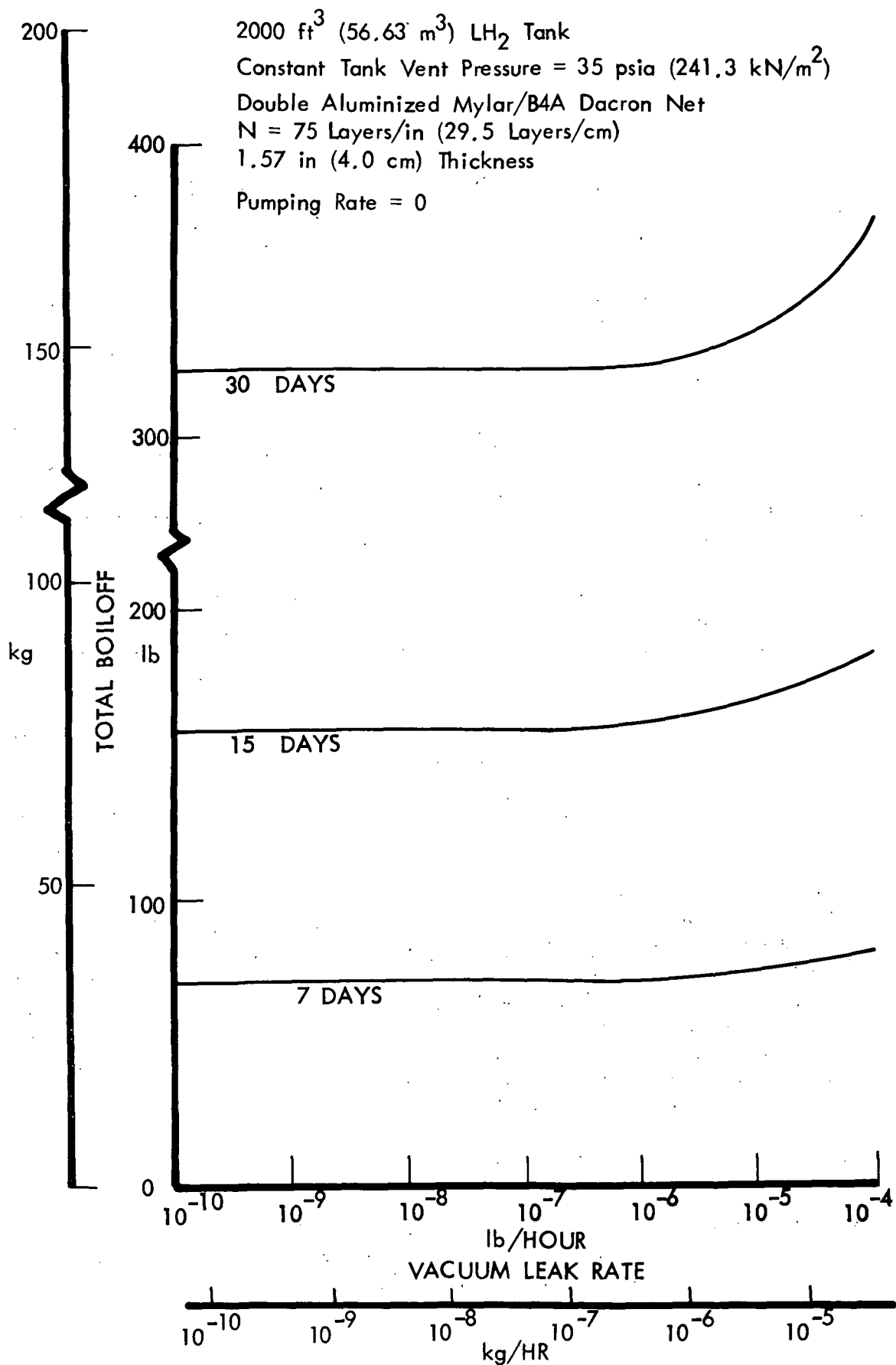


Figure 3-19: EFFECT OF N₂ GAS LEAKAGE ON LH₂ TANK BOILOFF

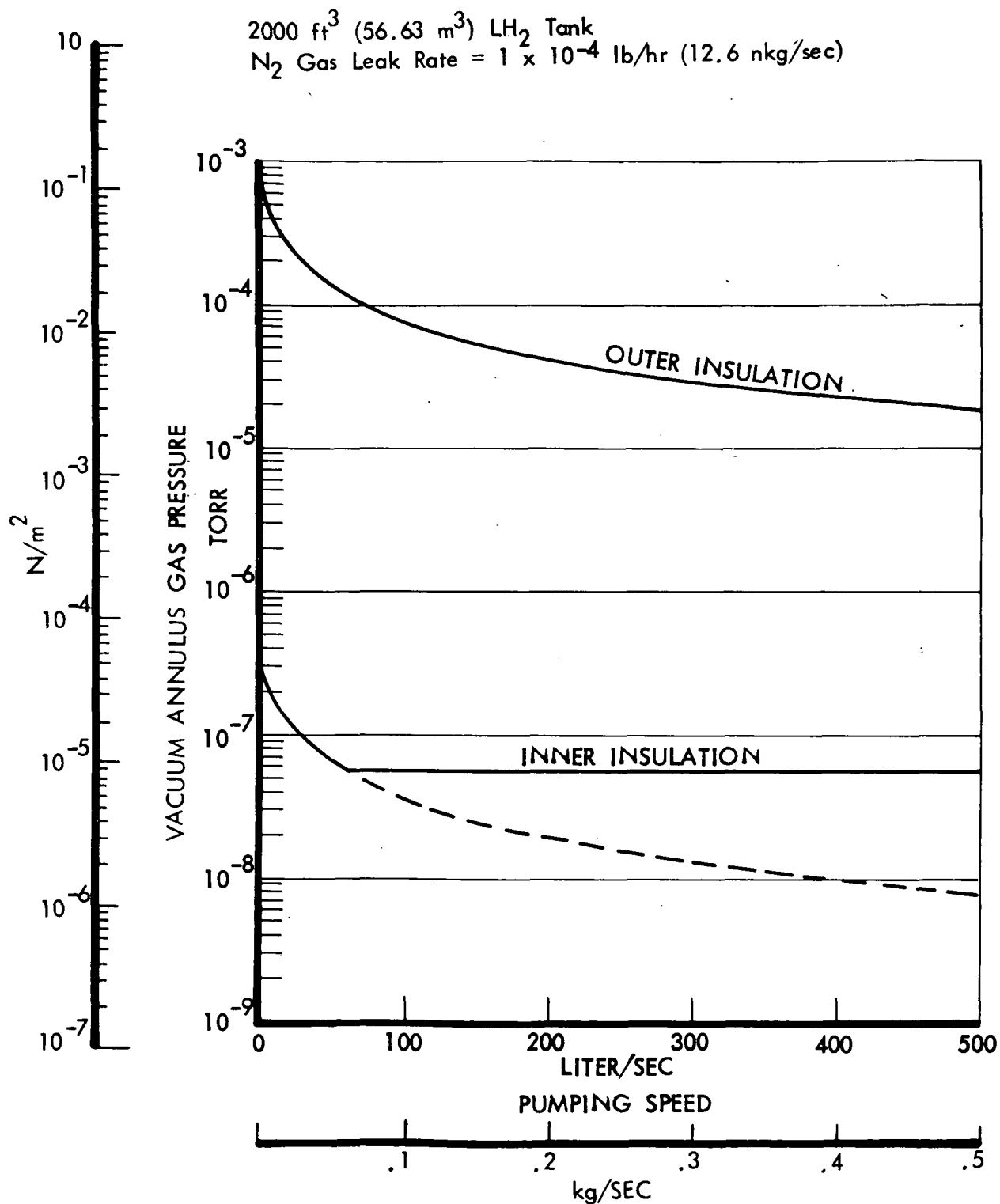


Figure 3-20: EFFECT OF INFLIGHT PUMPING ON VACUUM ANNULUS PRESSURE

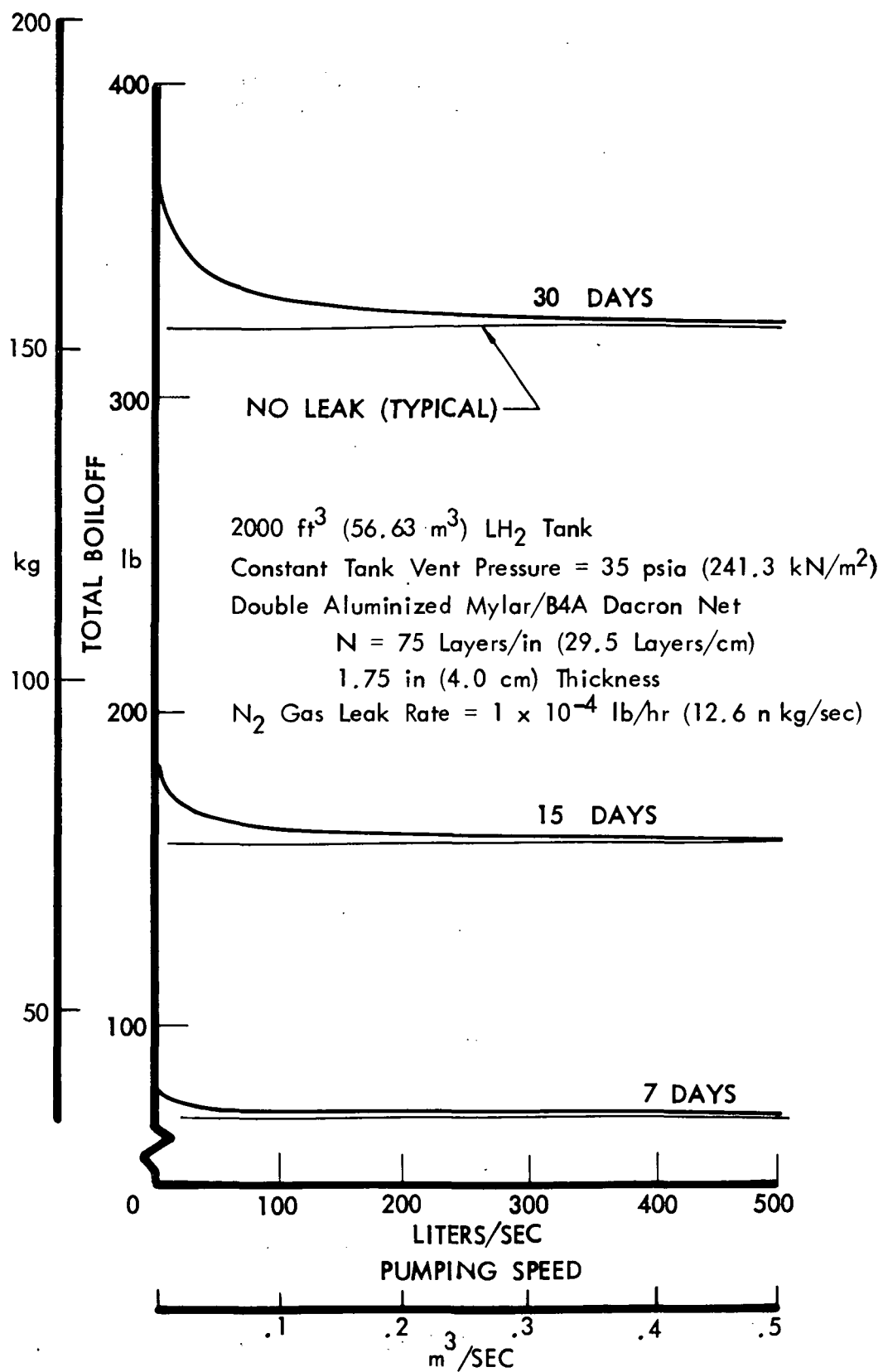


Figure 3-21: EFFECT OF PUMPING ON LH₂ TANK BOILOFF

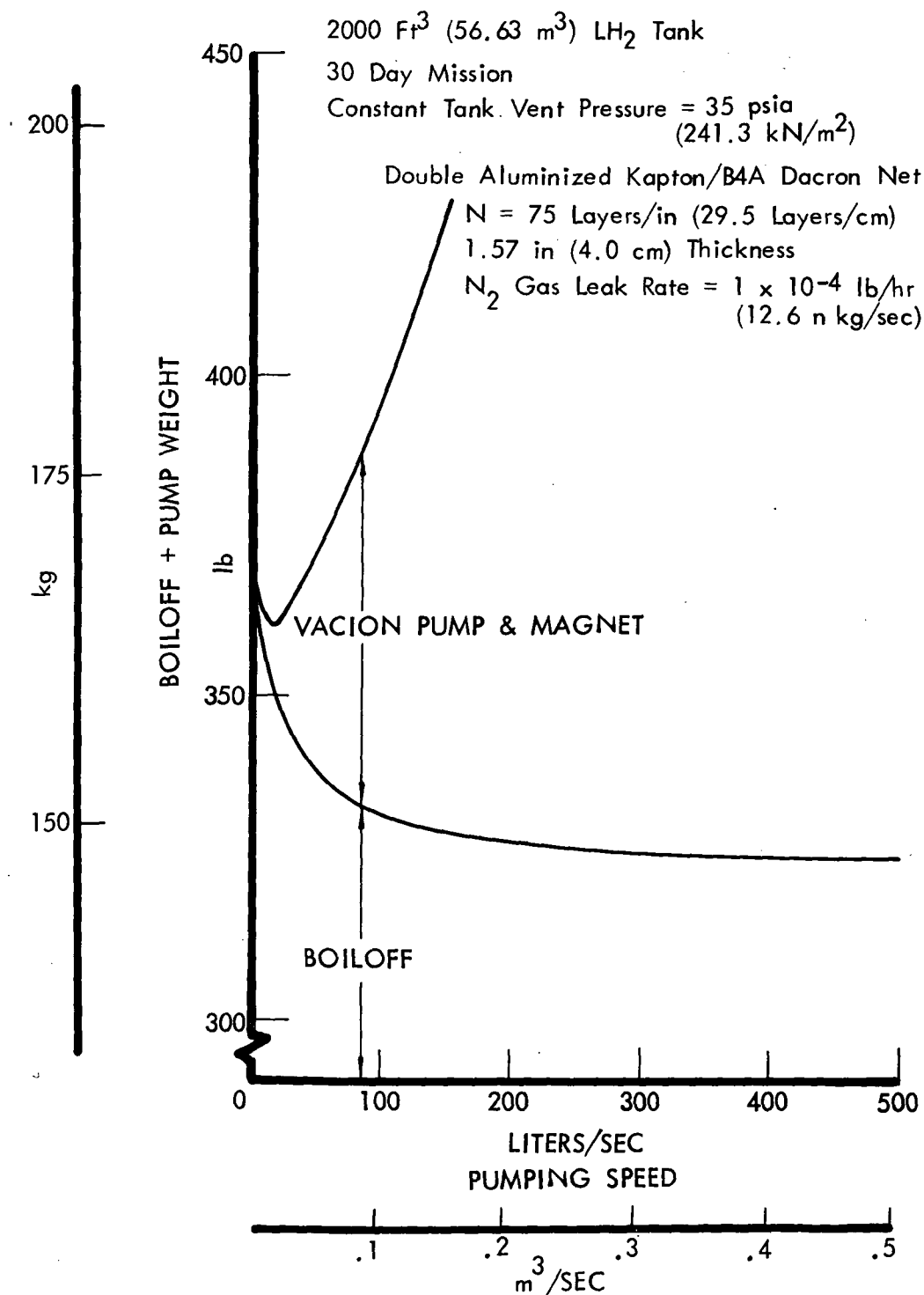


Figure 3-22: EFFECT OF INFLIGHT PUMPING ON SYSTEM WEIGHT

3.3.4 Propellant Leakage Isolation

Design Studies

The manhole access configurations shown in Figure 3-23 were studied to determine their merits as propellant leakage isolation methods. The bellows arrangement, shown in Figure 3-23 (a), is excellent for isolating propellant leakage from the vacuum annulus, but provides another heat leak path (through the bellows) to the cryogen. The metallic seal arrangement, shown in Figure 3-23 (b), eliminates the heat leak path problem, but depends on the reliability of the metallic seal to ensure no contamination of the vacuum annulus. There appears to be no data which indicates that a seal of this size, once having passed a leak test, will maintain that sealing capability through 100 flights. The welded annulus arrangement with a overboard vent, shown in Figure 3-23 (c), has the advantages of both preceding configurations. The one disadvantage is the welding required during manufacture and each time the cover is removed for access to the pressure vessel interior. This disadvantage is overcome by the arrangement shown in Figure 3-23 (d).

The plumbing line penetrations shown in Figure 3-24 have propellant leakage isolation methods similar to the manhole covers. Comments on arrangements shown in Figures 3-23 (a), (b) and (d) apply to Figures 3-24 (a), (b) and (c). The submerged valve arrangement is comparable to the arrangement in Figure 3-24 (c) with the added advantage of reducing the size of the protrusion in the vacuum jacket.

Thermal Analysis

The manhole access with bellows and the manhole access without bellows were evaluated. Both approaches optimized between 0.1 and 0.7 Btu/ft²-hr (0.32 to 2.21 W/m²) considering the heat leak and zero gas leakage. Figure 3-25 shows these optimization curves. A line representing 10% boil-off losses has been used for illustration. At the optimum insulation thickness there is a margin of 223 lb (101.2 kg) between the actual boil-off loss curve and the 10% line, for the concept with the bellows. An error of twice the estimated value of MLI thermal conductivity can consume this entire margin. There are also errors to be expected in estimating penetration heat leak which have not been included. The concept without the bellows and no gas leakage gives about double the margin.

The concept without the bellows but with gas leakage is shown in Figure 3-26. The seal leakage rates shown here are actually somewhat lower than that obtained on the 9 ft (2.74 m) tank fabricated on Contract NAS 3-7957. The seal leakage rate for metallic seals is based on vendor catalog data. The 10% maximum propellant loss rate for the 30 day mission could not be maintained with the leak rate from the 9 ft. (2.74 m) tank, (1×10^{-8} lb/hr) (1.24 ng/sec), using a Creavey Seal.

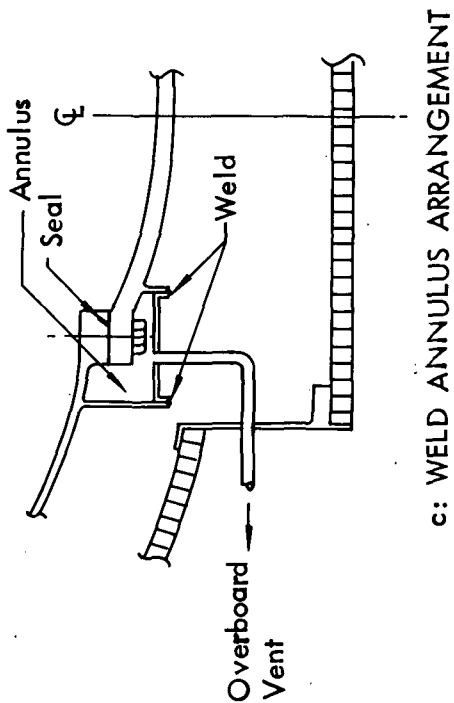
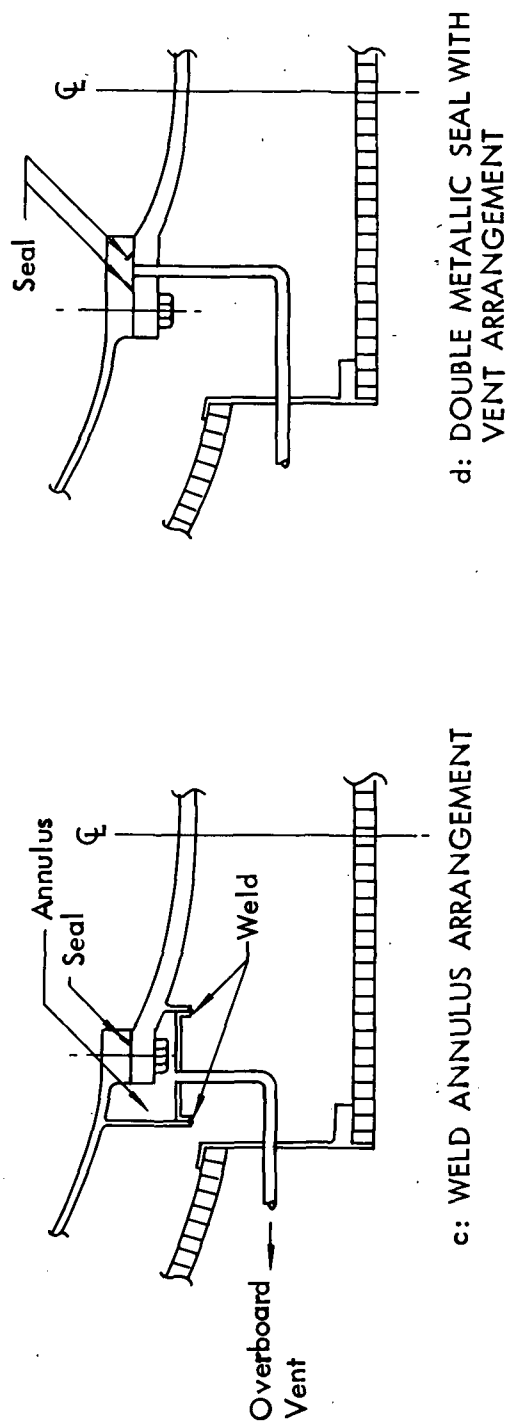
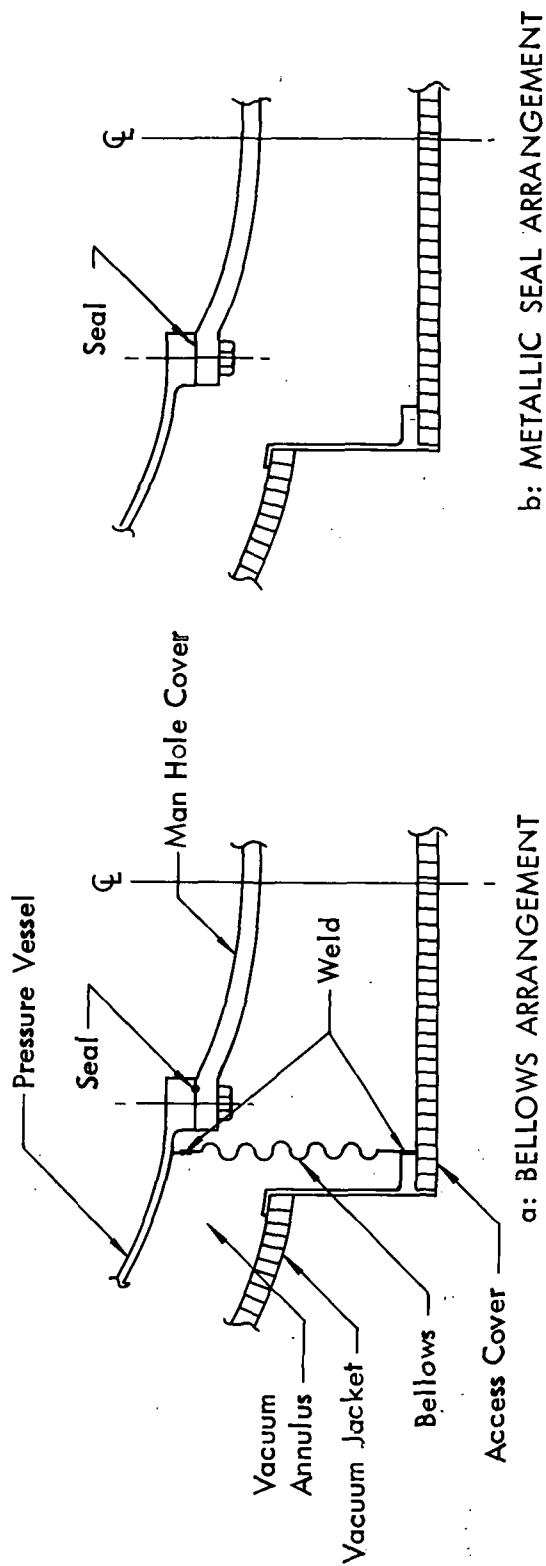


Figure 3-23: PROPELLANT LEAKAGE ISOLATION CONCEPTS - MANHOLE ACCESS

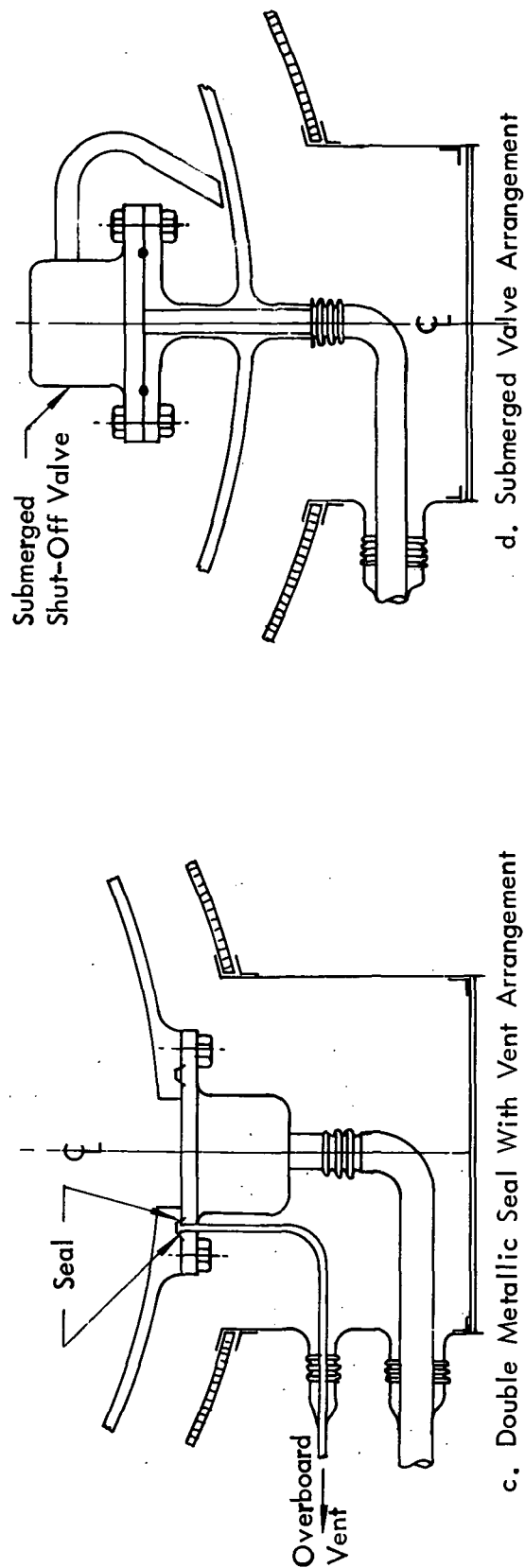
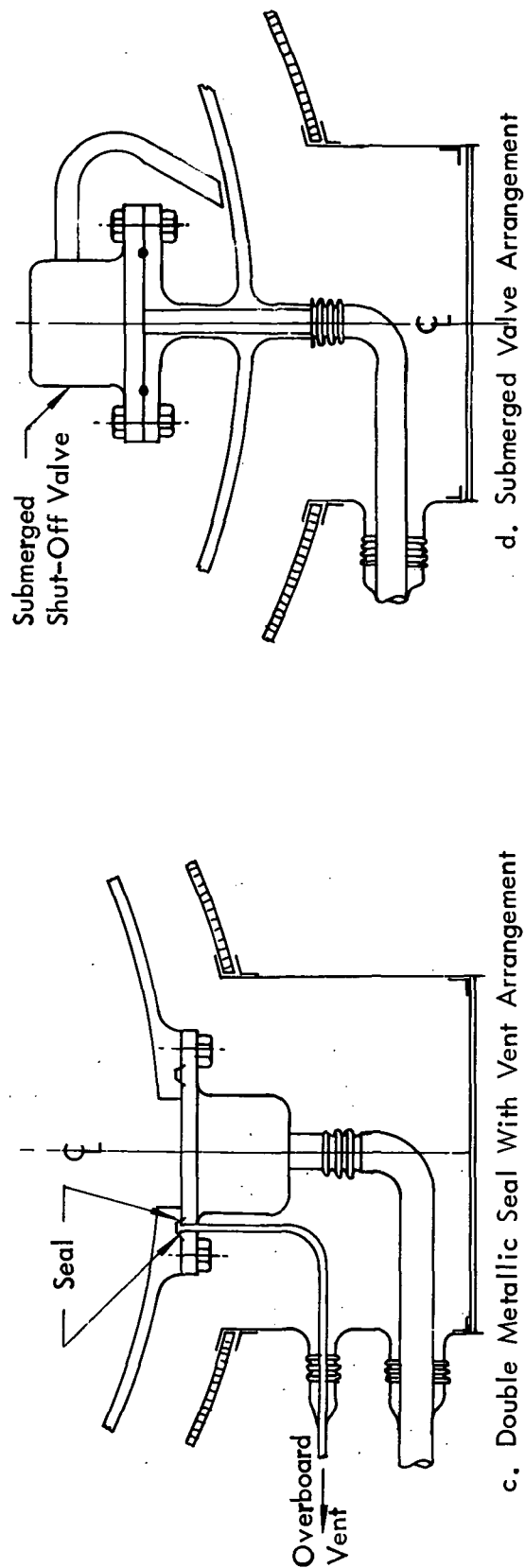
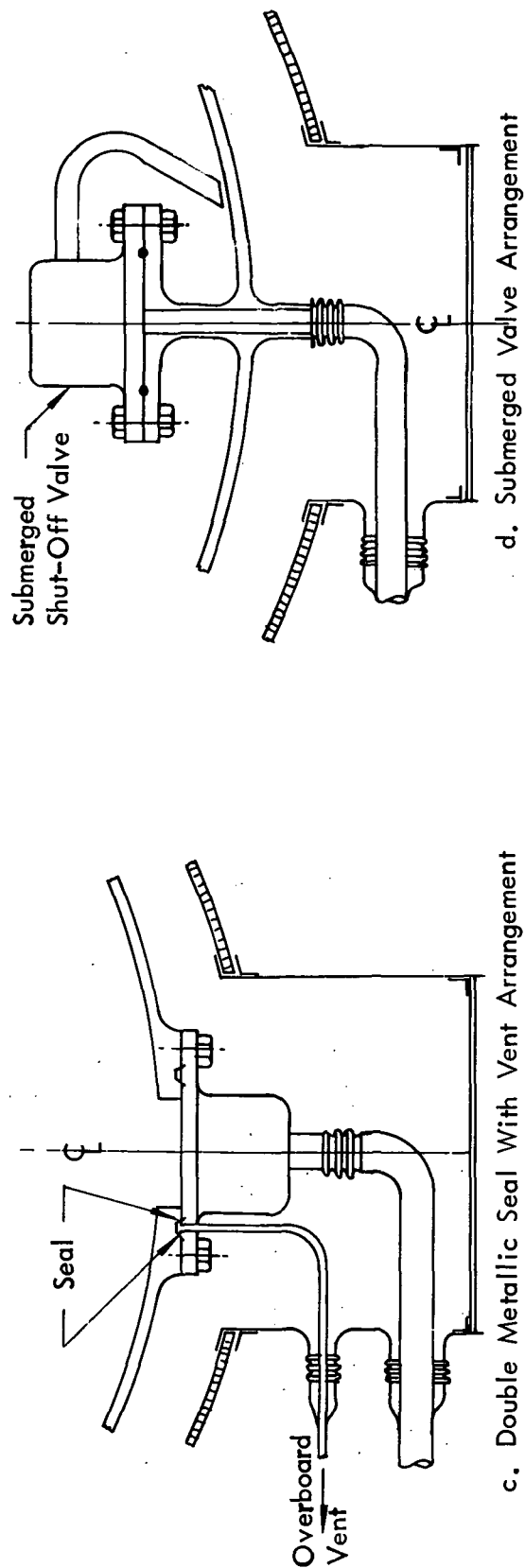
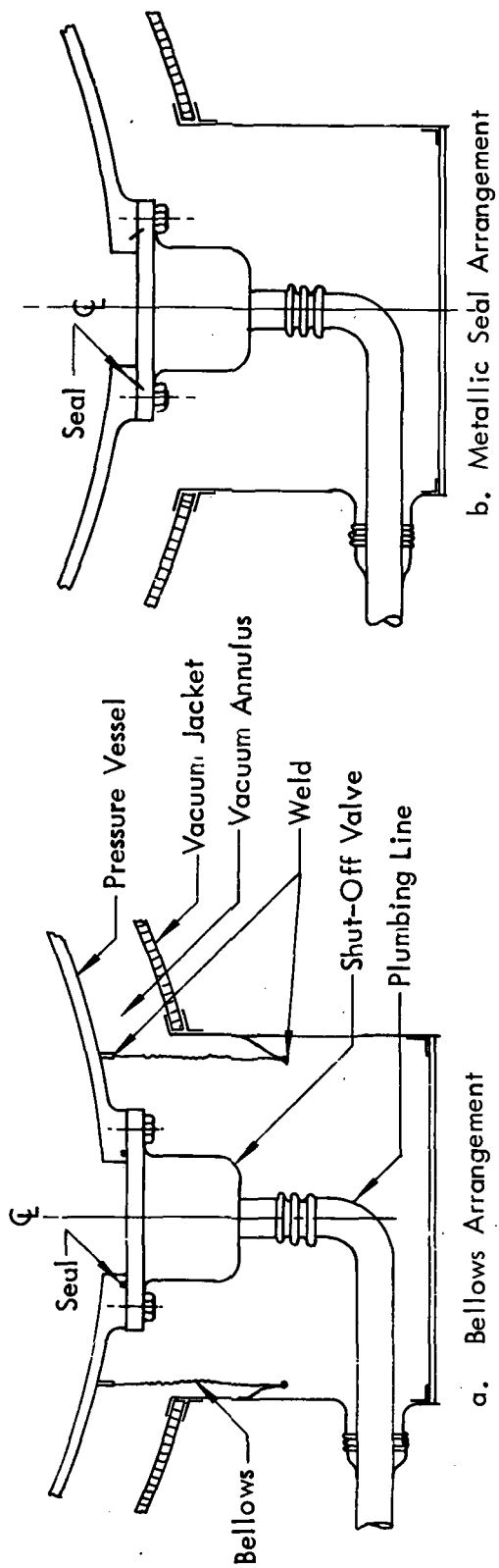


Figure 3-24: PROPELLANT LEAKAGE ISOLATION CONCEPTS - PLUMBING LINES

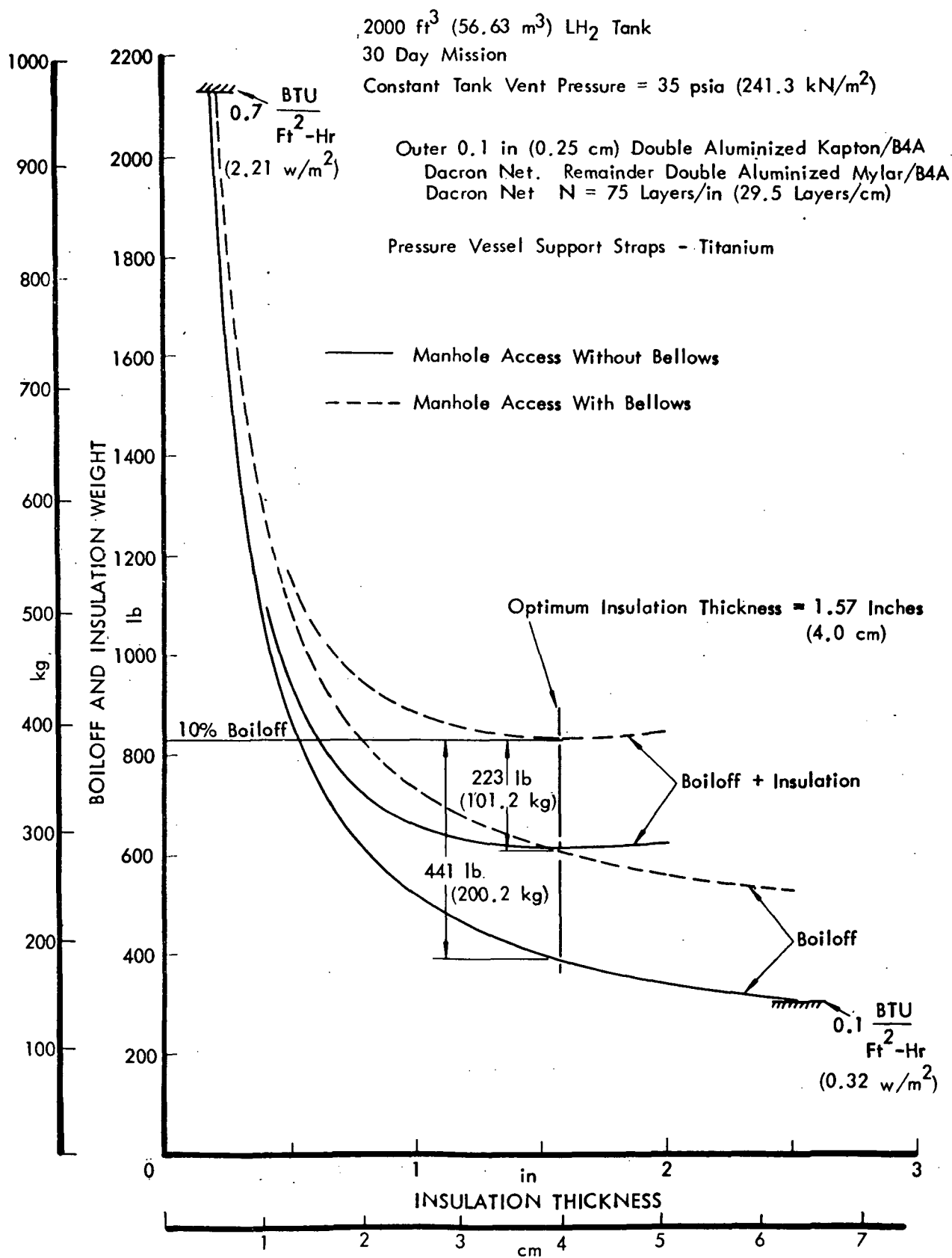


Figure 3-25: EFFECT OF MANHOLE CONFIGURATION ON WEIGHT

2000 ft³ (56.63 m³) LH₂ Tank

Constant Tank Vent Pressure = 35 psia (241.3 kN/m²)

Initial Vacuum Annulus Gas Pressure = 1×10^{-4} torr
(13.3 mN/m²)

Double Aluminized Kapton/B4A Dacron Net

N = 75 Layers/in (29.5 Layers/cm)

1.38 in (3.5 cm) Thickness

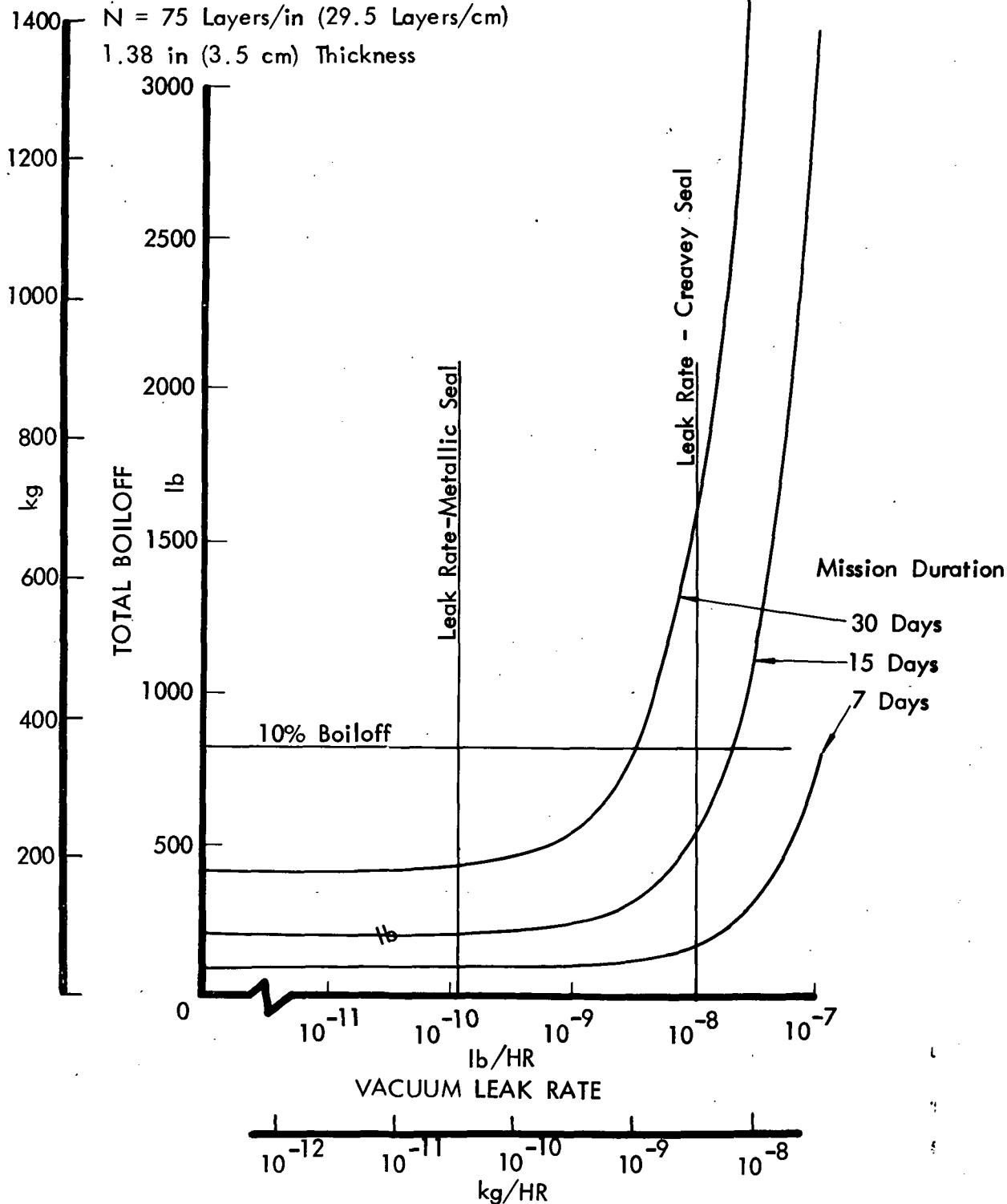


Figure 3-26: EFFECT OF H₂ GAS LEAKAGE ON BOILOFF WEIGHT

The margin between a 10% boil-off loss, 830 lb (376.5 kg), and the 30 day mission curve is 390 lb (176.9 kg). The MLI performance is extremely sensitive to gas pressure and if the leak rate increases to 5×10^{-9} lb/hr (0.62 ng/sec) the margin is eliminated. In view of the results obtained for the Creavey Seals it appears that exposing mechanical sealed joints to the vacuum annulus is risky. The concepts shown in Figures 3-23 (c) and (d) will remove most of this risk.

3.4 Shell Trade Studies

The purpose of the shell trade studies was to produce data for an optimum choice of the vacuum jacket. These data were evaluated with the data from the tank configuration and thermal studies to recommend an LH₂ and LO₂ design concept. The recommended designs are discussed in Section 3.6. Two types of shells were studied: Self Supporting Vacuum Shells and Semi-Rigid Vacuum Shells. The semi-rigid shells only resist circumferential loads; axial loads are resisted by other structure.

3.4.1 Self-Supporting Vacuum Shells

Three parameters were considered in this study: (1) tank shape, (2) materials, and (3) the method of shell construction. The results of the study are families of curves showing the weight of each shell versus L/D and tabulated data for optimum weight designs.

The study was conducted in four steps. It was assumed from preliminary trade studies that honeycomb sandwich would be a prime candidate for the optimum design concept. Material combination trade studies were conducted first to select the best combination of materials for honeycomb sandwich shells. The second step was an investigation of the design criteria for the girth rings used to join the hemispherical heads to the cylindrical sections of the shells. The third step was a study of shell construction methods including honeycomb sandwich. Total weights in this study included the end rings. The fourth step investigated the effect of design factors on the total weight of the vacuum jacket. Details of each step are discussed in the following four sections.

3.4.1.1 Sandwich Material Combinations

Sandwich material trade studies were conducted for a 2000 cu. ft. (56.63 m³) LH₂ tank and a 750 cu. ft. (21.24 m³) LO₂ tank to determine the vacuum jacket weight for several combinations of sandwich materials. A 4.5 in. (11.4 cm) vacuum annulus was provided between the pressure vessel and the vacuum jacket for the MLI. A maximum structural temperature of 350°F (450°K) was assumed for all the materials. At least one face of the sandwich was metallic for vacuum sealing purposes. The trade studies evaluated the effect of (a) HRP versus aluminum 5056 Flex-Core, (b) a single metal skin on the inside or outside face, and (c) material combinations of the best core material and face configurations for different face materials.

Design Conditions

The OPTRAN computer program described in Reference 9 was used to perform these trades. A limit design external pressure of 14.7 psi (101.4 kN/m²) was used with a 1.4 ultimate factor of safety. Launch loads were not considered, nor were weight allowances for fittings and joints made. A uniform shell temperature of 350°F (450°K) was assumed. Also, these studies assumed sandwich cylinders without stiffening rings. The analysis methods recommended by Sullins, et al, (Reference 10) were used for the sandwich vacuum jackets.

A 99 percent probability of the design not failing under the design ultimate external pressure was used for the hemispherical head and cylinder analyses to determine the knockdown factors. Boeing statistical knockdown factors were used in place of those recommended by Reference 10.

All weights presented include the face skins, core and bonding adhesive. An adhesive weight of 0.0006 lb/in² (0.42 kg/m²) for each surface was used.

Allowable Face Properties

The allowable face skin properties used in these analyses are shown in Table 3-5. It was assumed that one metallic face skin would be required on the sandwich shell designs as a vacuum sealing surface. Aluminum and titanium were selected as candidates for face skin. Aluminum alloy 2219 was selected for its high temperature strength properties and excellent weldability. Annealed titanium was used to minimize fabrication difficulties. The allowable material properties were determined from the Boeing Design Manual, Book 84.

Fiberglass was selected as a candidate since it is used extensively in the manufacture of sandwich structures. Both epoxy and polyimide resin systems were included to evaluate different resin systems, since the 350°F (450°K) requirement is at the upper limit of the modified epoxy systems.

Boron/epoxy laminates were also selected as a face skin material for these studies because of their high stiffness/density ratio. The composite allowable properties were estimated from the material specifications with manufacturer's technical data.

Allowable Core Properties

HRP* (Heat Resistant Phenolic) core and 5056 aluminum Flex-Core* were selected for their elevated temperature strength and stiffness. Table 3-6 lists the allowable HRP core properties. These properties were estimated from Hexcel data for the effect of material scatter, temperature, and thermal cycling. The factors used were:

- 80% of the Hexcel typical values to obtain allowable values
- 92% of room temperature properties for 1/2 hour at 350°F (450°K)
- 92.5% of 1/2 hour at 350°F (450°K) values for 100 hours at 350°F (450°K)

*Registered Trade Mark of Hexcel Aerospace

Table 3-5: ALLOWABLE FACE SKIN PROPERTIES AT 350°F (450°K)

MATERIAL	α		E_x		E_y		G_{xy}		Poisson's Ratio μ_x	F_{cy}		F_{creep}		ρ	
	$\ln/\ln_F \times 10^{-6}$	$\mu m/m/^\circ K$	psi	GN/m^2	psi	GN/m^2	psi	GN/m^2		psi	kN/m^2	psi	kN/m^2	lb/in^3	kg/m^3
2219-T81 Aluminum	10	18.0	9.5×10^6	65.5	9.5×10^6	65.5	3.53×10^6	24.3	0.33	34,000	234,422	20,000	137,895	0.10	2768
6Al-4V Titanium	5	9.0	14.7×10^6	101.4	14.7×10^6	101.4	5.7×10^6	39.3	0.30	100,000	689,476	100,000	689,476	0.16	4429
Fiberglass/Epoxy	5	9.0	2.2×10^6	15.2	2.2×10^6	15.2	$.35 \times 10^6$	2.41	0.125	22,000	151,685	15,000	103,421	0.072	1993
Fiberglass/Polyimide	2.4	4.32	2.8×10^6	19.3	2.8×10^6	19.3	$.47 \times 10^6$	3.24	0.125	36,000	248,211	24,000	165,474	0.072	1993
Rigidite 5505/4 Boron/Epoxy (0°/90°)	2.5	4.50	14.0×10^6	96.5	14.0×10^6	96.5	1.0×10^6	6.89	0.028	68,000	468,844	25,000	172,369	0.073	2021

1 Not to be exceeded under limit load conditions.

2 Assumed

3 AVCO Systems Division, ALLOWABLES ESTIMATED

Table 3-6: ALLOWABLE HRP CORE PROPERTIES AT 350°F (450oK)

Hexcel Size	Density		E_c		(G_L) ▢		(G_w) ▤		Core tension		Core compression	
	Inches	cm	lb/Ft ³	kg/m ³	psi	kN/m ²	psi	kN/m ²	psi	kN/m ²	psi	kN/m ²
3/16		.476	4.0	64	38,800	267,500	7,800	53,780	3,400	23,440	408	2810
1/4		.635	3.5	56	31,300	215,800	6,120	42,190	2,380	16,410	340	2348
3/8		.953	3.2	51.2	25,800	177,900	5,440	37,510	2,040	14,060	300	2070
3/8		.953	2.2	35.2	8,850	61,020	3,400	23,440	1,360	9,380	170	1170

▢ Shear Parallel to the Core Ribbon

▤ Shear Transverse to the Core Ribbon

Table 3-7 : ALLOWABLE 5056 ALUMINUM FLEX-CORE PROPERTIES AT 350°F (450°K)

Hexcel Designation	Approximate Honeycomb Cell Size		Density		E_c		(G_L) ▢		(G_w) ▤		Core Tension Strength		Core Compression Strength	
	Inches	cm	lb/Ft ³	kg/m ³	psi	kN/m ²	psi	kN/m ²	psi	kN/m ²	psi	kN/m ²	psi	kN/m ²
F40	.300	.762	2.1	33.6	32,300	222,701	8,920	61,501	4,960	34,198	129	889	129	889
F40	.300	.762	3.1	49.6	62,000	427,475	15,850	108,937	6,700	46,194	233	1606	233	1606
F40	.300	.762	4.1	65.6	92,000	634,318	22,300	153,753	8,430	58,123	342	2358	342	2358
F80	.150	.381	4.3	68.8	96,900	668,102	23,300	160,648	8,940	61,639	368	2537	368	2537

▢ Shear Parallel to the Core Ribbon

▤ Shear Transverse to the Core Ribbon

Table 3-7 lists the allowable 5056 aluminum Flex-Core properties at 350°F (450°K). These properties were factored from Table 3 in Reference 11. The factors were:

- 80% of the Hexcel typical values to obtain allowable factor
- 62% of the room temperature properties for 100 hours at 350°F (450°K)

Face Skin and Core Trade Study for LH₂ Vacuum Jacket

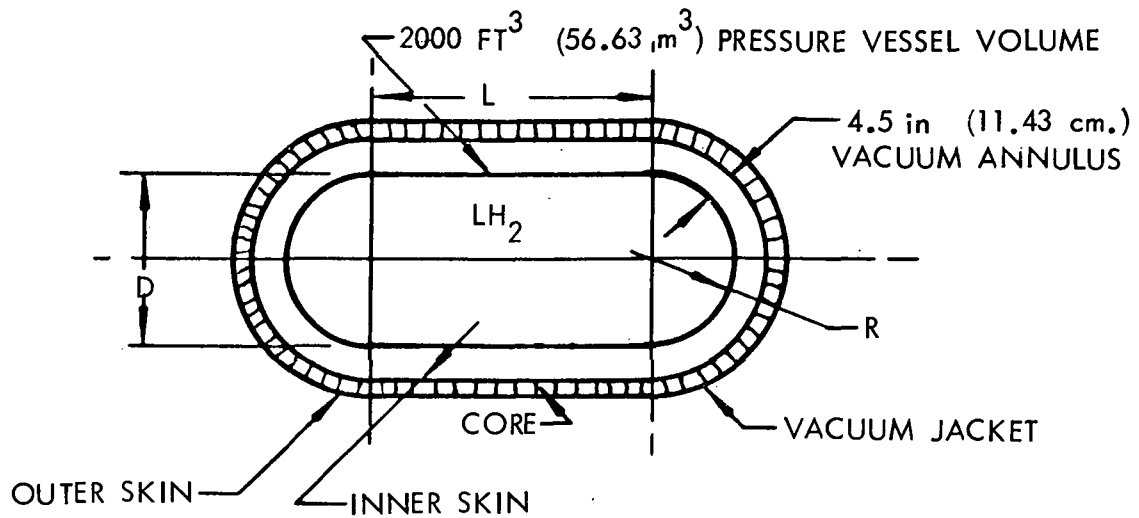
Four vacuum jacket geometries were studied to obtain four different L/D ratios. They are shown in Figure 3-27. The weight of the two heads and the cylinder were determined for each L/D ratio and material combination. The total jacket weights, exclusive of end rings, for these designs are plotted versus L/D for different combinations of core material and metal skin location in Figures 3-28, 3-29, 3-30, and 3-31. Detail data from these shell trade studies are tabulated in Appendix B.

Figures 3-28 and 3-29 are for sandwich material combinations with HRP core. Figure 3-28 shows the weight when the inner skin is used for vacuum sealing. Figure 3-29 uses the outer skin for vacuum sealing. There was no significant weight difference in using the inner or outer skin for vacuum sealing.

Figures 3-30 and 3-31 are for sandwich material combinations with 5056 aluminum core. Figure 3-30 shows weights when the inner skin is used for vacuum sealing; Figure 3-31 applies to the outer skin. Again, there was no significant difference in vacuum jacket weight for using either the inner or outer skin for vacuum sealing. However, there is a significant weight reduction when 5056 aluminum Flex-Core is used in place of HRP core for the LH₂ tank vacuum jacket. This applies to all material combinations studied.

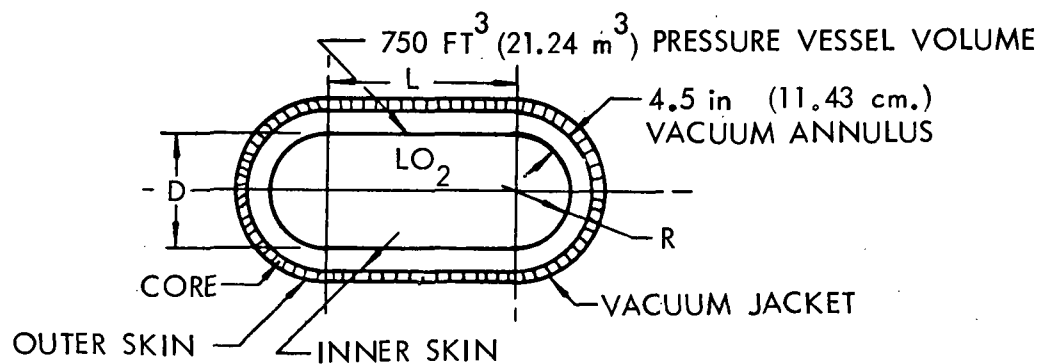
The discontinuities in Figures 3-28 through 3-31 are due to a change in the critical failure mode of some of the designs. When the individual design weights for the hemispherical heads and the cylinders were plotted against L/D as shown in Figure 3-32, the reason for the discontinuities was evident. The weight of the heads was a smooth function of L/D because all the designs were buckling critical. However, the cylinder designs were both buckling critical and crimping critical. This created discontinuities at an L/D of approximately 3. Since shear crimping was primarily dependent on the core shear moduli, the significant weight reductions for using 5056 aluminum Flex-Core were due to the better modulus/weight ratio of the Flex-Core. The conclusions of this trade study were:

- 1.0 The optimum face material has the greatest stiffness to density ratio.
- 2.0 There were no conclusive weight trends between using aluminum or titanium skins, nor in the placement of these metal skins either on the inside face of the sandwich or the outside face. The selection of L/D ratio, core material, and the remaining sandwich face skin will determine whether an aluminum or a titanium face skin is more efficient.



R		L		L/D
in	cm	in	cm	
36	91.44	801	2034.54	11.1
48	121.92	414	1051.56	4.31
63	160.02	193	490.22	1.53
90	228.60	16	40.64	0.09

LH₂



R		L		L/D
in	cm	in	cm	
24	60.96	684	1737.36	14.3
30	76.20	418	1061.72	6.97
42	106.68	178	452.12	2.12
60	152.40	35	88.90	0.29

LO₂

Figure 3-27: VACUUM JACKET GEOMETRIES FOR SANDWICH SHELL TRADE STUDIES

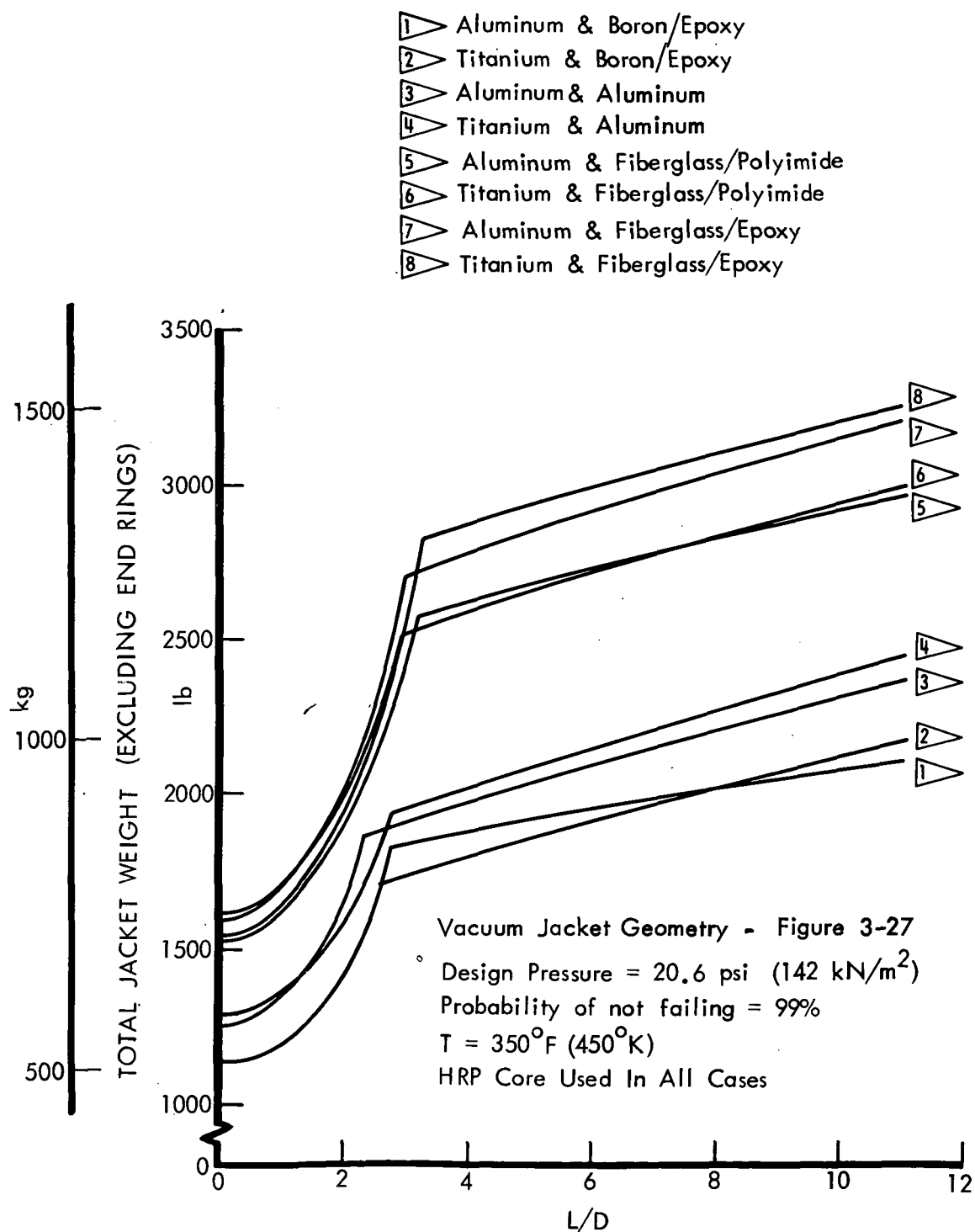


Figure 3-28: LH₂ VACUUM JACKET WEIGHT VS L/D FOR EIGHT FACE MATERIALS AND HRP CORE (METAL INNER SKIN)

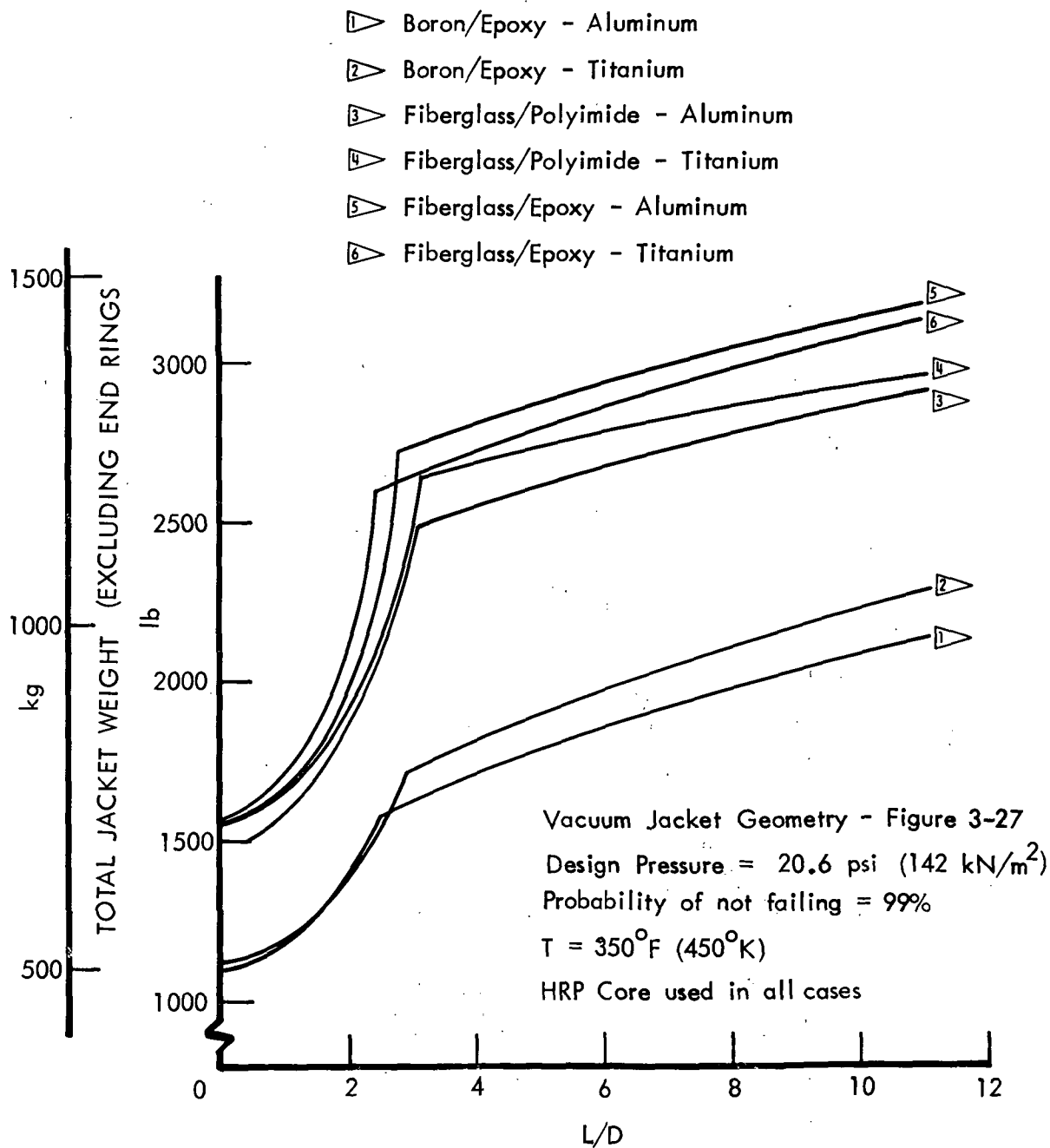


Figure 3-29: LH₂ VACUUM JACKET WEIGHT VS L/D FOR SIX FACE MATERIALS AND HRP CORE (METAL OUTER SKIN)

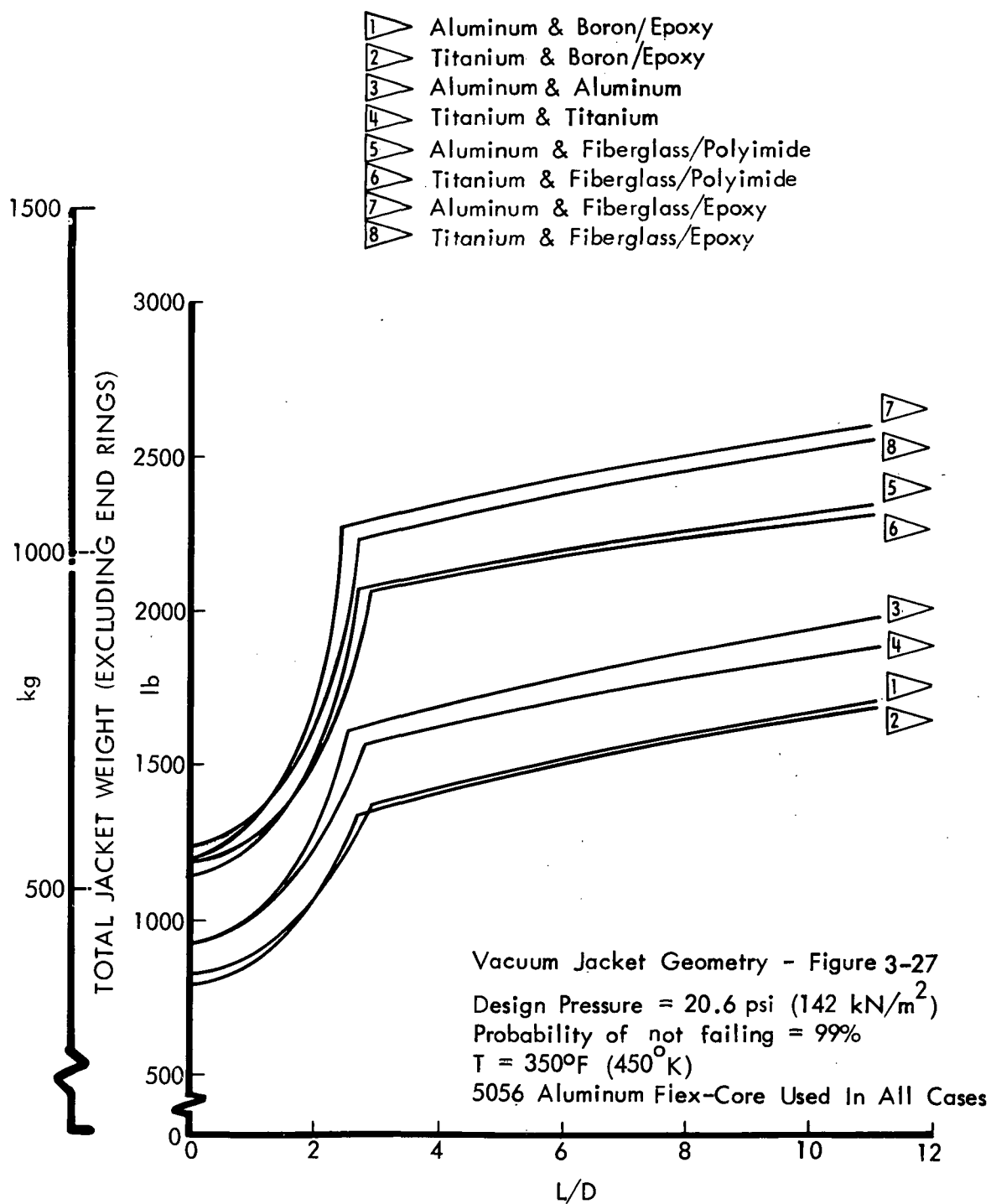


Figure 3-30: LH₂ VACUUM JACKET WEIGHT VS L/D FOR EIGHT FACE MATERIALS AND 5056 ALUMINUM FLEX-CORE (METAL INNER SKIN)

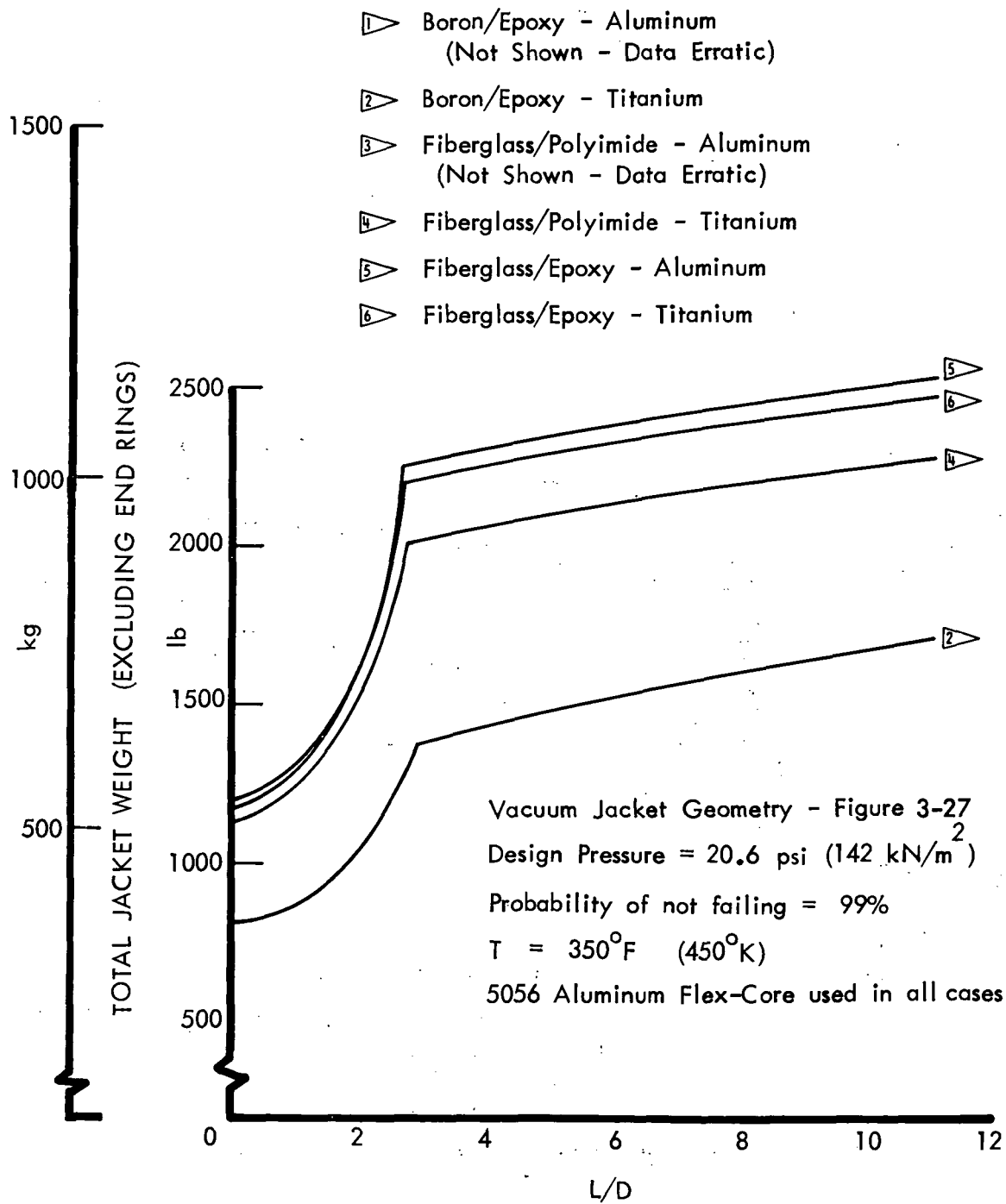


Figure 3-31: LH₂ VACUUM JACKET WEIGHT VS L/D FOR SIX FACE MATERIALS AND 5056 ALUMINUM FLEX-CORE (METAL OUTER SKIN)

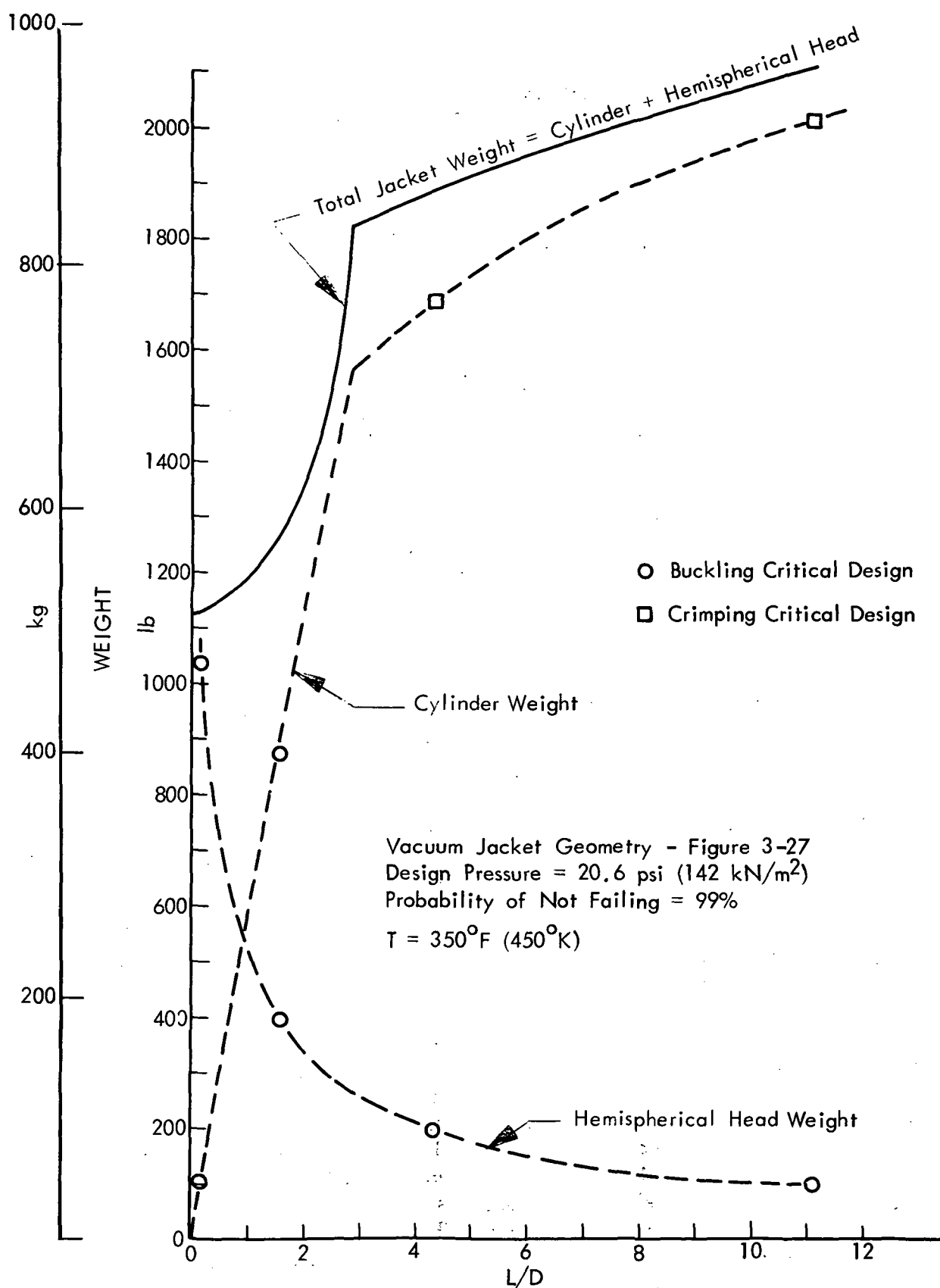


Figure 3-32: TYPICAL LH₂ VACUUM JACKET CYLINDER AND HEAD WEIGHT VS L/D

- 3.0 The optimum core material has the greatest shear stiffness-to-density ratio (GL or G_W/ρ). The use of a low shear modulus core, i.e., HRP, results in a severe weight penalty (17 to 30 percent) for the vacuum jacket designs.

Face Skin and Core Trade Study for LO_2 Vacuum Jacket

The vacuum jacket geometries for the L/D ratios studied are shown in Figure 3-27.

Total jacket weights, exclusive of end rings, for HRP and 5056 Aluminum Flex-Core are plotted versus L/D in Figures 3-33 and 3-34. These data show a significant weight reduction when 5056 Flex-Core was used in place of HRP core. About the same relationship exists for the weight efficiency of the material combinations studied. The materials with the largest stiffness to weight ratio produce the lightest vacuum jacket designs. Although the designs shown have the metal outer skin, the results of the LH_2 vacuum jacket trades should apply to the LO_2 vacuum jacket. The inner skin can be metal without a significant difference in total jacket weight. Detail data from these shell trade studies are tabulated in Appendix B.

The conclusions from this study were:

- 1.0 The high stiffness to weight ratio of boron/epoxy results in substantial weight savings over other face material combinations.
- 2.0 All material combinations with 5056 aluminum core were more efficient than with HRP core.

3.4.1.2 Girth Ring Design Criteria (LH_2 Vacuum Jacket)

A conservative ring design criteria was added to the initial OPTRAN design programs for the vacuum jacket. The criteria had the advantage of being simple, independent of the shell design, and required very little computer time. However, this conservative approach resulted in total end ring weights that were too heavy for lightweight vacuum jackets.

To establish a better ring design criteria the effect of the end ring area on the radial deflection of the shell was investigated for the design external pressure, 20.6 psi (142 kN/m²). This analysis used the OPTRAN sandwich designs for the aluminum and boron/epoxy faced sandwich constructions with 5056 aluminum Flex-Core. The results are shown in Figure 3-35. The radial deflection, "W" is plotted versus the end ring area for each vacuum jacket radius "a". The original "conservative" shell buckling analysis assumed a simple support at the edges of the hemisphere and the cylinder. The "↑" symbol shown in Figure 3-35 designates the ring areas that were used with the conservative criteria. Theoretically, simple support means no radial deflection, "W", at the edge of the shell. However, all practical designs have some radial deflection. Hence,

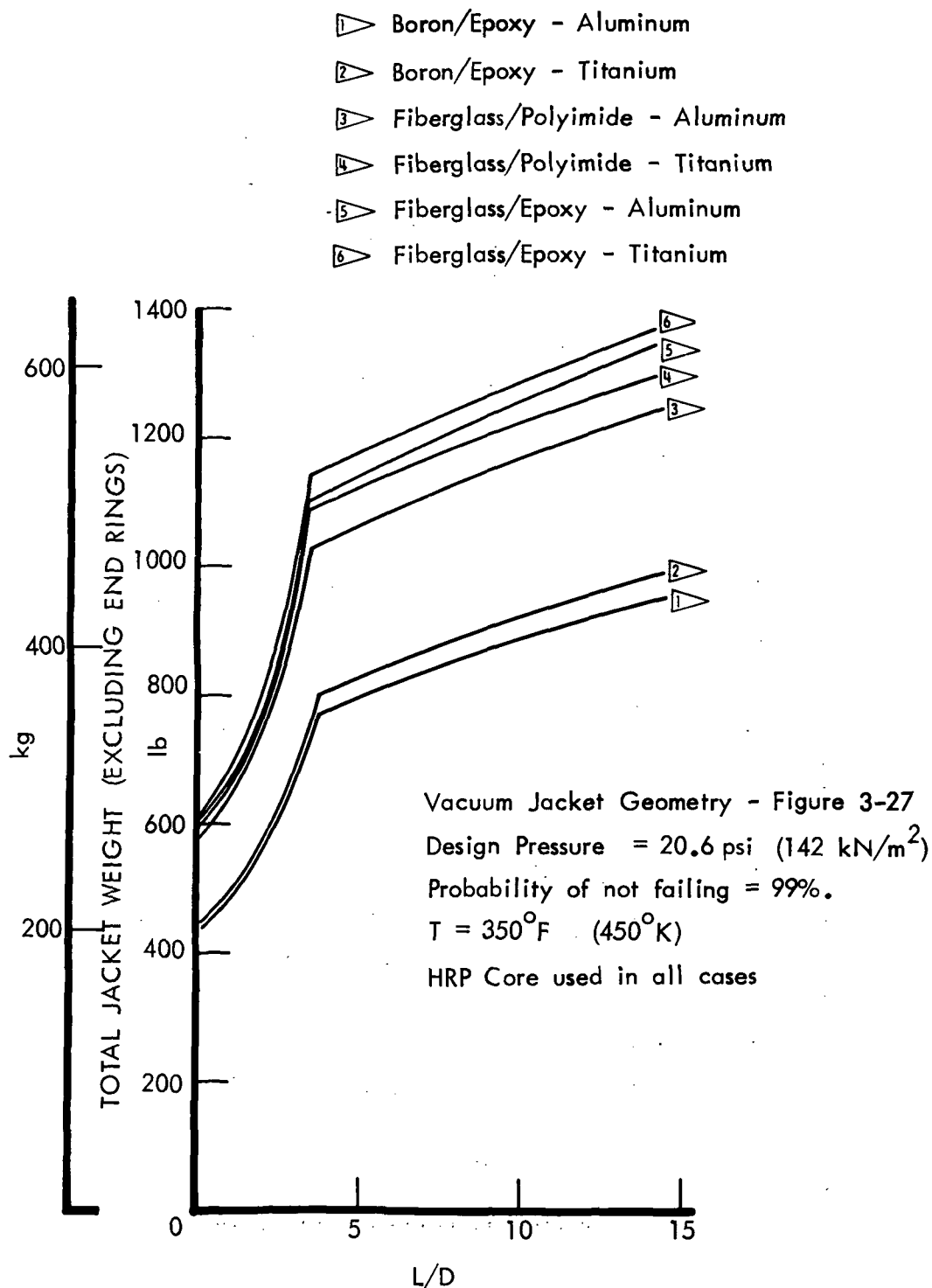


Figure 3-33: LO₂ VACUUM JACKET WEIGHT VS L/D FOR SIX FACE MATERIALS AND HRP CORE (METAL OUTER SKIN)

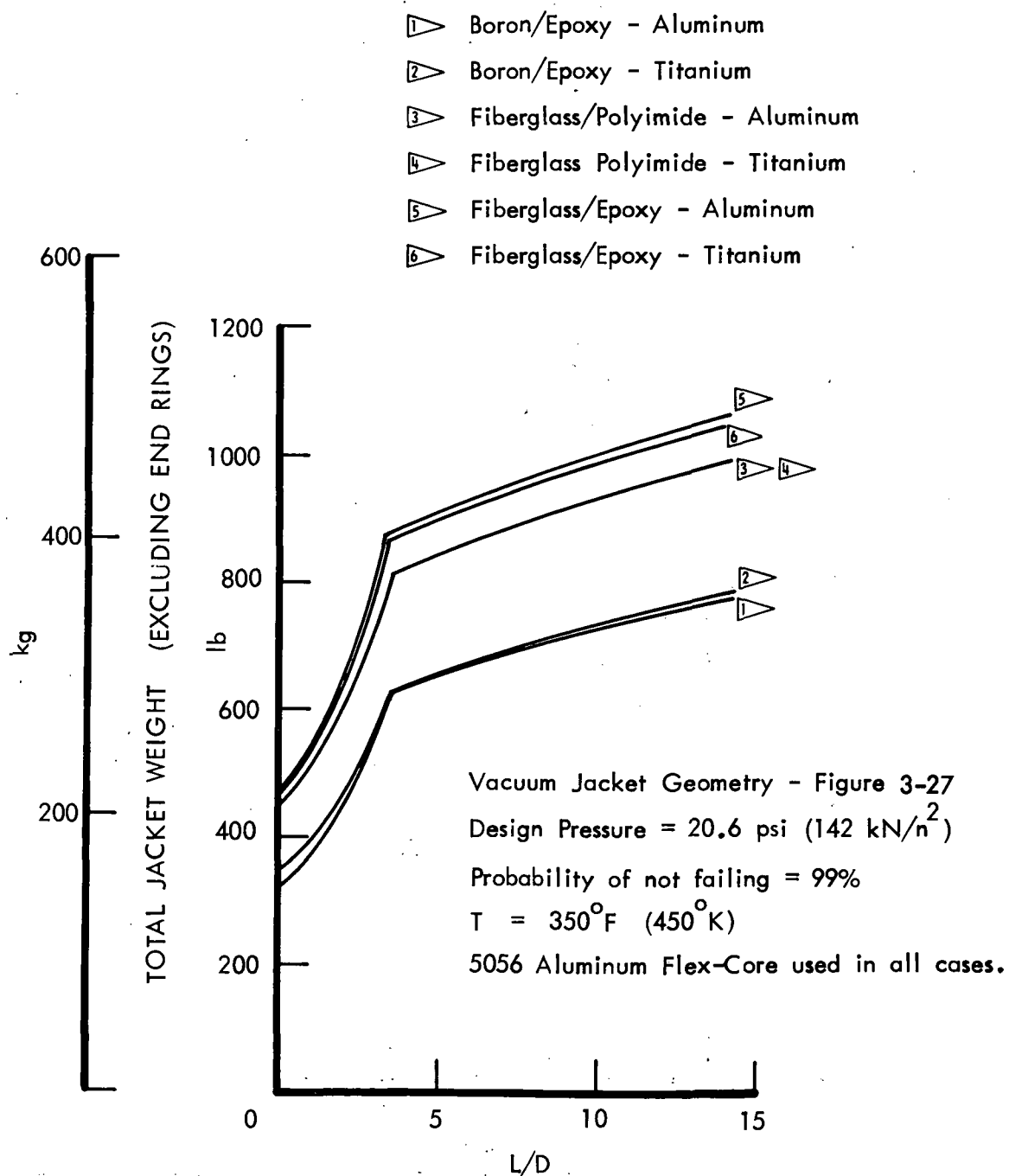


Figure 3-34: LO₂ VACUUM JACKET WEIGHT VS L/D FOR SIX FACE MATERIALS AND 5056 ALUMINUM FLEX-CORE (METAL OUTER SKIN)

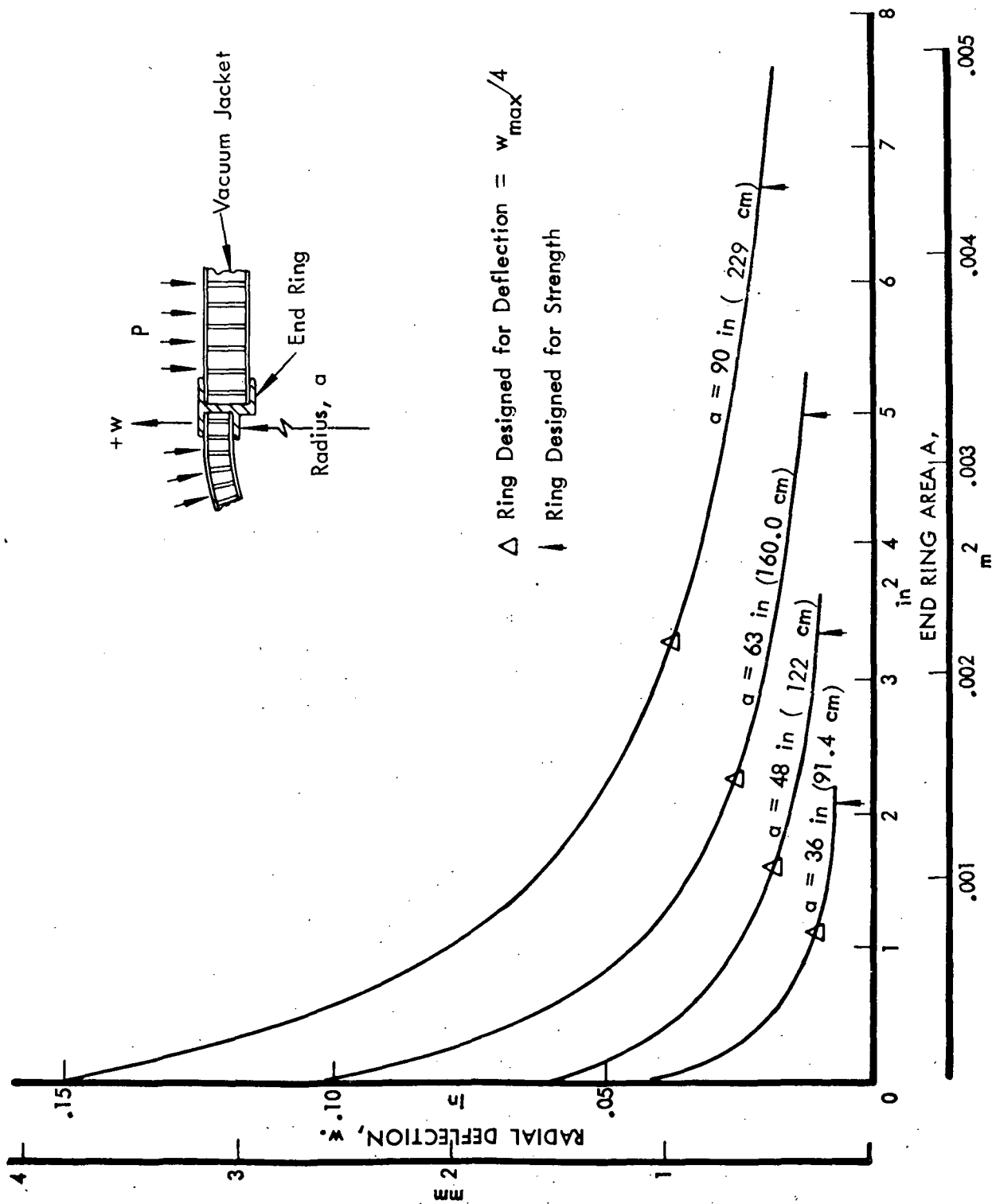


Figure 3-35: RADIAL DEFLECTION VS END RING AREA

it is likely that some radial deflection of the ring can be permitted with little effect on the shell buckling strength. Using this reasoning, a radial deflection of $W_{\max}/4$ was arbitrarily selected as a better criterion for sizing the rings. " $W_{\max}$ " is the radial deflection that would occur if there were no rings present and the edges of the head and cylinder were free to deflect under external pressure. This $W_{\max}/4$ criteria, shown as Δ in Figure 3-35 reduced the ring weight to approximately half that required by the conservative criteria.

The $W_{\max}/4$ criteria did not account for the effect of the end ring stiffening the sandwich shell. To study this effect and also to check the $W_{\max}/4$ criteria, a shell buckling analysis of the $L/D = 0.09$ vacuum jacket was made using the BOSOR 3 computer program, Reference 12.

Half of the segmented composite shell for the BOSOR 3 analysis is shown in Figure 3-36. Both faces of the sandwich construction were aluminum to simplify the hand calculations for the program input. The aluminum gages were selected from an OPTRAN design for aluminum faced sandwich. Figure 3-37 shows the ring geometry used in the analysis, and the sandwich dimensions. The ring area and bending stiffnesses were varied to determine the effect of ring area on the composite shell buckling strength.

Two buckling analyses were made. The first was a classical eigenvalue problem to investigate the linear analysis buckling modes and determine the classical buckling load. The second analysis included nonlinear prebuckling effects. Both symmetrical and antisymmetrical buckling modes were calculated. The theoretical, critical external pressures were plotted in Figure 3-38 as a function of external ring area for the $L/D = 0.09$ vacuum jacket. The results indicated that the end ring area could be reduced to 0.81 in^2 (5.14 cm^2) without significant effect on the composite shell buckling strength. The critical pressure decreased rapidly at ring areas less than this. Since two end rings were required, the total ring area per jacket was 1.62 in^2 (10.5 cm^2) which weigh approximately 100 lb. (45.4 kg).

Referring to Figure 3-35, it can be seen that the 1.62 in^2 (10.5 cm^2) total end ring area selected from the results of the BOSOR3 analysis was equivalent to a deflection criteria of $W_{\max}/1.7$ for $\alpha = 90$. This indicated that the arbitrary criteria of $W_{\max}/4$ was conservative for the $L/D = 0.09$ vacuum jacket.

Because the $L/D = 0.09$ vacuum jacket had a very short cylinder, the ring criteria study was extended to $L/D = 1.53$ vacuum jacket configuration. This configuration had a 193 in. (490.2 cm) long cylindrical section.

Figure 3-39 is a sketch of the vacuum jacket analyzed. The sandwich segments used in the analysis are also shown in Figure 3-39. The inner faces were aluminum and the outer faces were boron/epoxy prepreg. This construction was selected because it was the lightest weight based on the OPTRAN shell studies.

NOTE: See figure 3-37 for details AA, BB & CC.

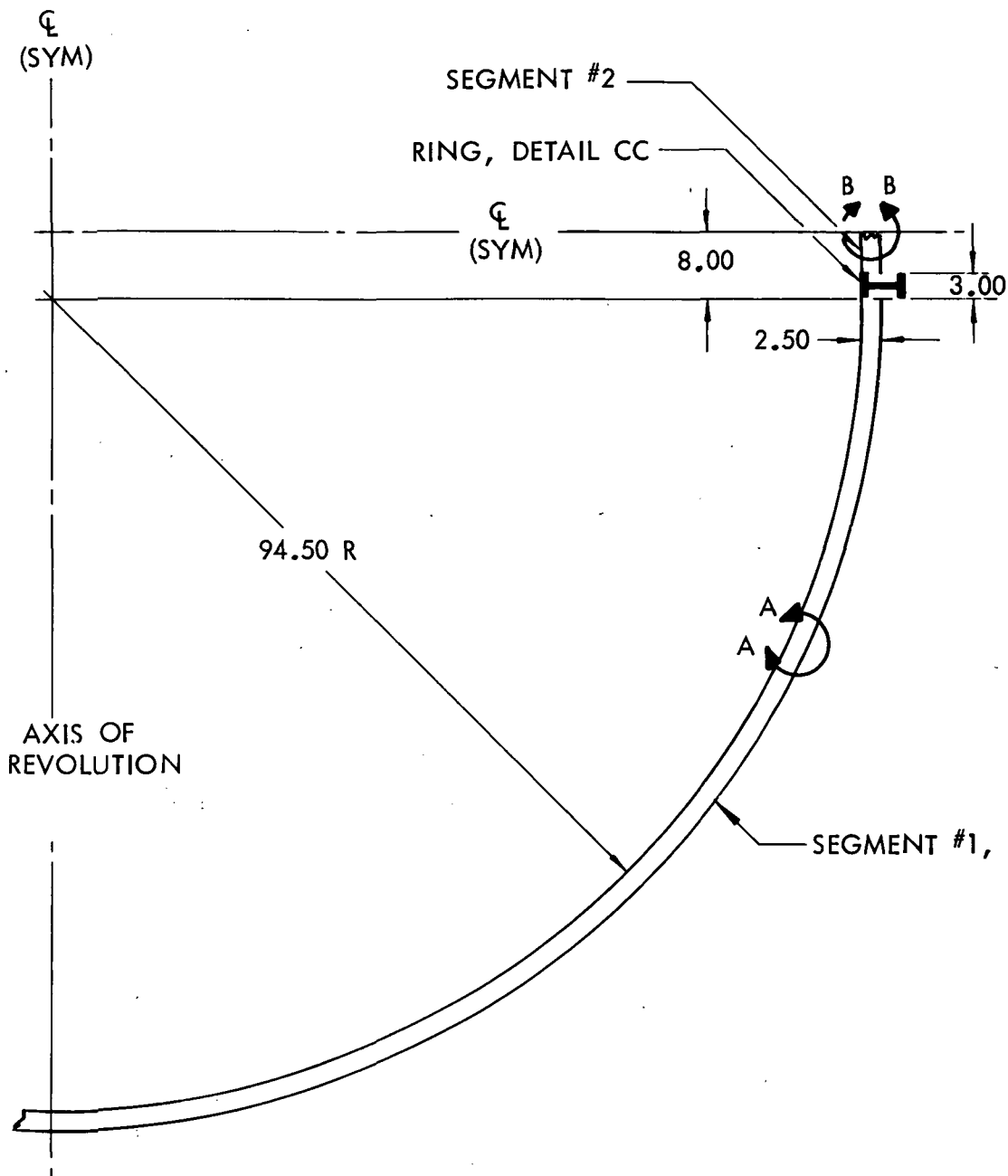


Figure 3-36: SEGMENTED COMPOSITE SHELL FOR ANALYSIS BY BOSOR 3

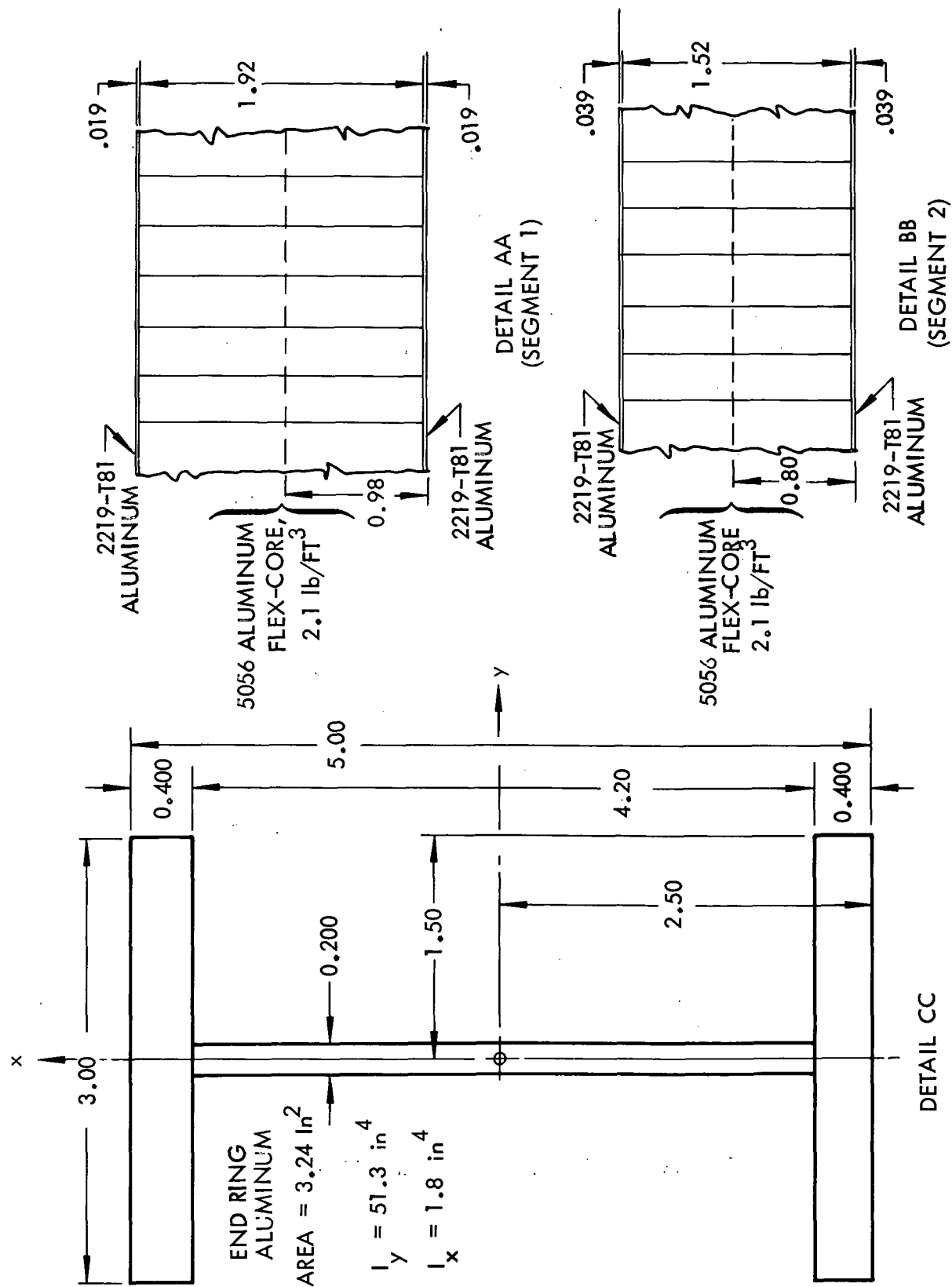


Figure 3-37: SEGMENTED COMPOSITE SHELL DETAILS

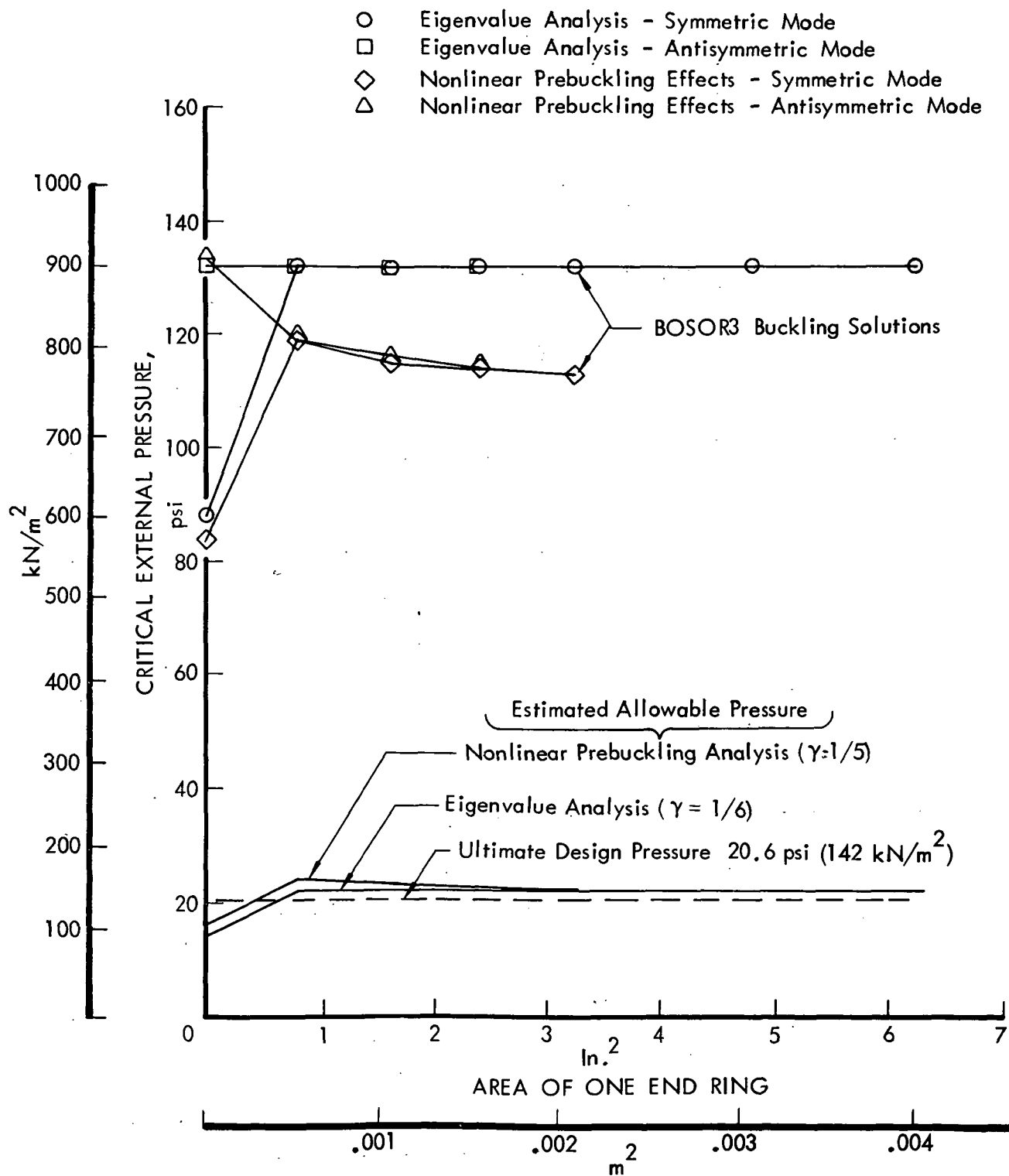


Figure 3-38: EFFECT OF END RING AREA ON THE CRITICAL EXTERNAL PRESSURE OF THE $L/D = 0.09$, LH_2 VACUUM JACKET

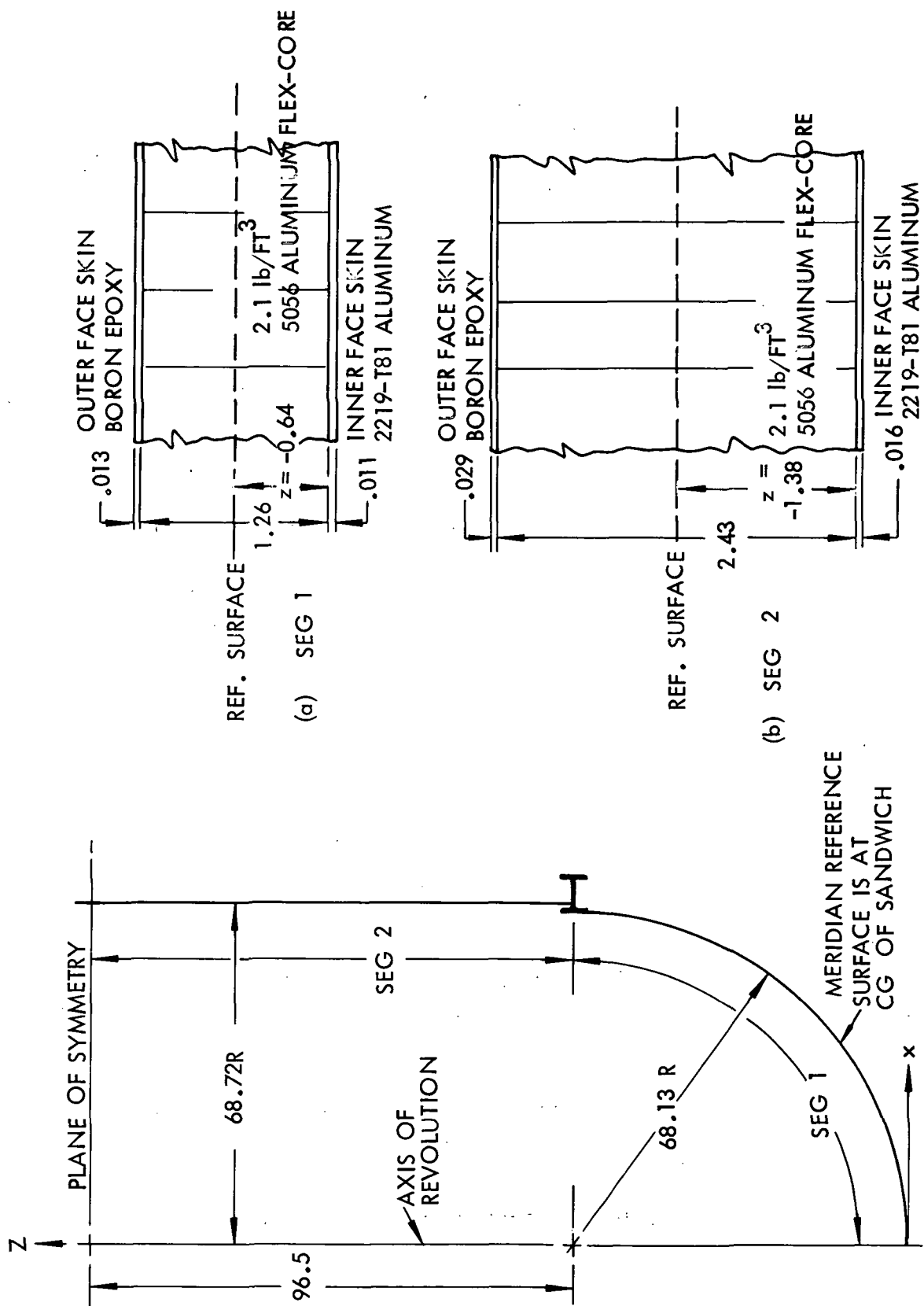


Figure 3-39: MERIDIAN REFERENCE SURFACE, GEOMETRY AND SEGMENT GEOMETRY

The stiffness coefficients for these segments were calculated using the SALC model (Reference 13) for composite materials.

The first case analyzed used 2.0 in^2 (12.9 cm^2) end rings. Analysis showed that the allowable external pressure was slightly less than the 20.6 psi (142 kN/m^2) required. Also, the end rings were not located in an effective position to resist buckling. The reason for this was that the cylinder buckling was the dominant failure mode and determined the critical pressure of the jacket. This is illustrated in Figure 3-40 a. The dashed line is the deflected shape of the buckling mode for 4 circumferential waves. The allowable external pressure was determined by multiplying the BOSOR3 solution (37.6 psi) (252 kN/m^2) by the knockdown factor for sandwich cylinders (0.523) with the same R/ρ . The knockdown factors for sandwich cylinders are plotted in Figure 3-41 b. The knockdown factors for a hemispherical mode are plotted in Figure 3-41 a. Increasing the end ring area (Figure 3-40 a) to 4.0 in^2 (25.8 cm^2) increased the allowable external pressure to only 20.2 psi (139 kN/m^2). This indicated the relative inefficiency of adding ring area to the ends of this configuration.

The effect of locating the ring at the mid-length of the cylinder is shown in Figure 3-40 b. The allowable pressure was increased to 26 psi (179 kN/m^2) and the cylinder was forced to buckle into two half waves for the same total ring area as Figure 3-40 a.

Two disadvantages to the single center ring configuration of Figure 3-40 are:

- (1) a heavier ring than necessary and
- (2) the displacement at the head to cylinder joint may overstress the sandwich locally.

The ring weight can be minimized by iterative design, but the displacement at the head to cylinder joint might cause the head to buckle prematurely.

A better configuration was to use the end and intermediate rings as shown in Figure 3-40 c. The end rings served to limit the displacement at the head to cylinder joint. The intermediate rings helped to support the pressure load. The total ring area shown in Figure 3-40 c is 2.0 in^2 (12.9 cm^2). Referring to Figure 3-35 it was seen that a 2.0 in^2 (12.9 cm^2) total area for end rings on the $a = 63$ configuration was equivalent to a deflection criteria of $W_{\max}/2.5$.

It appeared from this analysis that for the higher L/D configuration the $W_{\max}/2.5$ deflection criteria was reasonable for estimating ring areas. This was done for the $a = 48$ and 36 ($L/D = 4.31$ and 11.1) configurations shown in Figure 3-42. These weights were used in Section 3.4.1.3, Shell Construction Trade Studies to estimate the total vacuum jacket weights including end rings. It should be noted that a detailed analysis such as BOSOR3 was necessary to properly optimize the size and location of the rings.

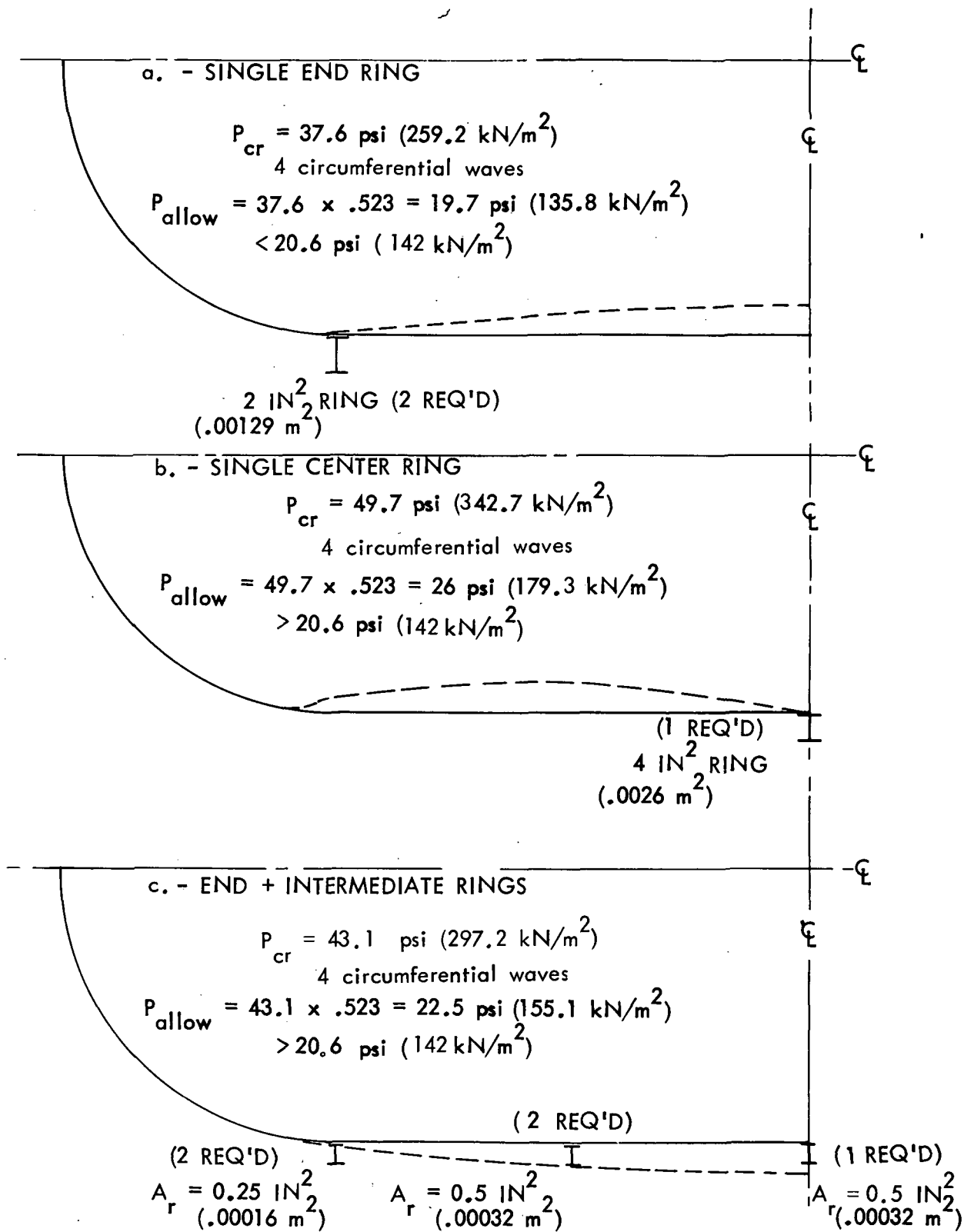


Figure 3-40: EFFECT OF RING LOCATION ON THE BUCKLING MODE OF THE
 $L/D = 1.53$, LH_2 VACUUM JACKET

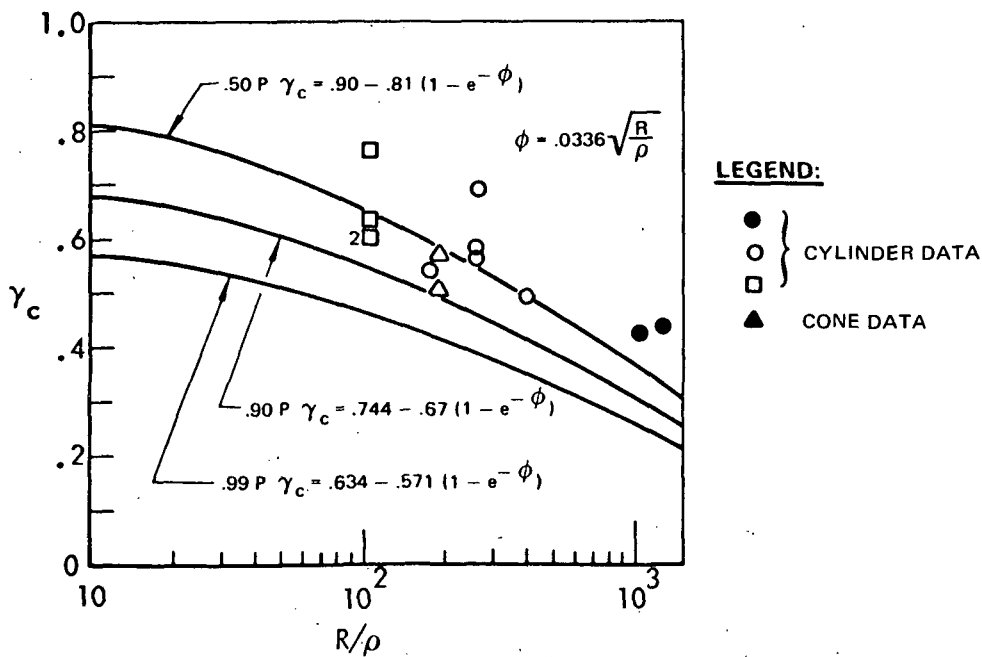
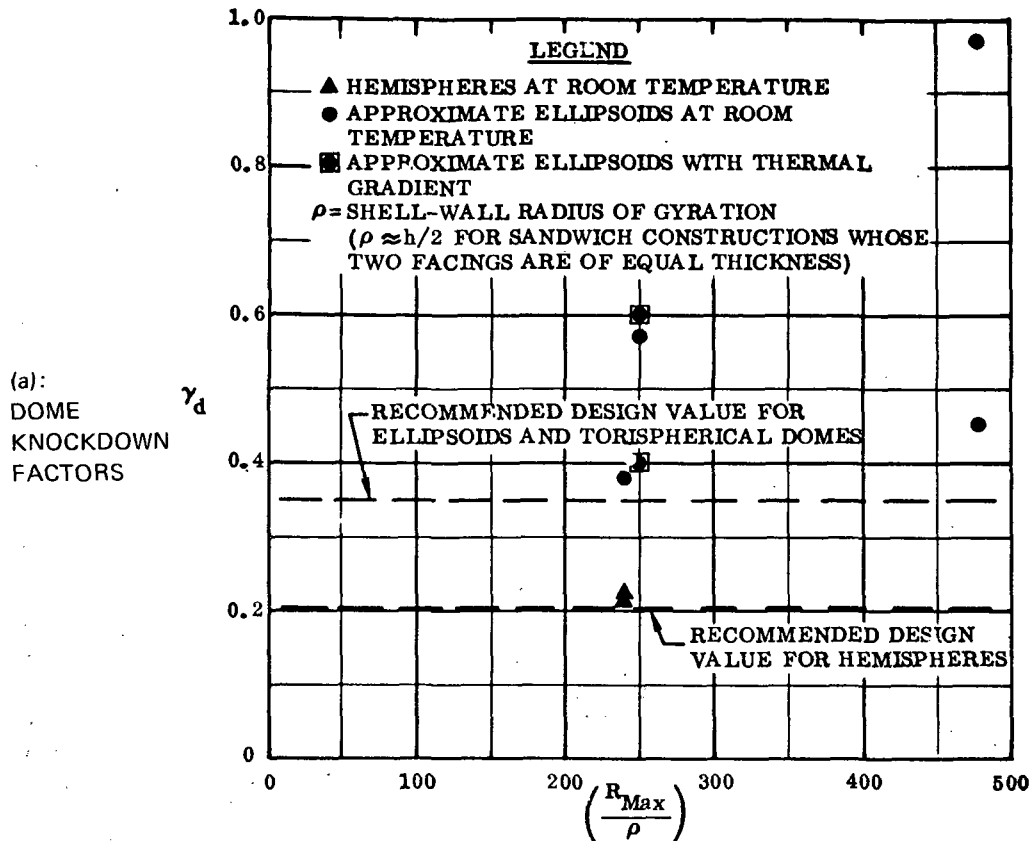


Figure 3-41: KNOCKDOWN FACTORS FOR SANDWICH DOMES & CYLINDERS
 SUBJECTED TO EXTERNAL PRESSURE

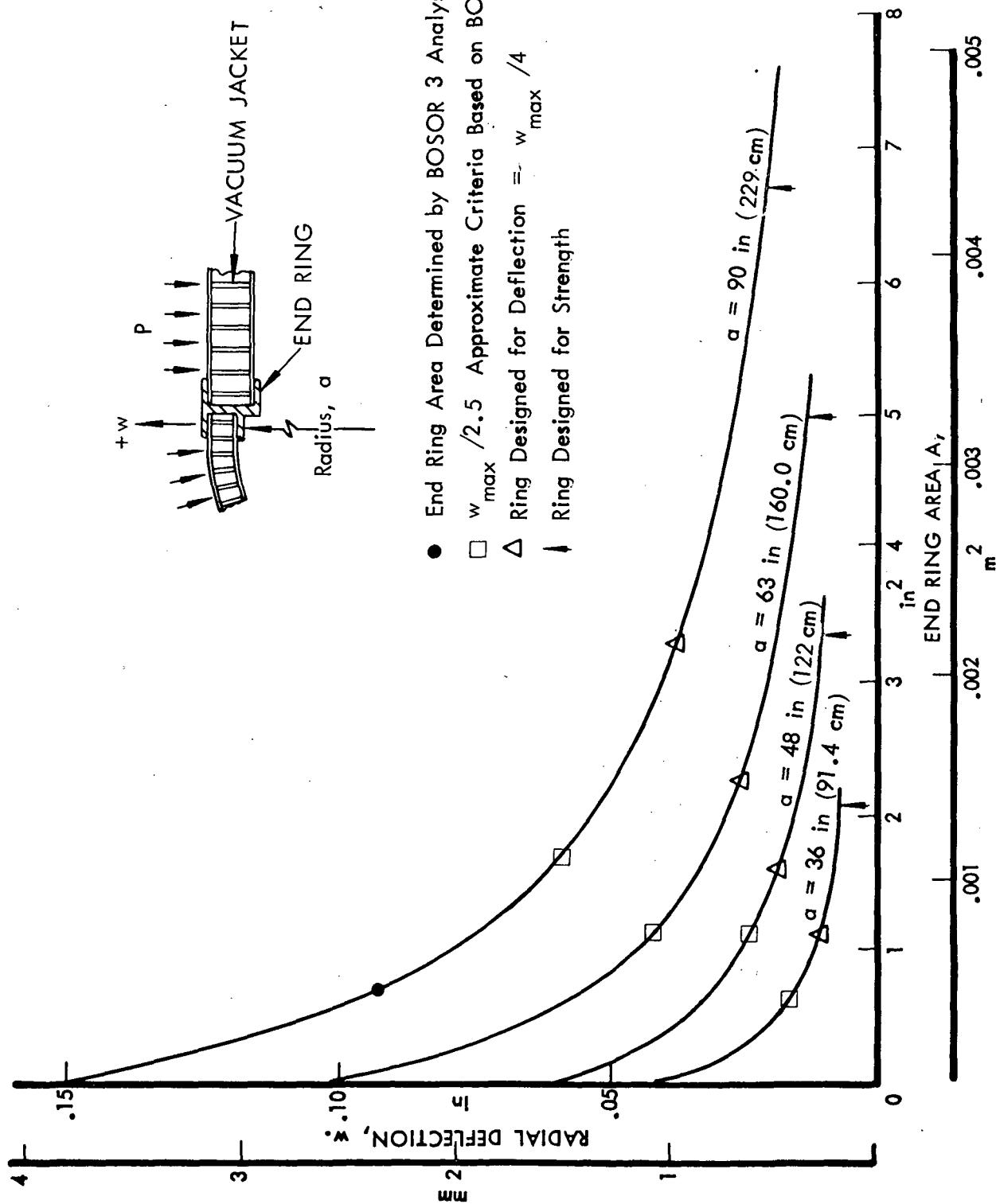


Figure 3-42: APPROXIMATE END RING AREAS FOR LH₂ VACUUM JACKETS

3.4.1.3 Shell Construction Trade Studies for the LH₂ Vacuum Jacket

These trades investigated other shell construction methods and compared their weights with the optimum sandwich shell weights. Properties for the 2219-T81 and 6Al-4V annealed materials were listed in Section 3.4.1.1.

The construction methods studied were:

- a) Honeycomb sandwich cylinder and heads
- b) Hoop corrugated cylinder with sandwich heads
- c) Waffle stiffened cylinder and heads
- d) Ring stiffened cylinder sandwich with sandwich heads
- e) I section ring and stringer stiffened cylinder (skin not buckled) with sandwich heads
- f) I section ring and stringer stiffened cylinder (skin buckled) with sandwich head.

The LH₂ vacuum jacket geometry for the four L/D ratios studied are shown in Figure 3-43. The total weight of the vacuum jacket includes the girth ring weights. The results from the girth ring investigation were used to estimate the ring weights.

Table 3-8 shows weights for the 5056 aluminum Flex-Core sandwich cylinders and heads with aluminum and boron/epoxy face skins, and with both face skins aluminum. The 100 lb (45.4 kg) aluminum end ring is used for the L/D = 0.09 configuration. The $W_{\max}/2.5$ deflection criteria was used to determine the weights for the other configurations.

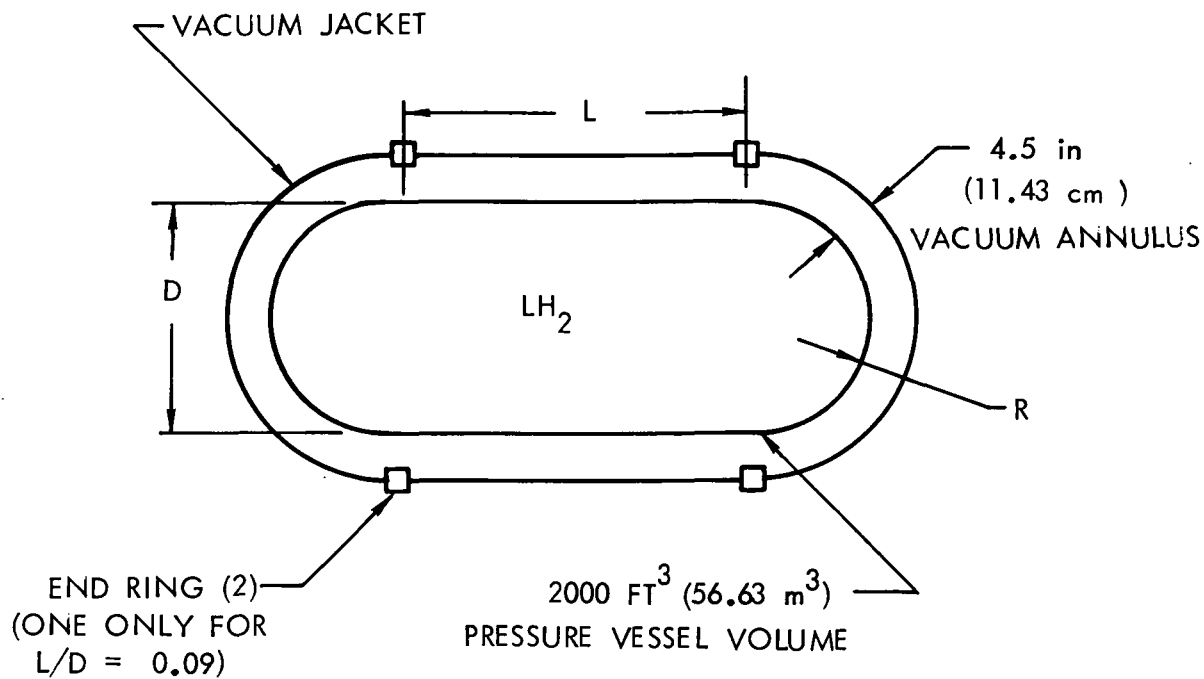
Also shown are the weights for the 5056 aluminum Flex-Core sandwich cylinders and heads with titanium and boron/epoxy face skin, and with both face skins titanium. The titanium end ring weights were factored from the aluminum end ring weights. This factor was determined by:

$$\text{Weight of ring} = \frac{\text{constant} \times \text{material density}}{\sqrt{E_{\text{material}}}}$$

Comparing aluminum and titanium then

$$W_{\text{t aluminum}} = \frac{k(.1)}{\sqrt{1.0 \times 10^7}}$$

$$W_{\text{t Titanium}} = \frac{k(.16)}{\sqrt{1.6 \times 10^7}}$$



R		L		L/D
in	cm	in	cm	
36	91.44	801	2034.54	11.1
48	121.92	414	1051.56	4.31
63	160.02	193	490.22	1.53
90	228.60	16	40.64	0.09

Figure 3-43: VACUUM JACKET GEOMETRIES FOR LH₂
SHELL TRADE STUDIES

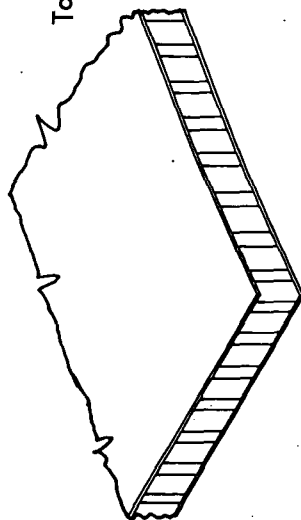


Table 3-8: OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET, HONEYCOMB SANDWICH CYLINDER AND HEADS

MATERIAL	PRESSURE VESSEL CYLINDER L/D	WEIGHT						TOTAL JACKET WEIGHT			
		TOTAL FOR TWO HEADS		CYLINDER		TOTAL FOR END RINGS					
		lb	kg	lb	kg	lb	kg	lb	kg		
ALUMINUM END RINGS 5056 ALUMINUM FLEX-CORE	INNER FACE SKIN ALUMINUM OUTER FACE SKIN BORON/EPOXY	11.1	77	34.9	1629	738.9	32	14.5	1738	788.3	
		4.31	146	66.2	1291	585.6	74	33.6	1511	685.4	
		1.53	283	128.4	661	299.8	97	44	1041	472.2	
		0.09	720	326.6	78	35.4	100	45.4	898	407.3	
	BOTH FACE SKINS ALUMINUM	11.1	85	38.6	1894	859.1	32	14.5	2011	912.2	
		4.31	161	73.0	1549	702.6	74	33.6	1784	809.2	
		1.53	318	144	815	369.7	97	44	1230	557.9	
		0.09	818	371	104	47.2	100	45.4	1022	463.6	
	TITANIUM END RINGS 5056 ALUMINUM FLEX-CORE	INNER FACE SKIN TITANIUM OUTER FACE SKIN BORON/EPOXY	11.1	88	39.9	1604	727.6	41	18.6	1733	786.1
			4.31	159	72.1	1272	577	94	42.6	1525	691.7
1.53			301	136.5	655	297.1	124	56.3	1080	490	
0.09			737	334.3	97	44	127	57.6	961	436	
BOTH FACE SKINS TITANIUM		11.1	105	47.6	1764	800.1	41	18.6	1910	866.4	
		4.31	189	85.7	1446	655.9	94	42.6	1729	784.3	
		1.53	341	154.7	765	347	124	56.3	1230	558	
		0.09	834	378.3	105	47.6	127	57.6	1066	483.5	

Therefore,

$$\text{Ring weight}_{ti} = 1.27 \times \text{Ring weight}_{al}$$

Table 3-9 shows optimum weights for the aluminum hoop corrugated sandwich cylinder with sandwich head and aluminum end rings. The width of the corrugation flat was allowed to vary from 0 to 10 inches (25.4 cm). Titanium hoop corrugated cylinder weights with titanium sandwich heads and titanium end ring weights are also listed in the table. Reference 14 describes the general instability analysis method used for this study. Classical plate buckling was used for the load instability modes.

A longitudinal corrugated cylinder was considered as an option to the hoop corrugated cylinder. However, it was reasoned that the hoop corrugation, stiffened cylinder, with its greater hoop stiffness was more efficient than a longitudinal corrugation stiffened cylinder. Also, a sandwich construction with two face skins was more efficient than a single skin construction such as the waffle.

The truss core sandwich concept was studied as a special case of the hoop corrugated sandwich. That is, the truss core had a 0 width corrugation flat. OPTRAN results showed that the hoop corrugated cylinder was lighter than the truss core.

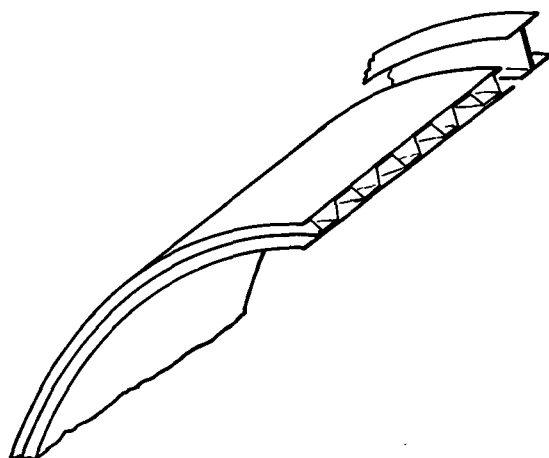
Table 3-10 shows weights for the aluminum ring stiffened sandwich cylinder with sandwich heads. This construction divides the cylinder into short bays using aluminum rings. The analysis methods of Reference 10 were used. For the L/D ratios = 11.1, 4.31 and 1.53 the aluminum end ring weight was reduced from that shown on Table 3-8 because the intermediate rings carried some of the end ring loads. This was not the case with the $L/D = 0.09$ since no additional rings were added to the cylinder. Therefore, the 100 lb (45.4 kg) weight from Table 3-8 was used.

Table 3-11 shows the weights for the aluminum waffle stiffened cylinder and waffle stiffened heads. The aluminum end ring weights are from Table 3-8. The analysis of Crawford and Schwartz, Reference 15, was used for the heads. The cylinder analysis method is described in Reference 14.

Titanium waffle heads were also studied; however, they were much heavier than the aluminum grid stiffened waffle heads.

Another form of the waffle grid stiffened construction was studied to find a lower weight design. This consisted of letting the skin buckle between the grids, which permitted the grid stiffeners to work to much higher stresses. The skin would operate as a pressure membrane and transfer the hoop and longitudinal stresses to the grid. The OPTRAN results showed a weight increase over the unbuckled skin. The reason for this was that the buckled skin further reduced the bending stiffness of the grid which in turn required heavier gages.

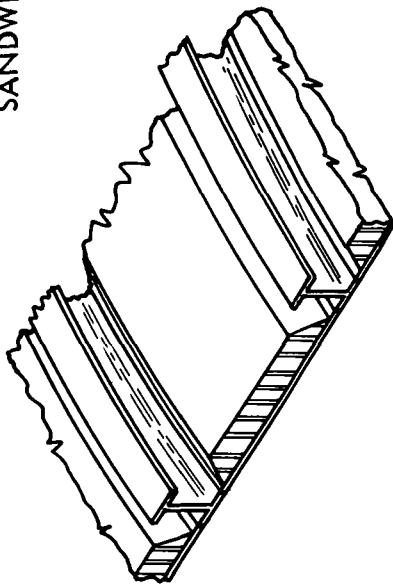
Table 3-9 : OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET
HOOP CORRUGATED SANDWICH CYLINDER WITH SANDWICH HEADS



- 1 ▷ AL-BORON/EPOXY FACE SKINS
- 2 ▷ TI-BORON/EPOXY FACE SKINS
- 3 ▷ ALUMINUM RINGS
- 4 ▷ TITANIUM RINGS

CYLINDER MATERIAL	PRESSURE VESSEL CYLINDER L/D	CYLINDER WEIGHT		TOTAL WEIGHT FOR TWO HEMISPHERICAL SANDWICH HEADS 5056 AL. FLEX-CORE		TOTAL WEIGHT FOR END RINGS		TOTAL JACKET WEIGHT	
		lb	kg	lb	kg	lb	kg	lb	kg
ALUMINUM	11.1	3125	1417.5	77 1 ▷	34.9	32 3 ▷	14.5	3234	1466.9
	4.31	2385	1081.8	146 1 ▷	66.2	74 3 ▷	33.6	2605	1181.6
	1.53	1401	635.5	283 1 ▷	128.4	97 3 ▷	44	1781	807.9
	0.09	117	53.1	720 1 ▷	326.6	100 3 ▷	45.4	937	425.2
TITANIUM	11.1	4452	2019.4	88 2 ▷	39.9	41 4 ▷	18.6	4581	2077.9
	4.31	3364	1525.9	159 2 ▷	72.1	94 4 ▷	42.6	3617	1640.6
	1.53	1963	890.4	301 2 ▷	136.5	124 4 ▷	56.3	2388	1083.2
	0.09	147	66.7	737 2 ▷	334.3	127 4 ▷	57.6	1011	458.6

Table 3-10: OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET, RING STIFFENED SANDWICH CYLINDER WITH SANDWICH HEADS



CYLINDER MATERIAL	PRESSURE VESSEL CYLINDER L/D	CYLINDER WEIGHT						TOTAL WEIGHT FOR TWO HEMISPHERICAL SANDWICH HEADS 5056 AL, FLEX-CORE AL-BORON/EPOXY FACE SKINS				TOTAL WEIGHT FOR ALUMINUM END RINGS				TOTAL JACKET WEIGHT	
		SANDWICH CYLINDER		INTER-MEDIATE RINGS		TOTAL											
		lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg
▷	11.1	1074	487.2	512	232.2	1586	719.4	77	34.9	24	10.9	1687	765.2				
	4.31	852	386.5	393	178.2	1245	564.7	146	66.2	58	26.3	1449	657.3				
	1.53	576	261.3	235	106.5	811	367.8	283	128.4	85	38.6	1179	534.8				
	0.09	78	35.4	0	--	78	35.4	720	326.6	100	45.4	898	407.5				

▷ ALUMINUM RINGS WITH 5056 AL, FLEX-CORE AND AL-BORON/EPOXY FACE SKINS

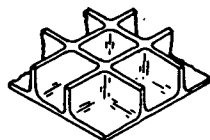


Table 3-11:
OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET,
GRID STIFFENED WAFFLE CYLINDER AND HEADS

MATERIAL	PRESSURE VESSEL CYLINDER L/D	WEIGHT						TOTAL JACKET WEIGHT	
		TOTAL FOR TWO HEADS		CYLINDER		TOTAL FOR END RINGS			
		LB	kg	LB	kg	LB	kg	LB	kg
ALUMINUM	11.1	130	59	--	--	32	14.5	--	--
	4.31	260	118	4101	1860	74	33.6	4435	2012
	1.53	516	234	2391	1085	97	44	3004	1363
	0.09	1330	603	102	46	100	45.4	1532	695

Table 3-12 shows the weights for the aluminum "I" section ring and stringer stiffened cylinder. This design used the constraint that the skin could not buckle within the squares formed by the intersecting rings and stiffeners. This constraint resulted in fairly heavy designs since a large portion of the total weight was in the skin. The aluminum end ring weights are reduced as shown in Table 3-10 since the ring stiffeners help carry the end ring pressure loads. Table 3-12 also shows the weights for the titanium "I" section ring and stringer stiffened cylinder in which the skin did not buckle.

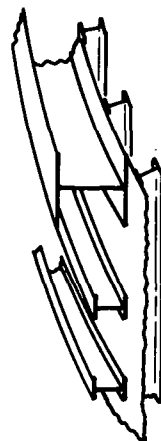
Table 3-13 shows the weights for both the aluminum and the titanium "I" section ring and stringer stiffened cylinder for the condition in which the cylinder skin was allowed to buckle. The weights shown are for the minimum skin gage of 0.020 in. (0.051 cm). The weights are considerably less than for the buckle resistant designs.

Figure 3-44 is a plot of the vacuum jacket weights versus L/D for twelve different construction methods. Honeycomb sandwich construction provides the least weight vacuum jacket for L/D's from .09 to about 7. Beyond an L/D of 7 the ring and stringer construction with buckled skin is more efficient.

The conclusions from these trade studies for the LH₂ vacuum jacket were:

- (1) The sandwich construction, including the necessary end rings, resulted in the least weight design for near spherical vacuum jackets with L/D ratios from 0 to 3.
- (2) The ring stiffened honeycomb sandwich construction had the least weight design for vacuum jackets with L/D ratios from 3 to 7.

Table 3-13: OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET, "1" SECTION
RING AND STRINGER STIFFENED CYLINDER WITH SANDWICH HEADS



NOTE:

SKIN IS ALLOWED TO BUCKLE

- 1 ▷ AL-BORON/EPOXY FACE SKINS
- 2 ▷ ALUMINUM RINGS
- 3 ▷ TI-BORON/EPOXY FACE SKINS
- 4 ▷ TITANIUM RINGS

CYLINDER MATERIAL	PRESSURE VESSEL CYLINDER L/D	CYLINDER WEIGHT								TOTAL WEIGHT FOR TWO HEMISPHERICAL SANDWICH HEADS 5056 AL, FLEX CORE				TOTAL WEIGHT FOR END RINGS		TOTAL JACKET WEIGHT	
		SKIN		INTER – MEDIATE RINGS		STRINGERS		TOTAL									
		lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg	lb	kg
ALUMINUM	11.1	408	185	628	285	308	139.7	1344	609.7	77	34.9	24	10.9	1445	655.4		
	4.31	273	124	861	391	264	120	1398	634	146	66.2	58	26.3	1602	726.7		
	1.53	164	74.4	711	322	339	154	1214	550.4	283	128.4	85	38.6	1582	717.6		
	0.09	19	8.6	33	14.9	33	15	85	38.5	720	326.6	100	45.4	905	410.5		
TITANIUM	11.1	652	296	534	242	136	61.7	1322	599.7	88	39.9	31	14	1441	653.6		
	4.31	437	198	816	370	128	58.	1381	626	159	72.1	74	33.6	1614	732		
	1.53	262	119	431	196	162	73.5	855	388.5	301	136.5	108	49	1264	572		
	0.09	30	13.6	--	--	15	6.8	45	20.4	737	334.3	127	57.6	909	412		

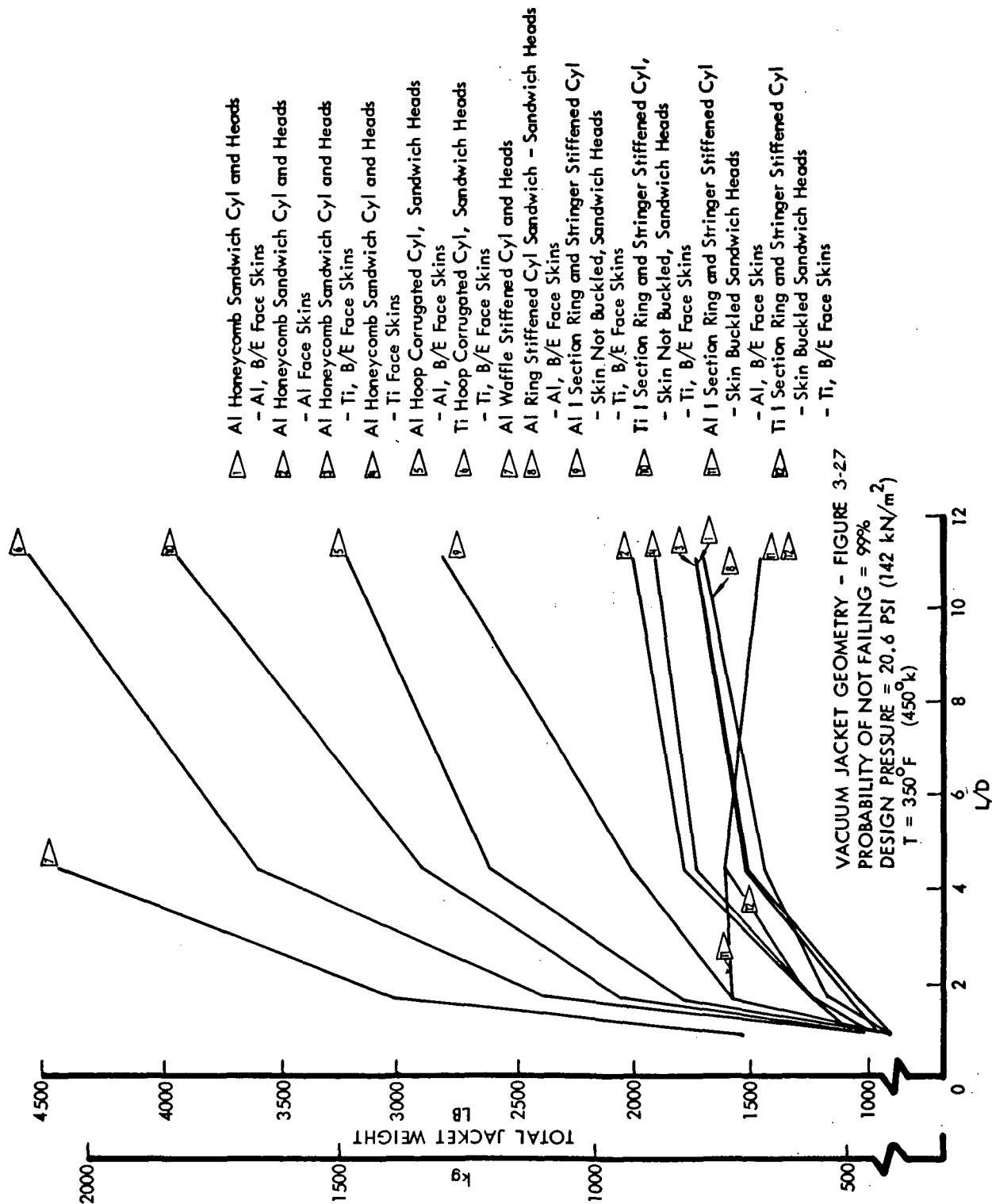


Figure 3-44: LH₂ VACUUM JACKET WEIGHT VS. L/D
 FOR TWELVE CONSTRUCTION METHODS

- (3) Above a L/D ratio of 7, the "I" section ring and stringer stiffened cylinder construction with honeycomb sandwich heads had the least weight.

3.4.1.4 Design Factor Considerations

The design criteria used for the Shell Trade Studies contain two accepted design factors: a 1.4 Factor of Safety and a 99% probability (of not failing) knockdown factor. The weight of the vacuum jacket designs are affected by these factors.

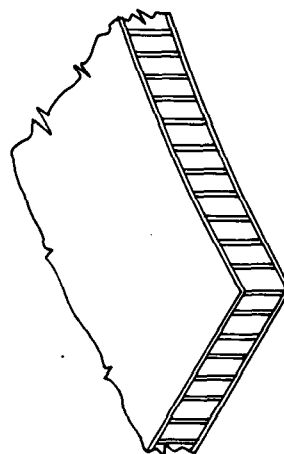
The 1.4 factor of safety is specified in Reference 1, Structural Design Criteria Applicable to a Space Shuttle. This value is at present, arbitrary. It is reserved for the accountability of only those uncertainties in the load-carrying capability of the structure which cannot be analyzed or otherwise accounted for in a rational manner. The uncertainties often arise from the inability to predict residual stresses or when fabrication processes are not ideal and cannot be controlled to produce ideal or identical structures. It is intended that these factors be verified or modified on the basis of the best available design techniques (e.g., fracture mechanics and statistical analyses as sufficient data become available) and that the values be consistent with the desired level of structural reliability.

Knockdown factors are defined separately from the factor of safety and applied separately in the design. The knockdown factors primarily account for "imperfections" in the shells which lower the buckling strength and the inability of the theoretical methods to predict the buckling strength of thin shells. The Boeing knockdown factors used in the Shell Trade Studies were obtained from a previous Boeing study using statistical analysis of all available shell buckling test data. Median, 90 percent and 99 percent probability (of not failing) knockdown factors were computed from the data. The 99 percent probability knockdown factors were selected as a conservative basis for the Shell Trade Studies. However, the Boeing data were all unstiffened shells; the Shell Trade Study shells were sandwich or stiffened shells. Therefore, the shell trade studies assume that the knockdown factors for unstiffened shells could be applied to sandwich and stiffened shells. This was evaluated by the external pressure tests described in Section 4.

The effect of using a different probability knockdown factor is shown in Table 3-14. Using a lower probability reduced the total weight of the LH₂ vacuum jacket, particularly if the design was buckling critical. If the 99 percent probability factors were used for design, the typical strength of the vacuum jacket would be considerably higher than the ultimate design pressure. The average jacket would be overweight with extra strength. However, if a 50 percent probability factor were used for design, the typical strength would about equal the ultimate design pressure. Some of the jackets would be understrength and some overstrength. By adding the requirement for a proof external pressure test, the minimum strength of each manufactured jacket can be determined. Weak jackets can be repaired and

Table 3-14:
THE EFFECT OF KNOCKDOWN
FACTORS ON THE LH₂
VACUUM JACKET WEIGHT

MATERIAL	PROBABILITY OF DESIGN NOT FAILING (Pd,n.f.)	PRESSURE VESSEL CYLINDER L/D	WEIGHT								TOTAL JACKET WEIGHT	
			TOTAL FOR TWO HEADS		CYLINDER		TOTAL FOR END RINGS					
			lb	kg	lb	kg	lb	kg	lb	kg	lb	kg
INNER FACE SKIN ALUM OUTER FACE SKIN BORON/EPOXY 5056 ALUMINUM FLEX-CORE	0.99	11.1	77	34.9	1629	738.9	32	14.5	1738	788.3		
		4.31	146	66.2	1291	585.6	74	33.6	1511	685.4		
		1.53	283	128.4	661	299.8	97	44.0	1041	472.2		
		0.09	720	326.6	78	35.4	100	45.4	898	407.3		
	0.90	11.1	69	31.3	1629	738.9	32	14.5	1730	784.7		
		4.31	125	56.7	1279	580.2	74	33.6	1478	670.5		
		1.53	237	107.5	661	299.8	97	44.0	995	451.3		
		0.09	596	270.3	72	32.7	100	45.4	768	348.4		
	0.50	11.1	69	31.3	1629	738.9	32	14.5	1729	784.3		
		4.31	124	56.3	1279	580.2	74	33.6	1477	670.1		
		1.53	225	102.1	661	299.8	97	44.0	983	445.9		
		0.09	537	243.6	70	31.8	100	45.4	707	320.8		
BOTH FACE SKINS ALUMINUM 5056 ALUMINUM FLEX-CORE	0.99	11.1	85	38.6	1894	859.1	32	14.5	2011	912.2		
		4.31	161	73.0	1549	702.6	74	33.6	1784	809.2		
		1.53	318	144.2	815	369.7	97	44.0	1230	557.9		
		0.09	818	371.0	104	47.2	100	45.4	1022	463.6		
	0.90	11.1	75	34.0	1888	856.8	32	14.5	1995	905.3		
		4.31	138	62.6	1507	683.6	74	33.6	1719	779.8		
		1.53	264	119.8	815	369.7	97	44.0	1176	533.5		
		0.09	659	298.9	101	45.8	100	45.4	860	390.1		
	0.50	11.1	74	33.6	1885	855.0	32	14.5	1991	903.1		
		4.31	131	59.4	1502	681.3	74	33.6	1707	774.3		
		1.53	251	113.9	815	369.7	97	44.0	1163	527.6		
		0.09	616	279.4	101	45.8	100	45.4	817	370.6		



strengthened. Strong jackets can be certified for use. This approach to the use of a lower probability design factor is discussed further in Section 4.

3.4.2 Semi-Rigid Vacuum Shells

Both the aluminum alloy and titanium alloy semi-rigid vacuum shells were studied. The weight results of the OPTRAN designs for these two materials are shown in Table 3-15. The heads in these design cases were 5056 aluminum Flex-Core sandwich construction. The aluminum alloy cylinder used heads with aluminum boron/epoxy face skins and aluminum alloy girth rings. The titanium alloy cylinder used heads with titanium boron/epoxy face skins and titanium alloy girth rings. The analysis method for buckling externally loaded rings, Reference 16, was used for the study. Bending of the flat portions of the corrugations was not included in the analysis.

It can be seen by comparing Table 3-15 with Tables 3-8 through 3-13 that there is a weight advantage with the semi-rigid vacuum shell over the self supporting shells for an $L/D = 11.1$ and 4.31 . Design details for the aluminum semi-rigid shell are shown in Figure 3-7 and are discussed in Section 3.2.2.

3.5 Outer Shell Studies

Outer shell studies were conducted to evaluate fabrication methods for the inner face skin of the sandwich shell. It was concluded that this skin should be a metal and be the vacuum sealing surface to stop outgassing by-products of the core from degrading the vacuum. The study objective was to determine which manufacturing techniques could achieve the skin thickness requirements and be vacuum tight. The three fabrication methods studied are described in Figure 3-45.

The head assembly shown consists of a 40.5 in. (102.9 cm) radius hemispherical head with a 10.0 in. (25.4 cm) cylindrical portion. There is a 2° slope to the cylinder wall to permit tool removal. This head is used in both configurations 4 and 5. The baseline sandwich arrangement consists of 5056 aluminum Flex-Core and a bonded 2024-T3 aluminum gore outer skin. The inner aluminum arrangements are described in Sections A₁ - A₁, A₂ - A₂ and A₃ - A₃.

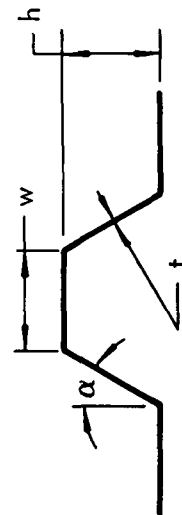
The first approach considered was spinning, bulge forming, or explosive sizing large blanks to contour, followed by selective chem-milling to meet thickness tolerances. 6061 or 2219 aluminum alloys are suitable for this skin. During the chem-milling, weld lands are left in the head at the apex and the girth. Close out rings are welded in place in these areas. The welds are helium leak checked and repaired as necessary prior to bonding the core and the outer face skin in place.

This manufacturing approach is feasible for heads up to approximately 10 ft. (3.05 m) in diameter. Experience has shown this approach to be costly and time consuming. Also as noted in Section 4.4, some aluminum alloys are susceptible to grain growth during spinning which cause subsequent chem-milling problems. In thinner gages

Table 3-15: OPTIMUM DESIGN WEIGHTS FOR LH₂ VACUUM JACKET, SEMI-RIGID HOOP CORRUGATED CYLINDER WITH SANDWICH HEADS

CORRUGATION MATERIAL	PRESSURE VESSEL CYLINDER L/D	CORRUGATION GEOMETRY								CORRUGATION WEIGHT		TOTAL WEIGHT FOR TWO HEMISPHERICAL SANDWICH HEADS 5056 AL. FLEX-CORE		TOTAL WEIGHT FOR END RINGS		TOTAL JACKET WEIGHT	
		α		w		h		t		lb	kg	lb	kg	lb	kg	lb	kg
ALUMINUM	11.1	51.7	0.90	2.73	6.93	2.21	5.61	0.053	0.135	1367.3	620.20	77	34.9	32	14.5	1476	669.6
	4.31	52.5	0.92	3.51	8.92	2.86	7.26	0.068	0.173	1183.1	536.7	146	66.2	74	33.6	1403	636.4
	1.53	60.8	1.06	4.13	10.49	3.69	9.37	0.08	0.203	906.8	411.3	283	128.4	97	44.0	1287	583.7
	0.09	62.7	1.09	5.58	14.17	5.16	13.11	0.11	0.279	147.0	66.7	720	326.6	100	45.4	967	438.7
TITANIUM	11.1	49.8	0.87	1.85	4.70	2.06	5.23	0.044	0.112	1846.9	837.7	88	39.9	41	18.6	1976	896.3
	4.31	63.2	1.10	2.79	7.09	2.56	6.50	0.049	0.125	1506.1	683.2	159	72.1	94	42.6	1759	797.9
	1.53	55.1	0.96	3.93	9.98	3.29	8.36	0.068	0.173	1165.0	528.4	301	136.5	124	56.3	1590	721.2
	0.09	55.2	0.96	5.47	13.89	4.61	11.71	0.095	0.241	188.5	85.5	737	334.3	127	57.6	1053	477.6

-  AL-BORON/EPOXY FACE SKINS
 TI-BORON/EPOXY FACE SKINS
 ALUMINUM RINGS
 TITANIUM RINGS



0.010 ± 0.0015

0.003 ± 0.0015

0.010
(REQUIRED THICKNESS)

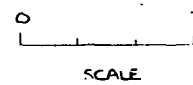
DETAIL I

VACUUM JACKET ARRANGEMENT

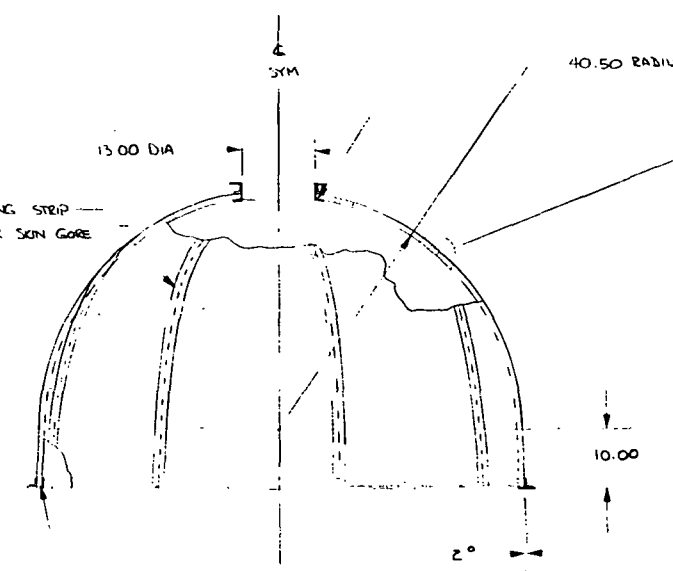
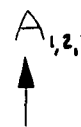
NOTES

- 1 SANDWICH CONSTRUCTION
- a) CORE 5056 ALUMINUM FLEX-CORE (2.1 LB. FT³)
- b) OUTER FACE SKIN - ALUMINUM ALLOY 2024 T3
- 2 6061 OR 2219 ALUMINUM ALLOY
- 3 CHEM-MILL TOLERANCES
- 4 WELD
- 5 WELDS IN 'AS WELDED' CONDITION
- 6 VACUUM SEAL
- 7 BOND
- 8 SHEET TOLERANCES 'AS RECEIVED'
- 9 SUPPLIER'S TOLERANCE
- 10 ALUMINUM ALLOY 2024 T3
- 11 2024, 2219 OR 6061 ALUMINUM ALLOY

SECTION C-C

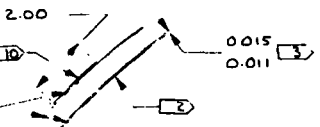


10 OUTER SKIN GORE JOINING STRIP
LOCATED AT 1/2 OF INNER SKIN GORE

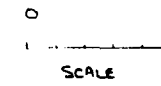


GIRTH CLOSEOUT RING

HEAD ASSEMBLY



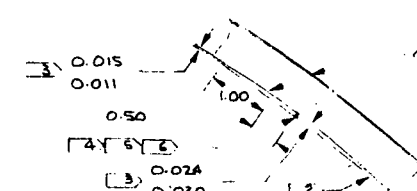
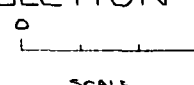
SECTION B-B



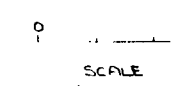
2 ACCESS CLOSEOUT RING



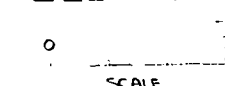
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SECTION E-E



SECTION F-F

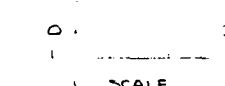


GIRTH CLOSEOUT RING

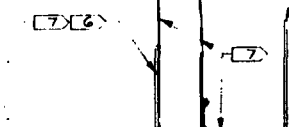
0.010 ± 0.0015

0.010 ± 0.0015

SECTION H-H



ACCESS CLOSEOUT RING



SECTION G-G



SECTION A-A

INNER SKIN - SPIN AND CHEM-MILLED



SECTION A₂-A₂

INNER SKIN - WELDED GORE



SECTION A₁-A₁

INNER SKIN - BONDED GORE



SANDWICH HEAD
CONSTRUCTION METHODS

Figure 3-45:

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there is the risk of developing pin holes during the chem-milling operation. Pinholes are difficult to locate. However, assuming no pinholes, this fabrication method does produce a highly reliable vacuum tight surface, which should not degrade from the thermal and pressure cycling imposed on these shells.

The second approach, described in Section A2-A2, was welding preformed gore sections together. The 6061 or 2219 aluminum alloy gores are stretch-formed, trimmed to shape, and chem-milled to provide weld lands at the gore joints, the apex and the girth. All welds are helium leak checked prior to bonding the core and the outer face skin in place. This manufacturing approach is feasible for the range of shells studied. The major disadvantage is the elaborate tooling necessary to align and hold the gores in place for welding. Weld shrinkage will cause some distortion in this skin. An explosive sizing operation after welding may be required to produce a satisfactory contour. A magnetic hammer can also be used to bring the skin back to contour. Effect of such rework on the vacuum integrity of the welds is uncertain. Another uncertainty with this approach is the ability to repair weld thin gages to achieve vacuum tightness.

The third approach, described in Section A3-A3, was a bonded gore arrangement for the inner skin. The 2024-T3 aluminum alloy gores are stretch formed and trimmed to shape. The joining strips, which bond adjacent gores together, transmit the skin loads. The gores, polar caps, joining strips, girth ring and apex ring are bonded together as an assembly and helium leak checked. Repairs are made as necessary to obtain vacuum integrity before the core and the outer face skins are bonded in place.

This approach is feasible for the range of shells studied and appears to offer better shell contour accuracy and lower fabrication costs than either the spun or the welded gore shell.

It was concluded from this study that the bonded gore approach offers the best opportunity of meeting the program objectives and is therefore selected for the recommended designs.

3.6 Recommended Designs

3.6.1 Design Features

The results of the design studies were used to develop the recommended LH₂ and the LO₂ tank designs. Those designs are shown in Figures 3-46 and 3-47. Both the LH₂ and the LO₂ recommended designs are low L/D ratios and use aluminum sandwich construction for the vacuum jacket. The use of boron/epoxy prepreg on the LH₂ tank vacuum jacket for the outer face skin is not recommended at this time. Although it is more efficient than aluminum, considerable work is required to define the best methods for laying up this material on a double curvature. In both designs the inner (vacuum sealing) face skin is bonded 2024-T3 aluminum gores.

ASSEMBLY AND QUALIFICATION PROCEDURES

1. DETAILS AND SUBASSEMBLIES
 - a) PRESSURE VESSEL WELD ASSEMBLY. LN₂ COLD SHOCK, PROOF PRESSURE TEST, AND HELIUM LEAK CHECK.
 - b) SUPPORT STRAP DETAILS
 - c) GIRTH RING MACHINING
 - d) VACUUM JACKET SUBASSEMBLIES
 - e) MLI BLANKET SUBASSEMBLIES
 - f) MANHOLE COVER ASSEMBLY (COVER, SHUT-OFF VALVE, AND FEED LINE)
2. ASSEMBLE PRESSURE VESSEL, SUPPORT STRAPS, GIRTH RING, VENT LINE, AND VENT RELIEF VALVE
3. INSTALL MLI BLANKET ON PRESSURE VESSEL. INSULATE SUPPORT STRAPS AND VENT LINE
4. INSTALL VACUUM JACKET SUBASSEMBLIES. SEAL JOINTS AT GIRTH RING AND VENT LINE OUTLET FOR VACUUM.
5. INSTALL MANHOLE COVER, FEED LINE, AND SEAL VENT LINE. INSULATE THE AREA. INSTALL VACUUM JACKET COVER. SEAL JACKET COVER AND FEED LINE OUTLET FOR VACUUM.
6. HELIUM LEAK CHECK ASSEMBLY. REPAIR AS NECESSARY. PUMPDOWN VACUUM MANIFEST TO 1.5×10^{-9} TORR. CONDUCT VACUUM DECAY TEST. PRECONDITION WITH HEAT AND VACUUM PUMPING AS NECESSARY TO REACH REQUIRED DECAY RATE. WIND OFF AND SEAL THE THREE FABRICATION PUMPDOWN PORTS. CLOSE VACUUM VALVES ON THE MAINTAINABILITY PUMPDOWN PORTS.
7. CONDUCT BOIL-OFF TEST

NOTES

- ① ALUMINUM ALLOY 2219-T81
- ② STAINLESS STEEL
- ③ WELDS IN 'AS WELDED' CONDITION
- ④ WELD
- ⑤ DIFFUSION BOND
- ⑥ MULTILAYER INSULATION (MLI) ⑦
 - a) ALUMINIZED MYLAR (0.15 MIL)/DACRON NET (B+A) - 75 LAYERS EACH
 - b) OUTER 0.10 INCH LAYER - ALUMINIZED KAPTON (0.30 MIL), DACRON NET (B+A) - 75 LAYERS EACH
- ⑧ SANDWICH CONSTRUCTION
 - a) FACE SHEETS - ALUMINUM ALLOY 2024 T3
 - b) CORE - SOSG ALUMINUM FLEX-CORE (2.1 LB/FT³)
- ⑨ VACUUM SEAL
- ⑩ ASSEMBLE WITH NYLON STUDS AND GROMMETS
- ⑪ ALUMINUM ALLOY ROLL FORGED RING 2219 T852

NAS 3-14369

D. L. BRADLEY

LH₂ ON ORBIT PROPELLANT
TANK - LOW L/D
RECOMMENDED DESIGN

SK2-5085-127

WTFB

Figure 3-46:

Continued on page 97



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ASSEMBLY AND QUALIFICATION PROCEDURES

- 1 DETAILS AND SUBASSEMBLIES
 - a) PRESSURE VESSEL WELD ASSEMBLY. LN₂ COLD SHOCK, PROOF PRESSURE TEST, AND HELIUM LEAK CHECK.
 - b) SUPPORT STRAP DETAILS.
 - c) GIRTH RING MACHINING.
 - d) VACUUM JACKET SUBASSEMBLIES.
 - e) MLI BLANKET SUBASSEMBLIES.
 - f) MANHOLE COVER ASSEMBLY (COVER, SHUT-OFF VALVE, AND FEED LINE)
- 2 ASSEMBLE PRESSURE VESSEL, SUPPORT STRAPS, GIRTH RING, VENT LINE, AND VENT RELIEF VALVE.
- 3 INSTALL MLI BLANKET ON PRESSURE VESSEL. INSULATE SUPPORT STRAPS AND VENT LINE.
- 4 INSTALL VACUUM JACKET SUBASSEMBLIES. SEAL JOINTS AT GIRTH RING AND VENT LINE OUTLET FOR VACUUM.
- 5 INSTALL MANHOLE COVER, FEED LINE, AND SEAL VENT LINE. INSULATE THE AREA. INSTALL VACUUM JACKET COVER. SEAL JACKET COVER AND FEED LINE OUTLET FOR VACUUM.
- 6 HELIUM LEAK CHECK ASSEMBLY. REPAIR AS NECESSARY. PUMPDOWN VACUUM ANNULUS TO 1.5×10^{-5} TORR. CONDUCT VACUUM DECAY TEST. PRECONDITION WITH HEAT AND VACUUM PUMPING AS NECESSARY TO REACH REQUIRED DECAY RATE. PINCH OFF AND SEAL THE THREE FABRICATION PUMPDOWN PORTS. CLOSE VACUUM VALVES ON THE MAINTAINABILITY PUMPDOWN PORTS.
- 7 CONDUCT BOIL-OFF TEST.

NOTES

- 1 ALUMINUM ALLOY 2219-T81
- 2 ALUMINUM ALLOY ROLL FORGED RING 2219 T852
- 3 STAINLESS STEEL
- 4 WELD
- 5 WELDS IN 'AS WELDED' CONDITION
- 6 DIFFUSION BOND
- 7 MULTILAYER INSULATION (MLI) 8
 - a) ALUMINIZED MYLAR (0.15 MIL)/ DACRON NET (B4A) - 75 LAYERS/INCH.
 - b) OUTER 0.10 INCH LAYER - ALUMINIZED KAPTON (0.30 MIL)/ DACRON NET (B4A) - 75 LAYERS/ INCH
- 8 SANDWICH CONSTRUCTION
 - a) FACE SHEETS - ALUMINUM ALLOY 2024 T3
 - b) CORE - 5056 ALUMINUM FLEX-CORE (2.1 LB/FT³)
- 9 VACUUM SEAL
- 10 ASSEMBLE WITH NYLON STUDS AND GROMMETS
- 11 TITANIUM GAL-4V ELI

NAS3-14369

DL BARCLAY

LO₂ ON-ORBIT PROPELLANT
TANK - LOW L/D
RECOMMENDED DESIGN

SK2-5085-128

NOTED

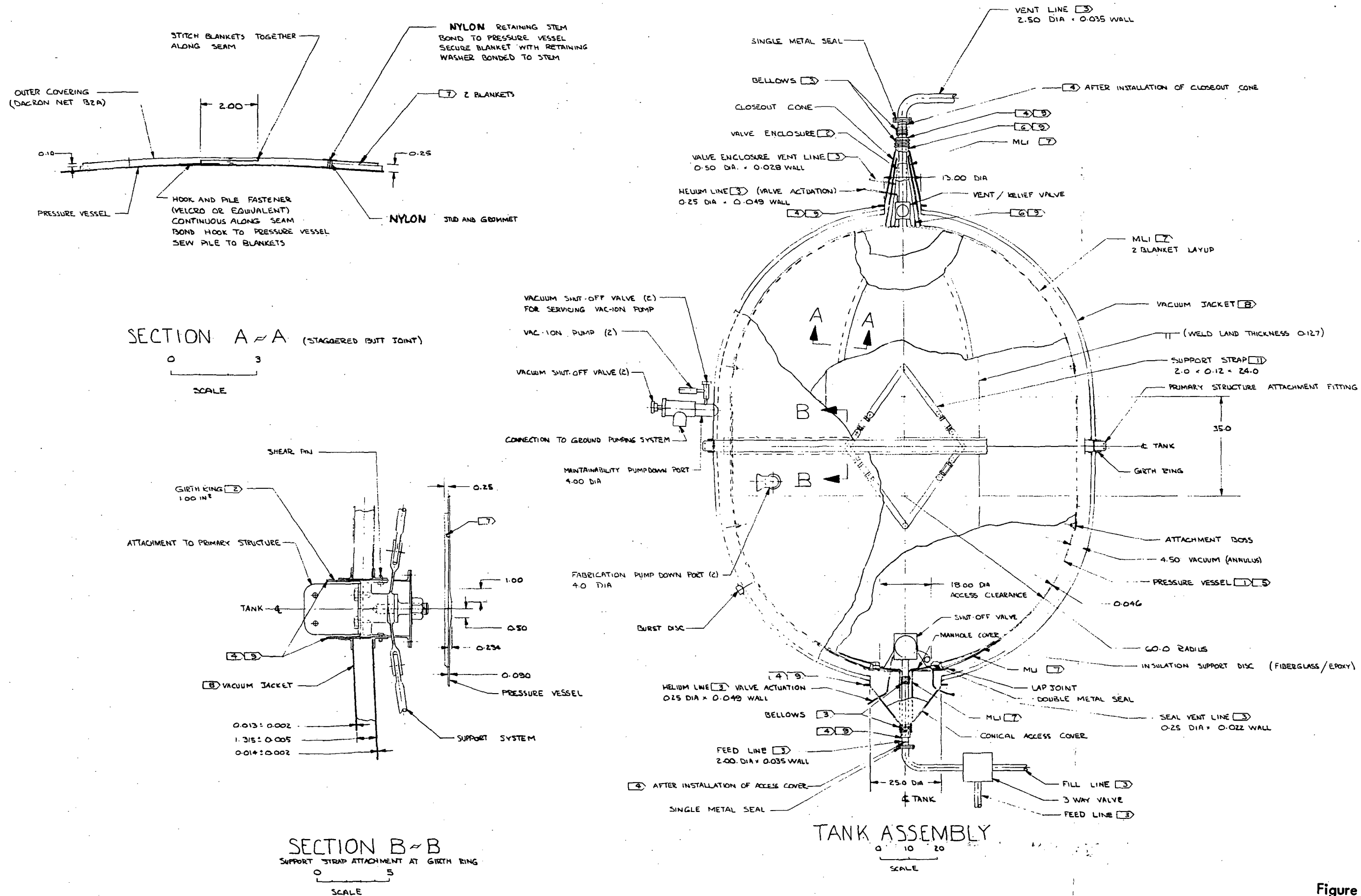


Figure 3-47:

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The MLI used in both designs is aluminized Mylar/dacron net (B4A) at 75 layers per inch (29.5 layer/cm) with the outer 0.10 in. (0.25 cm) layer, aluminized Kapton/dacron net (B4A) at 75 layers per inch (29.5 layer/cm).

Three 4.0 in. (10.16 cm) diameter fabrication pumpdown ports are provided for the LH₂ tank. Two are provided for the LO₂ tank. Three maintainability pumpdown ports, including shut-off valves and Vac Ion pumps, are provided for the LH₂ tank. Two are provided for the LO₂ tank.

Sixteen tension straps support the 2219-T81 pressure vessels from the girth rings. The LH₂ tank straps are made from fiberglass/epoxy. Titanium straps are used to support the heavier LO₂ tank. Small cross-sectional area straps are preferred to minimize disturbance of the MLI. Stainless steel turnbuckles pretension the straps after assembly.

The aluminum alloy roll forged girth ring is used to support the pressure vessel during MLI layup. The vacuum jacket heads are located to the ring by shear pins. Two circumferential welds seal the vacuum jacket assembly.

The feedline penetrates each tank at the manhole access. The submerged shut-off valve is mounted to the manhole cover. A double metal seal arrangement with a vent line between the seals protects the vacuum annulus from propellant leakage. The conical access cover is welded to the vacuum jacket and the feedline to vacuum seal this area. The 25.00 in. (63.5 cm) diameter opening with jacket at the manhole cover gives adequate access to the insulation.

The vent relief valve is mounted externally on the pressure vessel. A stainless steel valve enclosure seals off the valve from the vacuum annulus. This enclosure is vented overboard preventing propellant leakage at the valve from contaminating the insulation annulus. The 13.00 in. (33 cm) diameter opening in the vacuum jacket provides access to the valve and the surrounding insulation. The vacuum jacket is sealed with a conical closeout welded to the jacket and the vent line.

Stainless steel bellows are provided for both the vent and feed lines to allow for pressure vessel shrinkage and assembly misalignments.

3.6.2 Remaining Uncertainties

Initial pumpdown duration and the extent of preconditioning needed to achieve and maintain the required pressure levels in the vacuum annulus has not been determined.

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4.0 VACUUM SHELL STRUCTURAL TESTS AND VACUUM ACQUISITION TESTS

The objectives of these tests were:

- (1) Obtain material outgassing data for material selection
- (2) Obtain material property data for design of the sandwich shells
- (3) Determine the nature of any vacuum acquisition problems
- (4) Determine the structural adequacy of selected designs, and
- (5) Verify the proposed nondestructive proof test concept for sandwich vacuum jackets.

Five assemblies of candidate organic materials were tested in a vacuum at 350°F (450°K) to determine their outgassing characteristics. Three adhesive materials were tested in lap shear tensile tests at room temperature and 350°F (450°K) to select an acceptable adhesive for 350°F (450°K) service. Three sandwich beams were tested in flexure to evaluate the effect of fabrication techniques on the face material properties, especially stiffness.

Two "nondestructive" proof tests with external pressure were conducted on an 8 ft. (2.44 m) diameter ellipsoidal sandwich shell. One "nondestructive" proof test was conducted on a 45.0 in. (1.14 m) diameter hemispherical sandwich shell.

The F/S buckling prediction method has been proposed as a technique of non-destructively proof testing a structure to predict the critical strength. F/S stands for Force/Stiffness. The method consists of measuring the stiffness of the structure in response to a known force. By plotting F/S versus F the critical load can be predicted before the critical load is applied. If the F/S predicted is less than the design strength, reinforcement is added to the structure to bring the strength up to the design value. Structures which pass the proof test are structurally adequate. This method has been used extensively by Boeing on a NASA/Langley contract for Advanced Structural Panels. One test objective on this program was to evaluate the use of the F/S buckling prediction method for doubly curved shells loaded by external pressure. Although the test method was intended to be nondestructive, the loading in all three proof tests was continued until a buckle formed to experimentally verify the buckling prediction method proposed for future shell tests of this type.

Two 45 in. (1.14 m) diameter hemispherical sandwich heads were vacuum tested. The vacuum sealing inner skin on one head was spun and chem-milled, on the other it was a bonded gore construction.

Details of these tests and the results are described in the following sections. Evaluation of the structural test data is discussed in Section 5.0. The final conclusions are in Section 6.0.

4.1 Material Outgassing Tests

4.1.1 Purpose

These investigations were conducted to determine the effect of exposing potential sandwich shell construction materials to the MLI vacuum annulus. Results from these tests were used to select materials and sandwich configurations for the Task I study.

4.1.2 Specimen Selection

Specimens selected were representative of material classes suitable for the design environment. These were:

Core

- a) Fiberglass/Phenolic (HRP, Hexcel Aerospace)
- b) Fiberglass/Polyimide (HRH 327E, Hexcel Aerospace)
- c) 5052 Aluminum Flex-Core (Hexcel Aerospace)

Face Skins

- a) 2219 Aluminum alloy
- b) Titanium (Commercially pure)
- c) Fiberglass/Epoxy prepreg
- d) Fiberglass/Phenolic prepreg
- e) Fiberglass/Polyimide prepreg
- f) Boron/Epoxy prepreg (Narmco 5505/14)

Adhesives

- a) HT 424 (Bloomingdale Dept. American Cyanamid)
- b) Metlbond 329 (Narmco Materials Division, Whittaker Corp.)

4.1.3 Tests

A thermogravimetric analysis (TGA in helium), a differential thermal analysis (DTA), and an isotherm TGA in a vacuum at 350°F (450°K) were conducted on each of the candidate materials. Combinations of the materials were selected for five sandwich shell configurations. These specimens described in Figures 4-1, 4-2, and 4-3 were tested at 350°F (450°K) in a vacuum outgassing apparatus described in Figures 4-4 and 4-5.



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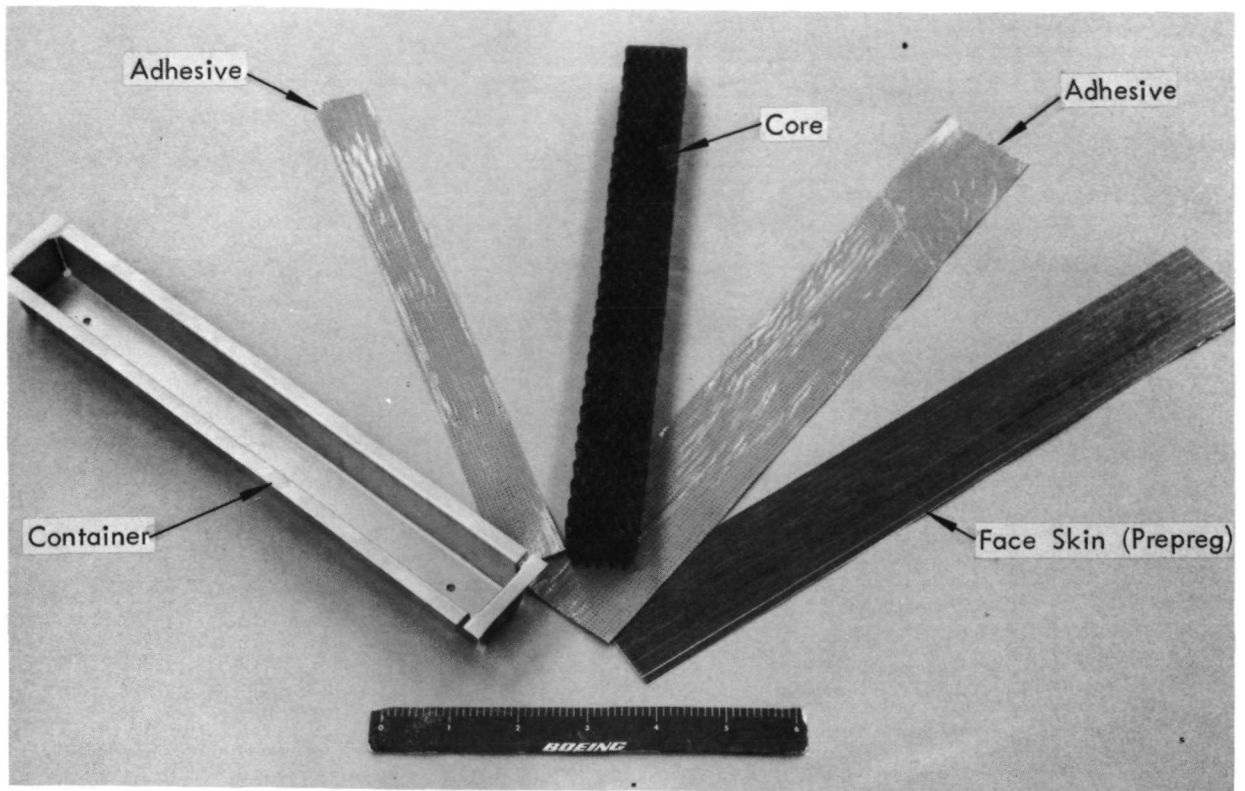


Figure 4-2: MATERIAL OUTGASSING SPECIMEN DETAILS

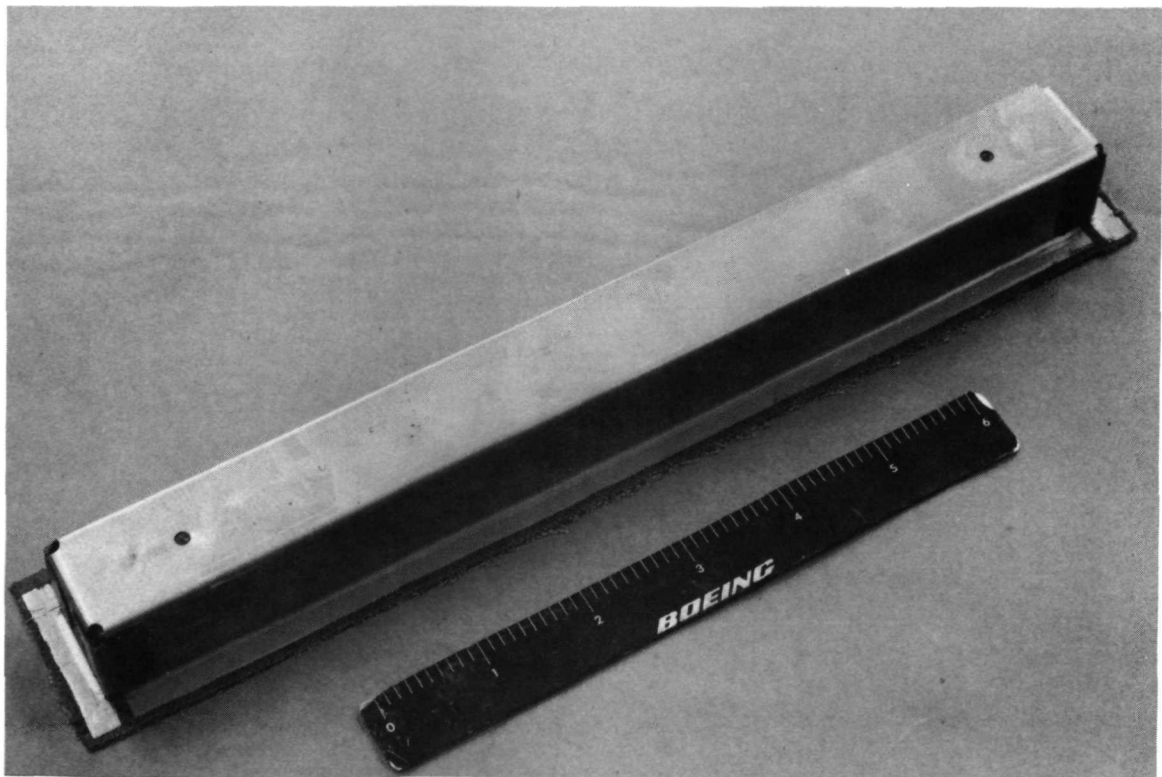


Figure 4-3 : MATERIAL OUTGASSING TEST SPECIMEN ASSEMBLY

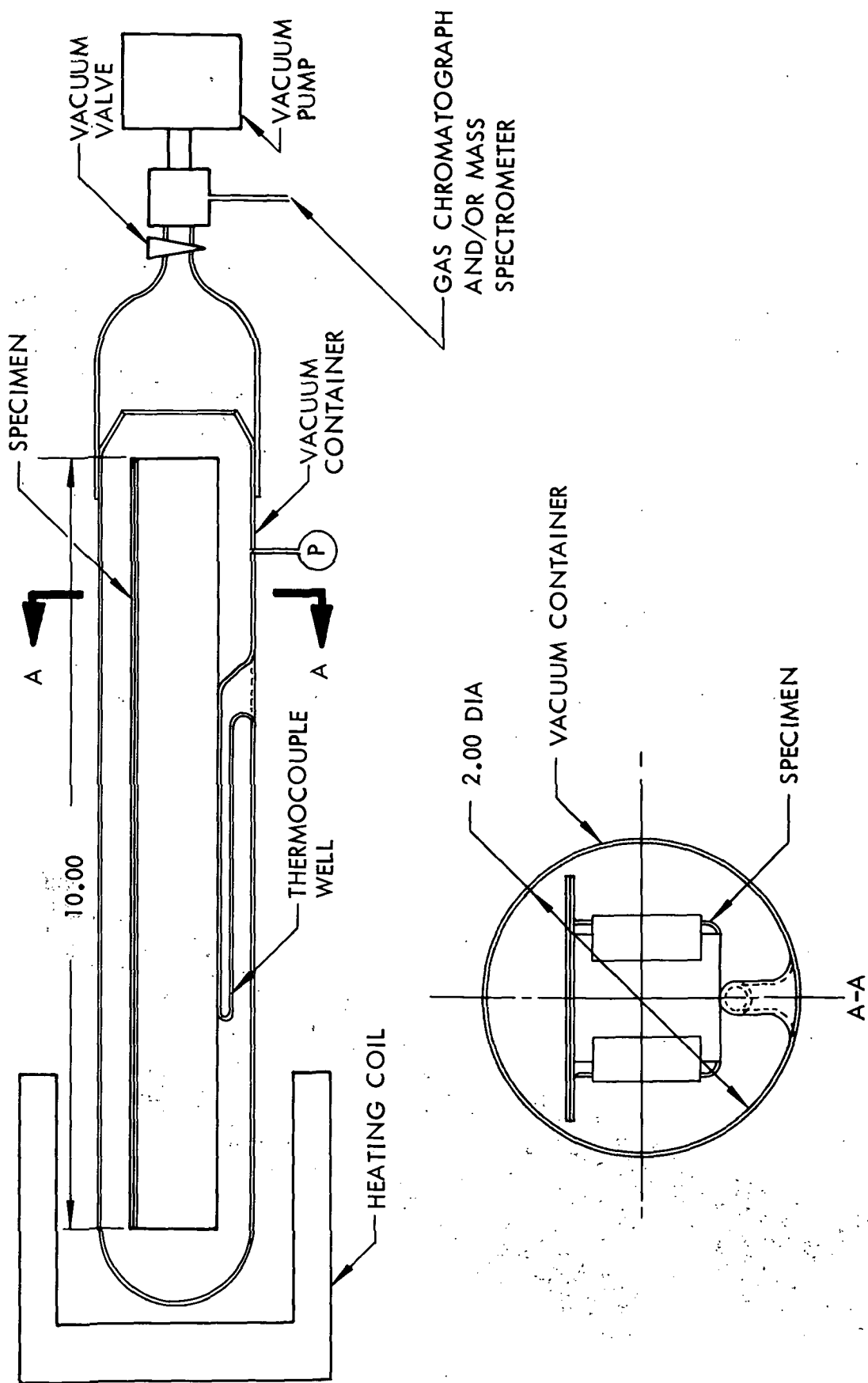


Figure 4-4: TEST SETUP SCHEMATIC - MATERIAL OUTGASSING TEST

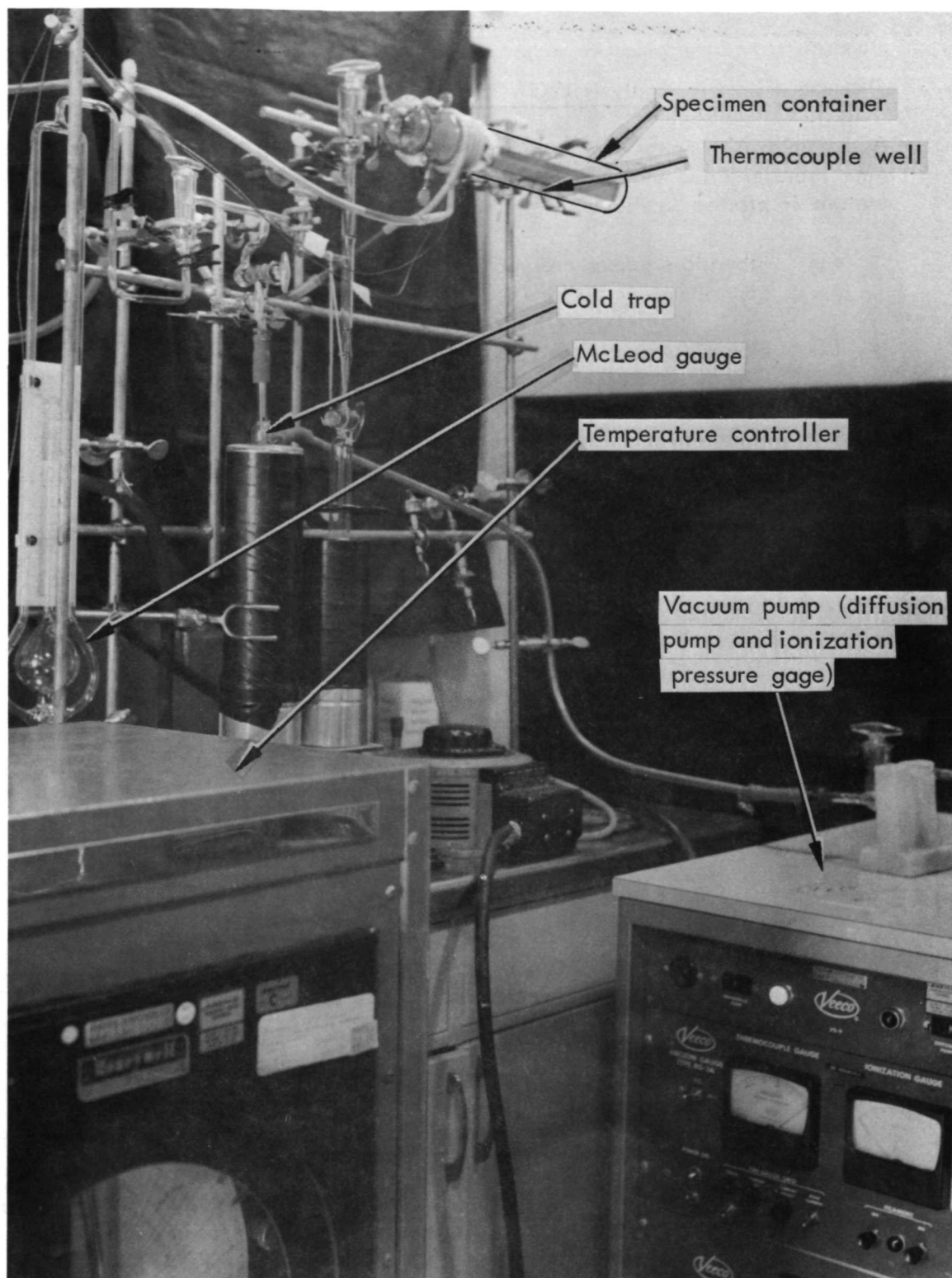


Figure 4-5 : VACUUM OUTGASSING APPARATUS - MATERIAL OUTGASSING TEST

Thermogravimetric Analysis (TGA in Helium)

A dynamic TGA in helium from room temperature to 1000°F (811°K) was run on nine representative material samples. The percentage loss in weight versus temperature is plotted in Figures 4-6 through 4-8. Results from these tests are:

- (1) Fiberglass/epoxy prepreg had no detectable weight loss up to approximately 500°F (533°K).
- (2) Fiberglass/phenolic prepreg shows no weight loss up to 155°F (342°K). Weight loss at 350°F (450°K) is 0.3%.
- (3) Fiberglass/polyimide prepreg shows a 0.3% weight loss of 100°F (311°K); which increases to 1.6% weight loss at 350°F (450°K).
- (4) Boron/epoxy prepreg (Narmco 5505/14) shows an unexplainable weight increase between 120°F (322°K) and 560°F (567°K).
- (5) Fiberglass/phenolic (HRP) honeycomb core shows a 0.4% weight loss at 110°F (317°K) which increases to 0.7% weight loss at 350°F (450°K).
- (6) Fiberglass/polyimide (HRH 327E) honeycomb core shows no weight loss up to 140°F (334°K). Weight loss at 350°F (450°K) is 0.2%.
- (7) 5052 Flex-Core had no detectable weight loss up to 1000°F (811°K). A slight weight increase is shown at 350°F (450°K) and above which suggests oxidation of aluminum by traces of oxygen.
- (8) Adhesive HT424 shows a 0.4% weight loss at 185°F (358°K) which increases to 0.6% weight loss at 350°F (450°K).
- (9) Adhesive metlbond 329 shows no weight loss up to 170°F (350°K). Weight loss at 350°F (450°K) is 0.2%.

Differential Thermal Analysis (DTA)

A DTA in nitrogen was run on nine representative material samples. The heat reaction vs. temperature results are plotted in Figures 4-9 through 4-11.

Isotherm TGA

An isotherm TGA in a vacuum at 350°F (450°K) was run on nine representative material samples. The percentage of original weight vs. time at 350°F (450°K) in a vacuum is plotted in Figures 4-12 through 4-14. Results from these tests are:

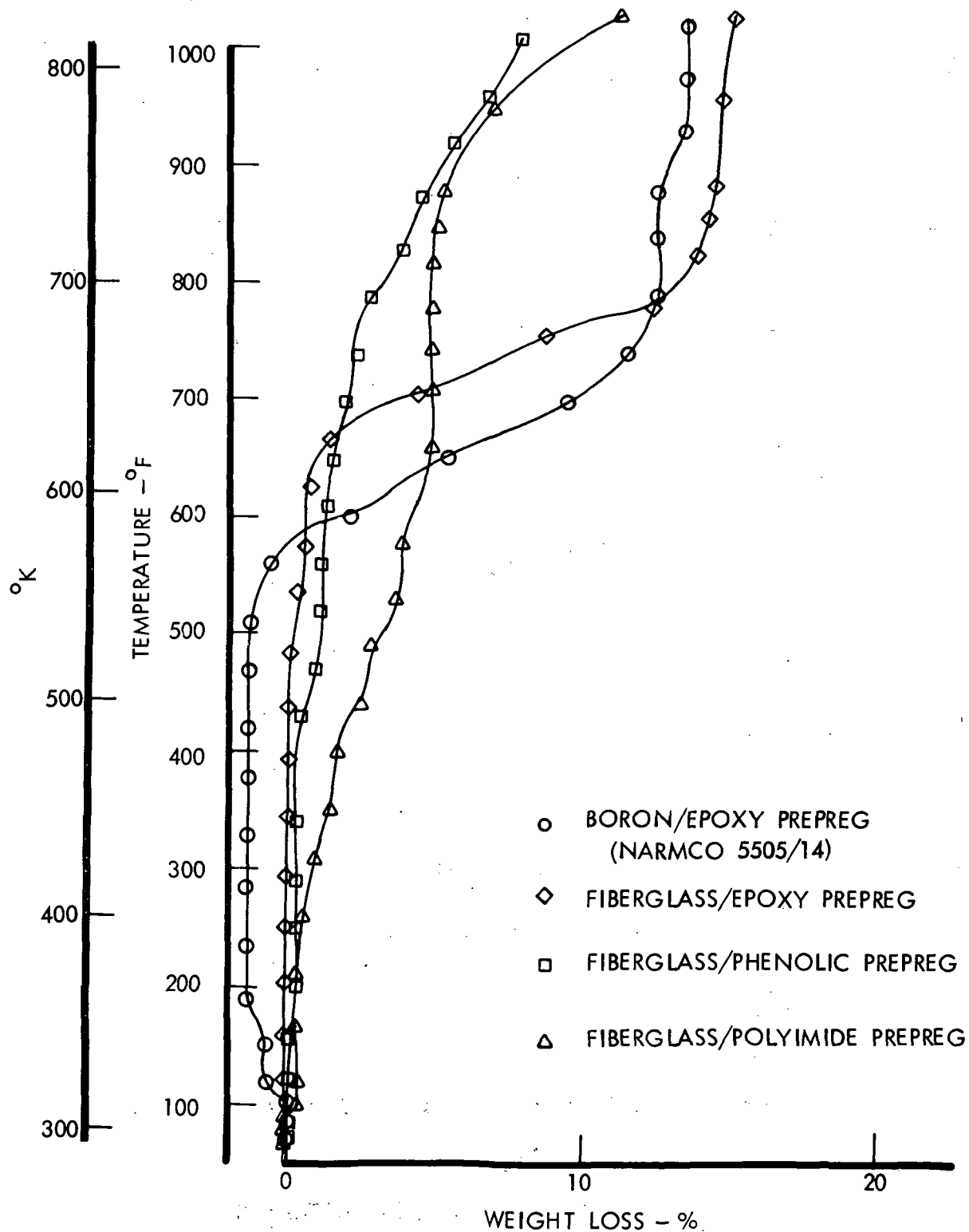


Figure 4-6: WEIGHT LOSS VS TEMPERATURE (TGA IN He)
4 FACE SKINS

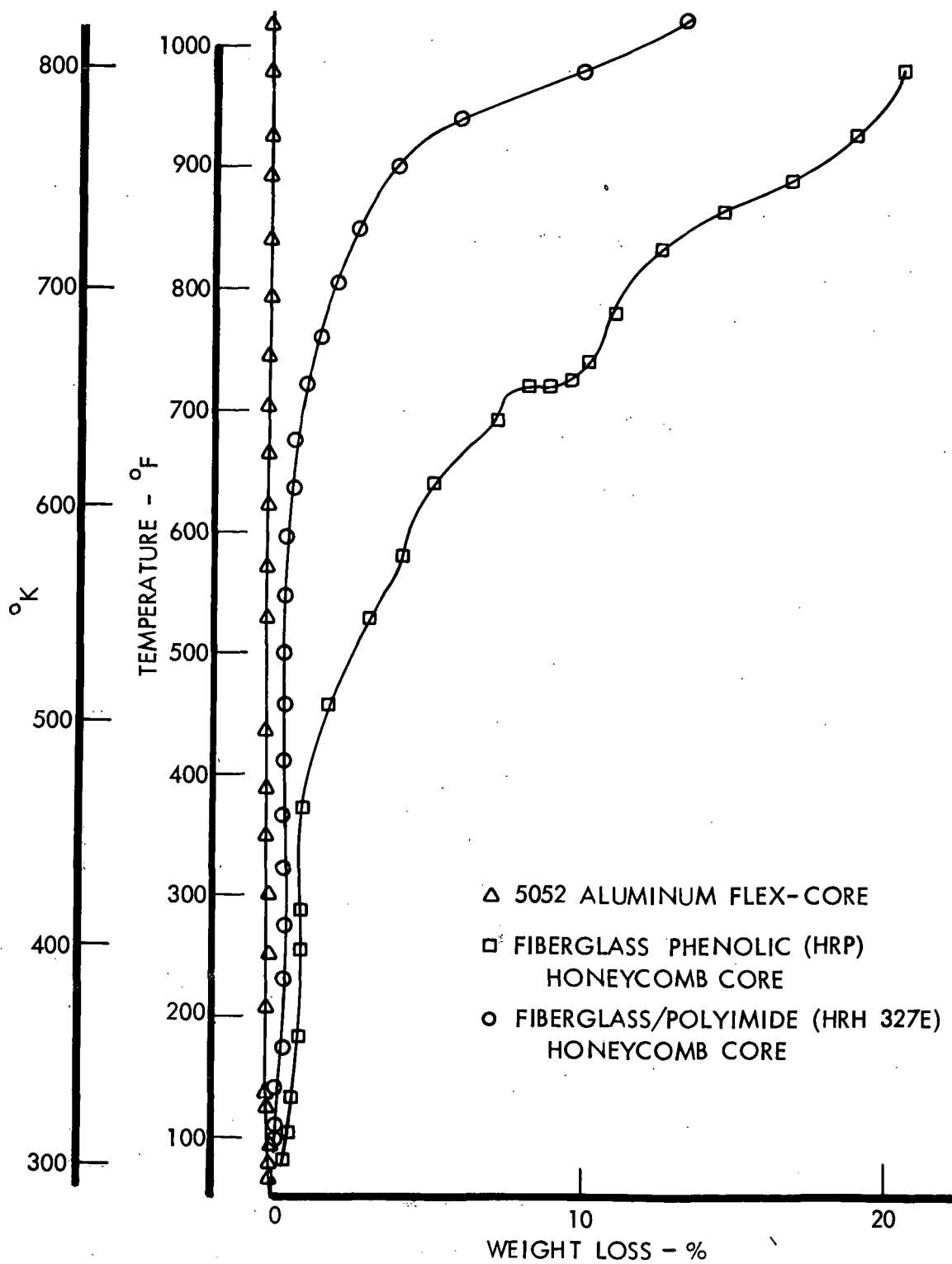


Figure 4-7: WEIGHT LOSS VS TEMPERATURE (TGA IN He)
3 CORES

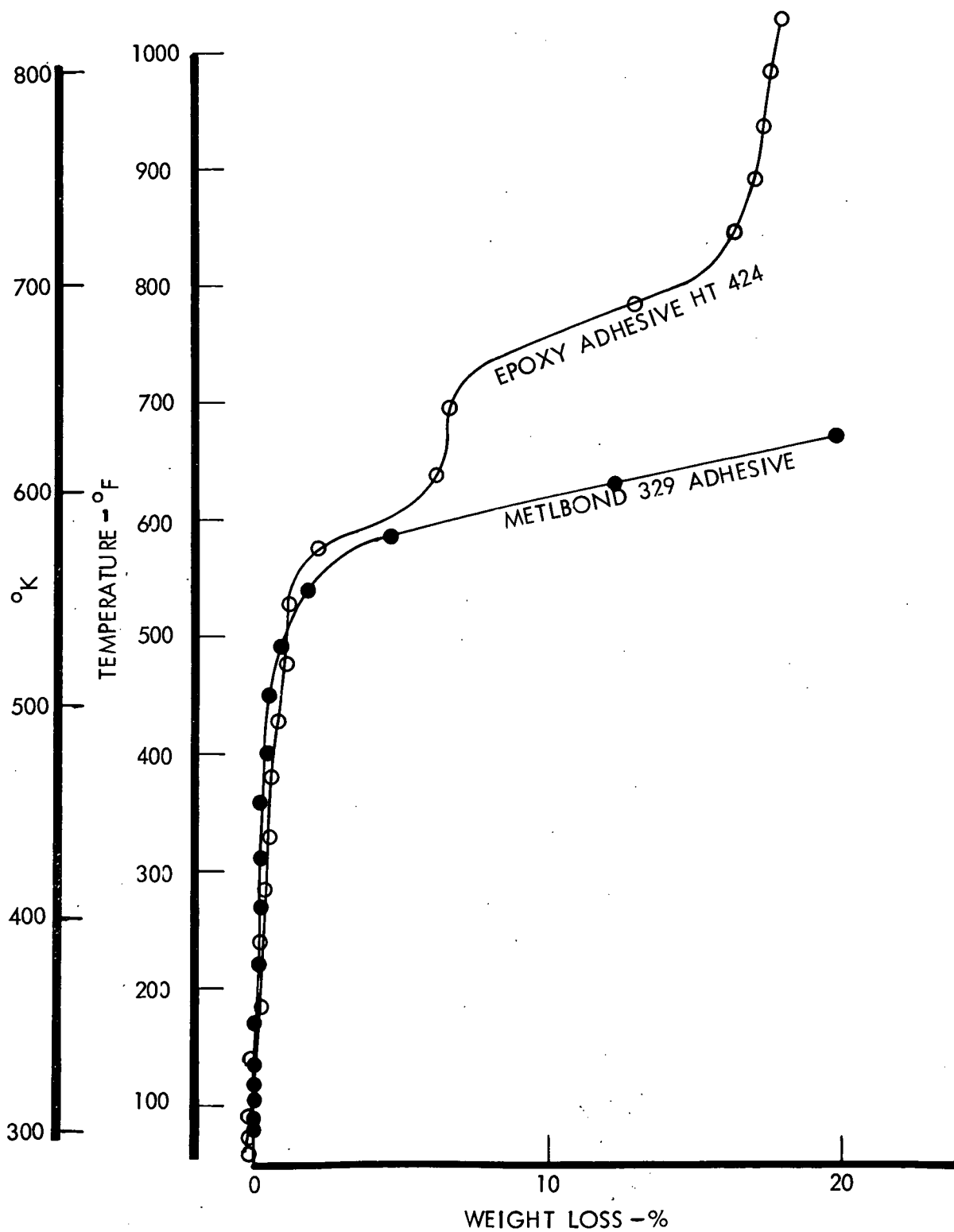


Figure 4-8: WEIGHT LOSS VS TEMPERATURE (TGA IN He)
2 ADHESIVES

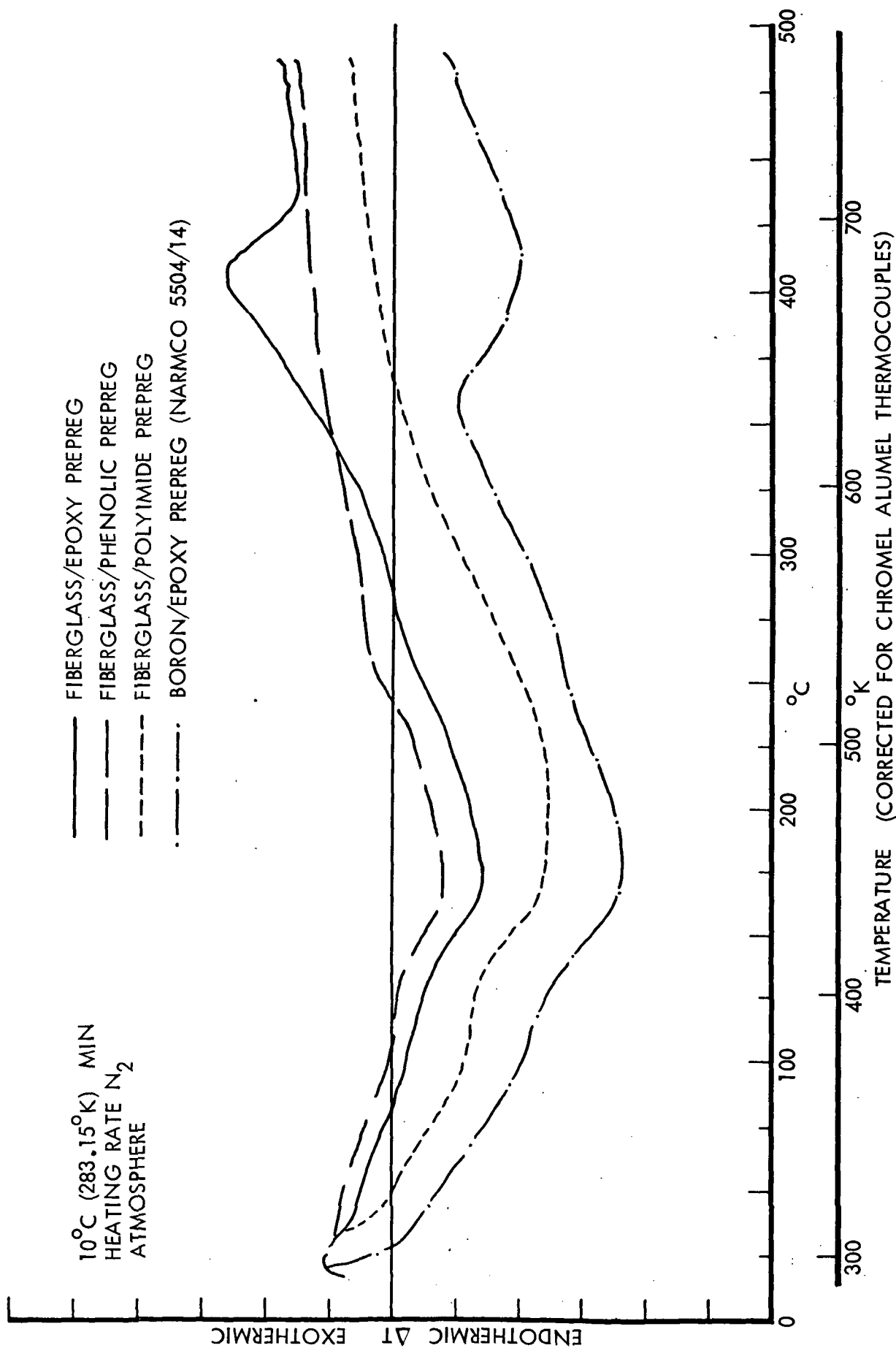


Figure 4-9: DIFFERENTIAL THERMAL ANALYSIS (DTA) 4 FACE SKINS

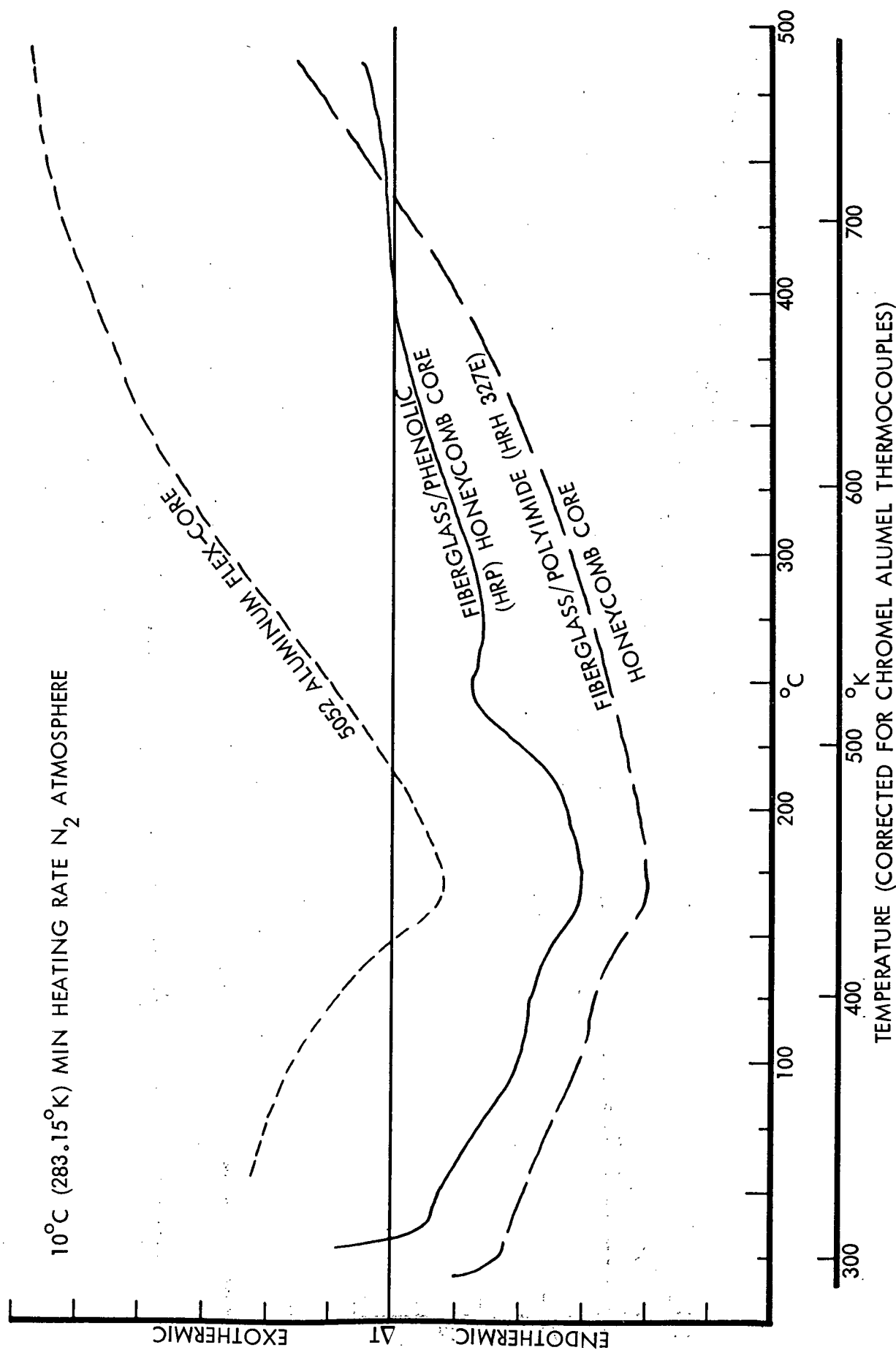
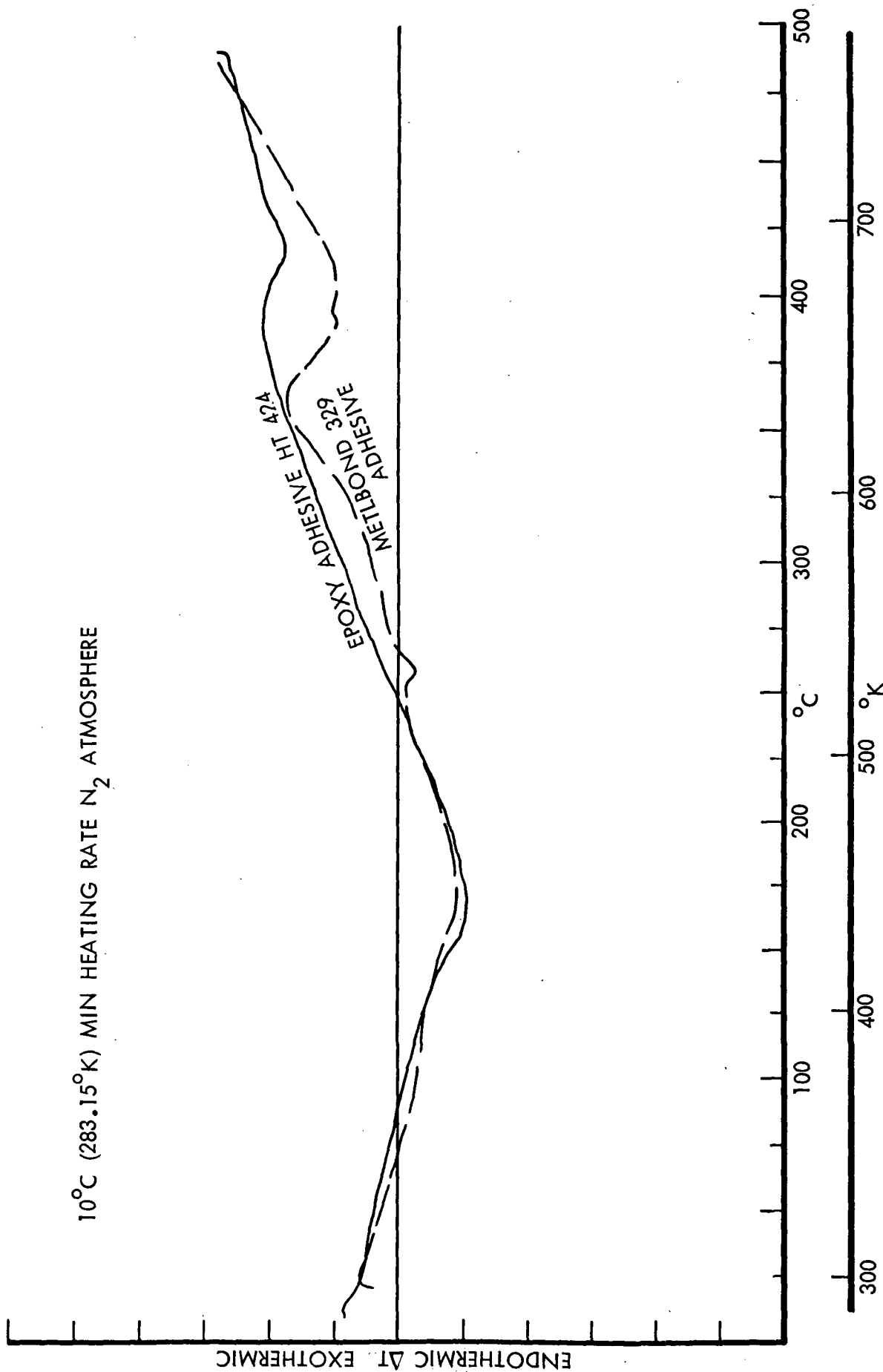


Figure 4-10: DIFFERENTIAL THERMAL ANALYSIS (DTA) 3 CORES

10°C (283.15°K) MIN HEATING RATE N₂ ATMOSPHERE



TEMPERATURE (CORRECTED FOR CHROMEL ALUMEL THERMOCOUPLES)

Figure 4-11: DIFFERENTIAL THERMAL ANALYSIS (DTA) 2 ADHESIVES

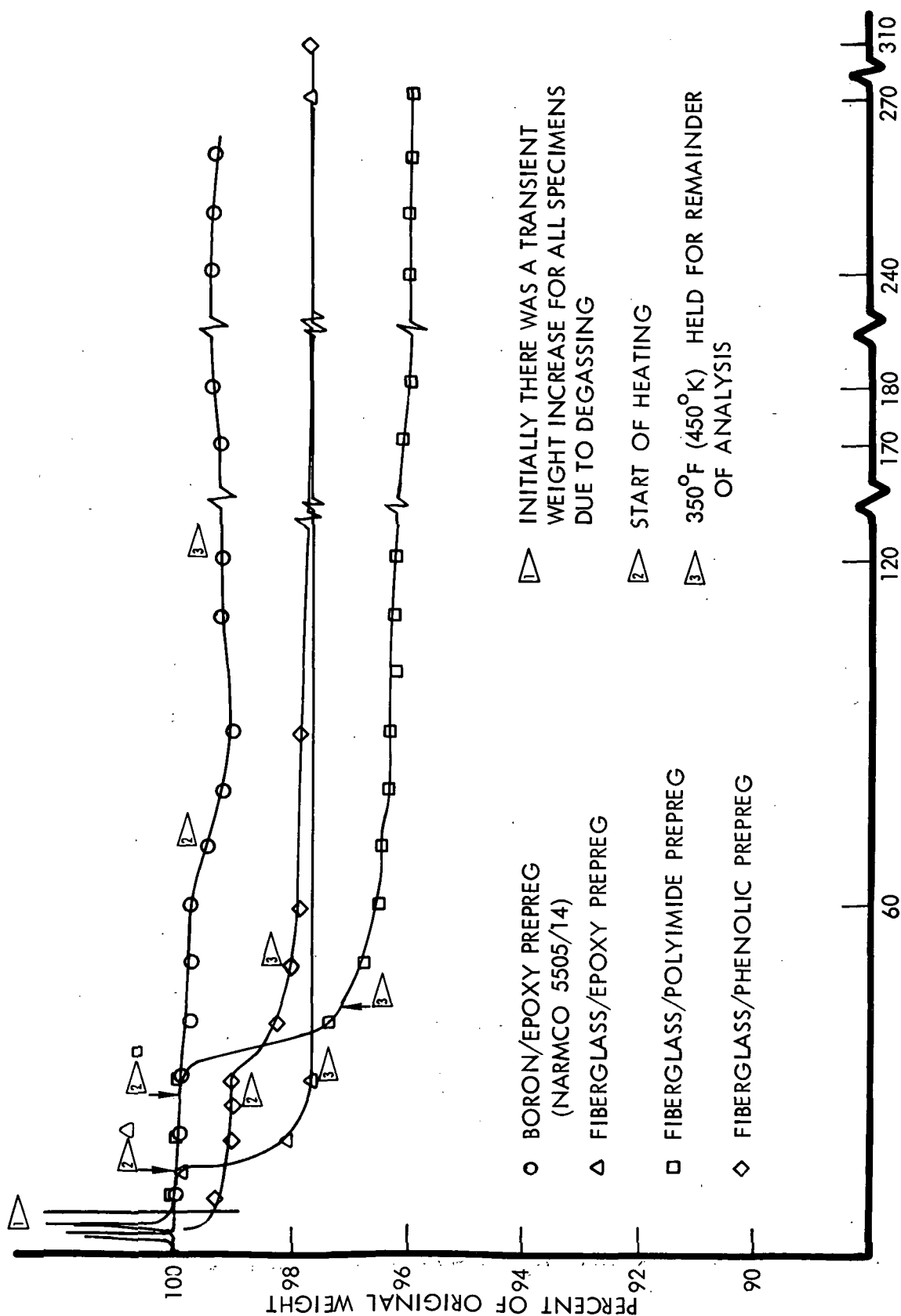


Figure 4-12: ISOTHERMAL GRAVIMETRIC ANALYSIS- 350°F (450°K) IN VACUUM
4 FACE SKINS

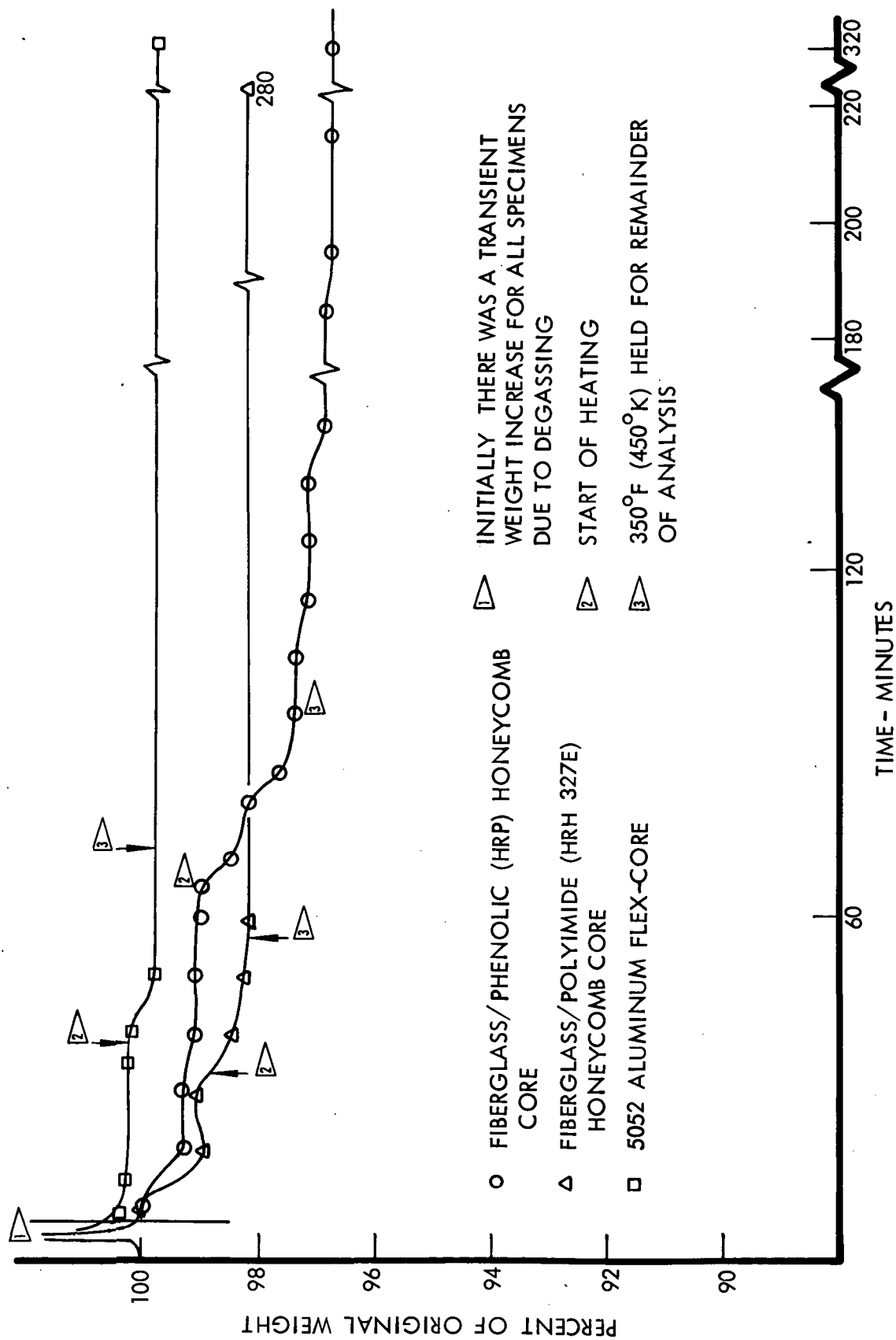


Figure 4-13: ISOTHERMAL GRAVIMETRIC ANALYSIS - 350°F (450°K) IN VACUUM
3 CORES

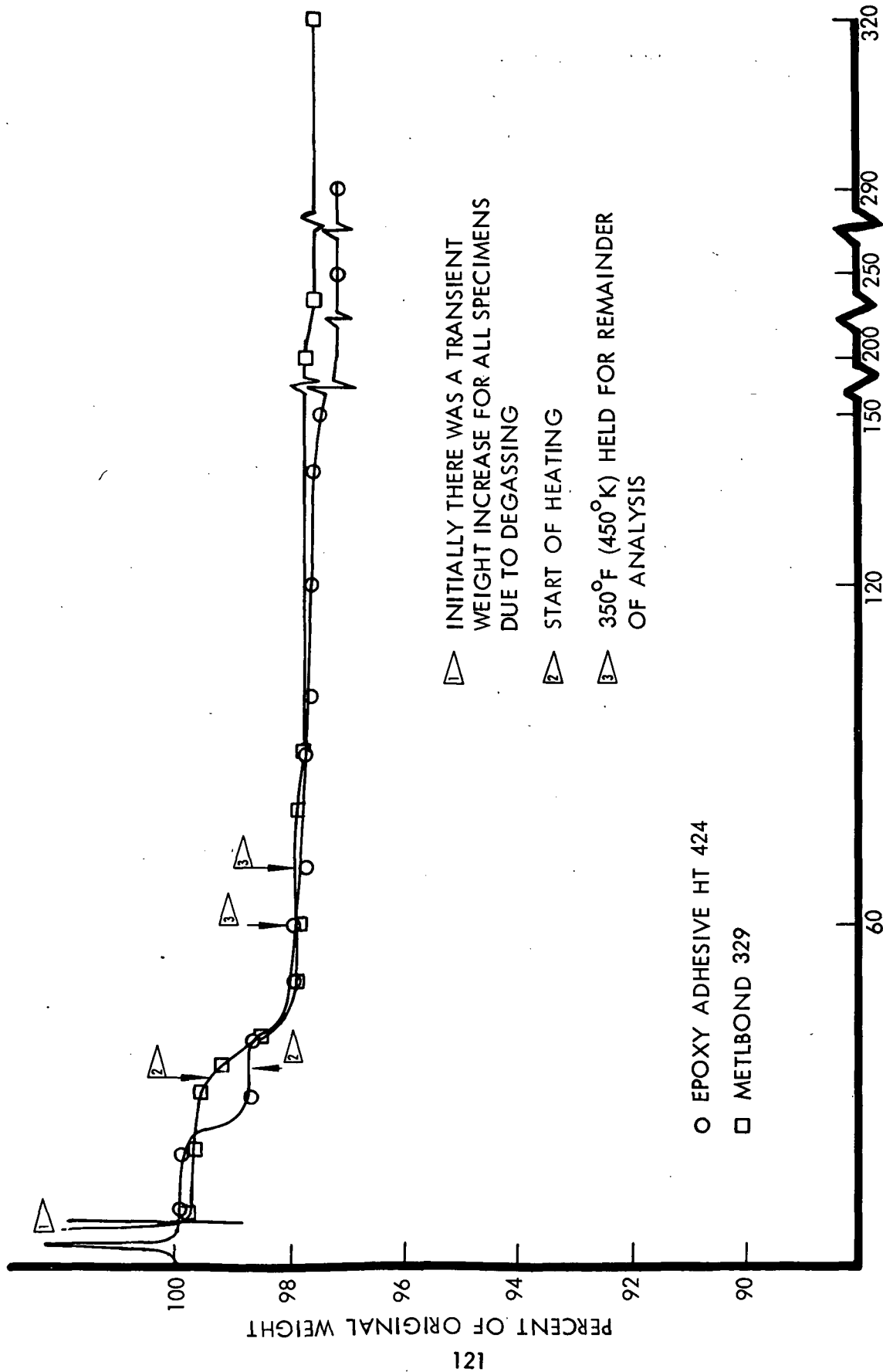


Figure 4-14: ISOTHERMAL GRAMMETRIC ANALYSIS - 350°F (450°K) IN VACUUM
2 ADHESIVES

- 1) Fiberglass/epoxy prepreg shows 97.7% of original weight after 270 minutes.
- 2) Fiberglass/phenolic prepreg shows 97.7% of original weight after 310 minutes.
- 3) Fiberglass/polyimide prepreg shows 96.0% of original weight after 270 minutes.
- 4) Boron/epoxy prepreg (Narmco 5505/14) shows 99.4% of original weight after 260 minutes.
- 5) Fiberglass/phenolic (HRP) honeycomb core shows 96.7% of the original weight after 320 minutes.
- 6) Fiberglass/polyimide (HRH 327E) honeycomb core shows 98.2% of the original weight after 280 minutes.
- 7) 5052 aluminum Flex-Core shows 99.7% of the original weight after 320 minutes.
- 8) Epoxy adhesive HT 424 shows 97.2% of original weight after 290 minutes.
- 9) Metlbond 329 adhesive shows 97.7% of the original weight after 320 minutes.

Sandwich Assembly Outgassing Tests

Tables 4-1 through 4-5 show the results of tests on the five sandwich assemblies using the outgassing apparatus shown in Figure 4-5.

Table 4-1: RESULTS OF VACUUM OUTGASSING TESTS AT 350°F (450°K)

-1 ASSEMBLY (Figure 4-1)

Container	2219 Aluminum Alloy
Core	Fiberglass/Phenolic (HRP)
Face Skin	Fiberglass/Epoxy Prepreg
Adhesive	Epoxy HT 424

Event No.	Cumulative Exposure Time at 350°F (450°K) (hr)	Pressure		Comments
		(Torr)	(N/m ²)	
1	-	1	1	2 3
2	0.75	6.5×10^{-3}	0.87	2
3	1.75	7.5×10^{-3}	1.0	2
4	2.75	6.0×10^{-3}	0.80	2
5	3.75	2.0×10^{-3}	0.27	2
6	-	-	-	4

1 The lowest pressure achieved during an initial 4 hours vacuum pumping at room temperature was 1.3×10^{-3} torr (0.17 N/m^2). During heat up to 350°F (450°K) the pressure increased to 6.5×10^{-3} torr (0.87 N/m^2) due to specimen outgassing.

2 These values are dynamic pressure values. The specimen-to-pump valve was maintained in the open position.

3 Heat and specimen-to-pump valve were shut off overnight. During this period the specimen experienced only cryogenic pumping at -110°F (317°K).

4 After cooling to room temperature a pressure of 5×10^{-5} torr (6.65 mN/m^2) was achieved. With specimen-to-pump valve closed, this vacuum decayed to 2×10^{-4} torr (26.6 mN/m^2) (at room temp) after a 45 minute period of time.

Table 4-2: RESULTS OF VACUUM OUTGASSING TESTS AT 350°F (450°K)

-2 ASSEMBLY (Figure 4-1)

Container	2219 Aluminum Alloy
Core	Fiberglass/Phenolic (HRP)
Face Skin	Fiberglass/Polyimide Prepreg
Adhesive	Epoxy HT424

Event No.	Cumulative Exposure Time at 350°F (450°K) (hr)	Pressure		Comments
		(Torr)	(N/m ²)	
1	-	1	1	2
2	0.75	1.2×10^{-2}	1.6	2
3	2.00	8.0×10^{-3}	1.28	2
4	4.00	2.8×10^{-3}	0.27	2
5	-	-	-	3
6	6.00	4	4	3

- 1 The lowest pressure achieved after 1 hour vacuum pumping at room temperature was 1.5×10^{-4} torr (20 mN/m²). During heat up to 350°F (450°K) (1/2 hour) the pressure increased to 6.5×10^{-2} Torr (18.65 N/m²) due to specimen outgassing.
- 2 These values are dynamic pressure values. The specimen-to-pump valve was maintained in the "open" position.
- 3 The specimen-to-pump valve was closed at this point and kept closed for the remainder of the test to obtain "static" pressures and vacuum decay rate at 350°F (450°K).
- 4 The pressure exerted due to specimen outgassing rose beyond the limit of the McLeod gauge scale which reads to a maximum of 1 torr (133 N/m²).

Table 4-3: RESULTS OF VACUUM OUTGASSING TESTS AT 350°F (450°K)

- 3 ASSEMBLY (Figure 4-1)

Container	2219 Aluminum Alloy
Core	Fiberglass/Phenolic (HRP)
Face Skin	Fiberglass/Phenolic Prepreg
Adhesive	Epoxy HT 424

Event No.	Cumulative Exposure Time at 350°F (450°K) (hr)	Pressure		Comments
		(Torr)	(N/m ²)	
1	-	1	1	2
2	1.25	8.2×10^{-3}	1.1	2
3	2.50	7.8×10^{-3}	1.05	2
4	-	-	-	3
5	4.00	8.0×10^{-3}	1.07	3
6	6.00	4	4	3

- 1 The lowest pressure achieved after 1 hour vacuum pumping at room temperature was 3.5×10^{-3} torr (0.47 N/m^2). During heat up to 350°F (450°K) (1/2 hour) the pressure increased to 9.0×10^{-3} Torr (1.2 N/m^2) due to specimen outgassing.
- 2 These values are dynamic pressure values. The specimen-to-pump valve was maintained in the "open" position.
- 3 The specimen-to-pump valve was closed at this point and kept closed for the remainder of the test to obtain "static" pressures and vacuum decay rate at 350°F (450°K).
- 4 The pressure exerted due to specimen outgassing rose beyond the limit of the McLeod gauge scale which reads to a maximum 1 torr (133 N/m^2).

Table 4-4: RESULTS OF VACUUM OUTGASSING TESTS AT 350°F (450°K)

-4 ASSEMBLY (Figure 4-1)

Container	Titanium
Core	Fiberglass/Polyimide (HRH 327E)
Face Skin	Fiberglass/Polyimide Prepreg
Adhesive	Metlbond 329

Event No.	Cumulative Exposure Time at 350°F (450°K) (hr)	Pressure		Comments
		(Torr)	(N/m ²)	
1	-	1	1	2
2	1.00	5.0×10^{-3}	0.67	2
3	2.00	3.8×10^{-3}	0.51	2
4	3.00	1.5×10^{-3}	0.20	2
5	-	-	-	3
6	4.00	4	4	3
7	5.00	4	4	3

1 The lowest pressure achieved after 1 hour vacuum pumping at room temperature was 2.2×10^{-3} torr (0.29 N/m²). During heat up to 350°F (450°K) (1/2 hour) the pressure increased to 5.5×10^{-3} Torr (0.73 N/m²) due to specimen outgassing.

2 These values are dynamic pressure values. The specimen-to-pump valve was maintained in the "open" position.

3 The specimen-to-pump valve was closed at this point and kept closed for the remainder of the test to obtain "static" pressures and vacuum decay rate at 350°F (450°K).

4 The pressure exerted due to specimen outgassing rose beyond the limit of the McLeod gauge scale which reads to a maximum of 1 torr (133 N/m²).

Table 4-5: RESULTS OF VACUUM OUTGASSING TESTS AT 350°F (450°K)

- 5 ASSEMBLY (Figure 4-1)

Container	2219 Aluminum Alloy
Core	5052 Aluminum Flex-Core
Face Skin	Boron/Epoxy Prepreg (Narmco 5505/14)
Adhesive	Epoxy HT 424

Event No.	Cumulative Exposure Time at 350°F (450°K) (hr)	Pressure		Comments
		(Torr)	(N/m ²)	
1	-	1	1	2
2	0.83	2.6×10^{-3}	0.35	2
3	1.83	2.0×10^{-3}	0.27	2
4	2.83	2.0×10^{-3}	0.27	2
5	-	-	-	3
6	3.83	4	4	3
7	4.66	4	4	3

1 The lowest pressure achieved after 1 hour vacuum pumping at room temperature was 5×10^{-4} torr (0.07 N/m^2). During heat up to 350°F (450°K) (1/2 hour) the pressure increased to 5.2×10^{-3} Torr (0.69 N/m^2) due to specimen outgassing.

2 These values are dynamic pressure values. The specimen-to-pump valve was maintained in the "open" position.

3 The specimen-to-pump valve was closed at this point and kept closed for the remainder of the test to obtain "static" pressure and vacuum decay rate at 350°F (450°K).

4 The pressure exerted due to specimen outgassing rose beyond the limit of the McLeod gauge scale which reads to a maximum of 1 torr (133 N/m^2).

4.1.4 Discussion of Results

The data presented shows that all the organic materials selected for testing exhibited outgassing. Use of materials that outgas should be avoided in the evacuated MLI annulus. Therefore, the data suggests that in sandwich shell vacuum jackets the vacuum sealing surface should be the inner face skin, and this skin should be metal.

The isothermal gravimetric analyses, to the extent of their duration (approximately 5 hours), on some materials exhibited negligible percentage weight loss after initial outgassing in a vacuum. The tests suggest that preconditioning of these materials could produce low outgassing organic - containing reinforcement components. The foregoing conclusion would need to be verified under simulated use conditions.

4.2 Material Structural Tests

4.2.1 Adhesive Tests

Purpose

Lap shear tensile tests were conducted on three adhesive candidates at room temperature and 350°F (450°K) to select a structurally adequate adhesive system for bonding the aluminum gore inner skin for the 45 in. (1.14 m) diameter test tank.

Tests

The lap shear specimens had a 1.00 in (2.54 cm) lap length. The aluminum plates used were alkaline cleaned. The 3 M - XA-3919 was the favored candidate since vacuum leak tests at 350°F (450°K) had been performed previously with good results. The lap shear test results are tabulated below:

<u>ADHESIVE</u>	<u>SPECIMEN</u>	<u>°F</u>	<u>°K</u>	<u>lb</u>	<u>kN</u>
3M-XA-3919	A	RT	RT	2260	10.05
	B	RT	RT	2350	10.45
	C	+350	450	1440	6.41
	D	+350	450	1430	6.36
3M-EC-2290	A	RT	RT	2855	12.70
	B	RT	RT	2910	12.94
	C	+350	450	135	0.60
	D	+350	450	132	0.59

<u>ADHESIVE</u>	<u>SPECIMEN</u>	<u>°F</u>	<u>°K</u>	<u>lb</u>	<u>kN</u>
3M-EC-1469	A	RT	RT	2330	10.36
	B	RT	RT	2290	10.19
	C	+350	450	748	3.33
	D	+350	450	613	2.73

Of the three candidates tested only EC 1469 and XA-3919 had acceptable strength at 350°F (450°K).

3M-XA-3919 adhesive is AF 130 in solution, and is applied by brush coating both surfaces. The surfaces are then air dried for 15 minutes followed by a 45 minutes drying cycle at 225°F. The bond is obtained by curing at 350°F (450°K) for 60 minutes at 50 psi (345 kN/m²). 3M-EC-1469 adhesive has no solvents but instead is a 100% solids paste. Its cure cycle is 350°F (450°K) for 2 hours at 50 psi (345 kN/m²).

The volatile content of EC-1469, as determined by vapor phase chromatography, was found to be zero for the uncured adhesive. No analysis of the cured adhesive was made. The relative volatile content of XA-3919 was determined as a function of flash-off time at 225°F (563°K). The table below shows the results.

<u>Test</u>	<u>Flash-off Time (min)</u>	<u>Approximate Volatile Concentration (% by Weight)</u>
1	45	1
2	90	0.2
3	180	0.1

It can be seen that a prolonged flash-off period reduces the solvent content significantly.

Conclusions

Based on the data presented, XA-3919 was selected for bonding the aluminum gore inner skin.

4.2.2 Sandwich Beam Tests

Three sandwich beam specimens were made and tested to evaluate the effect of the planned vacuum jacket fabrication technique on fiberglass face stiffness. The fiberglass was laminated so that dimples were not formed over the cells. Two specimens were 1.5 in. (3.81 cm) wide, 18.0 in. (45 cm) long. These specimens had two ply fiberglass face skins, exact representations of the 45 in. (1.14 m) diameter hemispherical shell wall. The third specimen was constructed with a single fiberglass ply face skin. The beams were simply supported on 17 in. (43 cm) centers and a load was applied at mid-span. Load versus deflection was recorded. The specimens were oriented so that the fiberglass face was in compression.

Table 4-6 lists the results for the three specimens. All the test values exceeded the requirements of the BMS 8-139 specification, confirming that the planned fiberglass face skin layup method for the vacuum jacket was acceptable.

4.3 Nondestructive Proof Test

4.3.1 Purpose

External pressure tests were conducted on an 8 ft (2.44 m) diameter ellipsoidal sandwich head to evaluate the nondestructive test method. Data obtained from these tests were used in the final evaluation of the sandwich shell analysis techniques.

4.3.2 Head Design, Analysis and Fabrication

Design and Analysis

Figures 4-15 and 4-16 show the 8 ft (2.44 m) diameter sandwich head assembly. The inner skin of this assembly was an existing 2219-T62 pressure vessel shell. This skin had a nominal thickness of 0.043 in. (0.092 cm). Locally it was thickened to 0.096 in. (0.244 cm) at the apex where a pickup lug was located, and to 0.073 in. (0.185 cm) at the equator.

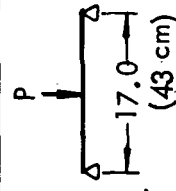
The core density and thickness and the fiberglass/epoxy face skin (outer) thickness were initially determined by an OPTRAN analysis then checked with a BOSOR 3 analysis.

OPTRAN Analysis

Reference 9 states that the theoretical and experimental results for thin oblate spheroidal shells are similar to those for a sphere of radius

Table 4-6: SANDWICH BEAM TEST RESULTS 

SPECIMEN NO.	t_1 		t_c 		t_2 (2 Plies) 		ULTIMATE LOAD		P/Y 		CALCULATED E_2		ULTIMATE COMPOSITE STRESS		BMS 8-139 SPEC. MINIMUM VALUES FOR 3 PLIES					
																	E		F _{cu}	
																				
in	cm	in	cm	in	cm	lb	N	lb/in	N/cm	psi $\times 10^6$	MN/m ²	psi	MN/m ²	psi $\times 10^6$	MN/m ²	psi	MN/m ²			
1	.010	.025	.410	1.04	.010	.025	63.4	282.0	70	122	3.2	22063	38,800	267.5	2.7	18615.8	27,000	186.2		
2	.010	.025	.410	1.04	.010	.025	59.3	263.8	70	122	3.2	22063	36,100	248.9	2.7	18615.8	27,000	186.2		
3	.010	.025	.410	1.04	.005	.012	33.2	147.6	50	87.5	3.9	26889	38,000	262.0	2.7	18615.8	27,000	186.2		



-  Single Point Load at Mid-Span, 17.0 in. (43 cm) Span
-  Alum. 2024-T3, $E = 10.5 \times 10^6$ psi (72394 MN/m²), $F_{ty} = 47,000$ psi (324 MN/m²)
-  5056/F40 -0014, 2.1 lb/ft³ Flex-Core
-  Fiberglass (BMS 8-139) 2 Plies or 1 Ply, 120 Glass Cloth @ 0.005 in. (.012 cm) Per Ply
-  Slope of Load-Deflection Curve Taken at Mid-Span

Figure 4-15:

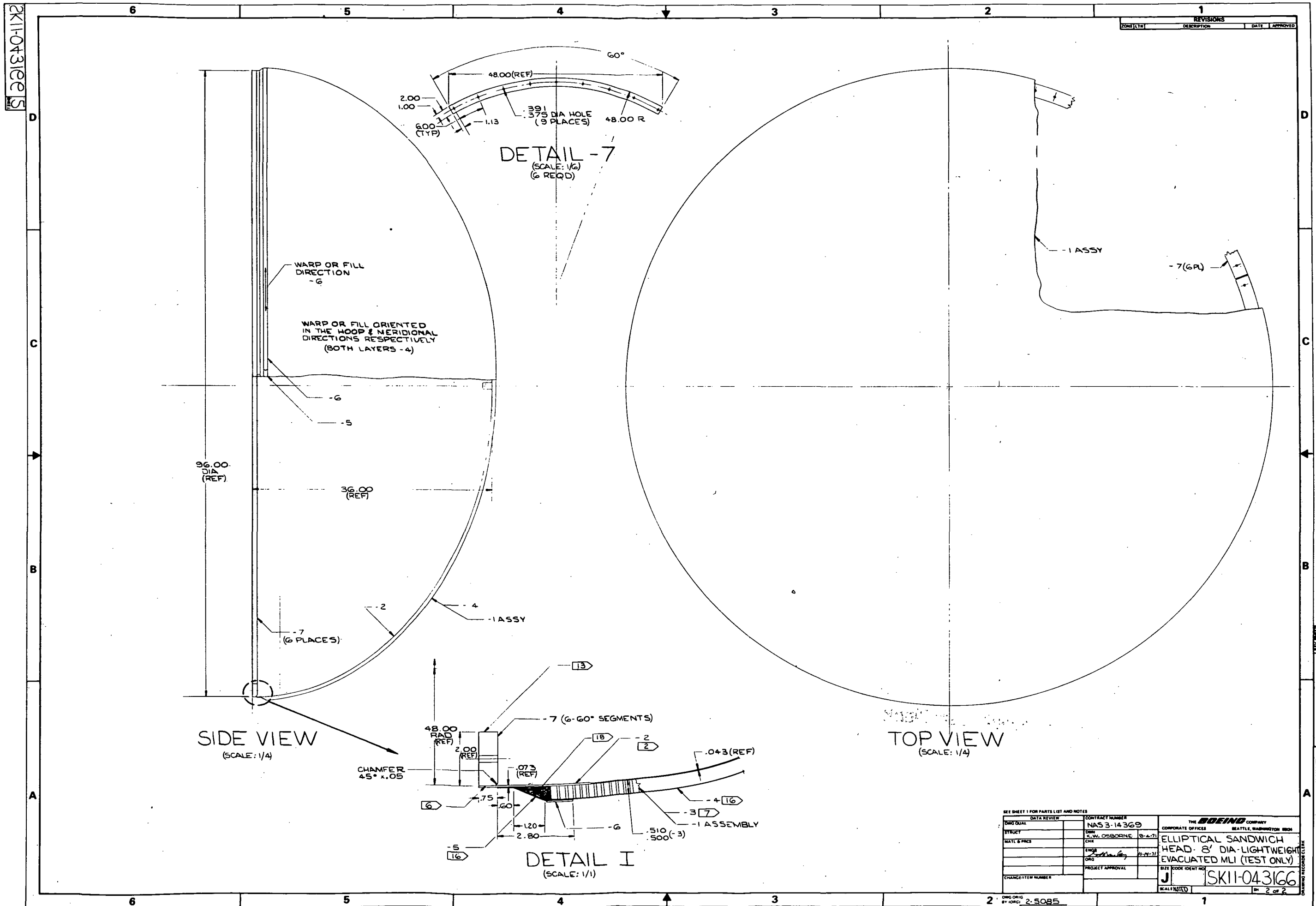


Figure 4-16:

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$$R_{\max} = B^2/A$$

where

B is the apex height and

A is the equatorial radius.

For the 2219-T62 shell

$$R_{\max} = 48^2/36 = 64 \text{ in. (1.63 m)}$$

Therefore, the design was handled as though the oblate spheroidal shell was a hemisphere with a 64 in. (1.63 m) radius. Sullins, Smith and Spier (Reference 10) treat the sandwich shell in the same manner. In addition, they have summarized test data to determine the knockdown factor for sandwich domes subjected to uniform external pressure. These data are summarized in Figure 4-17.

OPTRAN designs were made for this shell using probability (of not failing) factors of 0.5, 0.90, and 0.99. These factors were based on Boeing statistical data for externally pressurized domes. The OPTRAN designs assumed a factor of safety of 1.4 on a limit design pressure of 14.7 psi (101.4 kN/m²).

Optimum vacuum jacket designs from this OPTRAN study for the 8 ft (2.44 m) diameter shell for an ultimate design pressure of 20.6 psi (142.0 kN/m²) with R = 64 in. (1.63 m) and using the 2219 aluminum alloy inner face skin, a 5056 aluminum Flex-Core and a fiberglass/epoxy prepreg outer face skin were:

 Prob- ability of Not Failing	Fixed Inner Face Skin Thickness		Optimum Outer Face Skin Thickness		Optimum Core Thickness		Optimum Core Density	
	in.	cm	in.	cm	in.	cm	lb/ft ³	kg/m ³
0.5	0.043	0.092	0.010	0.025	0.345	0.876	2.1	33.64
0.90	0.043	0.092	0.010	0.025	0.625	1.588	2.1	33.64
0.99	0.043	0.092	0.010	0.025	0.963	2.446	2.1	33.64

 Using Boeing statistical data for externally pressurized domes

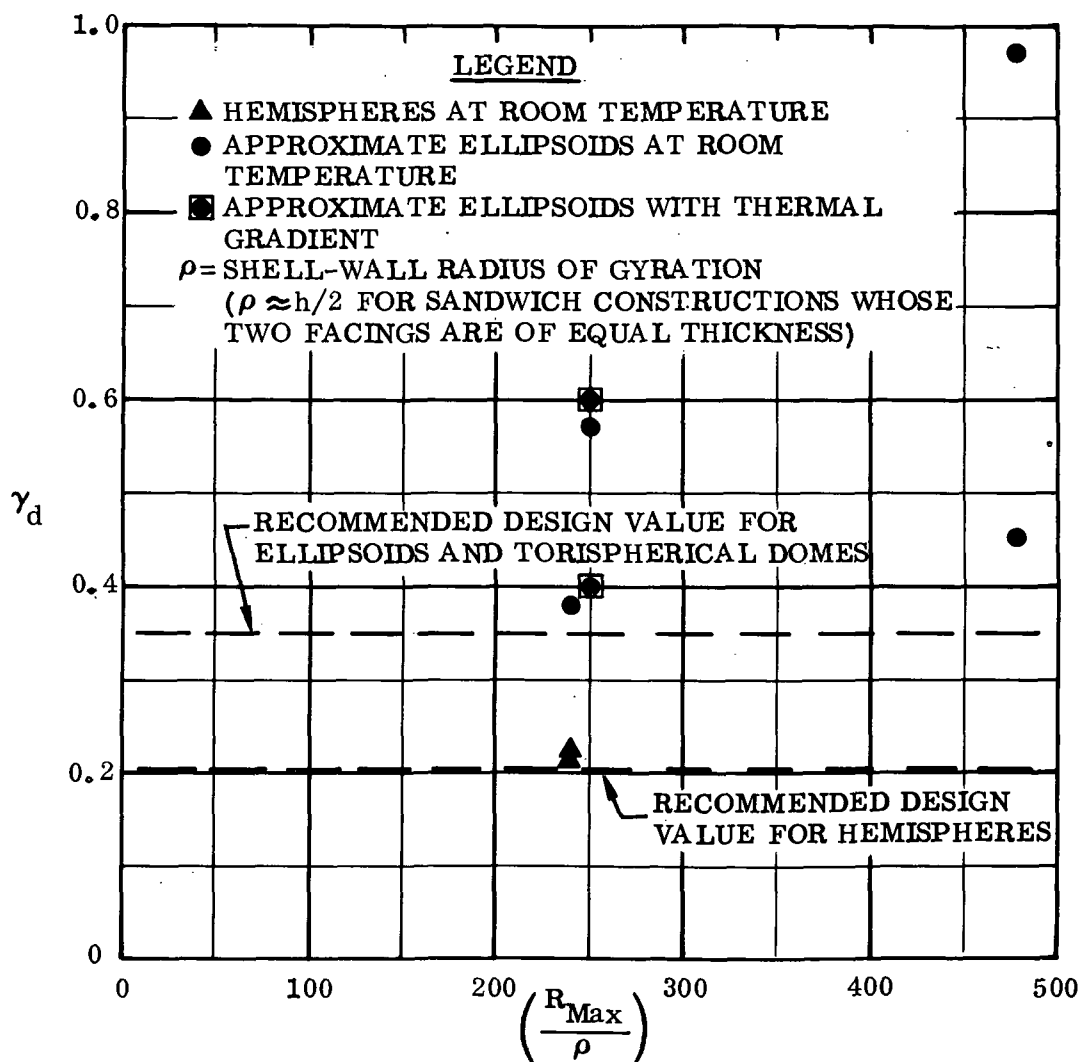


Figure 4-17: KNOCK-DOWN FACTOR γ_d FOR SANDWICH DOMES
 SUBJECTED TO UNIFORM EXTERNAL PRESSURE
 (REFERENCE 10)

A conservative probability factor of 0.99 was assumed in the trade study analyses in Section 3.4.1. However, since the nondestructive proof test was to demonstrate the ability to reinforce understrength shells, as well as to determine the level of conservatism in the probability factors, a lower probability factor was selected for the 8 ft. (2.44 m) shell.

The OPTRAN results showed a 0.345 in. (0.876 cm) core thickness for a 0.5 probability of not failing. The data for approximate ellipsoids at room temperature shown in Figure 4-17 are for a $R_{\max}/\rho = 250$ where ρ is one half the core thickness. In order to obtain a similar $R_{\max}/\rho$ value for the 8 ft. (2.44 m) shell a core thickness of $(64 \times 2/250)$ or approximately 0.500 in. (1.27 cm) is required. After considering the OPTRAN designs with the Boeing statistical analysis and the recommended design approach of Reference 10 with the test data shown in Figure 4-17, a core thickness of 0.500 in. (1.27 cm) was selected.

The selected design for the 8 ft. (2.44 m) diameter shell was:

Inner Face Skin - 2219-T62 aluminum alloy - 0.043 in. (0.092 cm)

Outer Face Skin - Style 120 Fiberglass/Epoxy prepreg, 2 plies material properties from Table 3-5, 0.010 in. (0.025 cm)

Core-5056 Aluminum Flex-Core - material properties from Table 3-7
0.500 in. (1.270 cm).

The OPTRAN predicted buckling strength was 54.5 psi (375.8 kN/m²). Based on test data shown in Figure 4-17, the knockdown factors used to predict shell strength were; (1) 0.6 for maximum external pressure and (2) 0.38 for minimum external pressure. The expected range of pressure using the OPTRAN results was:

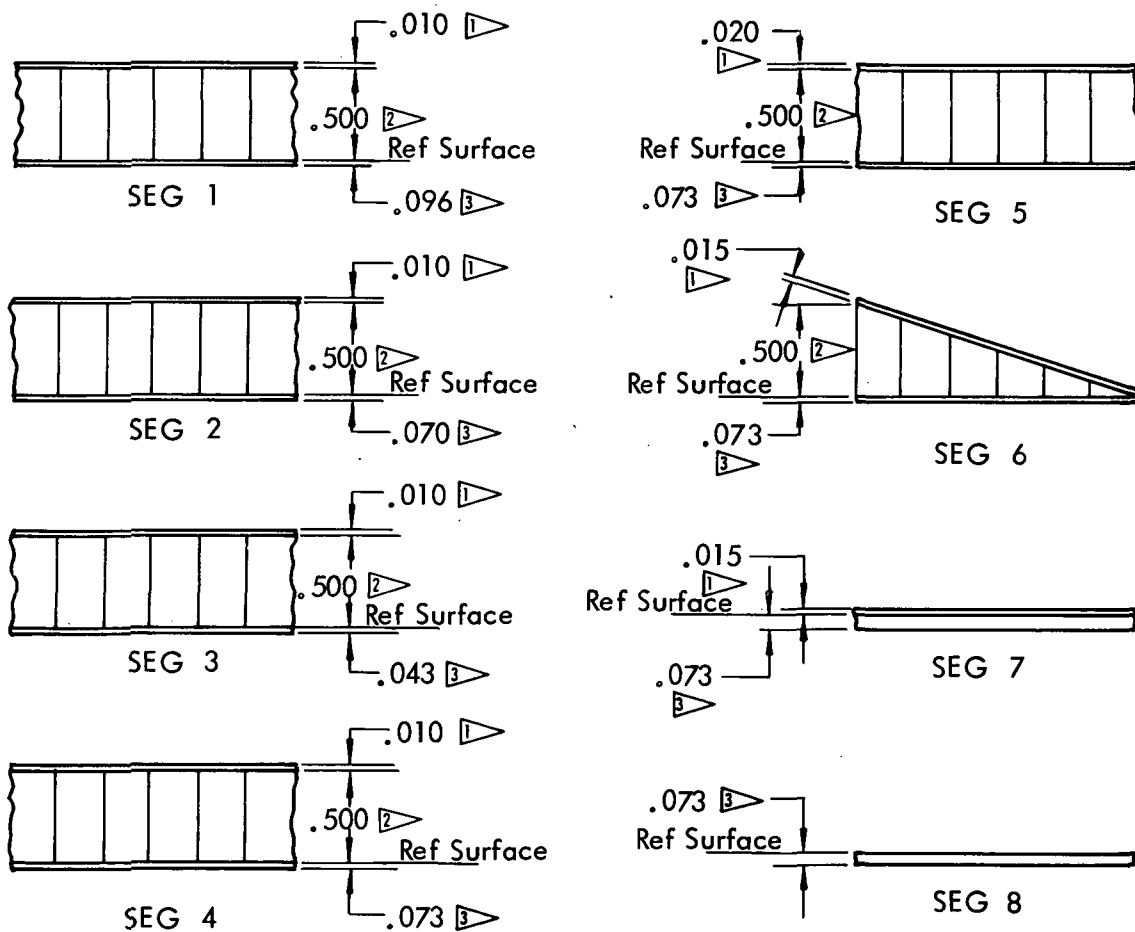
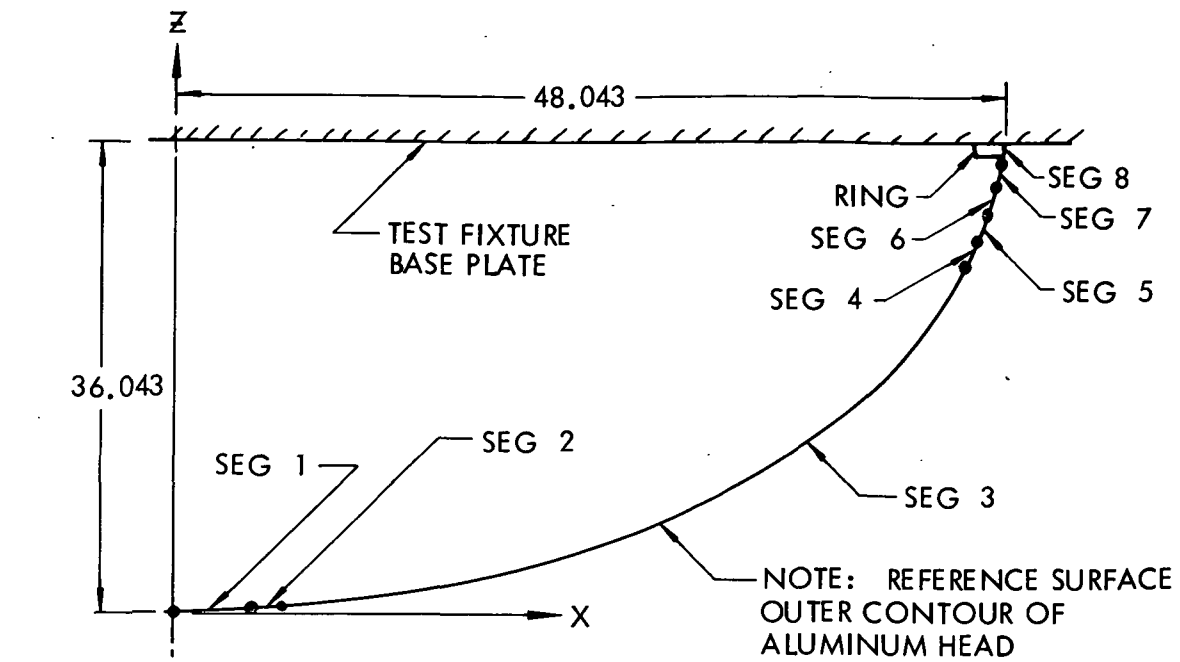
$$\text{Maximum external pressure} = 54.5 \times 0.6 = 32.7 \text{ psi (275.5 kN/m}^2\text{)}$$

$$\text{Minimum external pressure} = 54.5 \times 0.38 = 20.7 \text{ psi (142.7 kN/m}^2\text{)}$$

BOSOR 3 Analysis

The OPTRAN analysis assumed simply supported edge conditions for a hemisphere of 64 in. (1.63 m) radius. However, there is some edge moment on the test shell provided by the bending stiffness of the 0.073 in. (0.185 cm) thick edge and the fiberglass reinforcement. The BOSOR 3 analysis accounted for this local bending stiffness. More significant to the buckling analysis is the geometrical shape of the ellipsoidal head. This is also accounted for in the BOSOR 3 analysis.

The BOSOR 3 structural analysis model is shown in Figure 4-18. Eight segments were used to describe the different portions of the sandwich shell. Each segment



1 ▸ Style 120 Fiberglass/epoxy Prepreg 2 ▸ 5056 Aluminum Flex-Core, 2.1 lb/FT³
 3 ▸ 2219-T62 Aluminum Alloy

Figure 4-18: MERIDIAN REFERENCE SURFACE, SEGMENT LOCATION AND SEGMENT GEOMETRIES

accounts for a change in face gage or core thickness. The segment thickness coefficients were calculated using the SALC computer program (Reference 13) and the room temperature properties of the aluminum and fiberglass skins.

The BOSOR 3 predicted buckling strength was 77.7 psi (535.7 kN/m²) external pressure. This is about one and one-half times the OPTRAN prediction of 54.5 psi (375.8 kN/m²). The difference is due to the ellipsoid geometry and edge fixity included in the BOSOR 3 analysis. Applying the same knockdown factors as used in the OPTRAN analysis, the expected range of pressures from BOSOR 3 were:

$$\text{Maximum external pressure} = 77.7 \times 0.6 = 46.5 \text{ psi (320.6 kN/m}^2\text{)}$$

$$\text{Minimum external pressure} = 77.7 \times 0.38 = 29.6 \text{ psi (204.1 kN/m}^2\text{)}$$

The critical mode shape is plotted in Figure 4-19. " W_o " is the radial displacement. A negative sign denotes inward displacement. The arc length is measured from the apex of the dome. The BOSOR 3 analysis indicates that this mode is axisymmetric. That is, there are no circumferential waves. The deflection shape plotted in Figure 4-19 is expected during the test but the W_o values will be scaled down to correspond to the critical test load.

The BOSOR 3 analysis was also used to determine the placement of the electronic deflection indicators (EDI's) and the strain gages on the test head.

The Electronic Deflection Indicator (EDI) locations, shown in Figure 4-20, were selected using the data in Figure 4-19. EDI-E1 measured the displacement of the apex at arc length 0. EDI's E2-E13 monitored the W_o 's at arc lengths 7.5, 10.0, 12.5 and 13.0 at three locations around the shell 120° apart. EDI's E14-E19 were placed at arc lengths 60 and 62.5 where positive outward displacements are shown in Figure 4-19. Strain gages were included at some of these locations to measure membrane strains and bending strains associated with these displacements.

Fabrication

The use of an existing aluminum shell for the inner face skin simplified the fabrication of the 8 ft (2.44 m) diameter head. This shell served as a mold to cut the HT 424 adhesive and the Flex-Core to the desired gore shape. The fiberglass caul plates which were used to produce a good surface finish and contour on the fiberglass/epoxy outer face skin were laminated on the aluminum shell prior to the assembly of the Flex-Core to the shell.

The first assembly operation was to cut and stabilize the tapered aluminum Flex-Core edge pieces with the BMS 5-25 potting compound. The HT 424 adhesive was cut to shape and laid up on the 2219-T62 aluminum inner shell. The Flex-Core was cut to shape and along with the tapered edge pieces was positioned on

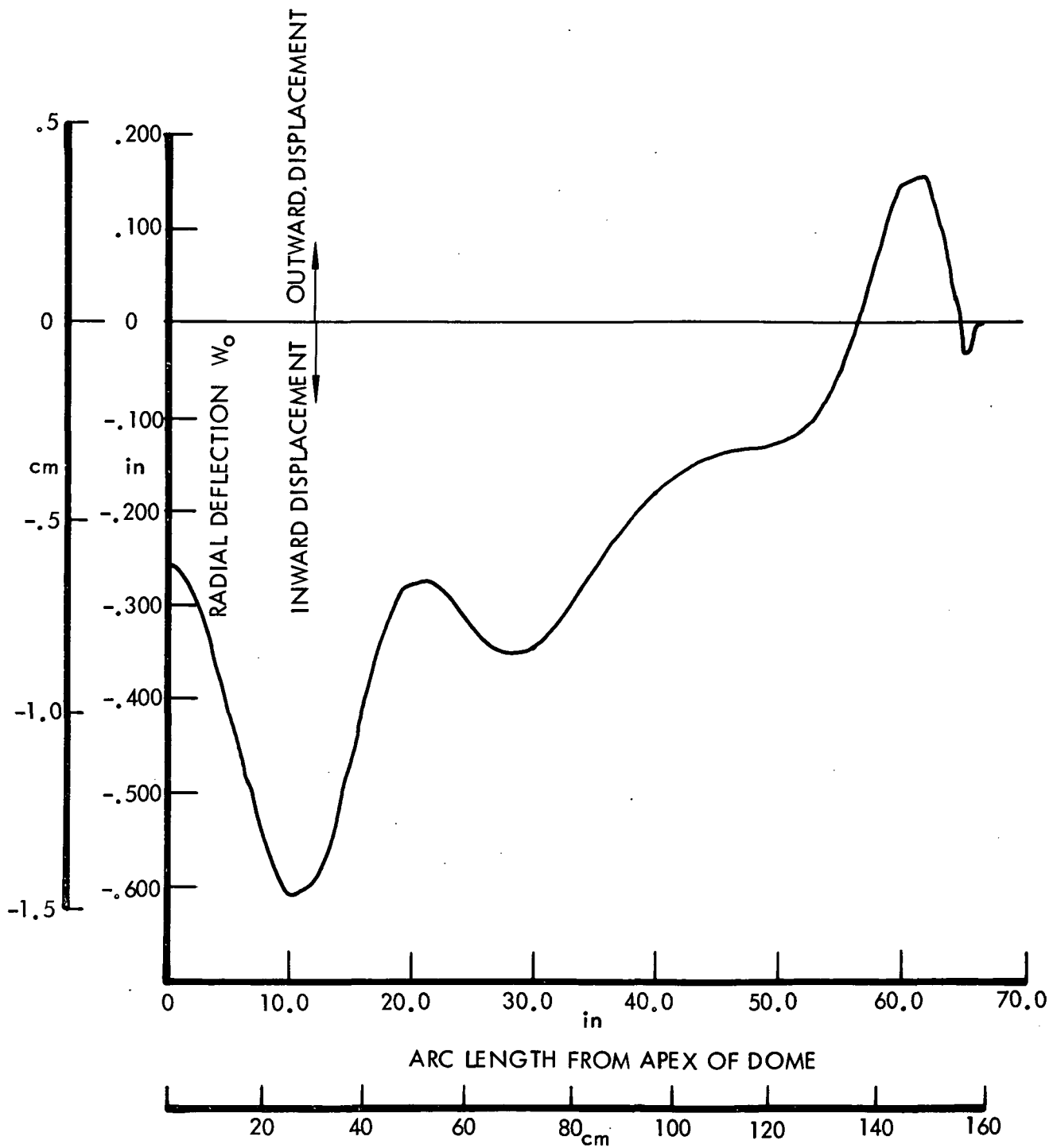


Figure 4-19: DEFLECTION MODE SHAPE FOR 8 FT (2.44 m) DIAMETER HEAD WITH 77.7 PSI (535.7 kN/m²) EXTERNAL PRESSURE

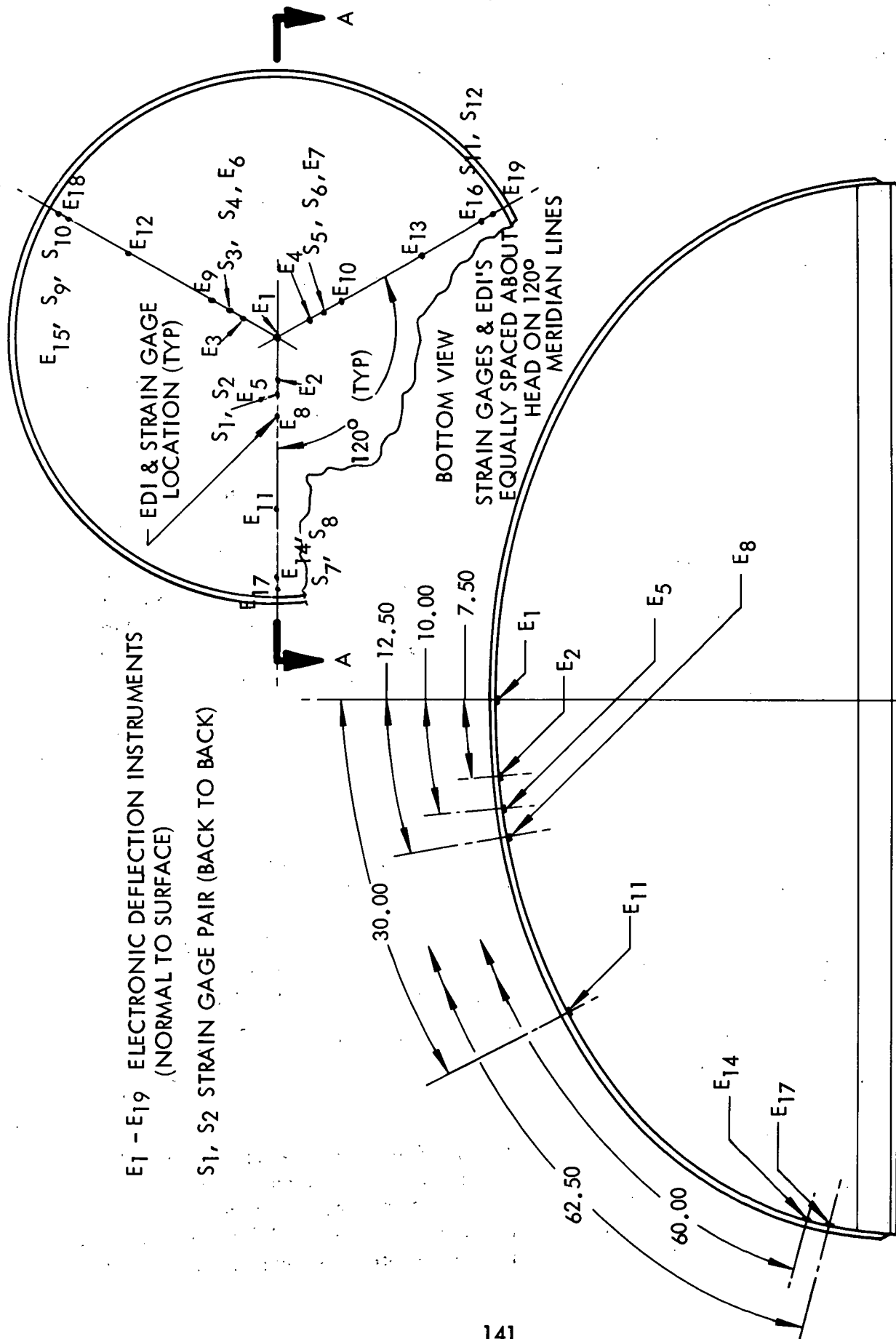


Figure 4-20: 8 FT (2.44 m) DIA HEAD INSTRUMENTATION

the adhesive. BMS 5-90 structural foaming adhesive was placed in the core joints. This assembly is shown in Figure 4-21a after curing in the autoclave.

Figure 4-21b shows the AF 131 adhesive being laid up on the Flex-Core. The outer skin fiberglass prepreg was trimmed and laid up on the adhesive. The caul plates were placed over the outer skin and this assembly was vacuum bagged and cured in the autoclave.

Inspection of the completed assembly showed a marked depression near the apex. Figure 4-22 describes this depression. It was suspected that the depression developed during curing from local oil-canning of the aluminum skin. It was decided to test this head without reworking this depressed area. An additional EDI (E 20) was located at the outer edge of the depression to monitor the shell's behavior in this critical location. Figure 4-23 shows the head assembly instrumented and mounted on the test fixture cover plate.

The measured weights of the adhesives used in bonding the inner and outer face skins to the core indicate that the 6.0×10^{-4} lb/in² (0.42 kg/m²) used in the Section 3.4.1 trade studies is a reasonable weight objective. The average weight of the HT 424 adhesive used to bond the inner aluminum skin to the Flex-Core was 8.55×10^{-4} lb/in² (0.62 kg/m²). This is a heavier adhesive than the AF 131 used for bonding the fiberglass/epoxy prepreg outer face skin to the Flex-Core. The average weight of the AF 131 adhesive used was 5.33×10^{-4} lb/in² (0.37 kg/m²). These weights are in close agreement with the nominal weights quoted in the material specifications.

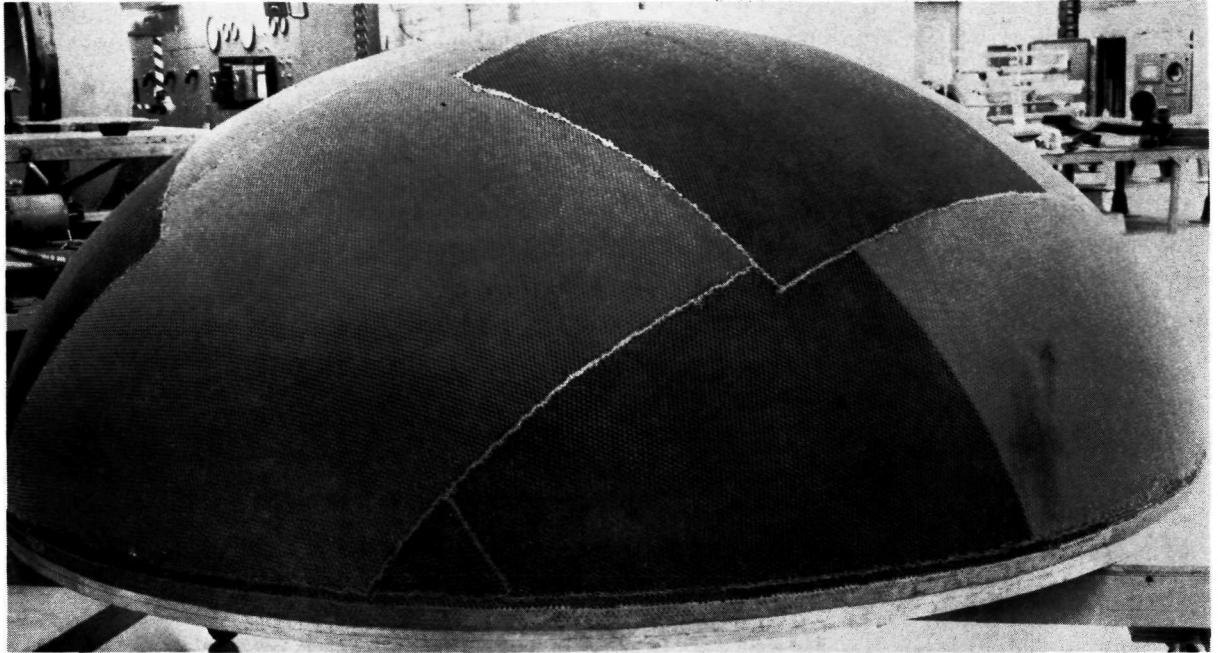
4.3.3 Tests

Test Setup

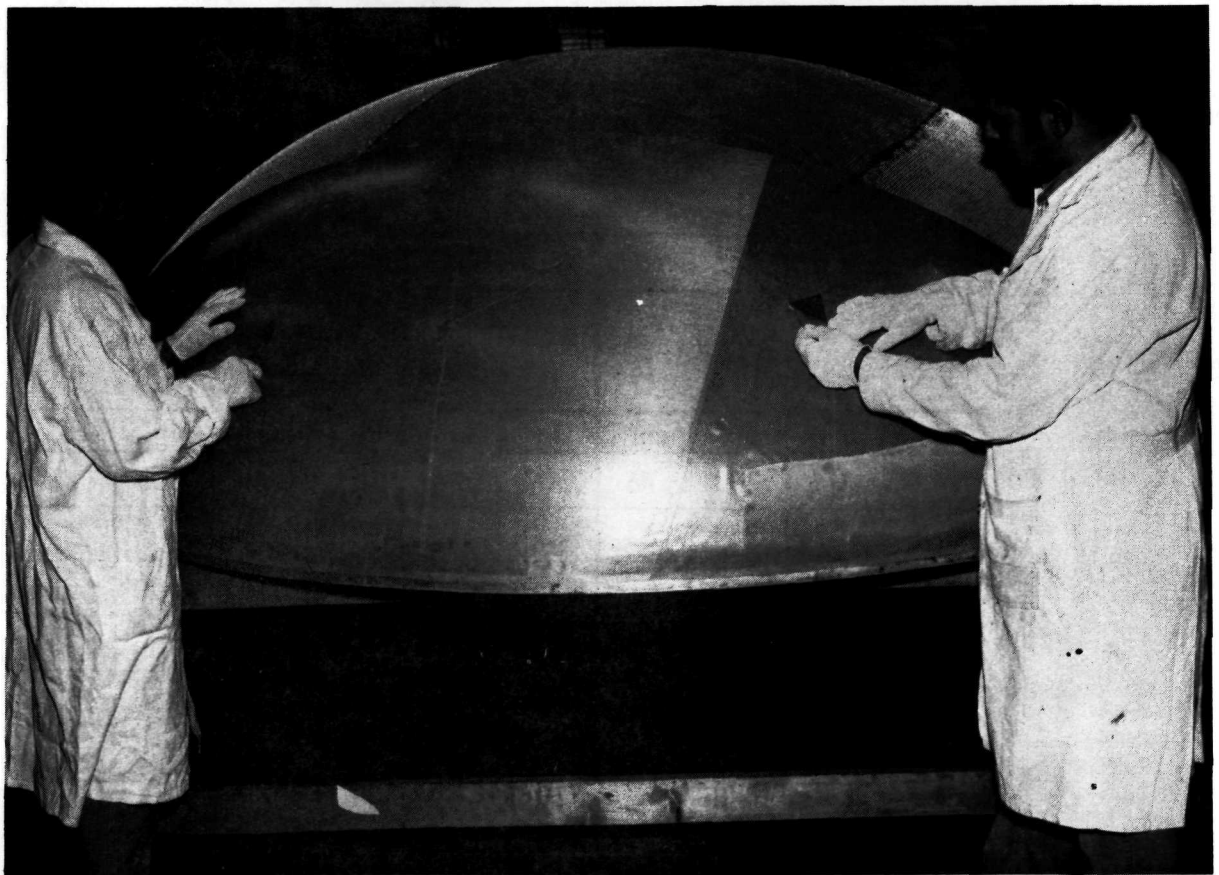
The test setup assembly is shown in Figure 4-24. The pressurant inlet was at the water tank apex and venting was through the cover plate.

The 8 ft. (2.44 m) ellipsoidal sandwich head was mounted to a 2 in. (5 cm) thick steel cover plate (Figure 4-24). A steel clamping ring around the exterior edge of the bonded head was used to provide reinforcement to the bonded joint. A bead of RTV 102 sealant was used along the exterior of the head-to-cover plate joint. The external surface (fiberglass/epoxy face skin) was waterproofed with three coats of a chem-mill maskant. An existing kirk site form die was used as the water pressure tank.

Figure 4-25 shows the overall test arrangement. Water pressure was controlled through a small capacity pump located in the pressure control cart, the bypass valve, and the vent shutoff valve. The hydrostatic pressure was measured at the top of the test fixture. There is an additional 1.7 psi (11.7 kN/m²), at the center of the head due to the difference in elevation between the gage and

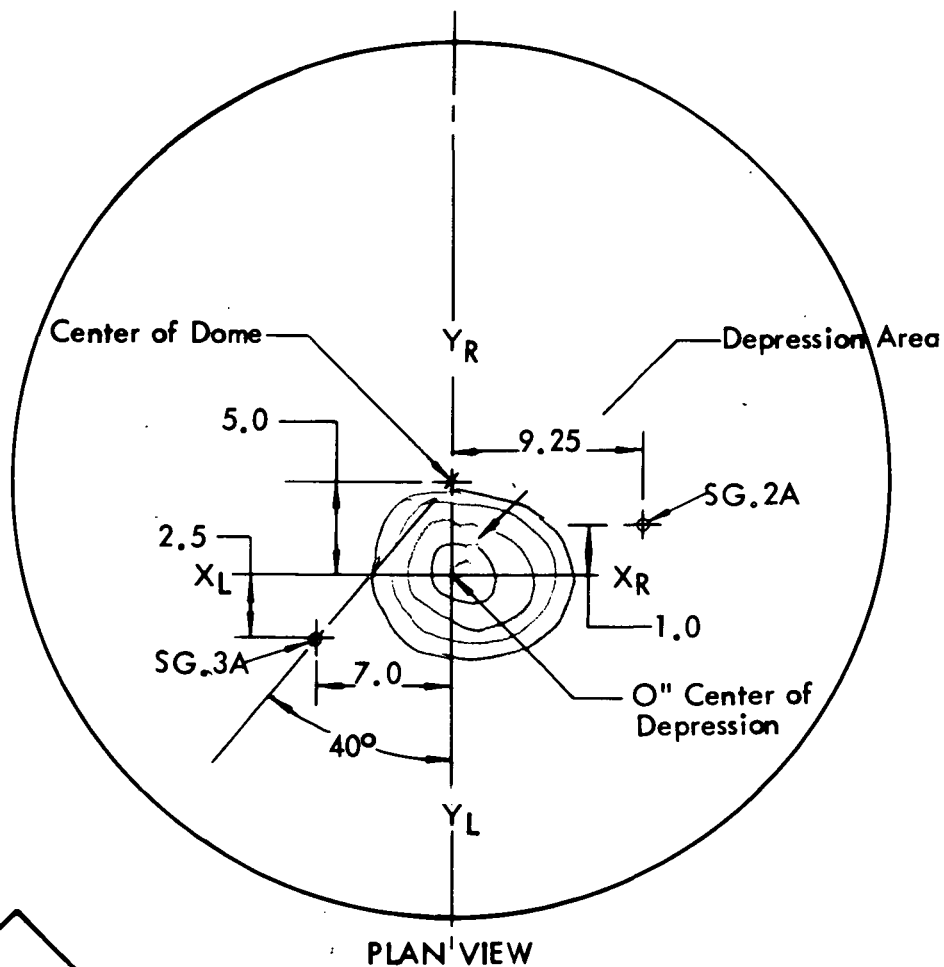


a: Inner skin and core assembly



b: Applying AF 131 adhesive to core

Figure 4-21: FABRICATION OF 8 FT (2.44 m) DIAMETER HEAD



IN DIRECTION OF DISTANCE FROM CENTER OF DEPRESSION		DEVIATION FROM CONTOUR							
		X _L		X _R		Y _L		Y _R	
in	cm	in	cm	in	cm	in	cm	in	cm
0	0	.300	.76	.300	.76	.300	.76	.300	.76
1	2.54	.250	.63	.250	.63	.250	.63	.225	.57
2	5.08	.200	.51	.200	.51	.200	.51	.200	.51
3	7.62	.150	.38	.125	.32	.125	.32	.150	.38
4	10.16	.050	.13	.050	.13	.050	.13	.050	.13
5	12.7	.025	.064	.000	-0	.000	-0	.000	-0

Figure 4-22: DEPRESSION CONTOUR MEASUREMENT

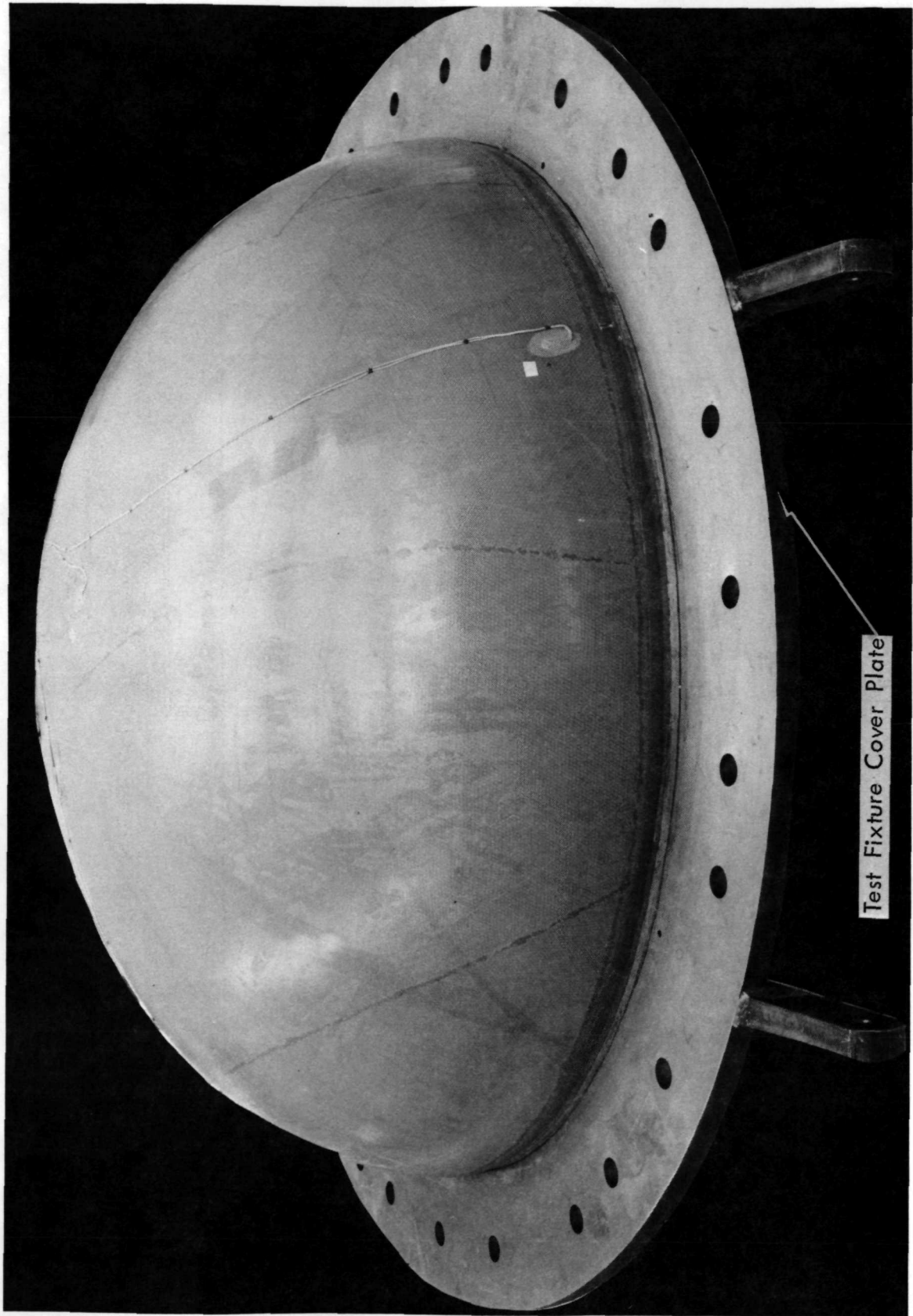


Figure 4-23: "AS MANUFACTURED" 8 FT (2.44 m) DIAMETER HEAD

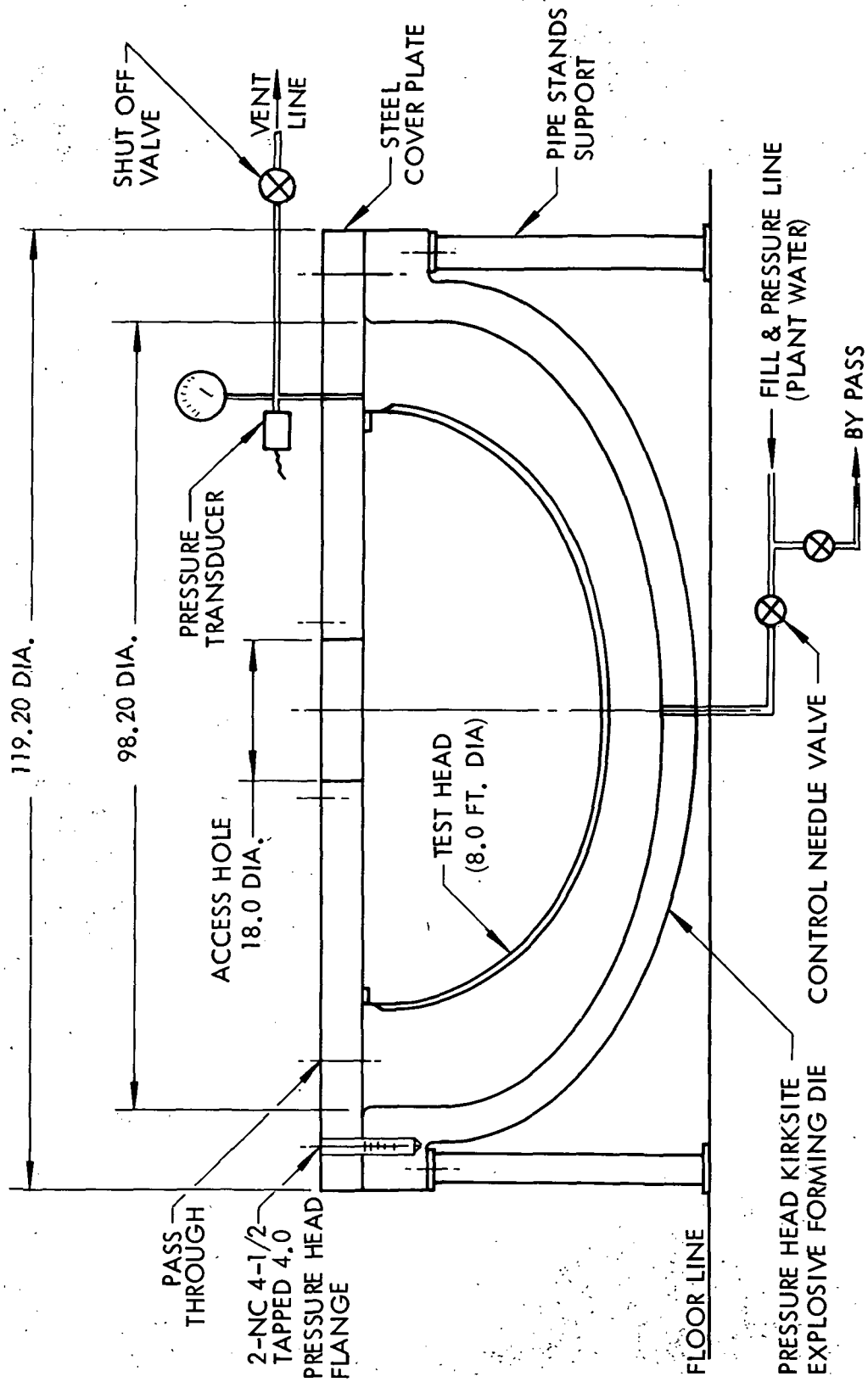


Figure 4-24: NONDESTRUCTIVE SHELL BUCKLING TEST SETUP

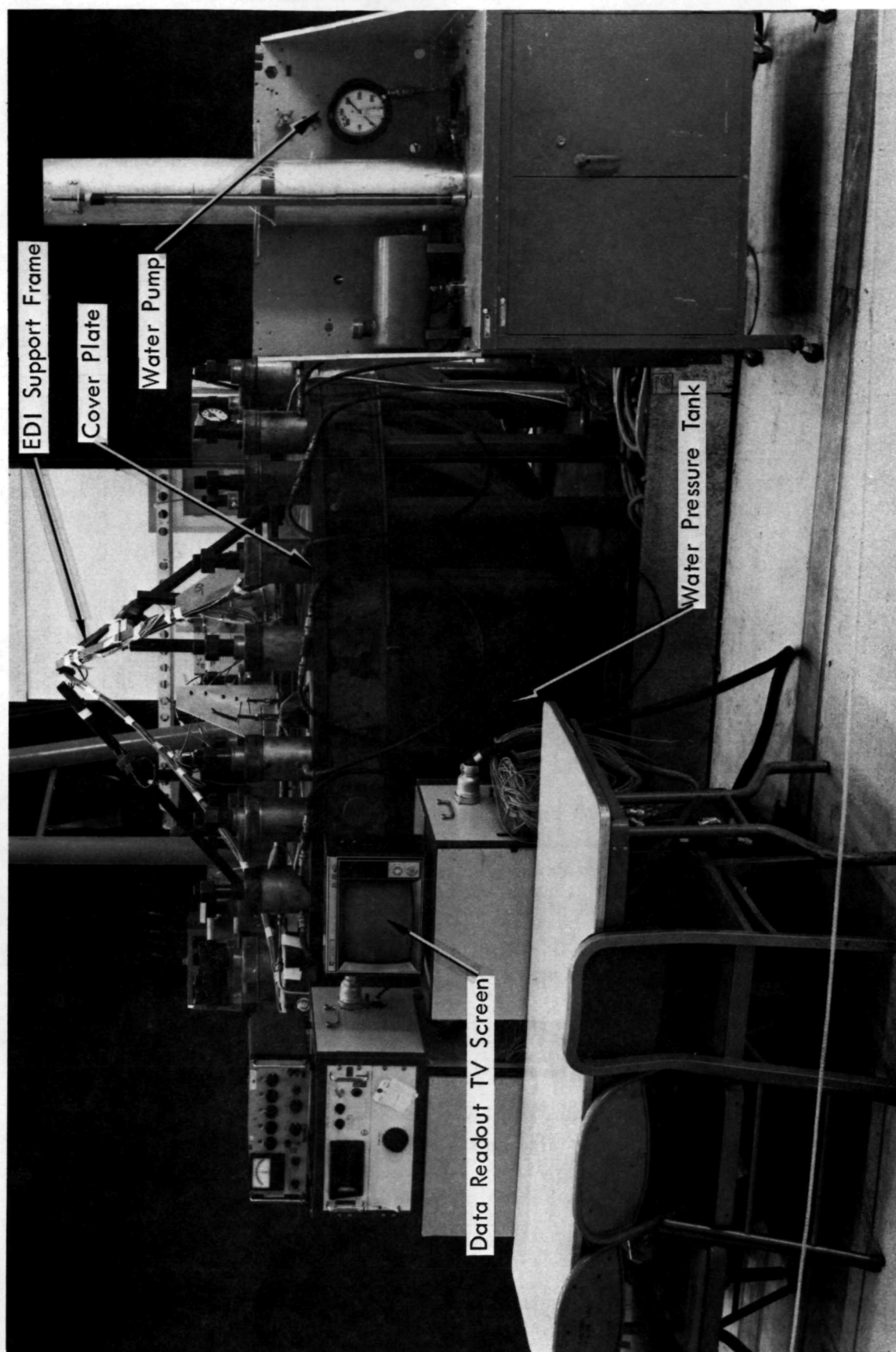


Figure 4-25: EXTERNAL PRESSURE TEST SETUP

the center of the head. Instrumentation data were automatically recorded on the SDS 910 data acquisition system and displayed for test monitoring and hand plotting on the data readout TV screen. During these tests, the procedure was to lock the screen at each pressure increment and shut off the pump until the data were manually recorded and the necessary plots made.

Buckling Prediction Test on "As Manufactured" Head

Preliminary Test

A preliminary test was conducted to check out instrumentation and testing procedures.

The test was stopped at 13.9 psi (95.5 kN/m^2) and the instrumentation adjusted. The test was rerun to a pressure of 14.8 psi (102 kN/m^2) when the shell buckled locally at the depression near the apex.

Buckling Prediction Test

The F/S plots predicted the failure load. However, an accurate interpretation of this data was not made during the test. As a result, the head buckled at a hydrostatic pressure of 14.8 psi (102 kN/m^2) with the buckle located around the local depression described earlier.

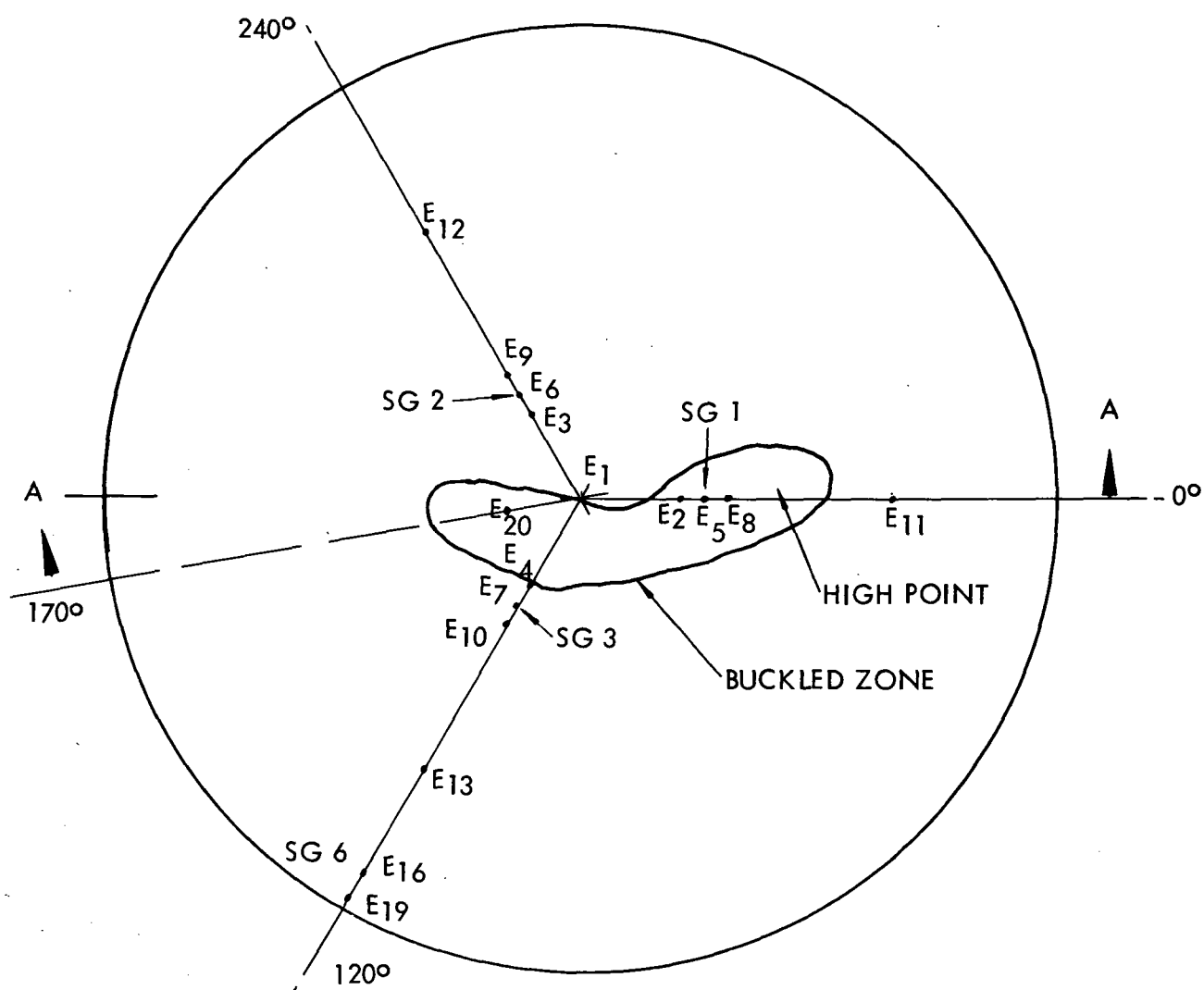
Failure Mode

Figure 4-26 shows the buckled zone as seen from the interior of the head. E1 is the center of the head. The center of the depression was located at E20. Figures 4-27a & b are photographs of the exterior surface of the buckled head. The solid line is the outline of the original depression. The dashed line traces the approximate shape of the buckled zone. The buckle was larger than the area shown; however, some of it popped back into shape when the water pressure was removed. The location of the three strain gages, SG1, SG2 and SG3 are noted as S1, S2 and S3.

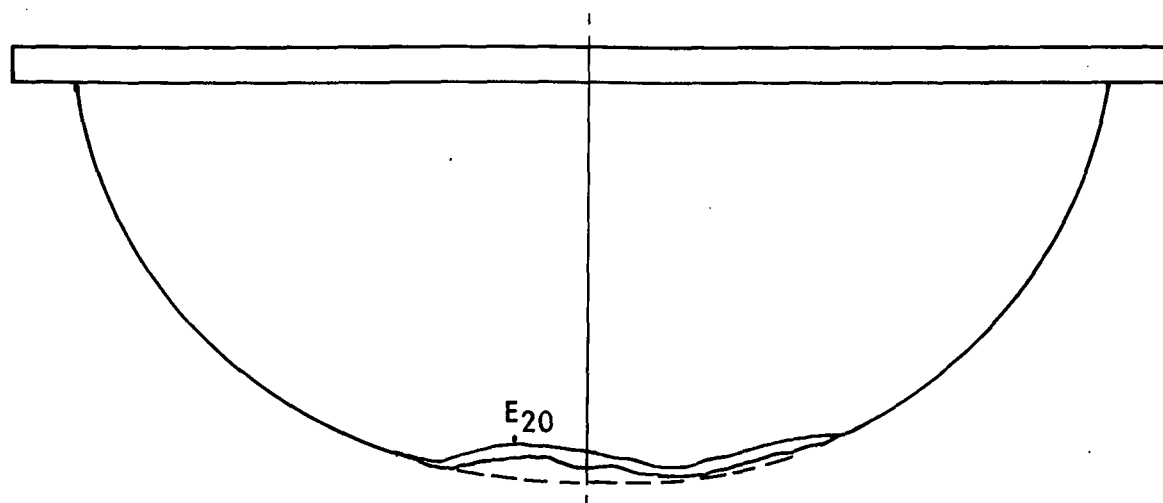
Figure 4-28 is a partial section to scale through the center of the depression at meridian 120° . EDI location for E1, E20, E4 and E10 are shown. This imperfection is quite serious and could not be permitted in a production shell. It was allowed to remain in the shell, to measure its effect on shell strength and to provide a basis for evaluating the F/S method for buckling predictions. The deflected shape of the inner surface at 14 psi (96.5 kN/m^2) external pressure is also shown.

F/S Plot Data

The F/S data was monitored and plotted continuously during the test. Pressure/deflection and pressure/ Δ strain (difference between inner and outer skin strains)

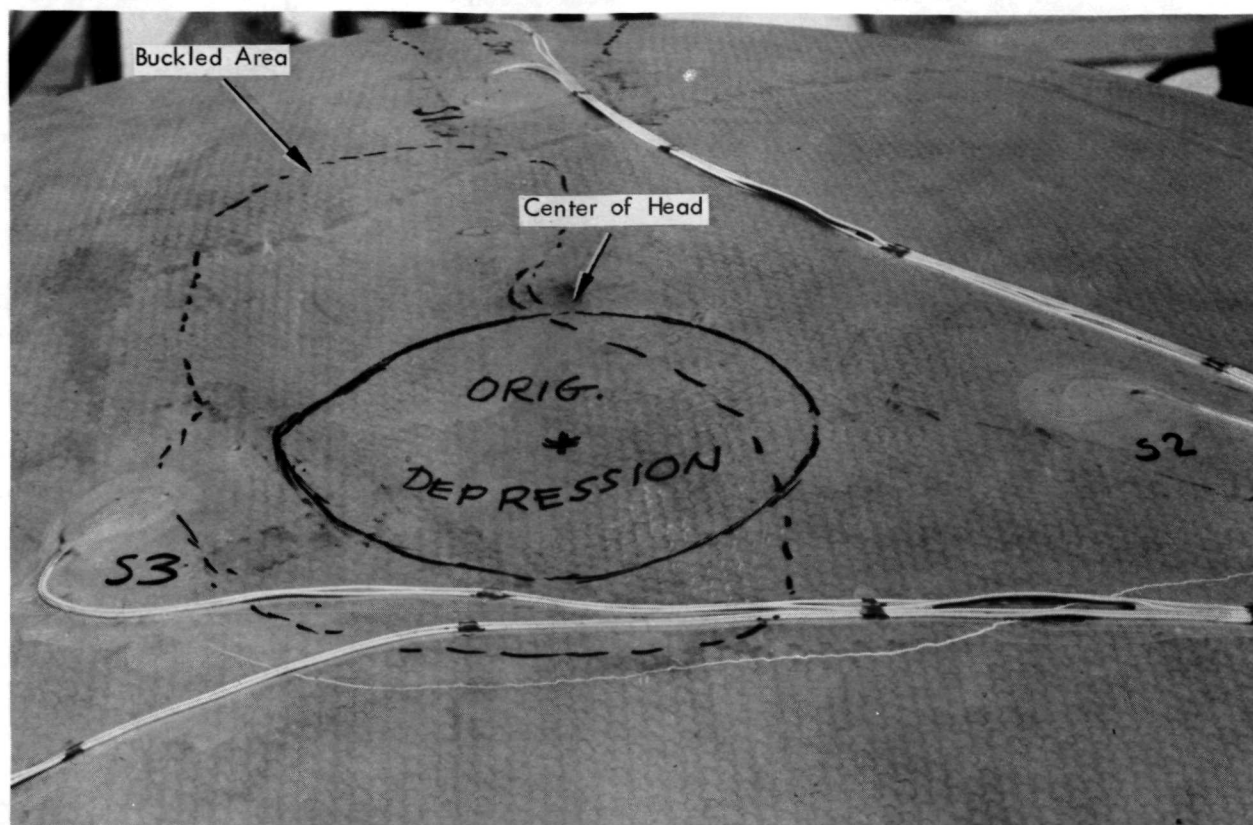


PLAN VIEW



SECTION A A

Figure 4-26: BUCKLED ZONE VIEWED FROM THE SHELL INTERIOR



a: Strain gage location in buckled area



b: Buckled area with maskant removed

Figure 4-27: "AS MANUFACTURED" 8 FT (2.44 m) DIAMETER HEAD AFTER TEST

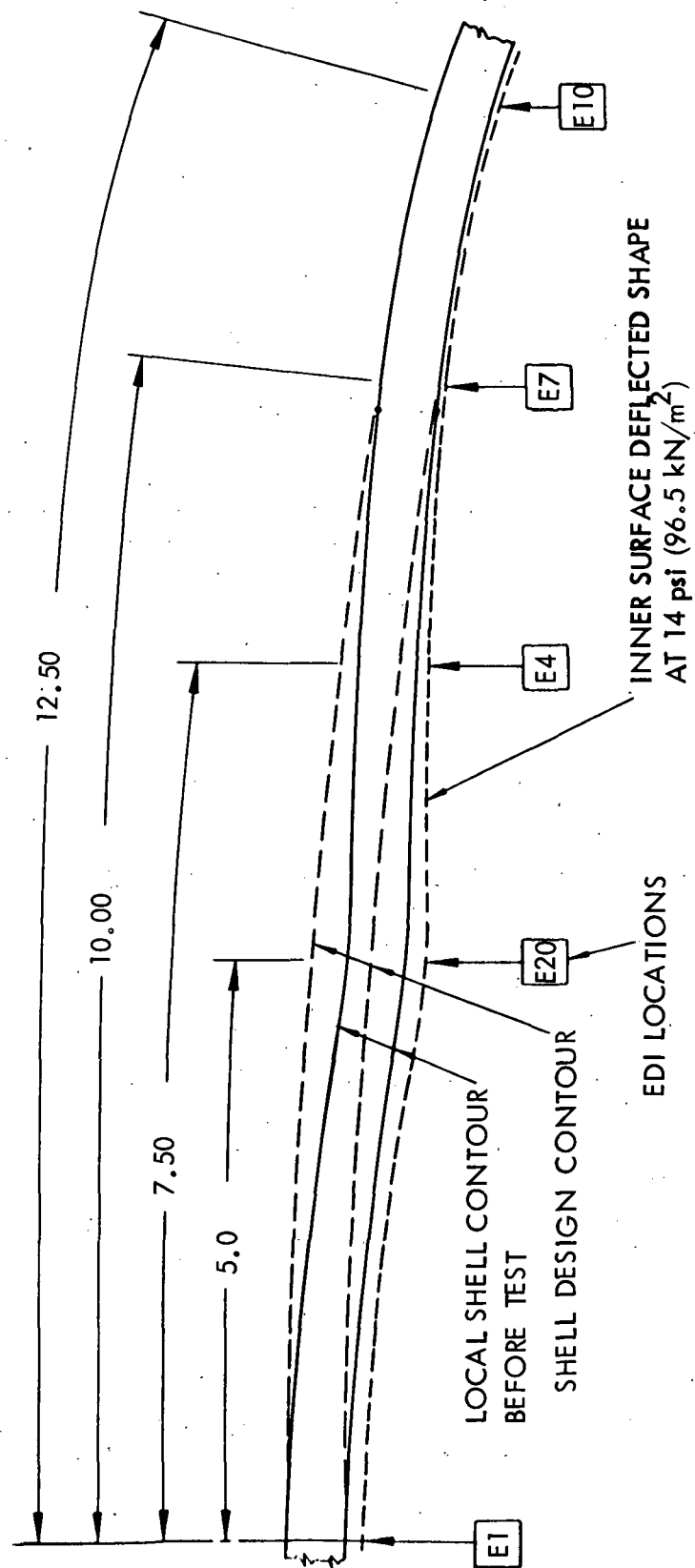


Figure 4-28: PARTIAL SECTION AT MERIDIAN 120° THRU THE DEPRESSED AREA

were plotted versus pressure. Some of these plots are shown in Figure 4-29. E2 and E20 show the F/S data from the EDI's located adjacent to the local buckle. It is apparent that the EDI's were not sensitive enough to detect the buckling mode. Their range and sensitivity were selected for a general instability failure at 77 psi (535 kN/m^2) with prebuckle deformations of 0.6 in. (1.52 cm). The local failure occurred at approximately a quarter of these values.

Figures 4-30 and 4-31 are the F/S data for the strain gages. Strain gages SG1 and SG6 predicted failures at 13.9 and 15.4 psi (96.0 and 106 kN/m^2) pressure. The F/S plot for SG1 is the critical plot since it was physically closest to the buckle. Referring to Figure 4-31, the value of F/S was determined at a pressure of 10.1 psi (69.6 kN/m^2) and 11.7 psi (80.7 kN/m^2), giving an extrapolated buckling pressure of 13.9 psi (96 kN/m^2) (the linear extension of the F/S plot to the abscissa). Since the preliminary test had loaded the shell to 13.9 psi (96 kN/m^2) without incident, it was decided to further increase the pressure to 12.8 psi (88.4 kN/m^2). The new linear extrapolation (using the last two data points) indicated a buckling pressure of more than 20 psi (138 kN/m^2). A further increase in pressure to 13.8 psi (95.2 kN/m^2) resulted in a predicted buckling pressure of 16.7 psi (115 kN/m^2). Finally, a further increase in pressure resulted in failure at 14.8 psi (102 kN/m^2).

The false prediction of 20.6 psi (142 kN/m^2) close to the actual buckling load is a typical behavior of the F/S prediction method. F/S test experience on Contract NAS 1-10749 "The Design and Testing of Advanced Structural Panels" exhibited this same behavior. However, the more linear portion of the F/S plot at lower load has been predicting failure loads quite accurately on the structural panels. If the prediction of 13.9 psi (95.9 kN/m^2) from the load region had been accepted, the error in the prediction would have been -6.1%, which is comparable to results in the panel program.

It is interesting to note that SG 6 was predicting a buckle pressure of 15.4 psi (106 kN/m^2). In fact, the F/S plot for SG 6 in Figure 4-30 twice showed a linear extrapolation to the same critical pressure. However, there were no signs of local buckling at SG 6. After the test some of the fiberglass and core were removed to examine the interior of the sandwich at SG 6. No damage was observed. Non-uniform loading near the center of the head may have contributed to a possible local buckling mode near SG 6 and the local collapse of the depression at 14.8 psi (102 kN/m^2) unloaded the area around SG 6 before the critical load was reached.

Deflection Data

In general, the EDI's showed the largest deflection at the center of the shell and smaller deflections at the edges. The maximum deflection occurred under the depressed area at E20.

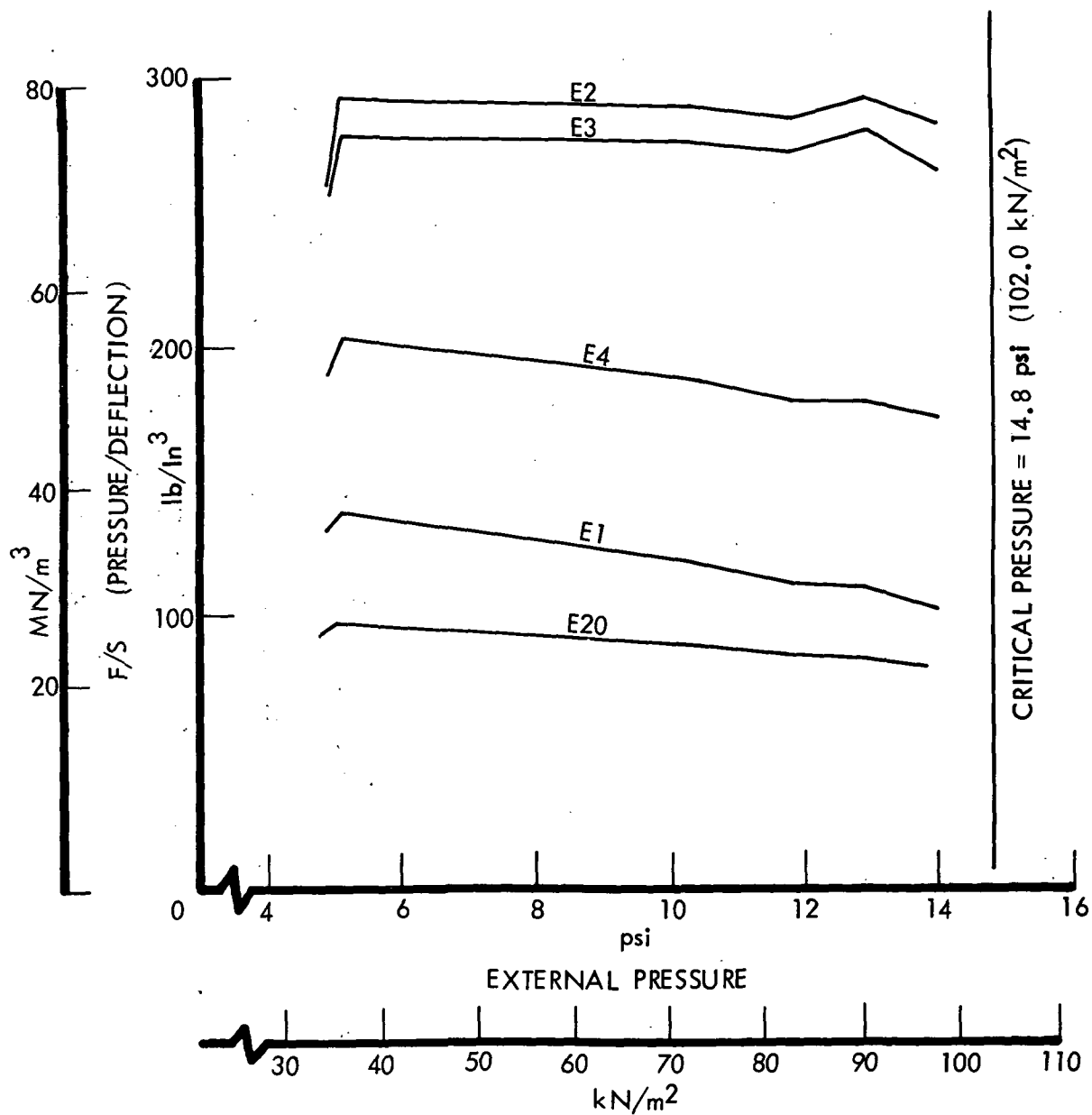


Figure 4-29: F/S PLOTS FOR EDI'S E1 - E4 & E20

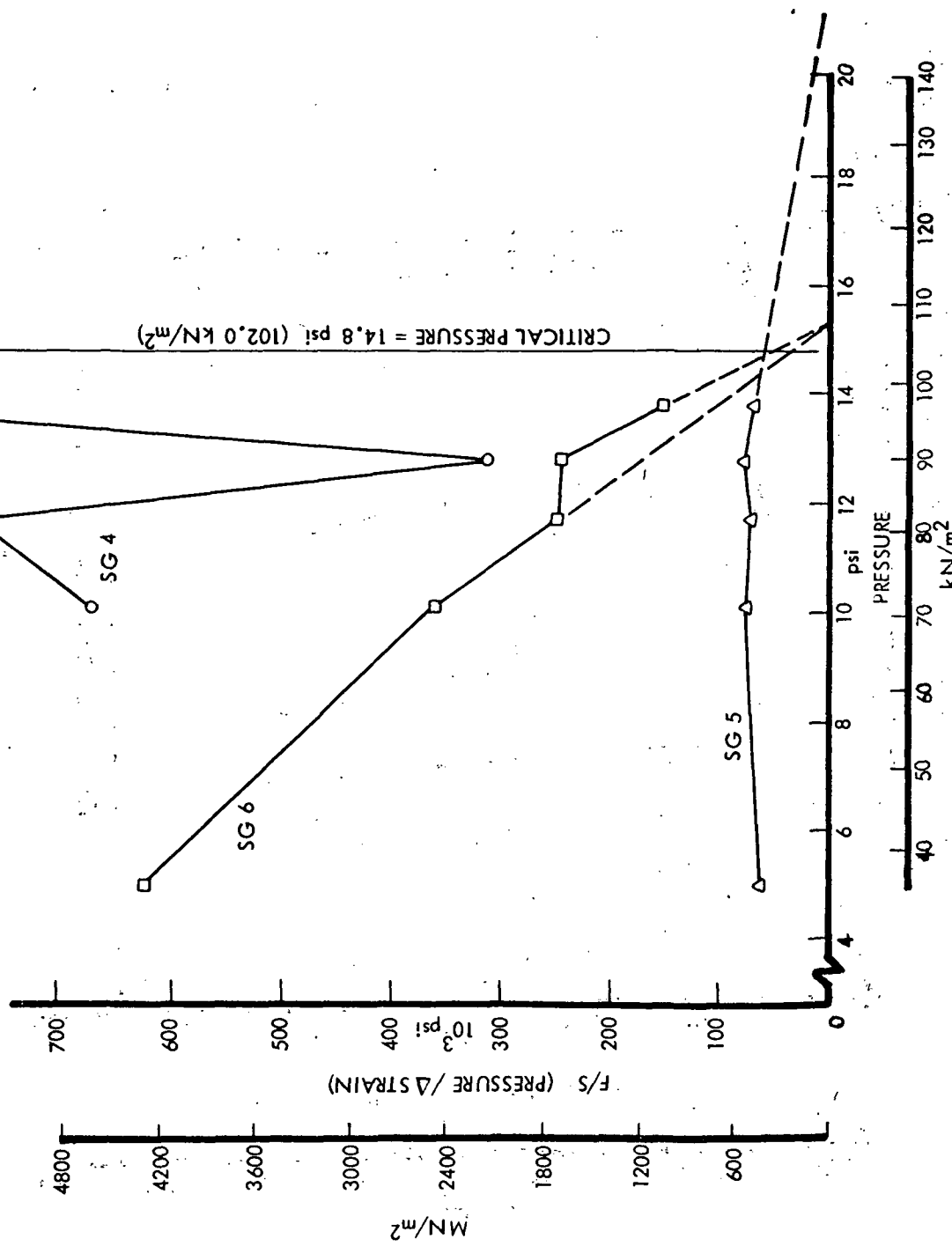


Figure 4-30: F/S PLOT FOR STRAIN GAGE SETS 4 - 6

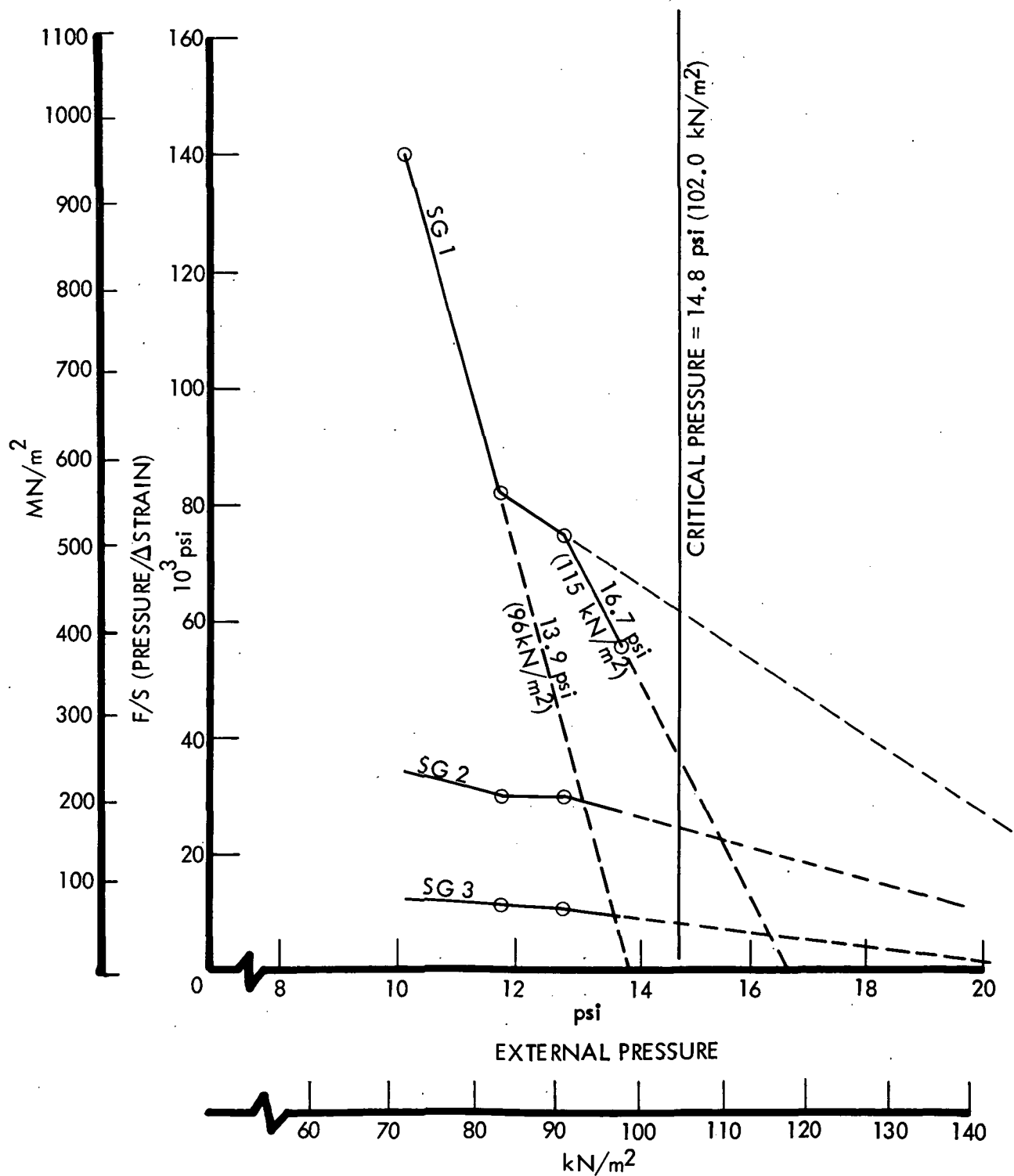


Figure 4-31: F/S PLOT FOR STRAIN GAGE SETS 1 - 3

Figures 4-32 through 4-35 are plots of external pressure versus deflection for 19 EDI's. EDI E18 did not operate during the test. The EDI plots show the relative effect of the depressed area on the radial deflection of the shell. For example, E4 has a greater deflection than either E2 or E3 at the 7.5 in. (19.2 cm) station. Further down the shell at 10 in. (25.4 cm), E7 has a smaller deviation from E6 and E5. By mid-shell height (E11, E12 and E13) all the deflections are about equal. At the edges of the shell, the total deflections are much smaller and the deviations are not as significant. Comparing the deflections for 14 psi (96.5 kN/m^2) at EDI's E1, E20, E3 and E4, it can be seen that the center of the depressed area deflects more than the surrounding shell. Thus, the depression did grow inward with increasing pressure and probably precipitated the buckle.

Strain Data

Figures 4-36 through 4-38 are plots of the strain gage data for the external pressure. Gages SG1, SG2 and SG3 are near the center of the head. SG4, SG5 and SG6 are near the edge of the head. The A gages are mounted on the exterior surface (fiberglass); the B gages are mounted on the interior surface (aluminum). Except for SG3A all the gages recorded compression strains. The SG3 gages were located near the edge of the depressed area. As the pressure increased, the local bending due to the depression caused tension in the outer fiberglass skin. The tension stress cracked the outer skin when the buckle occurred. This crack can be seen in Figure 4-27 running from S3 to the right of S2.

The shapes of the SG1A and SG2A are caused by the unsymmetrical elastic properties of the sandwich construction. Figure 4-39 shows the location of the neutral axis on a section of the shell wall.

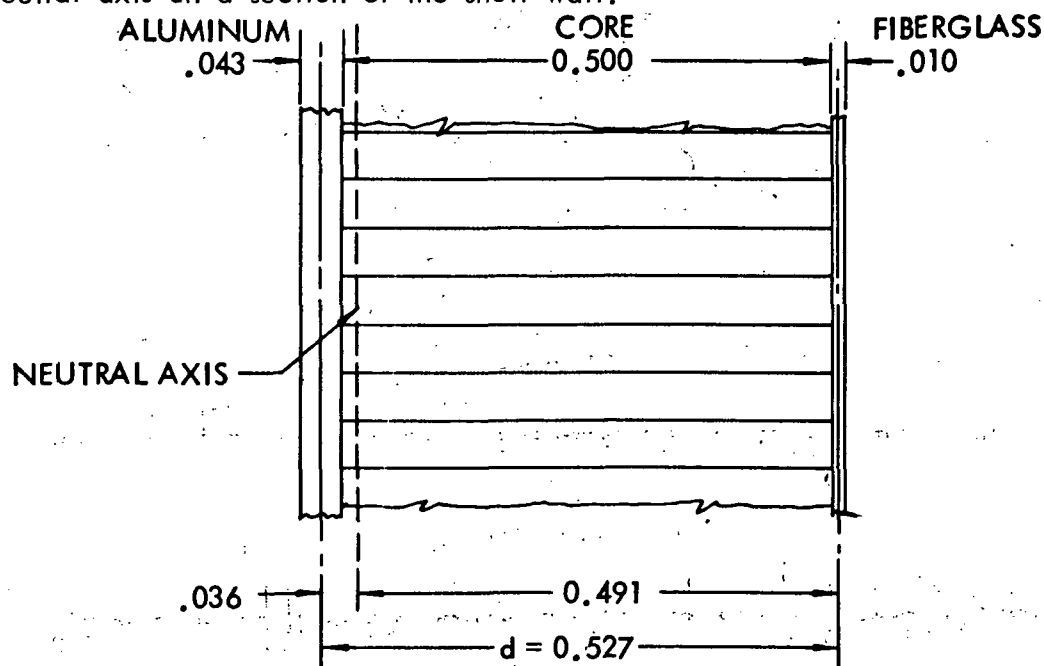


Figure 4-39: NEUTRAL AXIS LOCATION FOR 8-FT (2.44m) DIA. HEAD

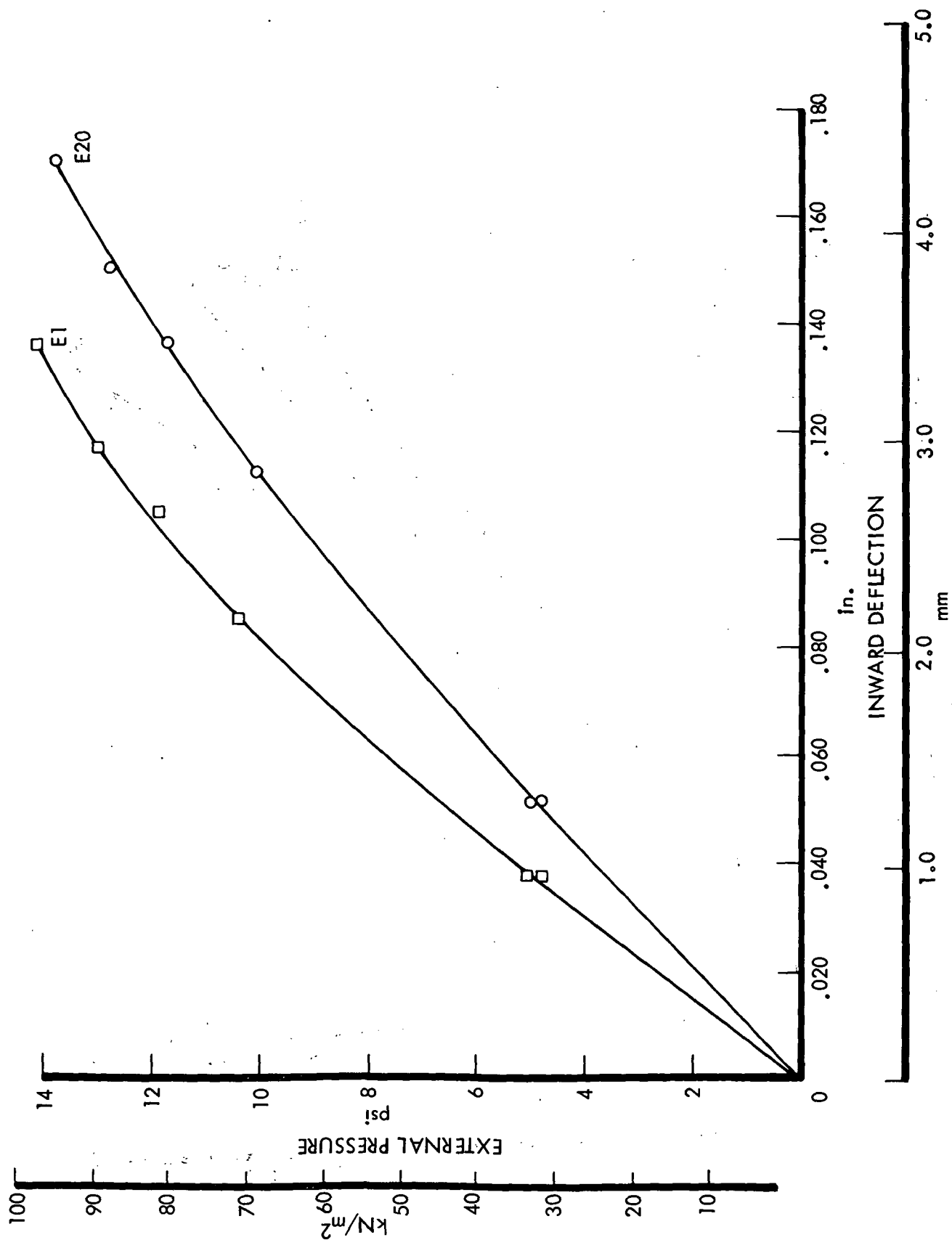


Figure 4-32: PRESSURE VS. DEFLECTION, E-1 & E-20

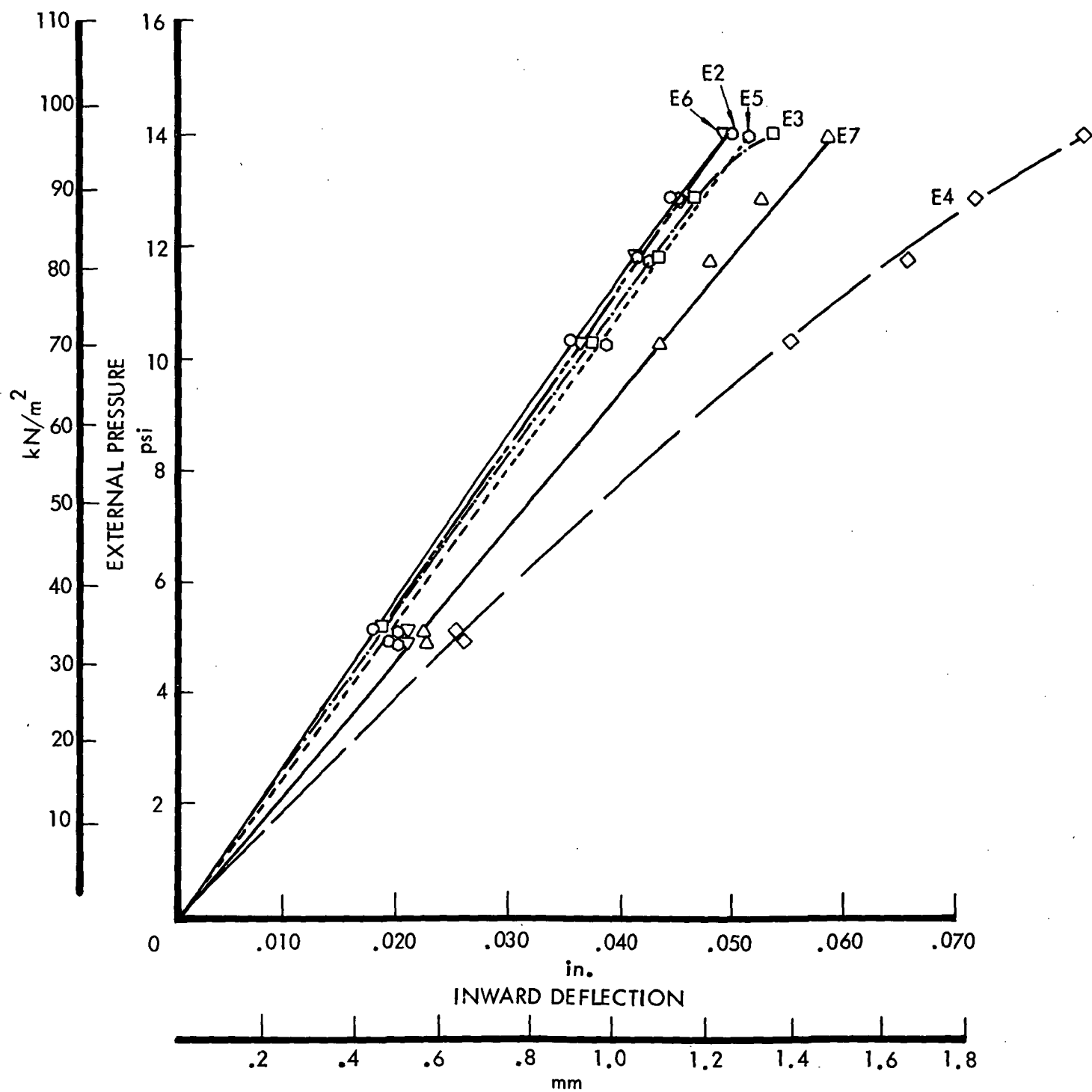


Figure 4-33: PRESSURE VS. DEFLECTION, E-2 - E-7

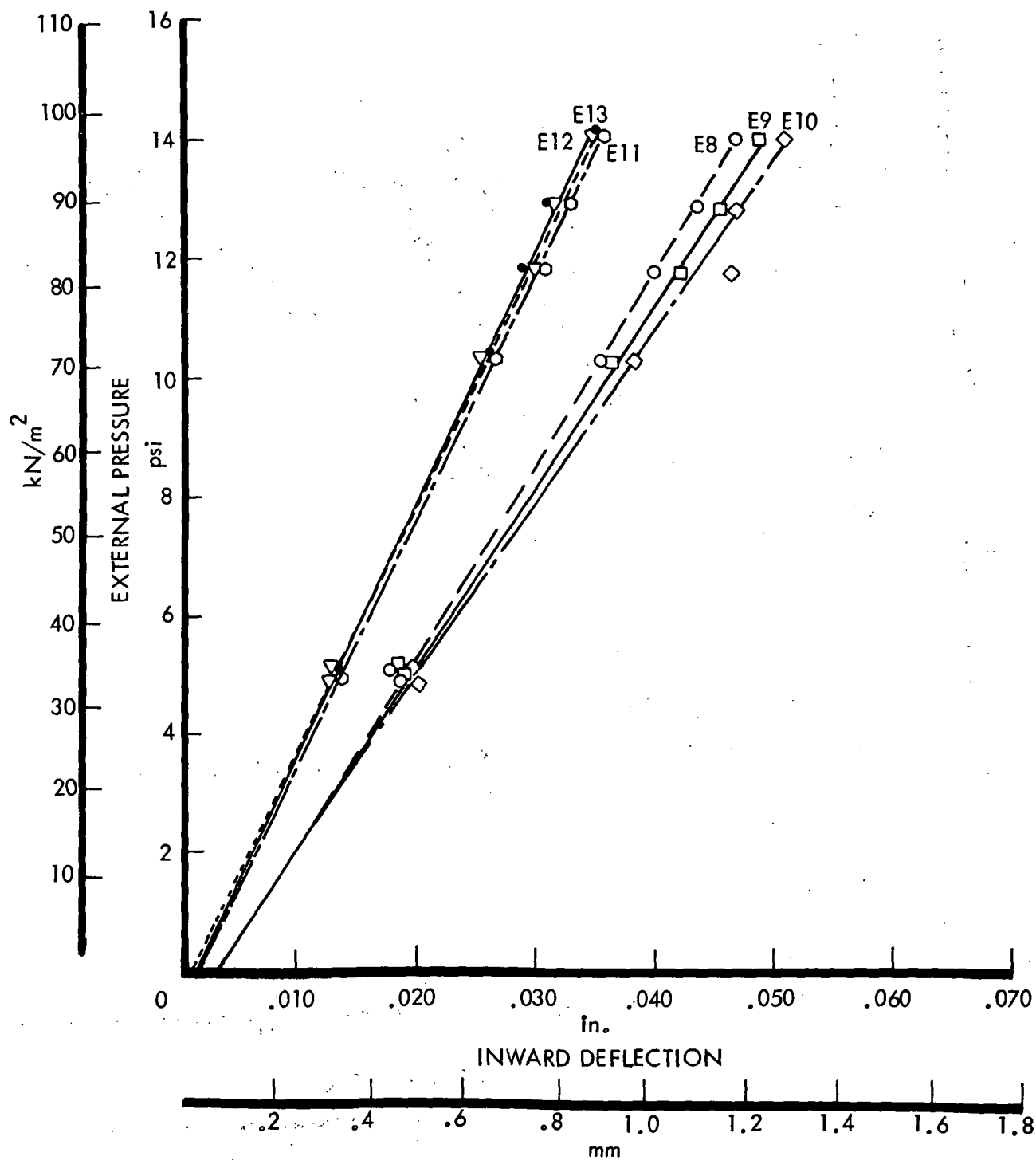


Figure 4-34: PRESSURE VS. DEFLECTION, E-8 - E-13

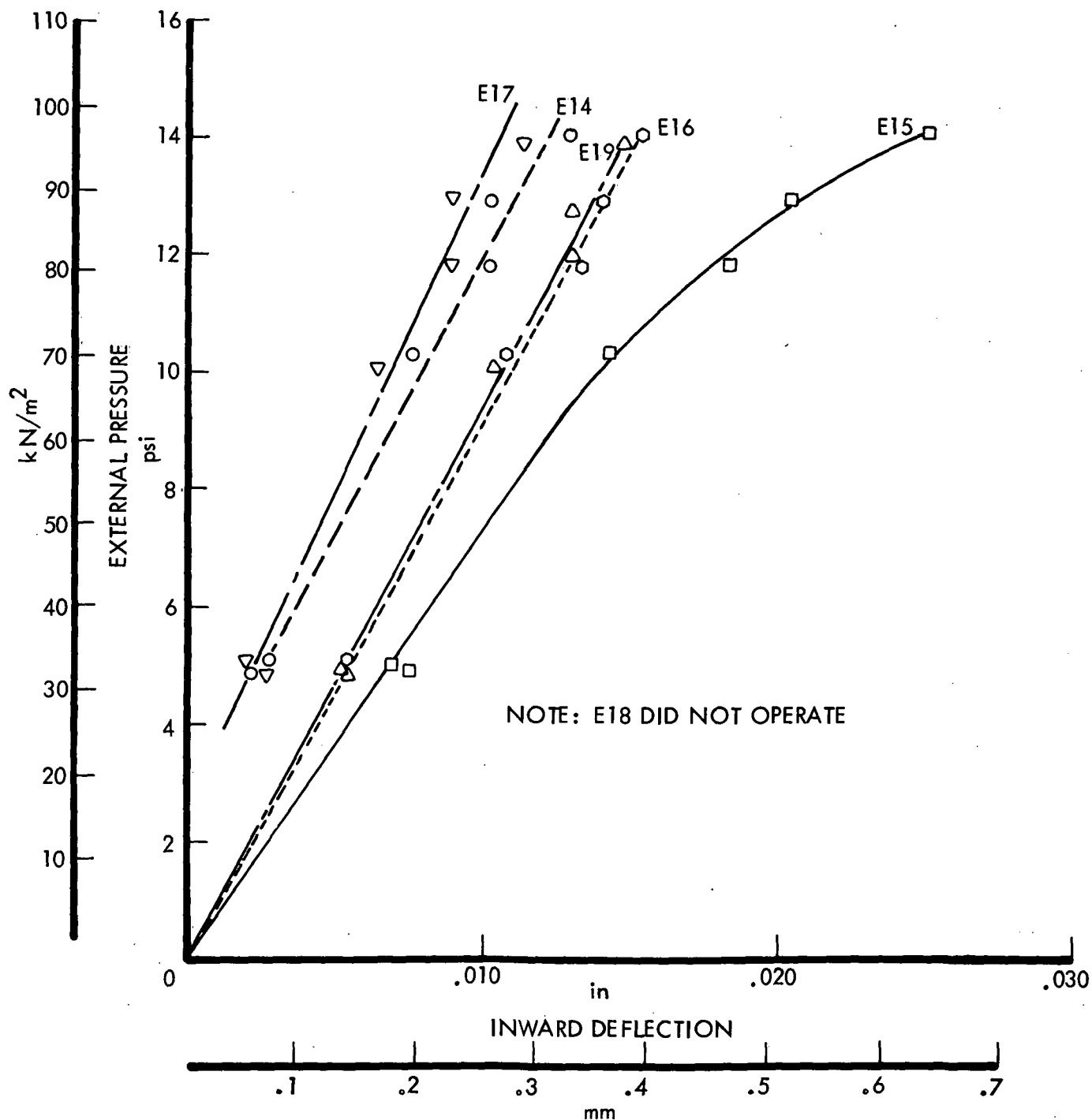


Figure 4-35: PRESSURE VS. DEFLECTION, E-14 - E-18

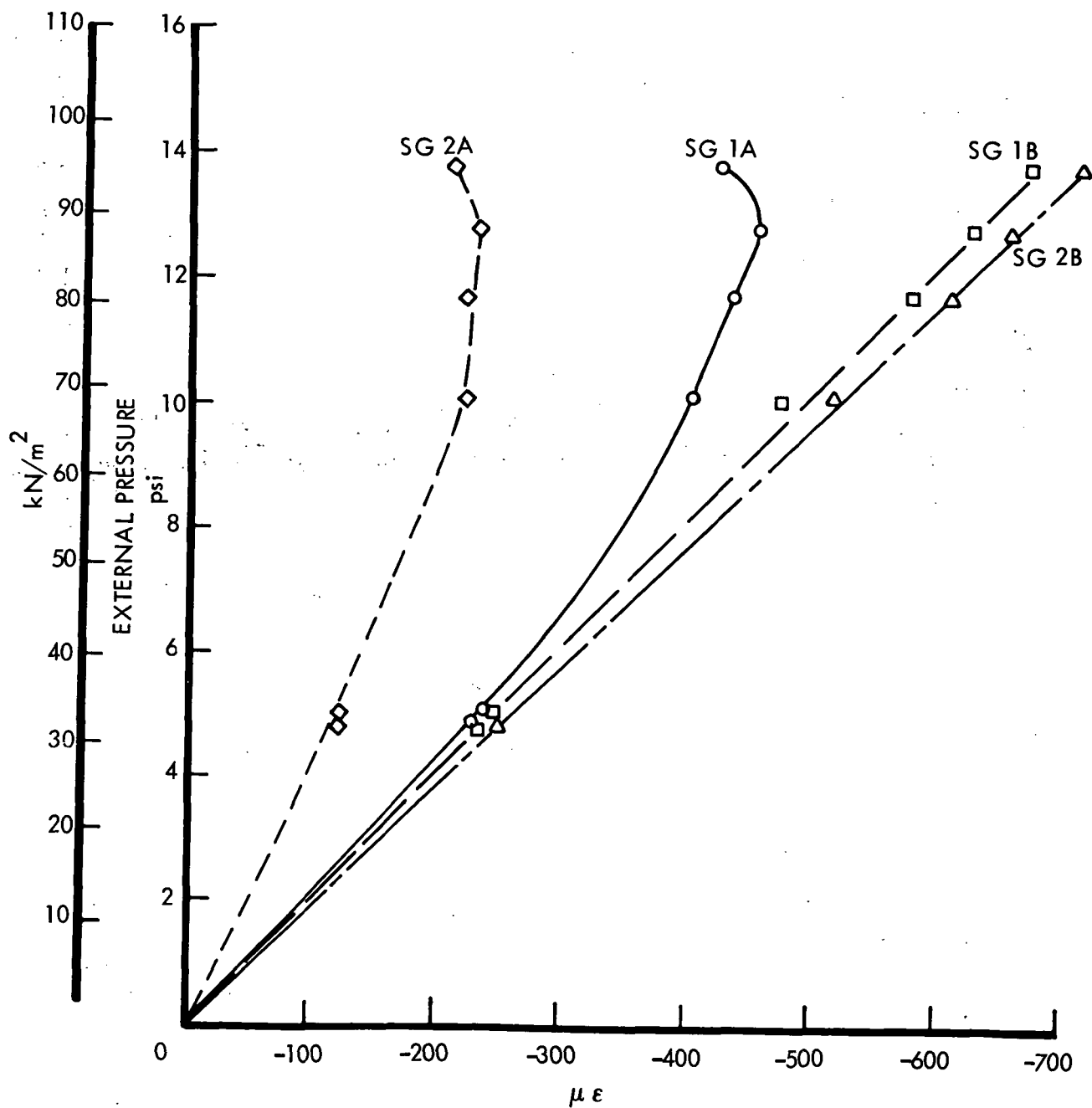


Figure 4-36: PRESSURE VS. STRAIN, SG 1 & SG 2

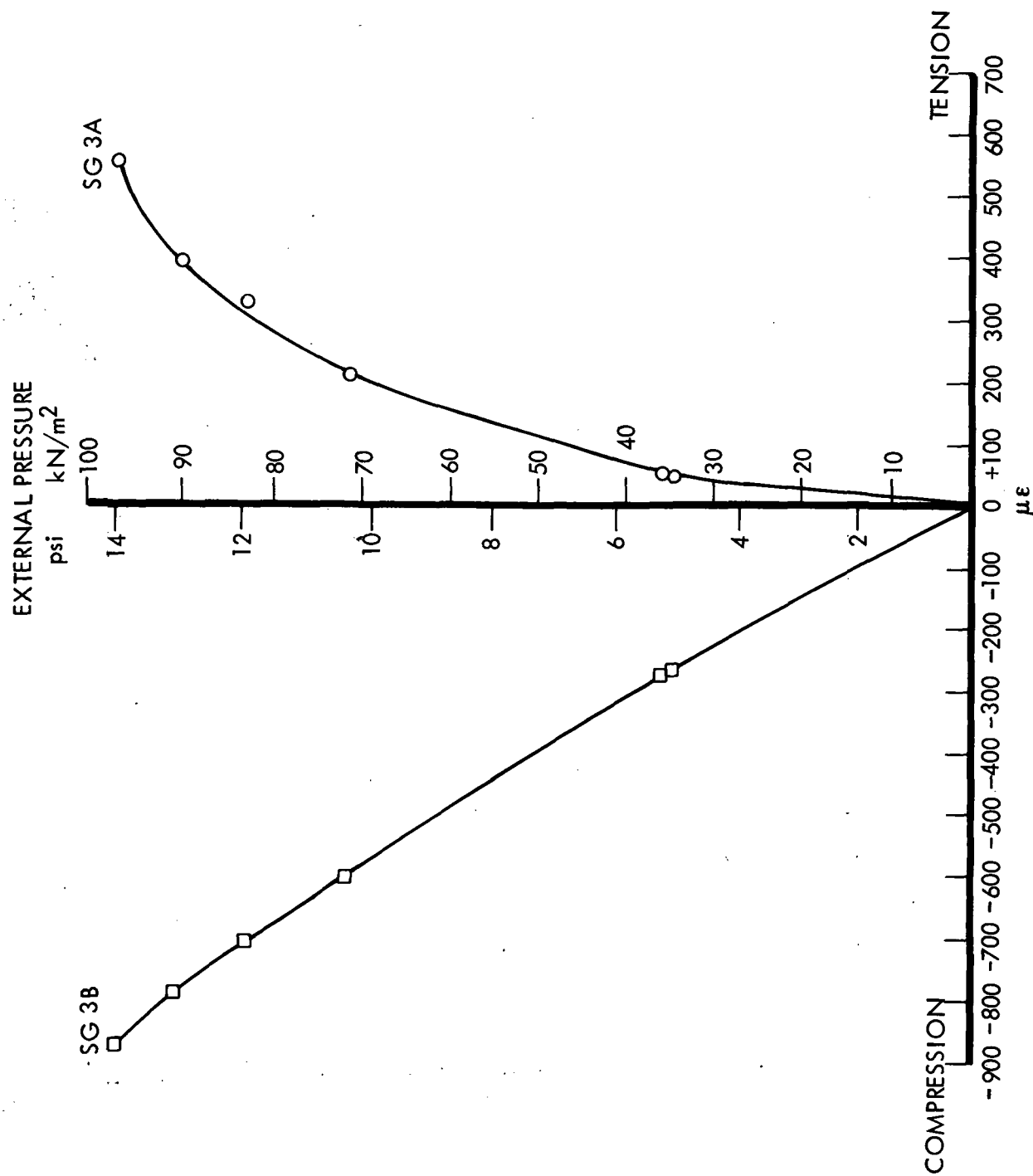


Figure 4-37: PRESSURE VS. STRAIN, SG3

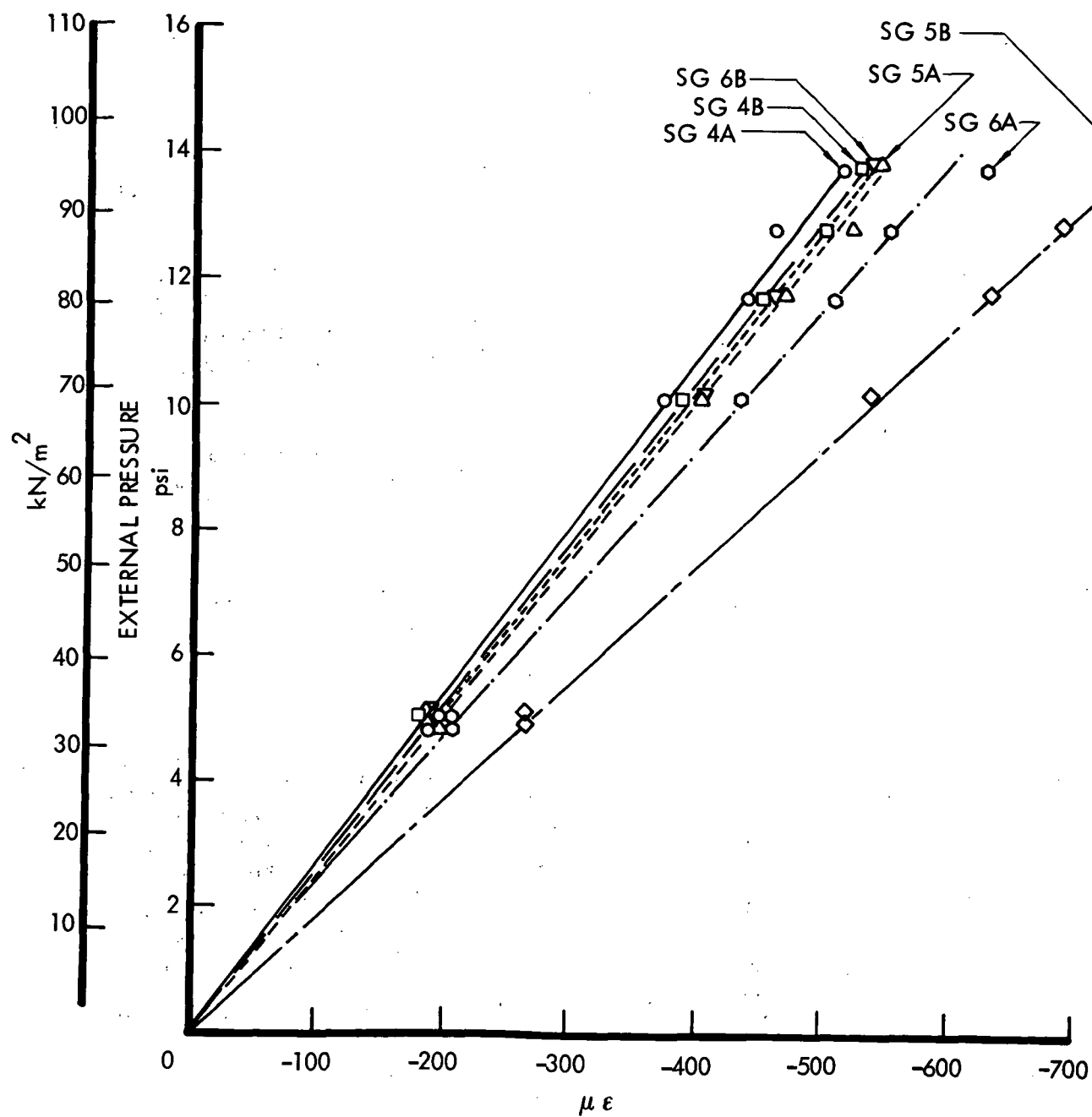


Figure 4-38: PRESSURE VS. STRAIN ,SG4 - SG6

The neutral axis is located very near the 0.043 in. (0.109 cm) thick aluminum face and about 0.49 in. (1.25 cm) from the fiberglass skin. Thus, the B gages on the aluminum skin mostly record the membrane compression strains. The A gages are very sensitive to bending strains. The bending is caused by the local perturbations in the shell wall such as the depressed area. Near the edges of the shell there is very little bending and the face strains are nearly equal, as shown in Figure 4-38.

Discussion of Results

Results of the first complete F/S test of a sandwich shell loaded by external pressure showed that

- 1) The F/S method for predicting buckling pressure on a shell is a valid proof test technique. The test indicated however that successful F/S buckling predictions require fairly extensive strain gage instrumentation, particularly when the location of buckling deformation is not known prior to testing.
- 2) The test procedure and the type of shell construction permitted a locally damaging failure to occur so that the shell could be locally repaired for subsequent use or test.
- 3) The EDI data were not as sensitive as the strain gage data, and did not predict the critical buckling pressure. The EDI data were useful in correlating the overall shell behavior under load.

Repair and Selective Reinforcement of the Head

The damaged area was shown in Figure 4-27. Repair was made by cutting out the damaged core and fiberglass face skin as shown in Figure 4-40a. This area measured approximately 37.0 x 17.0 in. (94.0 x 43.2 cm). The buckle in the aluminum was forced out to contour and held in place by support tooling during adhesive curing. A new section of core was bonded in place as shown in Figure 4-40b. The outer skin was repaired by placing two layers of style 120 fiberglass cloth over the damaged area and the additional two layers of style 120 fiberglass cloth in an elongated polar cap shape extending beyond the repaired area. These two layers were added as a selective reinforcement to assure an improved buckling load on the shell. The repaired head with instrumentation wires installed is shown in Figure 4-41.

Buckling Prediction Test on "Reinforced" Head

Instrumentation

Based on the analysis of the data from the first test, six sets of strain gages were added to the test head, for a total of twelve strain gage sets. Six of



a: Damaged core removed



b: Core replaced

Figure 4-40: REPAIR OF "AS MANUFACTURED" 8 FT (2.44 m) DIAMETER HEAD

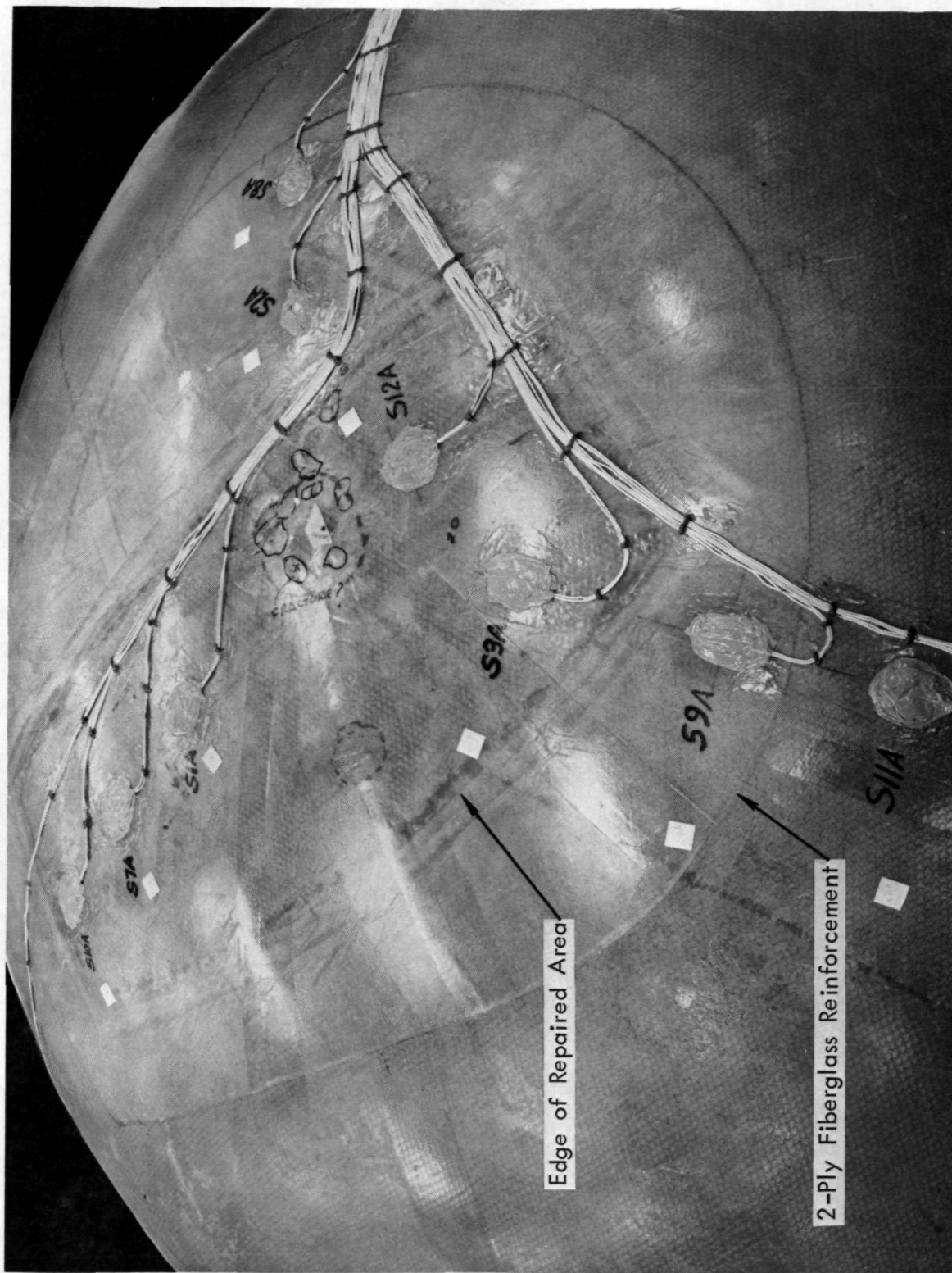


Figure 4-41: "REINFORCED" 8 FT. (2.44 m) DIAMETER HEAD

the EDI's were removed since their data was of little value in the analysis of the first test. The revised instrumentation plan is shown in Figure 4-42. One set of strain gages, SG 12A and B, was located at the center of the repaired "depression" area.

Buckling Prediction Test

Another test was performed on the shell after it was repaired. The procedure followed was the same as described for the first test. A large portion of the head buckled at a hydrostatic pressure of 24.8 psi (171 kN/m^2) or 119 percent of the ultimate design pressure, (20.6 psi) (142 kN/m^2). Two of the F/S plots clearly predicted the buckling pressure at approximately 24 psi (166 kN/m^2). That is within 3 percent of the actual buckling pressure. One F/S plot, strain gage set 12, predicted the failure pressure when the load was 21 psi (145 kN/m^2). The test could have been stopped at that load; however, the loading was continued to determine the experimental buckling load. The results demonstrate that the F/S test method can be used successfully to non-destructively determine the buckling strength of lightweight vacuum jacket shells.

Failure Mode

Failure resulted in a kidney shaped buckle much larger than the first test. Figures 4-43a and b are photographs of the buckled area. The buckle appeared to originate at location E20 (see Figure 4-42). It extended halfway around the center of the head and 19 in. (48.2 cm) away from the center. The buckled area is about twice as large as the first test and much deeper. The size of the buckled area is probably determined by the pressure loading and the strain energy stored in the shell.

Analysis of the Test Data

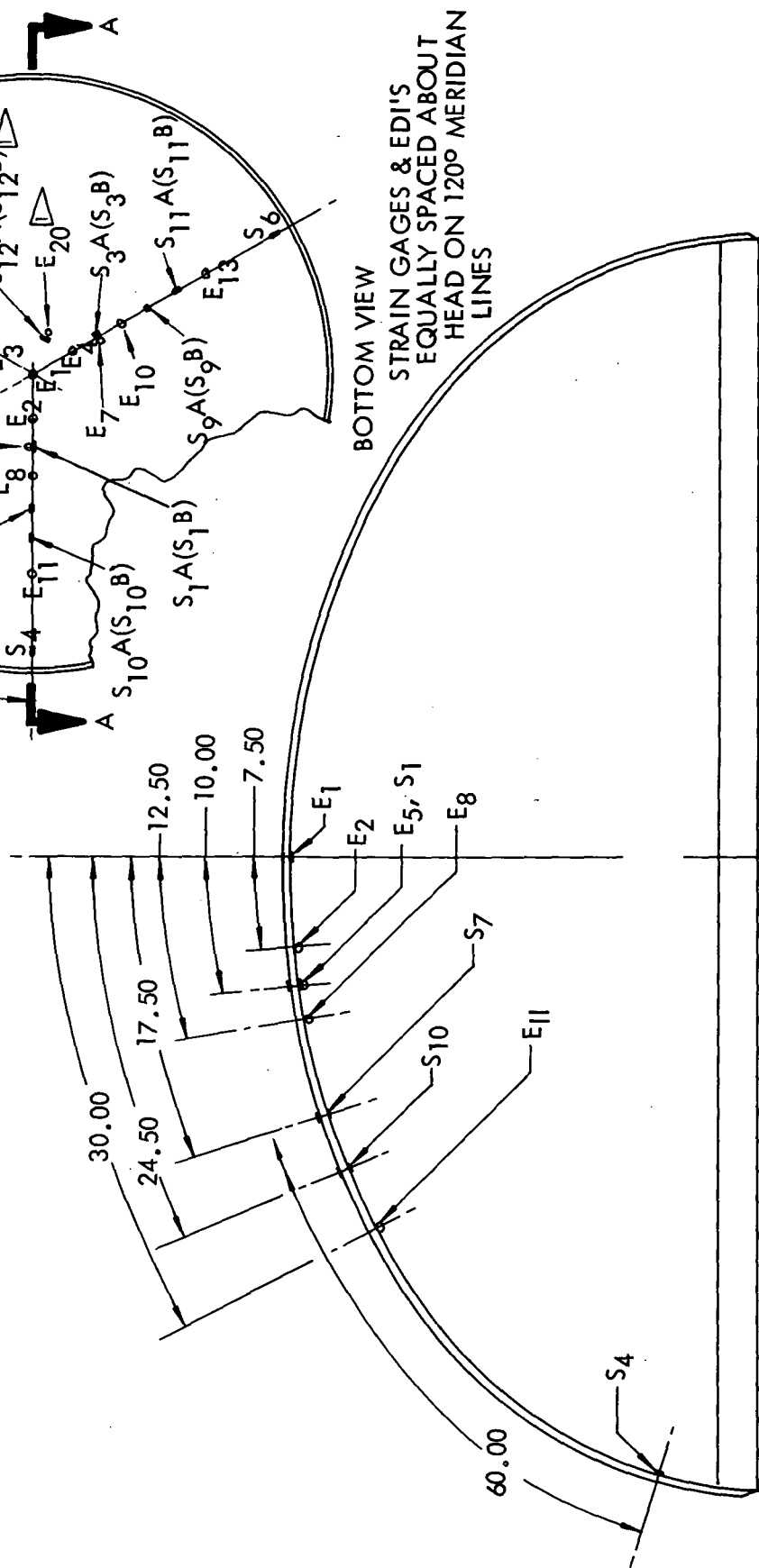
The F/S test data were monitored during the test to identify the "active" areas of the head and to predict the buckling pressure. The most active areas observed were strain gage set 12 and strain gage set 1.

F/S data versus pressure for SG 12A and B are plotted in Figure 4-44. The large scatter in the F/S values up to 20 psi (138 kN/m^2) pressures is apparently due to the small difference in A and B gage strains. This small difference is shown in Figure 4-45 where the microstrain difference between A and B gages is 50 at 14 psi (96.5 kN/m^2) and 100 at 20 psi (138 kN/m^2). Between these pressures the F/S data scatter in Figure 4-44 rapidly decreases. Between 20-22 psi (138 - 152 kN/m^2) the F/S data indicated a critical pressure of 23.9 psi (165 kN/m^2). At 22 psi (152 kN/m^2) the data predicted a critical pressure of more than 30 psi (206 kN/m^2). This is the same F/S test phenomena which was discussed earlier. It is apparently due to the local decrease in stiffness preceding the failure and should be ignored in predicting the critical pressure.

▷ LOCATE 4.9 FROM @ 40° FROM S₃ MERIDIAN

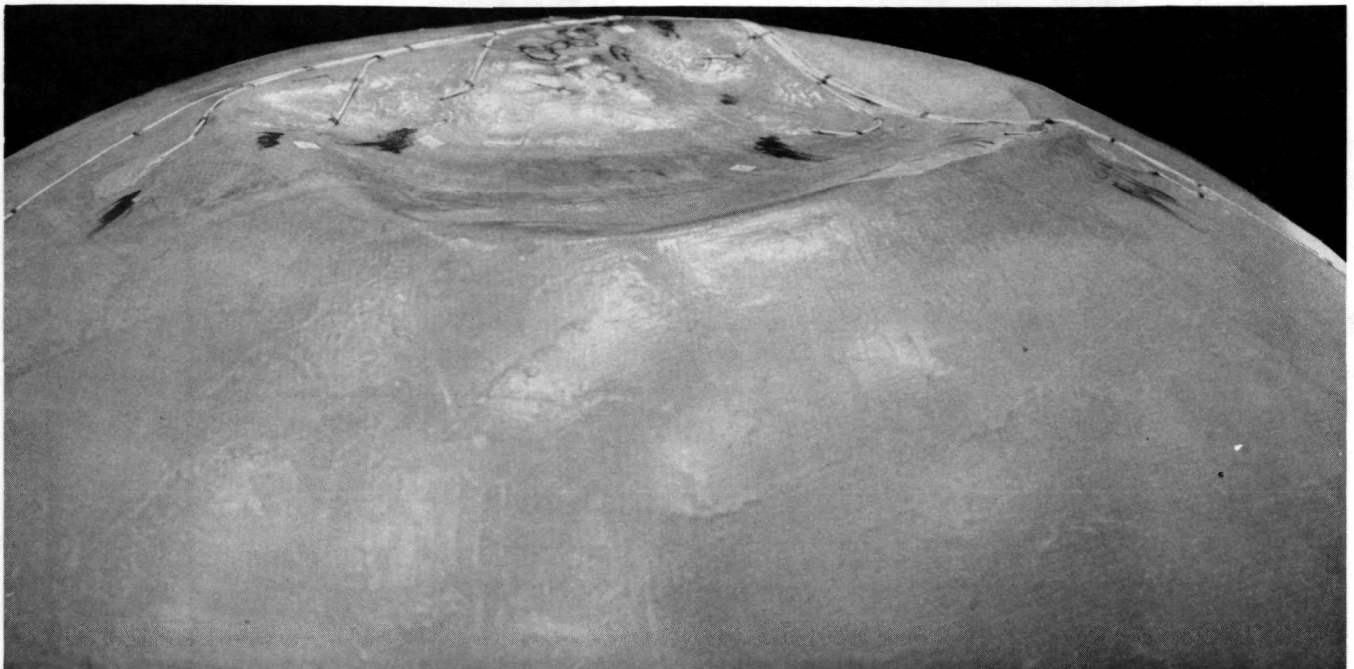
E₁ - E₂₀ ELECTRONIC DEFLECTION INSTRUMENTS
(NORMAL TO SURFACE)

S₁ - S₁₂ A&B STRAIN GAGE PAIR (BACK TO BACK)
A = OUTER B = INNER

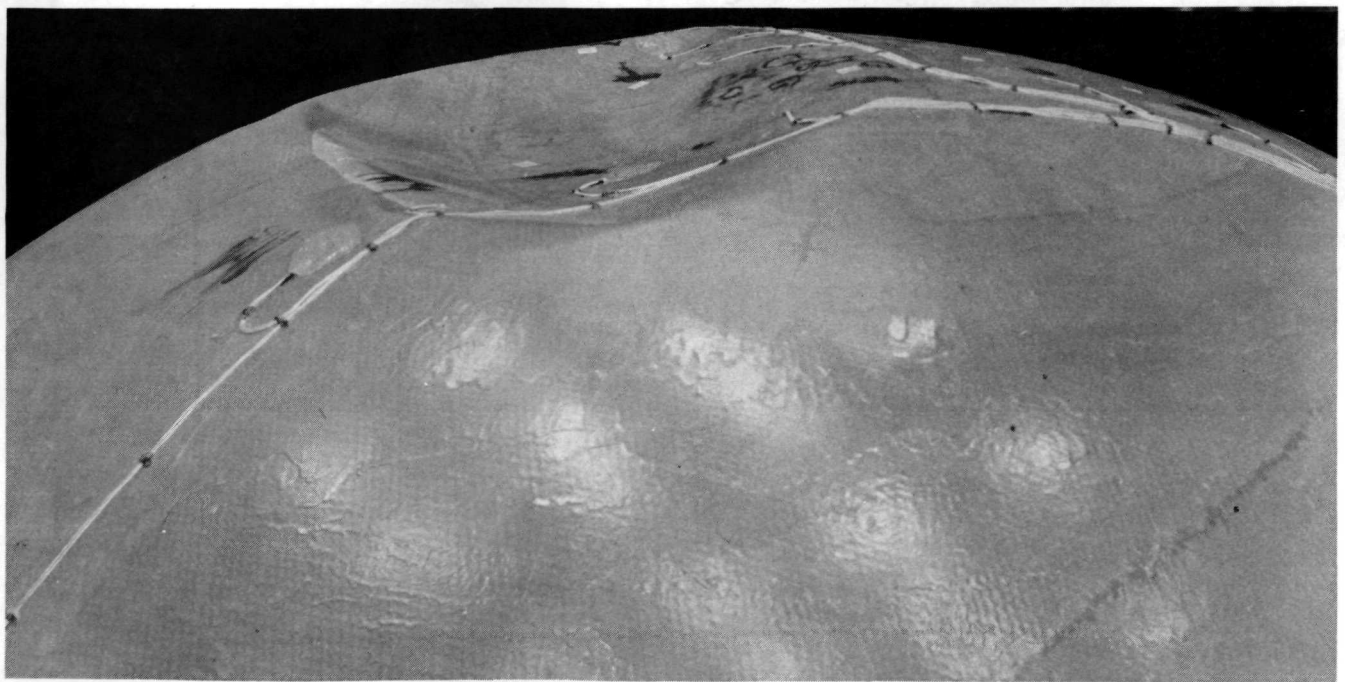


SECTION A-A

Figure 4-42: 8 FT (2.44 m) DIA HEAD INSTRUMENTATION



a: Oblique view of buckled area



b: Oblique view of buckled area

Figure 4-43: "REINFORCED" 8 FT (2.44 m) DIAMETER HEAD AFTER TEST

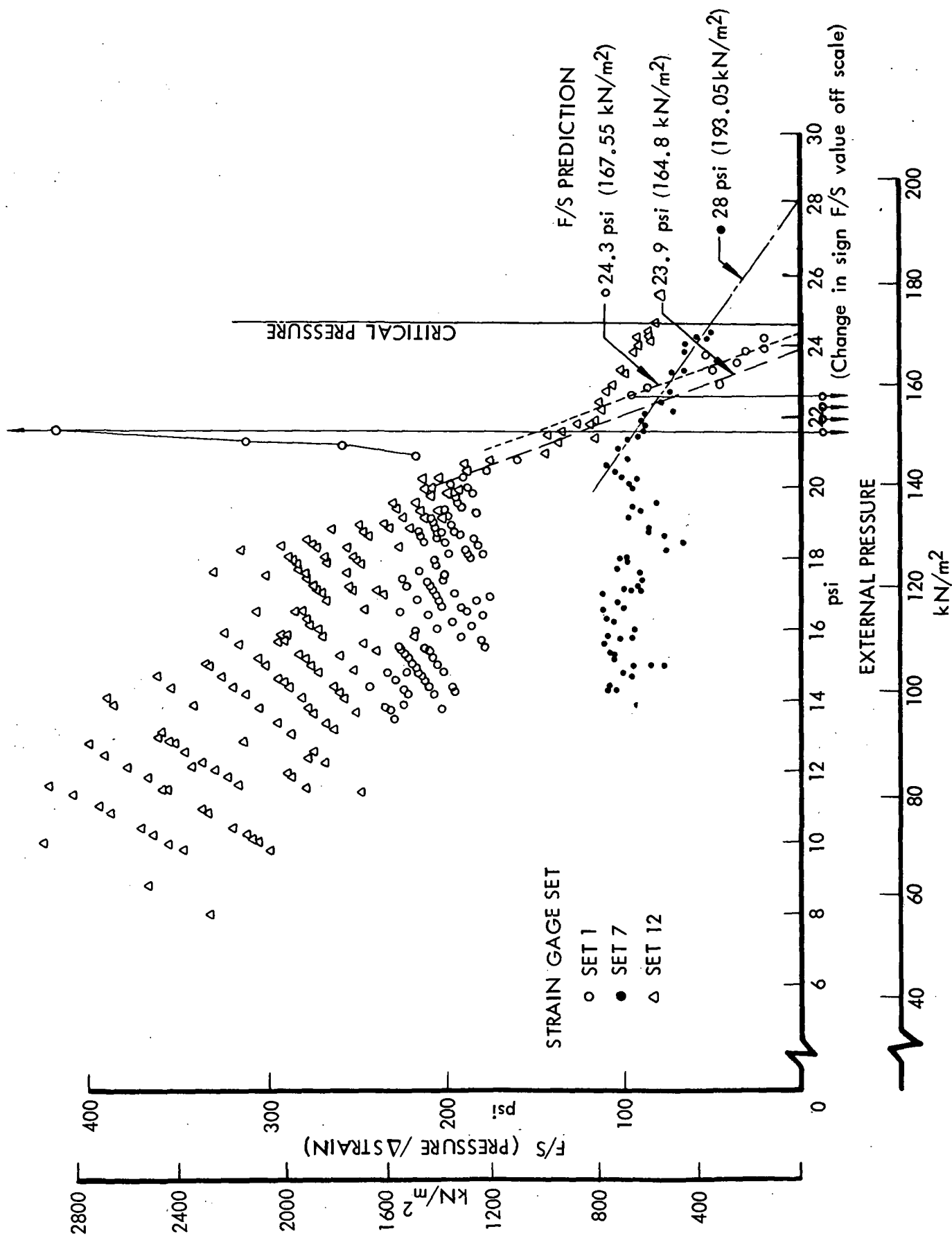


Figure 4-44: F/S PLOT OF STRAIN GAGE SETS 1, 7 & 12

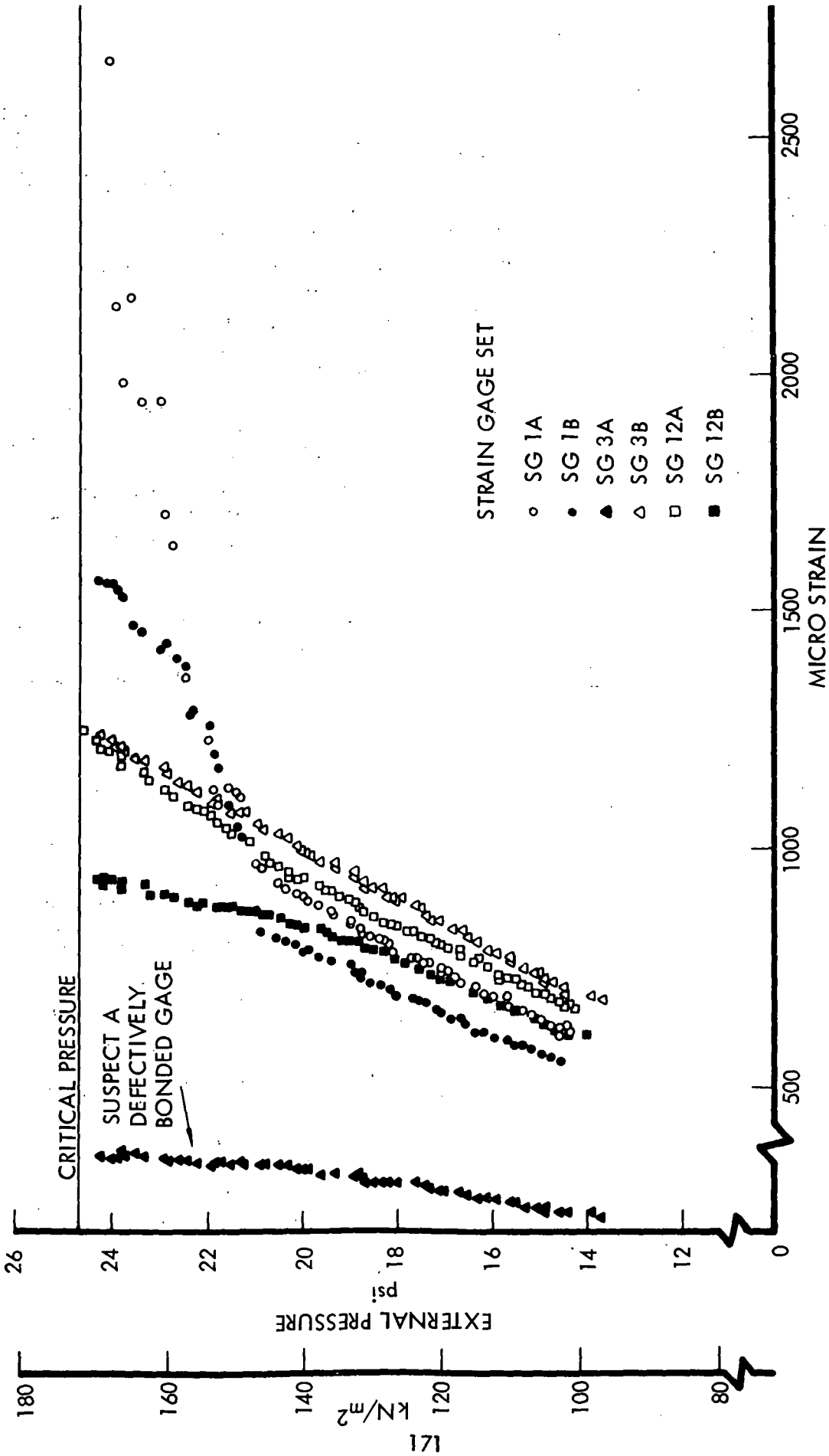


Figure 4-45: PRESSURE VS. STRAIN ,SG1, SG3 & SG12

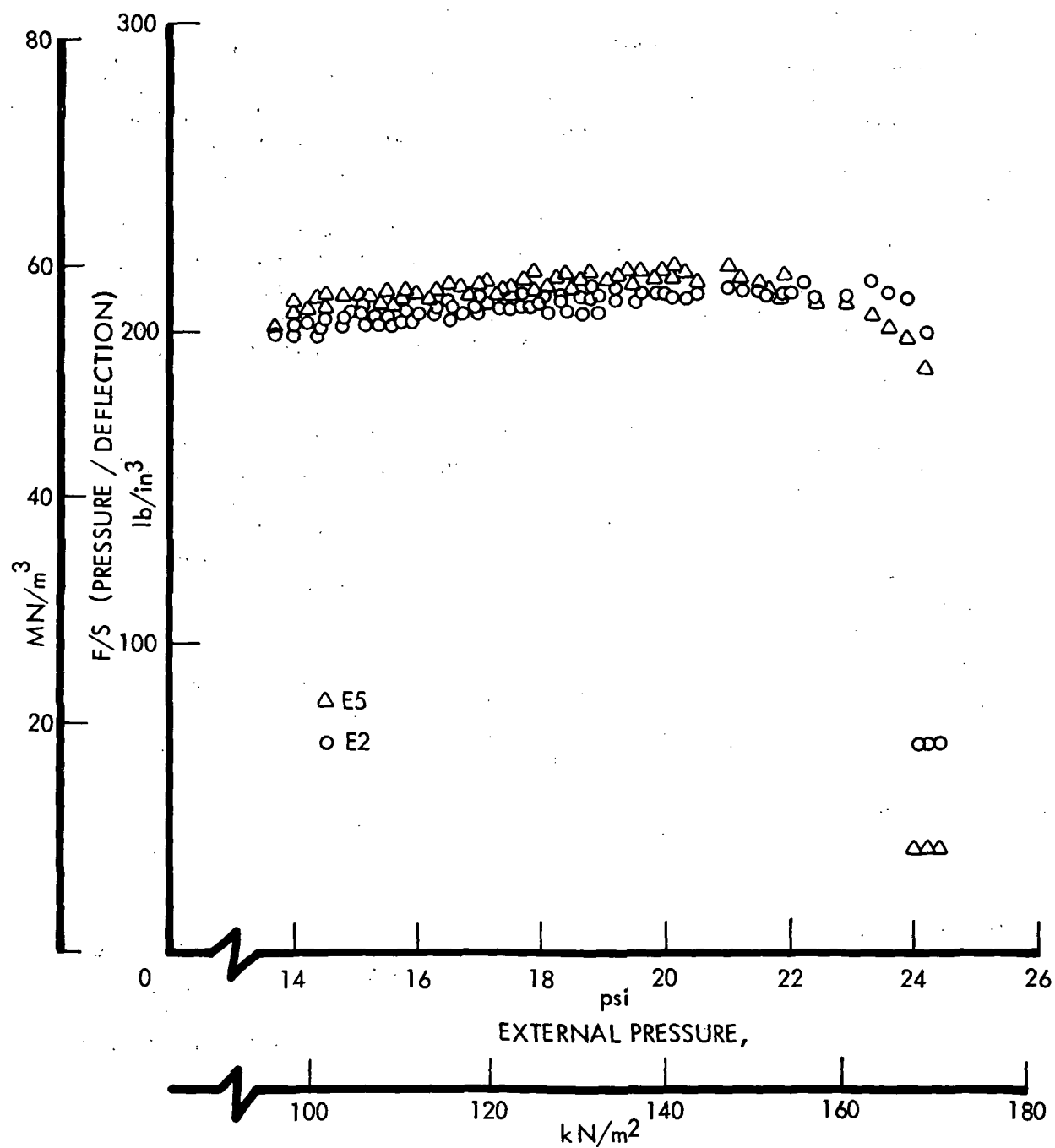
The F/S data plot for SG 1A and B have a large scatter for pressures 14-20 psi (96.5-138 kN/m²) due to small differences in strain. At 21 psi (145 kN/m²) the large scatter appears. Four of the F/S values have a change in sign and go off scale for the plot. The cause of this F/S activity is evident from the microstrain plots in Figure 4-45. Up to 21 psi (145 kN/m²) there is a difference in the A and B strains. From 21 to 22.5 psi (145-155 kN/m²) the strains are nearly equal and result in large values of F/S. For pressures above 22.5 psi (155 kN/m²) the strain difference is again large and the F/S values are less scattered. Apparently a local failure began in the area of strain gage set 1 at 21 psi (145 kN/m²); however, the head did not buckle because the surrounding shell was still capable of carrying higher loading. The F/S prediction for set 1 was 24.3 psi (168 kN/m²). It is interesting to note that the F/S value for set 1 continues to decrease right up to the critical pressure. This indicates that the final buckle began in the area of set 1 rather than 12. It appears that set 12 began to buckle at 20 psi (138 kN/m²) and that the redistribution of load (based on stiffness) may have caused the critical buckle to form near set 1. This would explain the change in the F/S prediction at set 12 before the critical pressure was applied.

Figure 4-44 also has a partial plot of the F/S data for strain gage set 7. These gages were 5.0 in. (12.7 cm) away from set 1 on the same meridian. The F/S prediction for set 7 is 28 psi (193 kN/m²) which indicates that the buckling was localized around set 1.

The F/S data for strain gage set 3 were apparently spoiled by a defectively bonded gage at SG3A. The very steep slope for this gage shown in Figure 4-45 does not correlate with any other strain gage data.

During this test the F/S data from the EDI's were very stable. No useful F/S data were obtained from the EDI's. Two EDI's, Nos. 2 and 5, did show a significant change in the F/S values just prior to the buckling pressure as shown in Figure 4-46. Although the F/S plots do turn downward at 24 psi (166 kN/m²) this prediction is too close to the critical pressure to be useful. The EDI data from this test confirm the conclusion drawn from the first test that the EDI data are not sensitive enough to predict the critical load.

During this test the F/S data from strain gage set 6 were closely monitored since the gages in that area indicated a low critical pressure during the previous test. These gages indicated stable F/S values during the test. A post-test examination did not indicate any buckling or physical damage. Apparently the previous test F/S data were the result of the scatter described earlier for small differences in the measured strains. In fact, several sets of gages suffered from this.



4.3.4 Discussion of Results

The conclusions drawn from the second complete F/S test of a sandwich shell loaded by external pressure were:

- 1) The repair and reinforcement procedures used on the head following the first test increased the critical external pressure from 14.8 to 24.6 psi (102-170 kN/m²). Part of this increase is due to removing the major imperfection noted in the first test. The remainder is due to the two ply reinforcement added to the critically stressed area of the head.
- 2) The F/S method predicted the critical external pressure load within 3 percent using strain gage data. There was sufficient warning that the loading could have been stopped at least 2 psi (13.8 kN/m²) below the critical pressure. Clearly, this demonstrates the use of F/S plots as a nondestructive proof test method for the development of lightweight vacuum jackets.
- 3) The EDI instrumentation was not as sensitive as the strain gage instrumentation for the F/S data. Pairs of strain gages should be used in all subsequent shell buckling tests. Caution must be exercised when interpreting the F/S data from strain gages since small differences in strain may lead to large variations in the F/S value. This condition can be identified by noting the strain values used to calculate the F/S value.
- 4) The location of the F/S instrumentation is critical to the successful prediction of the critical load. Known areas of structural imperfection and highly stressed areas should be sufficiently instrumented to detect and measure all anticipated modes of failure.

4.4 45-Inch (1.14 m) Diameter Sandwich Head Tests

4.4.1 Purpose

The purpose of these tests was to provide comparative design and manufacturing data on two configurations and to assess their vacuum and structural integrity. The external pressure tests were to provide data for the F/S method. Two hemispherical heads were designed and fabricated. One head was external pressure tested to failure.

4.4.2 Head Design, Analysis, and Fabrication

Head Design

The sandwich heads were designed for an ultimate external pressure of 20.6 psi (142 kN/m²) at a uniform structural temperature of 350°F (450°K) to meet the space shuttle design criteria.

Six sandwich configurations were designed with OPTRAN for the 45 in. (1.14 m) diameter heads. Three of the designs used a 6061-T6 aluminum inner skin for vacuum sealing. The outer skins were: (a) boron/epoxy, (b) fiberglass/polyimide, or fiberglass/epoxy laminates. The core in all six cases was 5056 aluminum Flex-Core. It was selected on the basis of the shell trade studies conducted earlier. The second three designs used 2219-T81 aluminum inner skins with the three combinations of outer skin. In each case the OPTRAN designs for minimum weight optimized at the minimum gage for each face with different core thicknesses required for different probabilities of the design not failing. From these studies two configurations were selected for fabrication. The fiberglass/epoxy prepreg was selected for the outer skin of both heads because of lower cost.

Two methods of construction were selected for fabrication of the vacuum sealing inner skin; (1) spinning with chem-milling to the required minimum gage and (2) adhesive bonding of minimum gage gores. Aluminum 6061-T6 was selected for the spun and chem-milled head because of adhesive bonding uncertainties with 2219 (Reference 17). 2024-T3 was selected for the inner skin on the bonded gore configuration. Both heads used the same configuration for the core splices and outer face skin construction.

The gages selected for both heads were:

inner face:	t_1	=	.010 inches	(0.025 cm)
core:	t_c	=	.420 inches	(1.07 cm)
outer face:	t_2	=	.010 inches	(0.025 cm)

2 plies of Style 120 fiberglass/epoxy prepreg

A 99 percent probability of the design not failing was used to guarantee that the jacket would not fail during the vacuum acquisition test at 350°F (450°K).

Analysis

The OPTRAN stress analysis results are tabulated in Table 4-7a. The most probable modes of failure are general instability (M.S. = +0.03) and intracell buckling (M.S. = +0.11).

A BOSOR 3 buckling analysis of the test head mounted on the test fixture was made to predict the buckling load and failure mode at room temperature. The results are listed in Table 4-7b. The results of the BOSOR 3 analysis were also used to place the strain gages required for the F/S Buckling Prediction Method. The same knockdown factor was assumed for the BOSOR 3 analysis and the OPTRAN analysis. The external pressure predicted by BOSOR 3 is higher than the OPTRAN prediction because the room temperature material properties were used and the edge fixity of the test fixture was included in the analysis.

Table 4-7: STRESS ANALYSIS OF 45 INCH (1.14m) DIAMETER HEAD

($t_1 = .010$ in (.025 cm) Aluminum, $t_2 = .010$ in (.025 cm) Fiberglass, $t_c = .420$ in (1.1 cm) 2.1lb/ft³(33.6 kg/m³) Flex-Core)
 (Ultimate Design Pressure = -20.6 psi (-142.0 kN/m²), Probability of Not Failing = 99 Percent)

a: 350°F DESIGN (OPTRAN)

MODE OF FAILURE	ALLOWABLE STRESS OR PRESSURE		APPLIED STRESS OR PRESSURE		MARGIN OF SAFETY Δ	COMMENTS
	psi	kN/m ²	psi	kN/m ²		
GENERAL INSTABILITY	- 21.2	-146.2	-20.6	-142.0	+0.03	Probability of Not Failing = 99 Percent, Knockdown Factor = 0.165
INTRACELL BUCKLING STRESS	- 22,100	-151,685.0	-19,400	-133,700.0	+0.14	Assumed a Square Cell Size of 0.3 in (0.76cm)
FIBERGLASS	- 6,560	-45,230.0	-4,470	-30,820.0	+0.46	
FACE WRINKLING STRESS	- 52,200	-359,910.0	-19,400	-133,700.0	+1.7	
FIBERGLASS	- 32,200	-222,008.0	-4,470	-30,820.0	>+1.0	
SHEAR CRIMPING STRESS	-160,000	-1,103,161.0	-19,400	-133,700.0	Large	
FIBERGLASS	- 48,600	-336,470.0	-4,470	-30,820.0	Large	

b: 70°F DESIGN (BOSOR3)

GENERAL INSTABILITY	-31.2 at $n = 8$	-215.12	-20.6	-142.0	+0.51	Knockdown Factor = 0.165
INTRACELL BUCKLING STRESS	-26,000	-179,264.0	-19,400	-133,700.0	+0.34	
FIBERGLASS	-7,800	-53,779.0	-4,470	-30,820.0	+0.74	
FACE WRINKLING STRESS	—	—	—	—	Not Critical	
SHEAR CRIMPING STRESS	—	—	—	—	Not Critical	

$$\Delta > 1 \text{ M.S.} = \frac{\text{Allowable Stress}}{\text{Applied Stress}} - 1$$

It appears from both the 350°F (450°K) and 70°F (170°K) analyses that the head could fail by intracell buckling or general instability. Since there was very little experimental data for shells with sandwich construction, it was difficult to determine which mode would govern the failure. Intracell buckling has rarely been a problem with aluminum faced sandwich.

Fabrication

Sandwich Head - Spun and Chem-milled Inner Face Skin

Figures 4-47 and 4-48 show the assembly arrangement for one of the 45 in. (1.14 m) diameter hemispherical sandwich heads. The 6061 aluminum alloy inner face skin was spun in the annealed condition then heat treated to the T6 condition. There was no significant warpage. The shell thickness was mapped using a Vidigage technique. Selective chem-milling was used to even out thickness variations. The part was then reduced in thickness by overall chem-milling. Preferential etching (pitting) problems developed during the chem-milling operation due to the enlarged grains which developed in the material during spinning. (The enlarged grains can be seen in Figure 4-49 a.) Final thickness of this shell ranged between 0.032 to 0.034 in. (0.081 to 0.086 cm). This exceeded the minimum design gage of 0.010 in. (0.025 cm). Further chem-milling presented too great a risk that local thin spots due to preferential etching would go unnoticed during gage inspection and result in pinhole penetrations during subsequent chem-milling. Pinholes would be difficult to locate and repair.

The tapered aluminum Flex-Core edge pieces were cut and stabilized with BMS 5-25 potting compound.

The HT 424 adhesive was cut to shape and laid up on the 6061-T6 aluminum inner shell. The Flex-Core including the tapered edge pieces was cut and positioned on the adhesive. BMS 5-90 structural foaming adhesive was placed in the core joints. After curing this assembly, AF 131 adhesive was laid up on the Flex-Core. The fiberglass prepreg was trimmed and laid up on the adhesive. Caul plates were placed over the outer skin and this assembly was vacuum bagged and cured in the autoclave. The caul plates were used to eliminate dimpling and folding of the fiberglass skins. These caul plates were cut from a fiberglass dome which was laminated on a plaster mold shaped to the skin contour. The dome was cut in several sections so that it could "follow" the skin when curing pressure was applied. The gaps between adjacent sections was bridged with very thin shim stock so that the skin was not pinched between adjacent caul sheet edges. The smooth surface that resulted after curing can be seen in Figure 4-49b. The head is shown in place on the base plate in Figure 4-50.

Sandwich Head - Bonded Gore Inner Face Skin

Figures 4-47 and 4-51 show the assembly arrangement for the other 45 in. (1.14 m) diameter sandwich head.

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REVISIONS

DATE APPROVED

30 BMS 5-25, GRADE 3.

29 2024-T3 ALUM SHEET .003 THICK STOCK, PER QQ-A-250/4.

28 FILL BUTT SEAMS BETWEEN GORES(-9), GORES & POLAR CAP(-15) WITH BMS 5-25, GRADE 3.

27 KEENSERT 1/2-20UNF, FURNISHED BY ENGR. (REF. NAS 1395-8 OR BAC 87-4300-1600)

26 3 LAYERS GLASS EPOXY PREPREG, PER BMS 8-139, TYPE 120.

25 6061-T4 OR T6 ALUM. PER QQ-A-250/11. (SIZE OPTIONAL FOR FABRICATION)

24 2024-T3 ALUM SHEET, .010 THICK STOCK, PER QQ-A-250/4.

23 BOND METAL LAMINATES WITH XA 3919 (3M Co.)

22 CLEAN AND BOND PER BAC 5529. ADHESIVE BMS 8-145, TYPE 1 (AF-13), MINNESOTA MINNING AND MFG CO. APPLIED TO FAYING SURFACES NOTED.

21 HELIUM LEAK CHECK WELD JOINT AT 1×10^{-5} TORR.

20 PENETRANT INSPECT WELDMENT PER BAC 5423.

19 HEAT TREAT TO T6 CONDITION PER BAC 5602.

17 THE ASSEMBLY SHALL BE PACKAGED AND SEALED IN A CLEAN POLYETHYLENE BAG PER BAC 5216, PARA 4.1 K(1), PRIOR TO STORING OR DELIVERY FOR TEST.

16 ALL RAW MATERIALS, SUB ASSYS, & ASSYS SHALL BE PROTECTED FROM OIL & PARTICLE CONTAMINATION. ASSEMBLY SHALL BE DONE IN A DUST FREE ROOM.

15 RUBBER STAMP PART NUMBER AS SHOWN.

13 STRUCTURAL FOAMING ADHESIVE PER BAC 5-90, TYPE 2, CLASS 350, GRADE 50.

10 FUSION WELD PER BAC 5935, CLASS A. EXCEPT RADIOGRAPHIC INSPECTION IS WAIVED. INSPECT WELD PER 20 21

9 6061-T4 OR T6 AL PLATE 1.25 x 48.00 x 48.00, PER QQ-A-250/11.

8 CLEAN & BOND PER BAC 5540, EXCEPT ADHESIVE BMS 5-17, TYPE 1, GRADE B, (HT 424 BLOOMINGDALE DEPT, AMERICAN CYANAMID) SHALL BE APPLIED TO FAYING SURFACES NOTED.

7 DENSIFY CORE WITH 30 COMPLETELY AROUND PERIMETER TO LENGTH SHOWN, PER BAC 5514-590.

6 GLASS EPOXY PREPREG, PER BMS 8-139, TYPE 120, 1 PLY.

5 GLASS EPOXY PREPREG, PER BMS 8-139, TYPE 120, 2 PLY. OVERLAP AT SEAMS, .10 TO .25 INCHES.

4 FLEX CORE, 5056/F40-.0014, 2.1 LB/FT³ ALUMINUM HONEYCOMB. SPlice CORE GORES WITH 13 HEXCEL PRODUCTS INC.

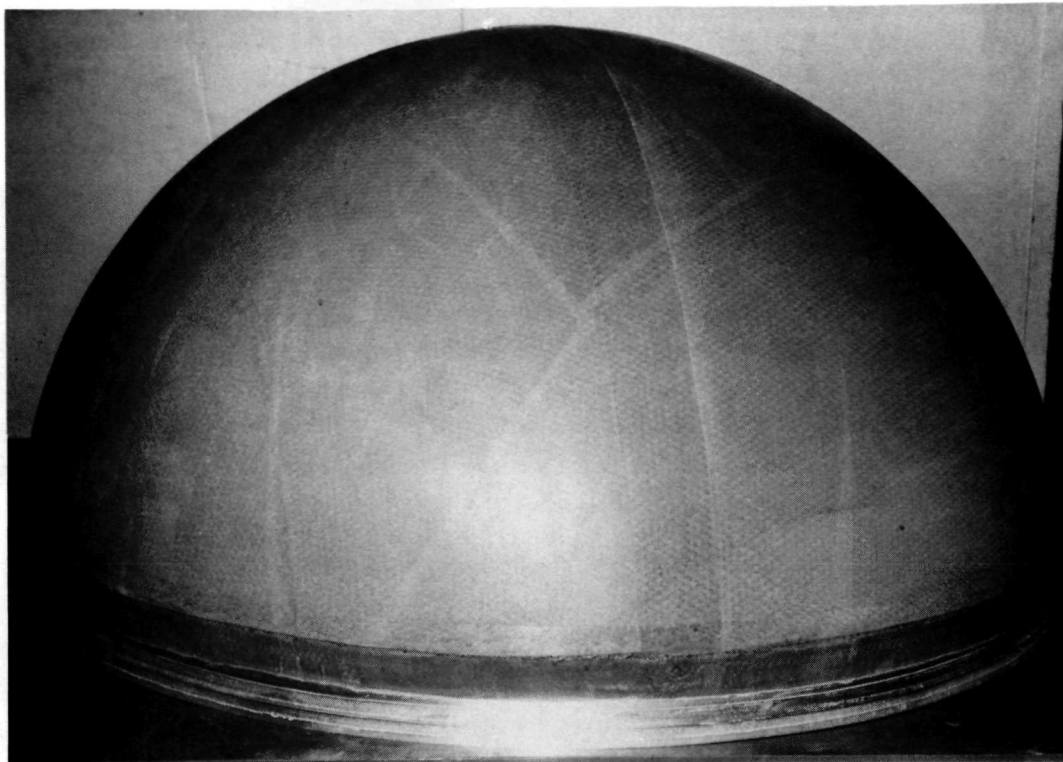
3 CLEAN PER BAC 5514

2 6061-O AL SHEET, .375 x 80.00 x 80.00, PER QQ-A-250/11.

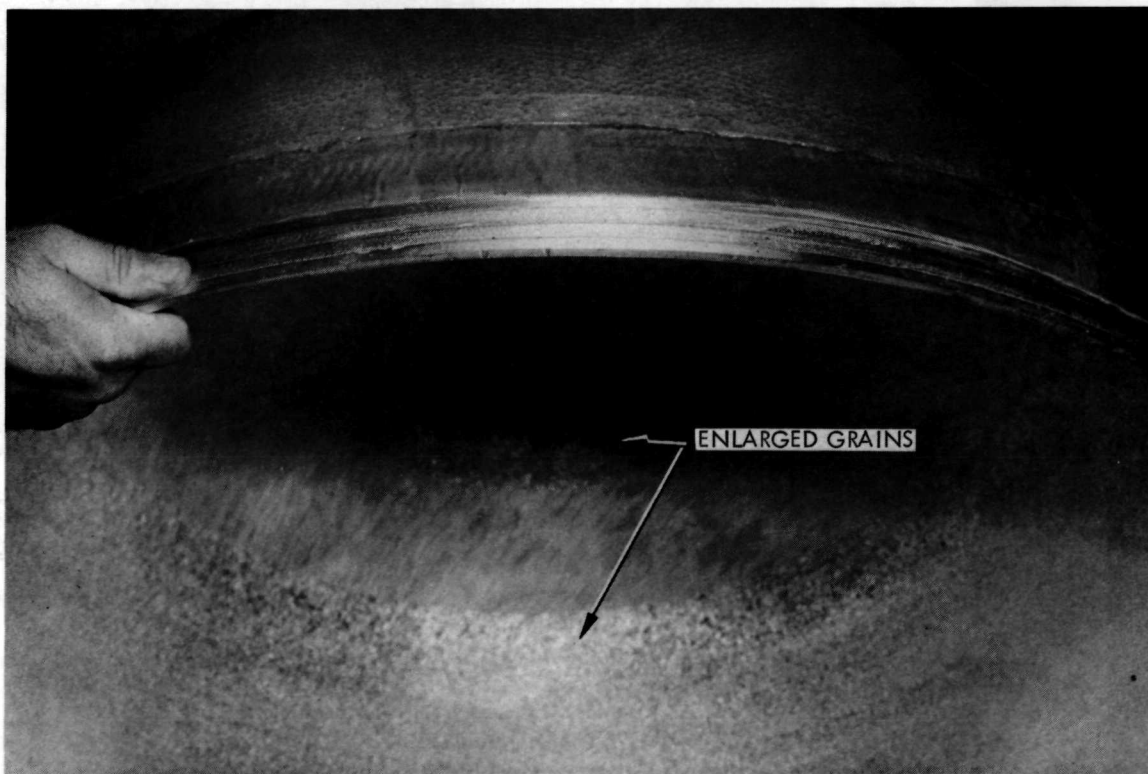
1 DEVIATION FROM RADIUS $\pm .02$, PROVIDED THE DISCONTINUITIES DO NOT EXCEED .02 IN 10.00 LENGTH IN ANY DIRECTION ALONG THE SURFACE.

RELEASE	QTY	DEC	PART OR IDENTIFYING NUMBER	OR DESCRIPTION	ZONE	MATERIAL AND SPECIFICATION	HT-TR	FINISH	PT-MK	REV
			8	-25	FILLER	B5-3	24	.010 x 1.0 x 4.00		
			8	-24	GUSSET	B5-3	29	.003 x 4.0 x 4.0		
			1	-23	ANNULUS RING	B2-3	29	.003 x 2.00 x 37.7		
			8	-22	STRIP, INNER	C2-3	24	.010 x 2.00 x 31.9		
			8	-21	PATCH	C2-3	24	.010 x 2.00 x 3.60		
			8	-20	STRIP, OUTER	C2-3	29	.003 x 2.00 x 28.9		
			-13	-6	INSERT	E8-2	27			
			1	-18	REINFORCEMENT	B5-2	26			
			1	-17	RING	A5-3	29	3	19	
			1	-15	POLARCAP	C1-3	24	3	.010 x 12.1 DIA	
			1	-13	BASE PLATE	A4-3	9			
			1	-12	REINFORCEMENT	A4-3	26			
			1	-11	CORE	A4-3	4			
			1	-10	FACE SKIN OUTER	A4-3	5			
			8	-9	FACE SKIN GORE INNER	A4-3	24	3	.010 x 18.0 x 30.9	
			1	-8	DOUBLER	A4-3	6			
			-	-	HEMISPHERICAL SANDWICH HEAD ASSEMBLY	B2-3				15
			1	-6	BASE PLATE	B7-2	9			
			1	-5	CORE	B5-2	4			
			1	-4	FACE SKIN OUTER	B5-2	5			
			1	-3	FACE SKIN INNER	B5-2	2	3	19	
			1	-2	DOUBLER	B6-2	6			
			-	-	HEMISPHERICAL SANDWICH HEAD ASSY	C3-2	8			15
			-7	-1	QTY DEC					

RELEASE	QTY	DEC	PART OR IDENTIFYING NUMBER	OR DESCRIPTION	ZONE	MATERIAL AND SPECIFICATION	HT-TR	FINISH	PT-MK	REV
			8	-25	FILLER	B5-3	24	.010 x 1.0 x 4.00		
			8	-24	GUSSET	B5-3	29	.003 x 4.0 x 4.0		
			1	-23	ANNULUS RING	B2-3	29	.003 x 2.00 x 37.7		
			8	-22	STRIP, INNER	C2-3	24	.010 x 2.00 x 31.9		
			8	-21	PATCH	C2-3	24	.010 x 2.00 x 3.60		
			8	-20	STRIP, OUTER	C2-3	29	.003 x 2.00 x 28.9		
			-13	-6	INSERT	E8-2	27			
			1	-18	REINFORCEMENT	B5-2	26			
			1	-17	RING	A5-3	29	3	19	
			1	-15	POLARCAP	C1-3	24	3	.010 x 12.1 DIA	
			1	-13	BASE PLATE	A4-3	9			
			1	-12	REINFORCEMENT	A4-3	26			
			1	-11	CORE	A4-3	4			
			1	-10	FACE SKIN OUTER	A4-3	5			
			8	-9	FACE SKIN GORE INNER	A4-3	24	3	.010 x 18.0 x 30.9	
			1	-8	DO					



b: Outer Face Skin Fiberglass/Epoxy Prepreg



a: Spun Inner Face Skin, 6061-T6 Aluminum

Figure 4-49: 45 INCH (1.14 m) DIAMETER HEAD

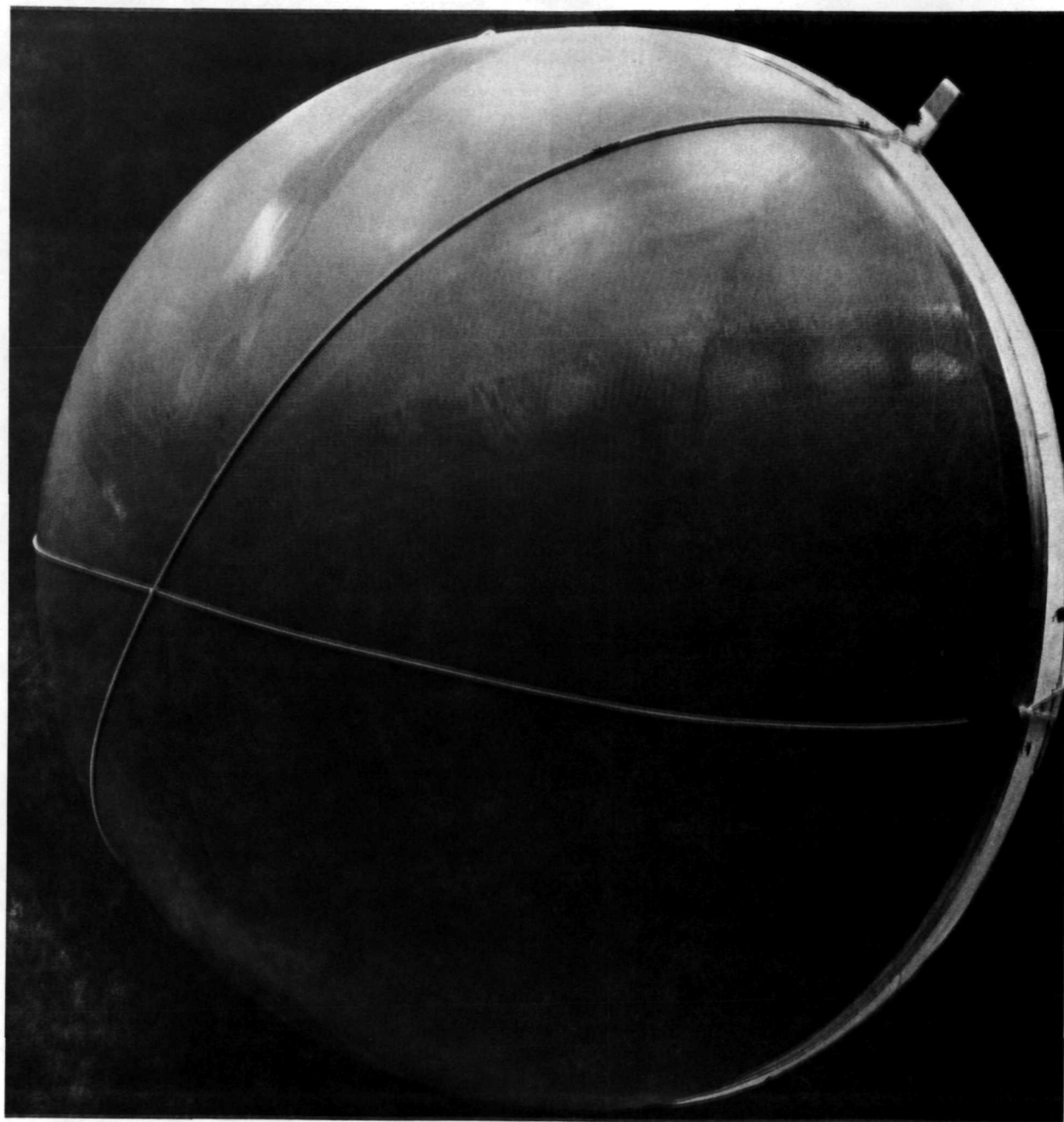


Figure 4-50: 45 IN. (1.14 m) DIAMETER HEAD ASSEMBLED TO BASE PLATE

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A high temperature fiberglass mandrel was used for stretch forming the 0.010 in. (0.025 cm) 2024 T3 aluminum gores and for layup of the hemisphere. The stretched form gore and the trim tool are shown in Figure 4-52a. Figure 4-52b shows the gores in place on the mandrel for final fitup. The first step in assembling this head was to place the core segments together on the mandrel with the girth ring in place. The foaming adhesive was placed in the joints and then these items were bonded together using a vacuum bag to hold the core in place on the mandrel. The fiberglass outer skins were then bonded in place with AF 131 adhesive. Figure 4-53a shows the fiberglass skin after cure. The meridional lines in the photograph are joints in the AF 131 adhesive. Figure 4-53b is an inside view of the shell showing the aluminum Flex-Core and one core splice. Figure 4-54a shows one of the gores in place prior to bonding. An epoxy/phenolic film adhesive (HT 424) was used to bond the aluminum skins to the core. Once the gores were placed, the vacuum bag was applied and the cure cycle was initiated. The gap between the gores was filled with a room temperature adhesive.

Aluminum seal strips were bonded over the gore segment joints. A liquid adhesive was used to obtain a very thin bond line and minimize vacuum leakage. The adhesive used was 3M-XA-3919 which is AF 130 in solution. It was applied by brush coating both surfaces. The surfaces were then air dried for 15 minutes at room temperature followed by a 45 minute drying cycle at 225°F (325°K). The bond was obtained by curing at 350°F (450°K) for 60 minutes at 50 psi (346 kN/m²). Figure 4-54b shows the inner face skin after final curing.

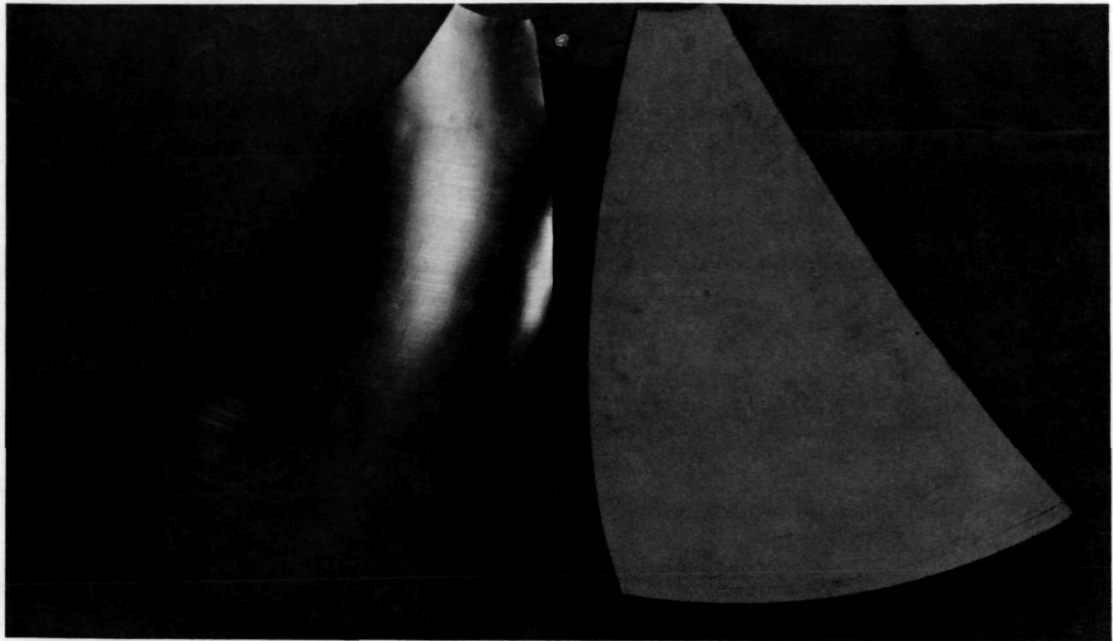
Contour measurements were made on the shell at four positions. The results are shown in Figure 4-55. The data show that the shell surface deviates from a perfect hemisphere by less than 8/100th of an inch (0.203 cm).

4.4.3 Vacuum Acquisition Tests

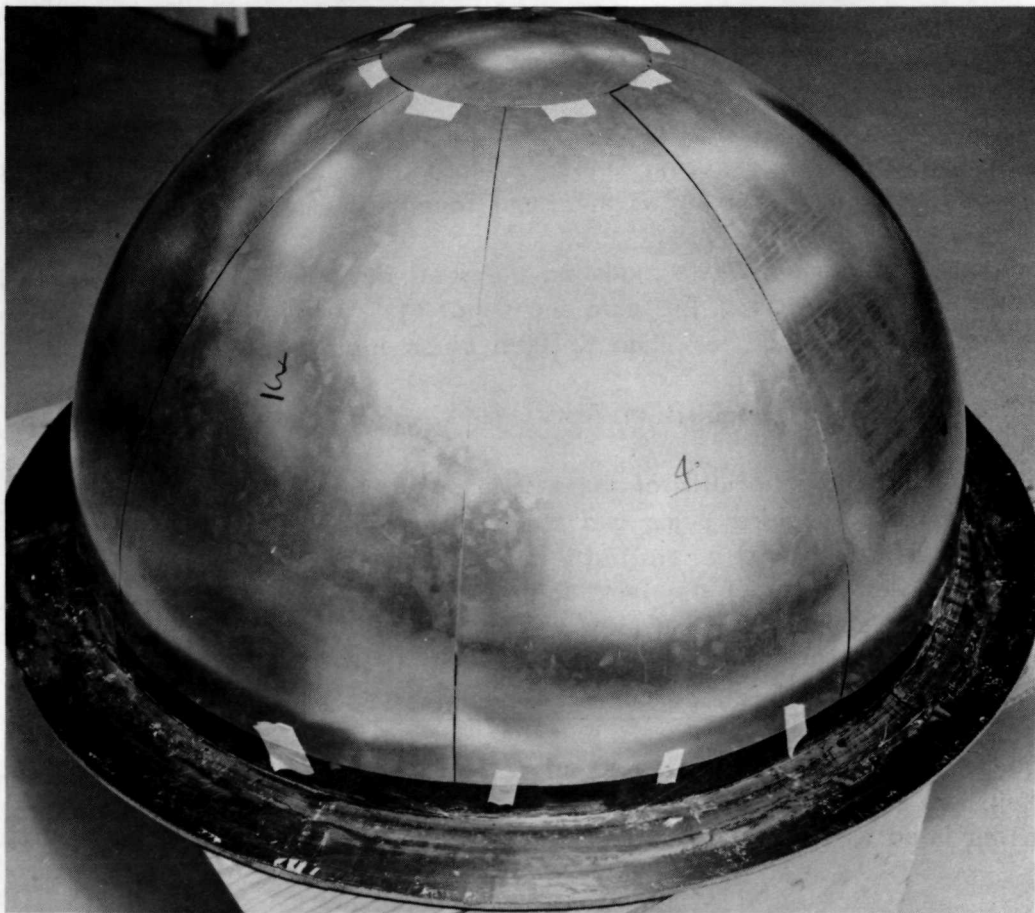
Figure 4-56 is a schematic of the test setup. Vacuum pumpdown equipment consisted of a roughing pump and a diffusion pump. The mass spectrometer used for leak detection had a sensitivity of approximately 2×10^{-10} atm cc helium/sec. The sensitivity of the leak detector was determined at the time of test by a calibrated standard helium leak of approximately 1×10^{-6} atm cc helium/sec.

Vacuum Acquisition Test on Bonded Gore Inner Face Skin Head

The head was welded to the base plate and this assembly bolted to the pumpdown equipment as shown in Figure 4-57. Figure 4-58 shows the instrumentation used on the head.

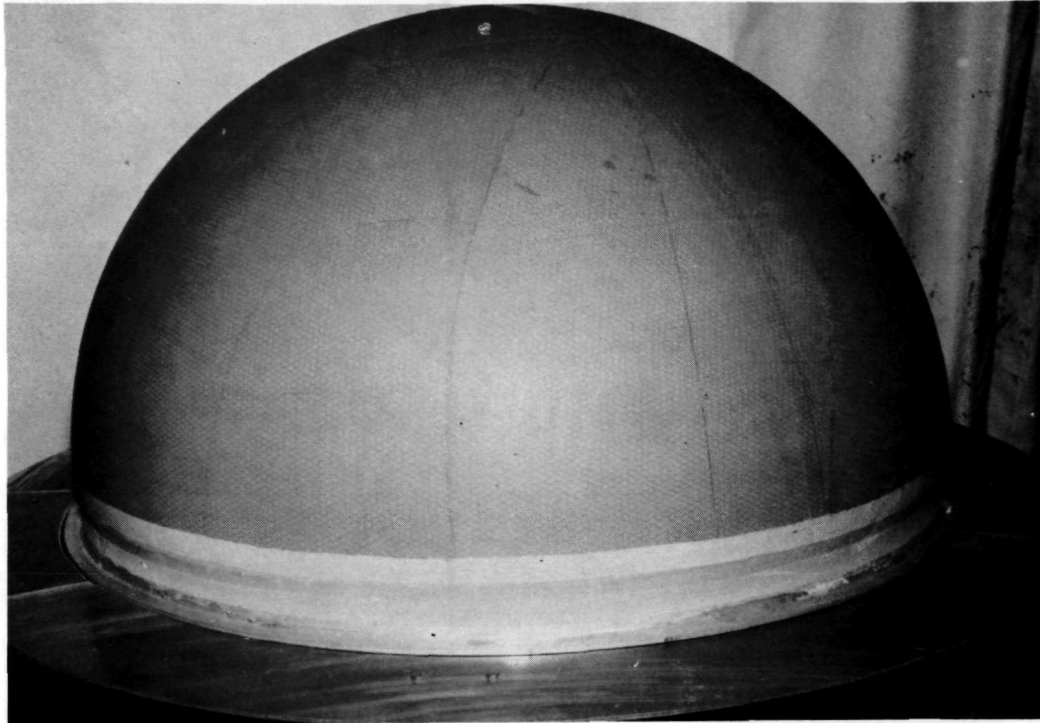


a: Trimmed Gore



b: Fitup On Mandrel

Figure 4-52: STRETCH FORMED GORES FOR 45 IN. (1.14 m) DIAMETER HEAD

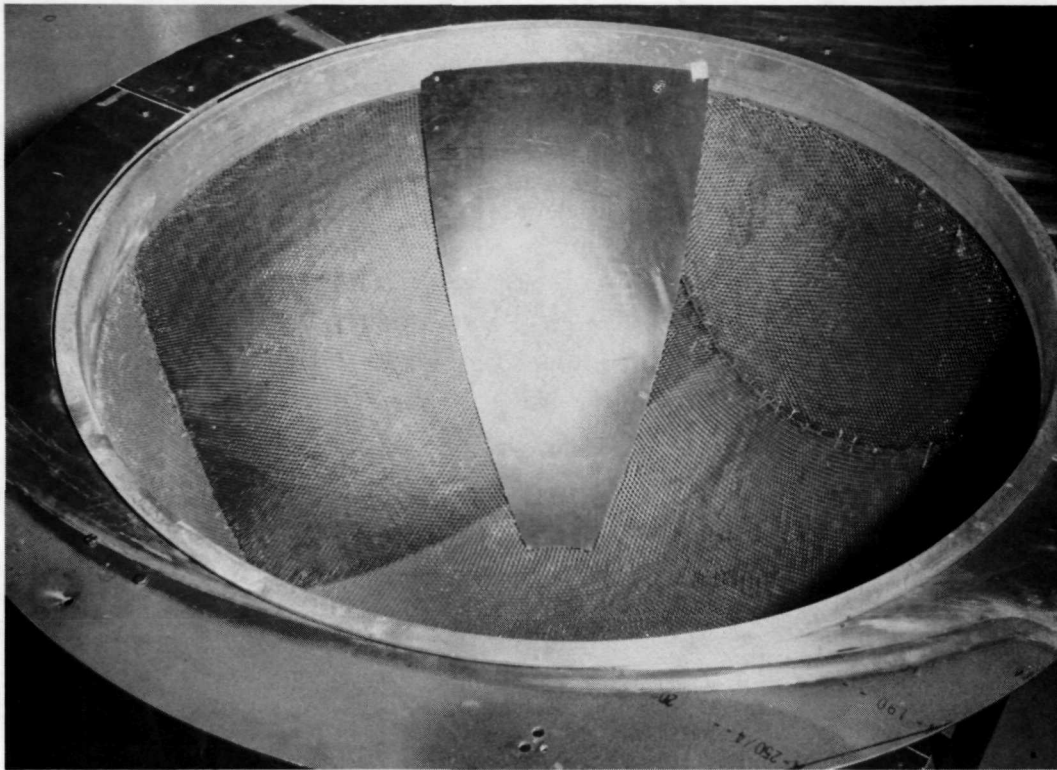


a: Outer Face Skin, Fiberglass/Epoxy Prepreg

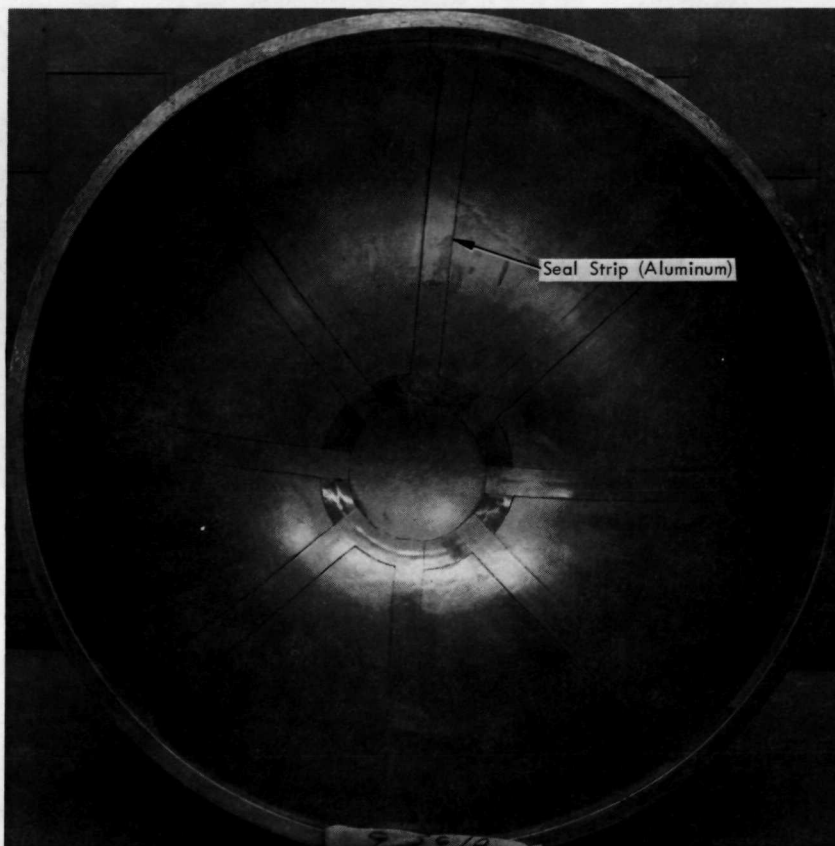


b: core

Figure 4-53: OUTER FACE SKIN AND CORE ASSEMBLY-45 IN. (1.14 m) DIAMETER HEAD



a: One Gore In Place



b: After Final Cure

Figure 4-54: ASSEMBLY OF INNER FACE SKIN TO 45 IN. (1.14 m) DIAMETER HEAD

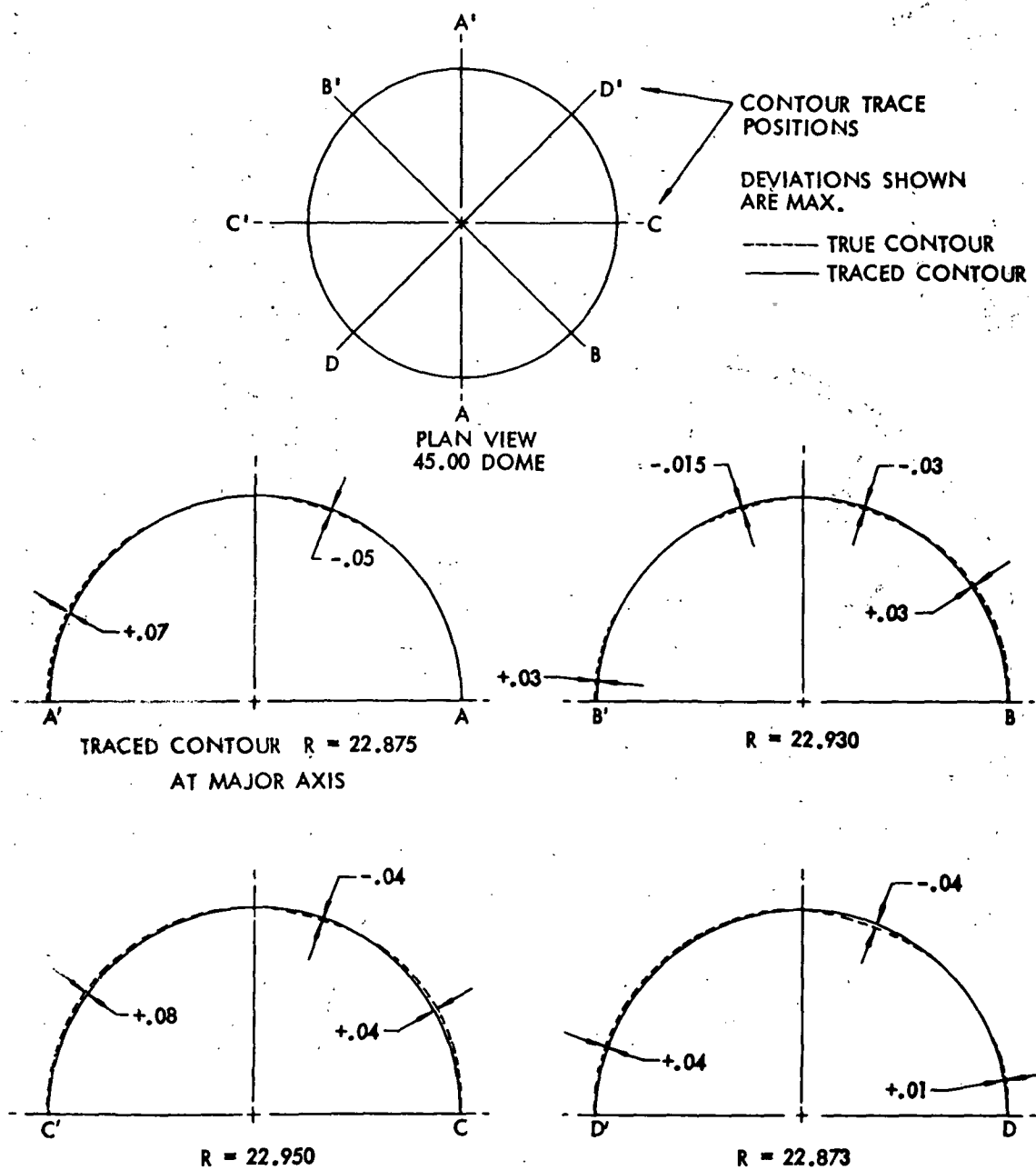


Figure 4-55: CONTOUR MEASUREMENTS - 45.0 IN. (1.14 m) DIAMETER HEAD

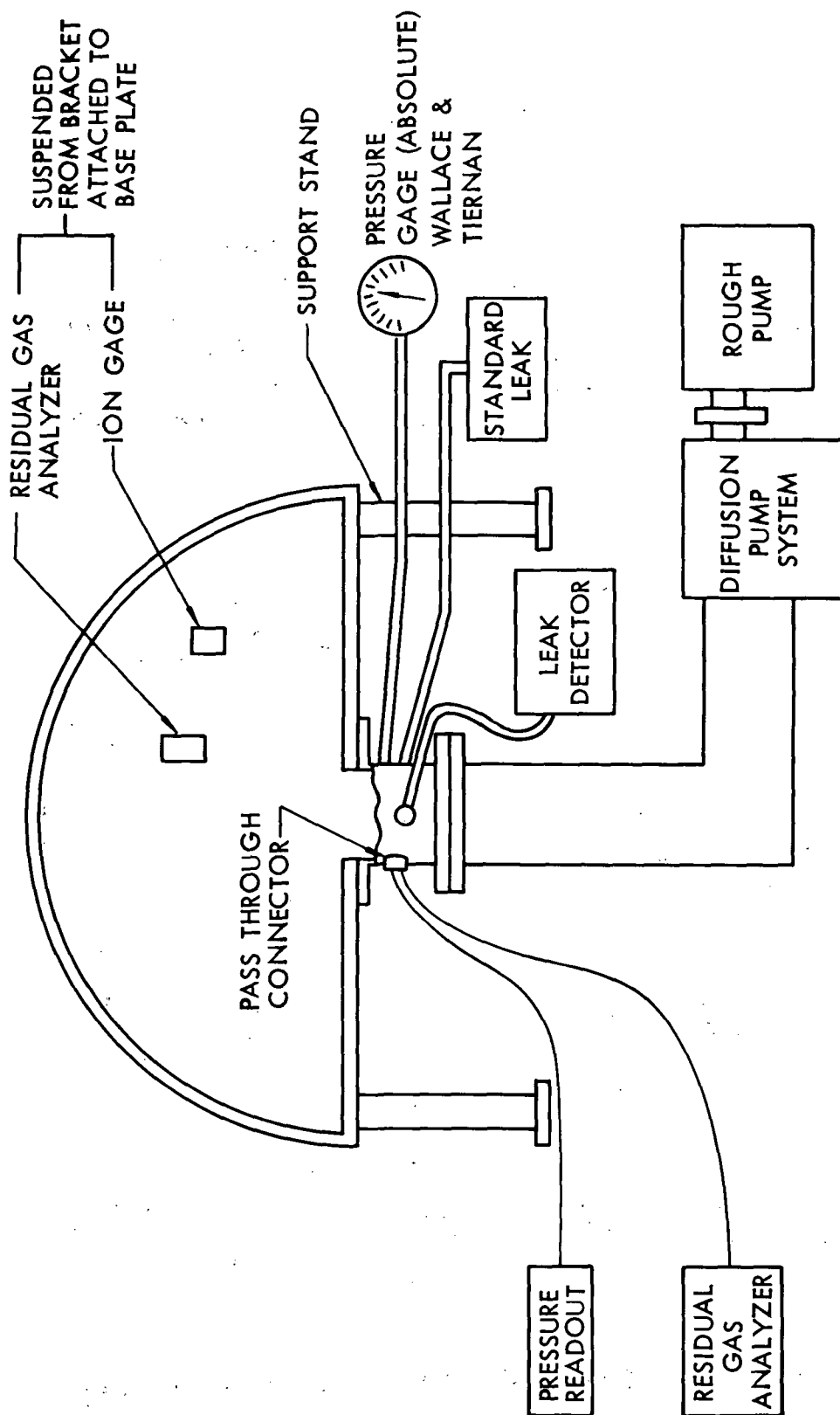


Figure 4-56: VACUUM ACQUISITION AND LEAK DETECTION TEST SETUP FOR 45.0 IN. (1.14 m) DIAMETER HEAD

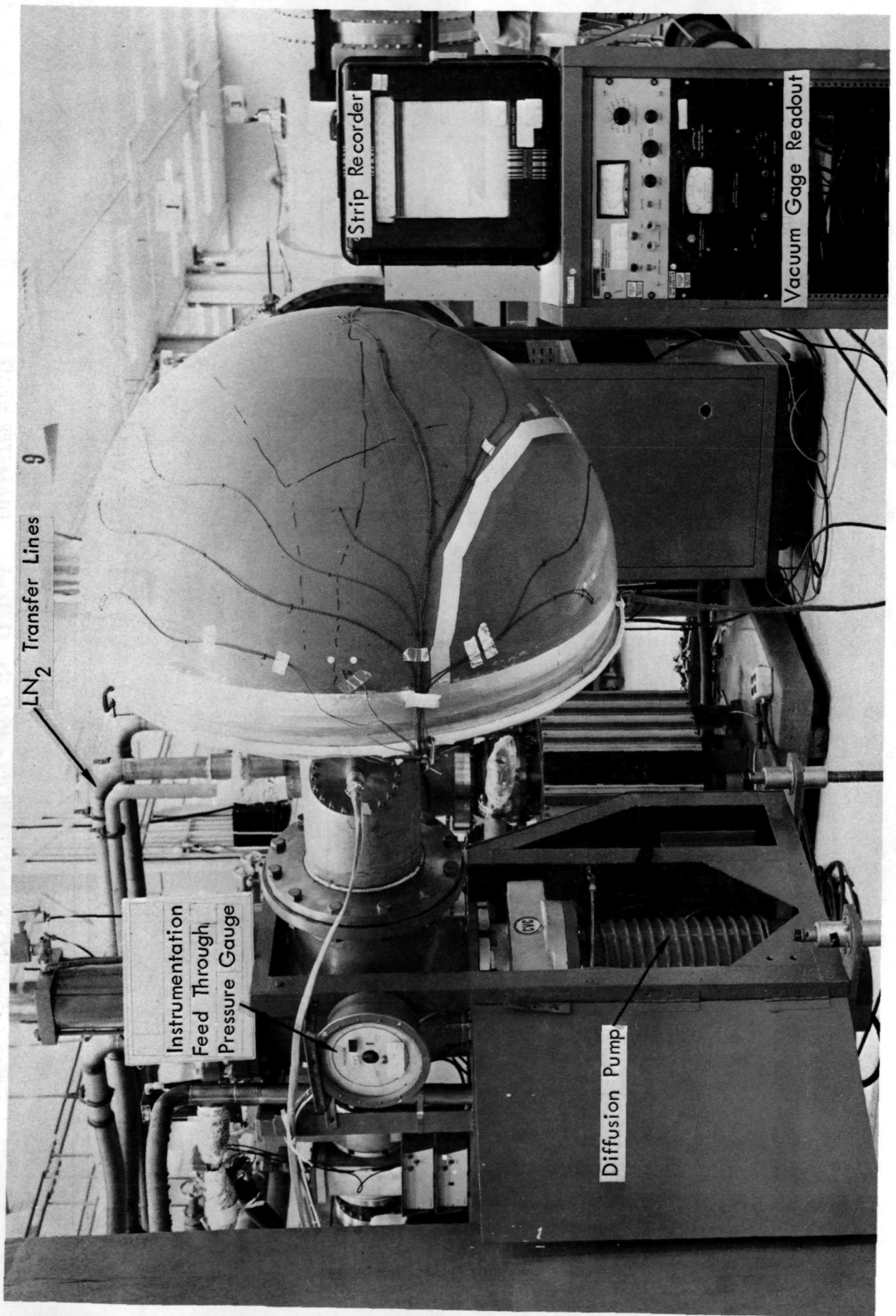


Figure 4-57: VACUUM ACQUISITION TEST ARRANGEMENT-45 IN. (1.14 m) DIAMETER HEAD

▷ FOUR EQUAL SPACES ALONG MERIDIAN

• T_1 THRU T_6 - THERMOCOUPLE

▲ E_1 THRU E_9 - ELECTRONIC DEFLECTION INSTRUMENTS (NORMAL TO SURFACE)

- S_1, S_2, S_3 - STRAIN GAGE PAIRS

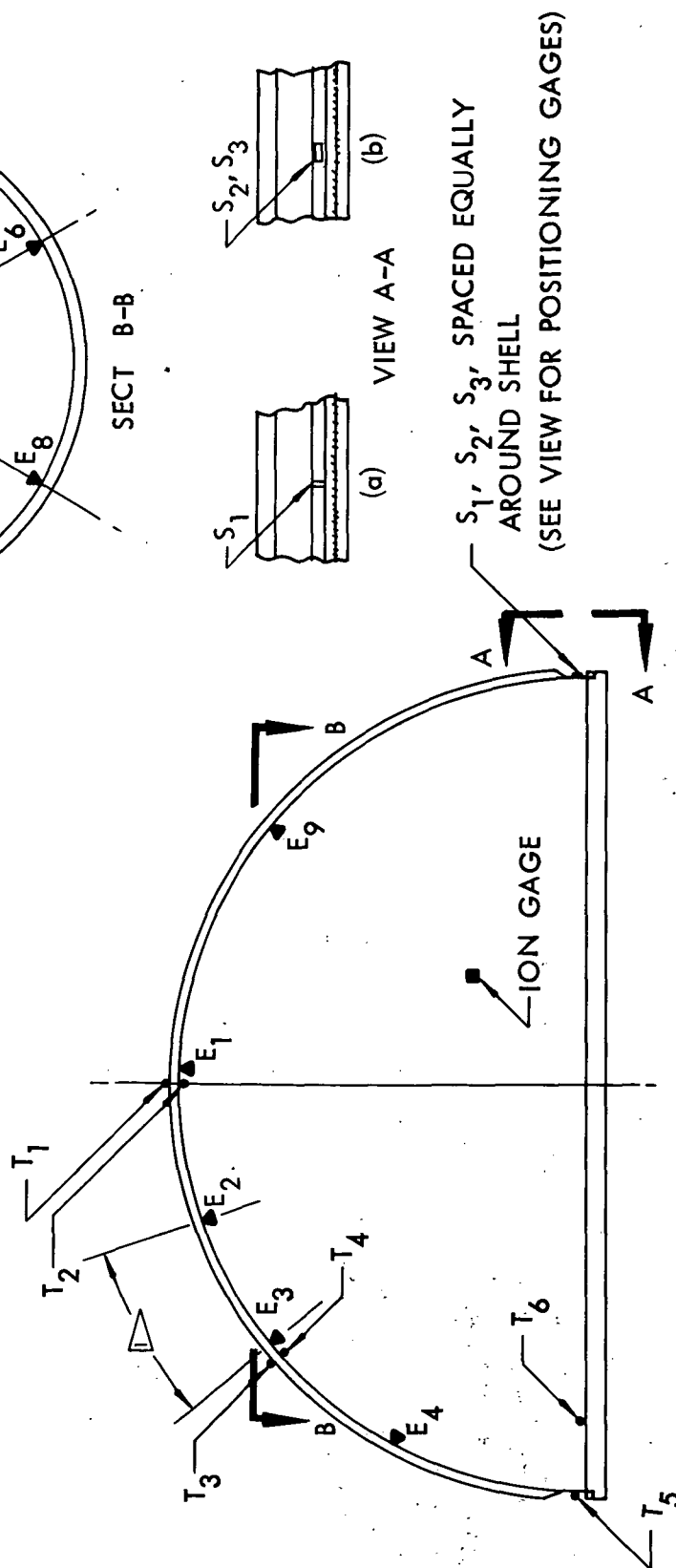


Figure 4-58: INSTRUMENTATION - 45.0 IN (1.14m) DIAMETER HEAD

F/S Buckling Prediction

During the initial pumpdown, data from the EDI's, strain gages, and vacuum pressure gage was plotted for F/S buckling prediction to assure no premature failure of the head up to 14.7 psi (102 kN/m²) external pressure. The data taken during this test indicated that failure was not imminent when an external pressure of 14.7 psi (102 kN/m²) was reached.

Acceptable Helium Leak Rate

From the thermal studies in Section 3.3 it was concluded that for the near spherical LH₂ tank (L/D = 0.09) a leakage rate of 1×10^{-6} lb/hr of nitrogen was acceptable.

Assuming free molecular flow the equivalent leak rate of helium is

$$\sqrt{\frac{M_{N_2}}{M_{He}}} \times N_2 \text{ leak rate (Reference 18)}$$

where M is the molecular mass, then the helium leak rate equivalent to 1×10^{-6} lb/hr of nitrogen is

$$\frac{1 \times 10^{-6} \times 2.832 \times 10^4}{3.6 \times 10^3 \times 0.0724} \times \sqrt{\frac{28.02}{4.003}} \\ = 2.9 \times 10^{-4} \text{ atm cc helium/sec.}$$

The ratio of the surface area of the near spherical LH₂ tank (Figure 3-46) to the surface area of the test head was

$$189(189 + 16) \times \frac{2}{45^2} = 38.3$$

Therefore, the maximum acceptable leak rate for the test head was

$$\frac{2.9 \times 10^{-4}}{38.3} = 7.5 \times 10^{-6} \text{ atm cc helium/sec}$$

for the LH₂ tank application.

Helium Leak Check and Repair

The preliminary leak check showed some leaks in the head to base plate weld joint. Local weld repairs sealed these leaks. The next overall check showed helium leakage through the head. Zone type leak checking was used to isolate leakage areas. The girth ring to shell attachment region was a major suspected leak area. There were indications of other leaks nearby which may have been caused by helium migrating to the major leak area.

The lowest pressure reached at this time with the vacuum pumping system operating was 5.9×10^{-4} torr (79 mN/m²).

A vacuum leak sealant was applied on the external shell surface in the suspected areas. During application and curing of the sealant, a vacuum was maintained in the head/base plate cavity to encourage migration of the sealant into the leak areas. Several coats proved ineffective. The head and base plate assembly was then removed from the vacuum pumping system and the sealant was applied to the inner skin joints. Access to the inner skin was through the 18.0 in. (0.46 m) diameter hole in the base plate. The leak check of the head after this repair showed no improvement in vacuum tightness. The vacuum level remained in the 5.9×10^{-4} torr (79 mN/m²) range with the pumps operating.

The base plate was then removed for better visibility and accessibility to the inner surface of the shell. Approximately 14.0 in. (0.36 m) of the inner skin was debonded at the girth ring. The reason for the debonding was not established. However, ring expansion and high local heating during welding were possible causes. The area was repaired by inserting chilled HT 424 adhesive between the skin and the girth ring. The adhesive was cured by applying vacuum bag pressure and heat locally. The leak sealant previously applied to the inner skin was removed since this proved to be a significant outgassing source.

Selective leak checking in the repaired area continued to show a high leak rate. A mylar/aluminum/mylar (MAM) sealing strip was bonded to the interior of the head in this region. The MAM (Zeroperm) sealing strip reduced the leak rate sufficiently to warrant reassembly of the base plate to allow a more conclusive vacuum test. Rather than welding the girth ring to base plate joint, these items instead were clamped together and sealed with a MAM (Zeroperm) strip covered with polyester adhesive.

With head remounted on the vacuum pumpdown equipment (Figure 4-57) the flange to base plate joint was leak checked. Two areas in this joint showed excessive leakage and were repaired using the MAM sealing strip and the polyester adhesive. The leak rate of this joint was then found to be no greater than the measurement system sensitivity of 5.6×10^{-9} atm cc helium/sec.

The head, exclusive of the flange to base plate joint, was then bagged and the bag filled with helium. For approximately 80 seconds, there was negligible movement of the leak detector needle. After 19 minutes the leak detector showed a leak rate of 9.6×10^{-5} atm cc helium/sec and was rising.

The test was stopped. The MAM (Zeroperm) strip and polyester adhesive were used to seal around the circumference with region joining the head to the girth ring which was a suspected leak area. The two core joint areas were also covered with the MAM strip as a possible prevention of helium channeling along these seams. The head was then leak checked in selected areas. Two areas representing a total of approximately one-quarter of the head surface area showed a maximum leak rate of 6.2×10^{-6} atm cc helium/sec on the detector after 8 minutes and held steady for 2 minutes. These areas excluded the girth ring to head joint area and the core seam areas. This test indicated that a portion of the head was reasonably leak tight.

The head, exclusive of the girth ring to head joint and the base plate to flange joint, was then bagged and the bag filled with helium. After 58 minutes the detector showed a leak rate of 2.8×10^{-4} atm cc helium/sec and rising.

The lowest pressure reached at this time with the vacuum pumping system operating was 3.6×10^{-5} torr (4.7 mN/m^2).

Vacuum testing on this head was discontinued since the leak rate was in excess of the maximum acceptable for the LH₂ tank application. The repairs produced an improvement in the leak tightness of the head as evidenced by the decrease in the dynamic pressure of the head to base plate cavity from 5.9×10^{-4} torr (79 mN/m^2) to 3.6×10^{-5} torr (4.7 mN/m^2).

Vacuum Acquisition Test on Spun and Chem-Milled Inner Face Skin Head

Helium Leak Check

After mounting the head to the vacuum pumpdown equipment, the head to base plate joint was bagged and the bag filled with helium. A leak rate of 5.6×10^{-7} atm cc helium/sec was measured.

Next, the bag was installed on the head exclusive of the head to base plate joint and the bag filled with helium. A leak rate of 1.48×10^{-7} atm cc helium/sec was measured.

With vacuum pumps operating, the vacuum level reached the 2.8×10^{-7} torr (0.37 mN/m^2) range. Since the vacuum level had not stabilized throughout the test it appeared that it would continue to improve with continued pumping. This was an indication of a relatively tight vacuum system.

Residual Gas Analysis

Figure 4-59 shows the results of the residual gas analysis using a partial pressure gage. These results can be compared with the typical partial pressure presentation from the user's manual (Reference 19) shown in Figure 4-60. It can be seen that a large amount of water vapor contaminated the system. This is expected in an unbaked vacuum system.

Pressure Decay

The results of the pressure decay test on a vacuum system consisting of the 45 in. (114 cm) diameter head and the pumpdown line to the shut-off valve is plotted in Figure 4-61. At the start of the test the pressure in the vacuum system was 2.8×10^{-7} torr (0.37 mN/m^2). Immediately after closing the shut-off valve, the pressure rose to 1.2×10^{-5} torr (16 mN/m^2). This pressure rise was caused by exposing a new surface (the valve's) to the vacuum system.

From Figure 4-61 it can be seen that the vacuum decay rate for the head with the system was 0.019 micron/min.

The vacuum decay rate of the system exclusive of the head was determined by removing the head from the system and blanking off the flange. The system (without the head) was then pumped down, the shut-off valve closed and the pressure decay rate established. This decay rate was found to be 0.0312 micron/min.

The pump adaptor leak rate which can be subtracted from the decay rate of the head with system shown in Figure 4-61 is ratioed from the two volumes concerned:

$$\begin{aligned}\text{head plus pump adaptor} &= 27,090 \text{ in}^3 \quad (.44 \text{ m}^3) \\ \text{pump adaptor} &= 3,290 \text{ in}^3 \quad (.05 \text{ m}^3)\end{aligned}$$

The subtractable decay rate is

$$\frac{0.0312 \times 3290}{27090} = 0.004 \text{ micron/min}$$

Therefore the decay rate of the head

$$\begin{aligned}&= 0.019 - 0.004 \\ &= 0.015 \text{ micron/min.}\end{aligned}$$

However, the head decay rate of 0.015 micron/min is higher than the decay expected to result from the total measured leak rate of 7.08×10^{-7} atm cc helium/sec. The discrepancy of the actual versus the expected vacuum decay rate appears

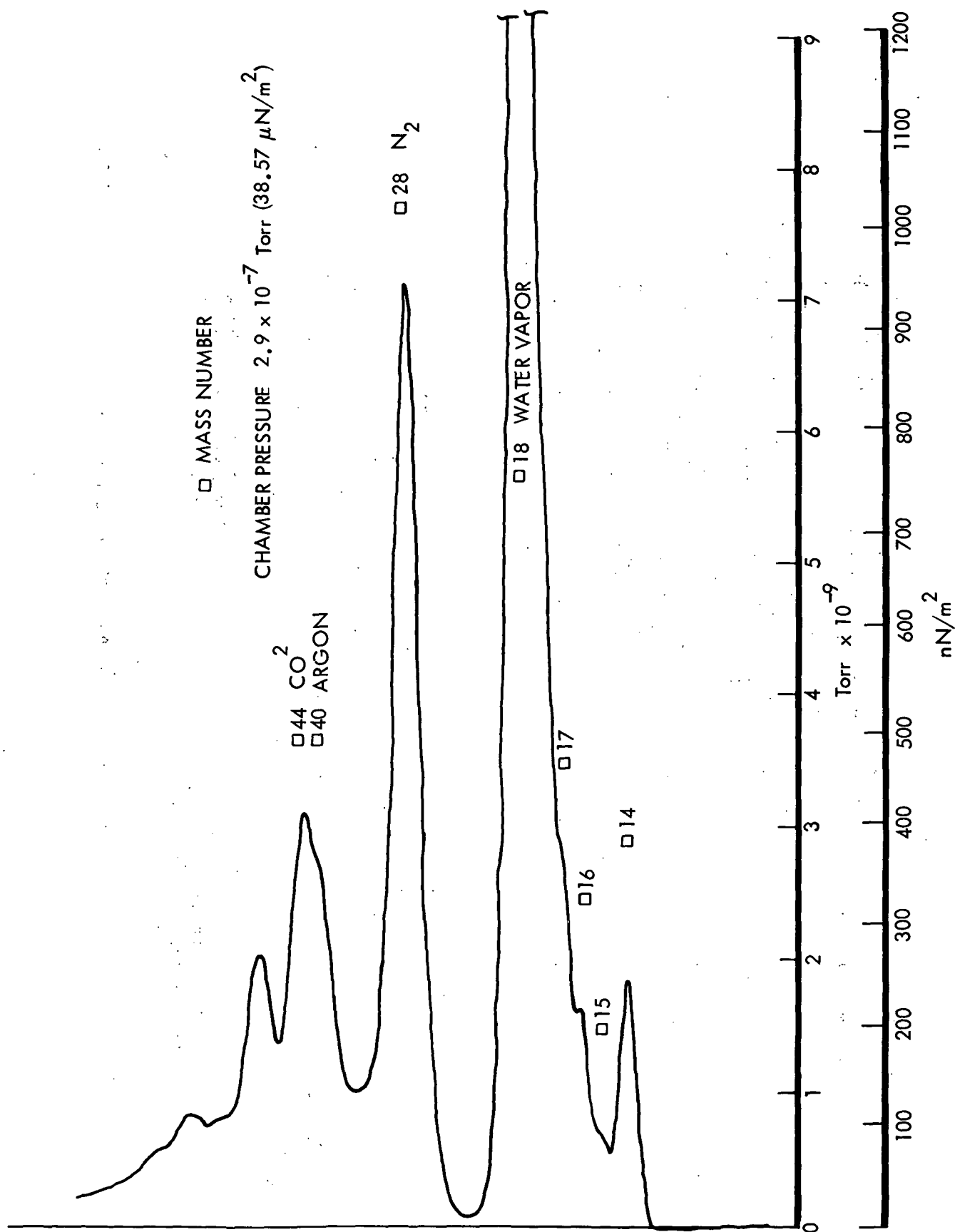
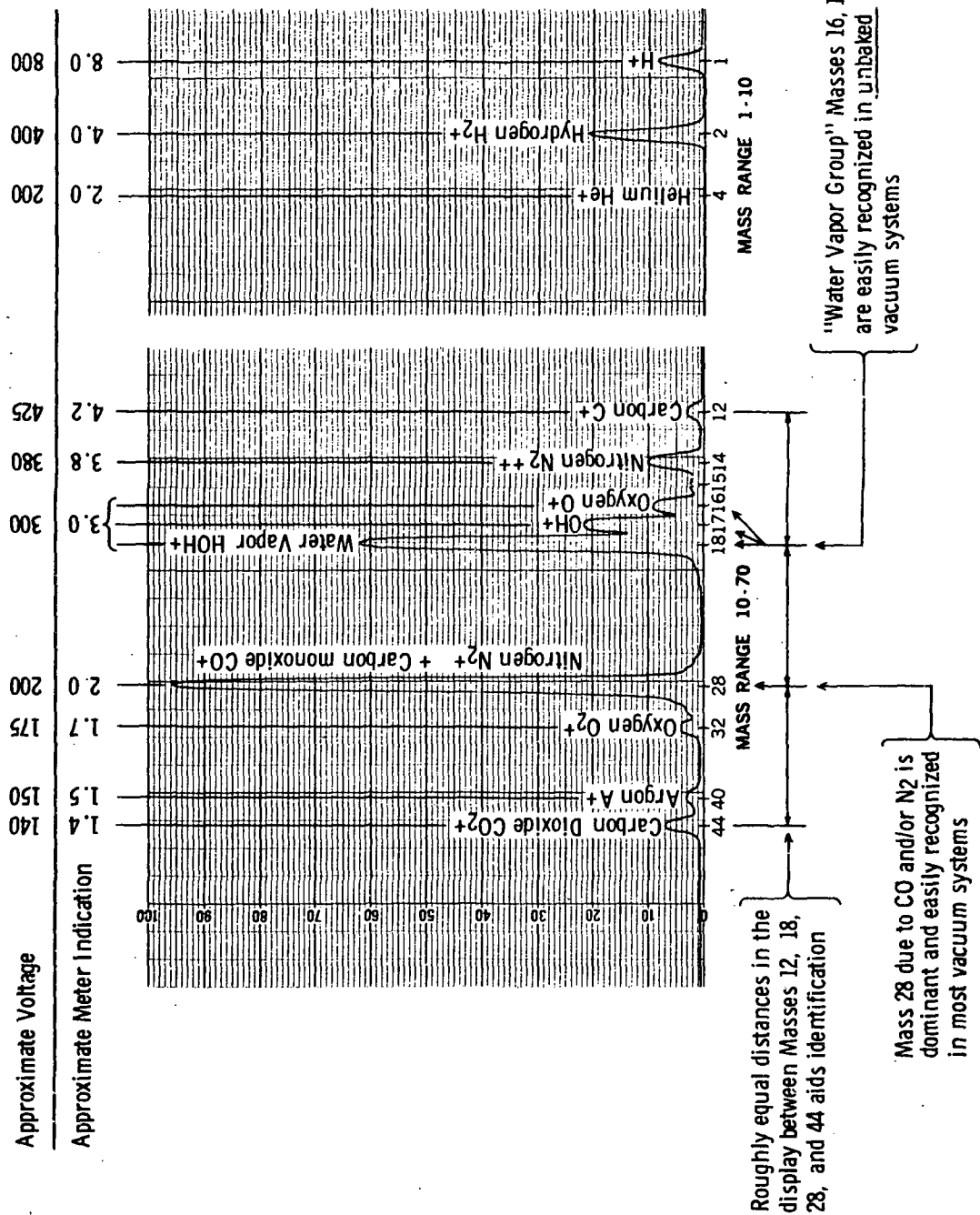


Figure 4-59: RESULTS OF PARTIAL PRESSURE MEASUREMENTS-45.0 in (1.14m) DIAMETER HEAD



Reference 19

Figure 4-60: TYPICAL PARTIAL PRESSURE PRESENTATION

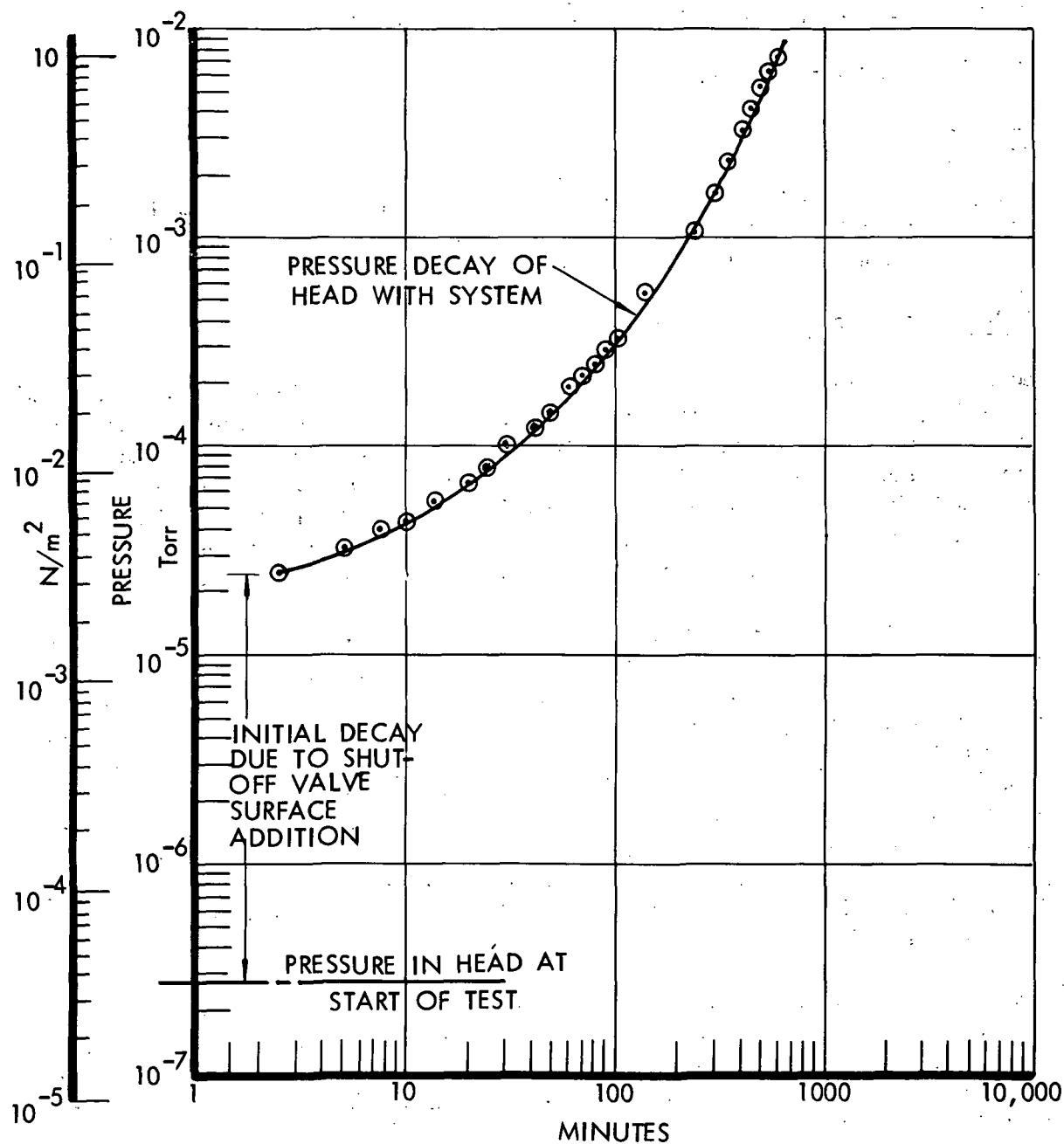


Figure 4-61: PRESSURE DECAY OF 45 INCH (1.14 m) DIAMETER HEAD WITH SYSTEM

to result from the water vapor contamination indicated by the gas analysis in Figure 4-59. It is expected that the decay rate of this system would substantially improve by preconditioning with heat and vacuum.

4.4.4 External Pressure Test

Purpose

The purpose of this test was to determine the buckling strength of the 45 in. (1.14 m) diameter head with the bonded gore inner skin, its mode of failure, and obtain data for the F/S Buckling Prediction Method.

Instrumentation and Test Procedure

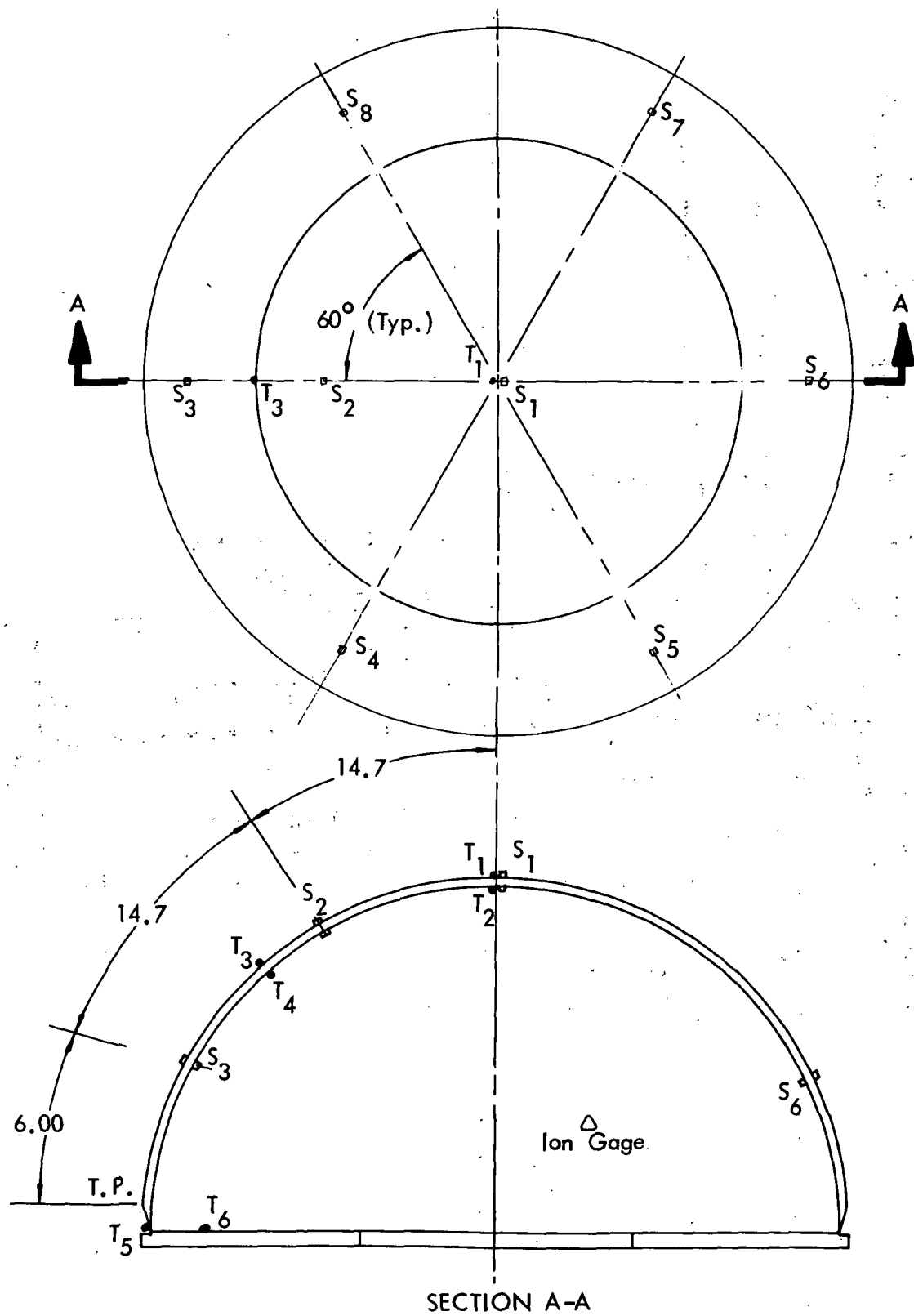
Based on the BOSOR 3 analysis for general instability and the experience obtained by testing the 8 ft. (2.44 m) head, eight pairs of strain gages were selected as the instrumentation for the 45 in. (1.14 m) diameter head test. The location of the eight pairs is shown in Figure 4-62. The "A" gages were located on the outer skin (fiberglass); the "B" gages were located on the inner skin (aluminum). The individual strains were recorded at various test pressures and the A and B pairs were used to compute the F/S values for bending during the test.

The test fixture was the same one described in Section 4.3.3. The test procedure was to slowly increase the external pressure without stopping to record the F/S data. This was attempted to eliminate some of the scatter recorded in the 8-foot (2.44 m) head tests. During the test selected values of the F/S data were plotted to identify the active areas of the head and to predict the probable buckling pressure in advance of the critical load. The F/S data were calculated by dividing the external pressure, F , by the difference in the A and B strains for each pair of gages. This calculation was performed by the data acquisition system. The results were displayed "testside" on a television monitor. This procedure appeared to yield acceptable F/S data without the large scatter observed in the 8 ft. (2.44 m) head F/S data.

One irregularity was observed during the test. Some of the F/S data became erratic at an external pressure of 25-28 psi (172-193 kN/m²). As a safeguard the pressure was reduced and the gage values examined. There were no definite signs of impending failure so the monotonic load was restarted at 0 psi and continued to the failure pressure of 49.1 psi (339 kN/m²).

4.4.5 Discussion of Results

The most active set of gages during the test were S8A and S8B. The F/S data for these are plotted in Figure 4-63. The first load cycle to 28 psi (193 kN/m²) is labeled Load Cycle No. 1. The second load cycle did not repeat the same F/S data up to 28 psi (193 kN/m²). Apparently the initial load cycle is not repeatable.



- T_1 Thru T_6 - Thermocouple
- ◻ S_1 Thru S_8 - Strain Gage Pairs

Figure 4-62: INSTRUMENTATION - 45 INCH (1.14 m)
DIAMETER HEAD

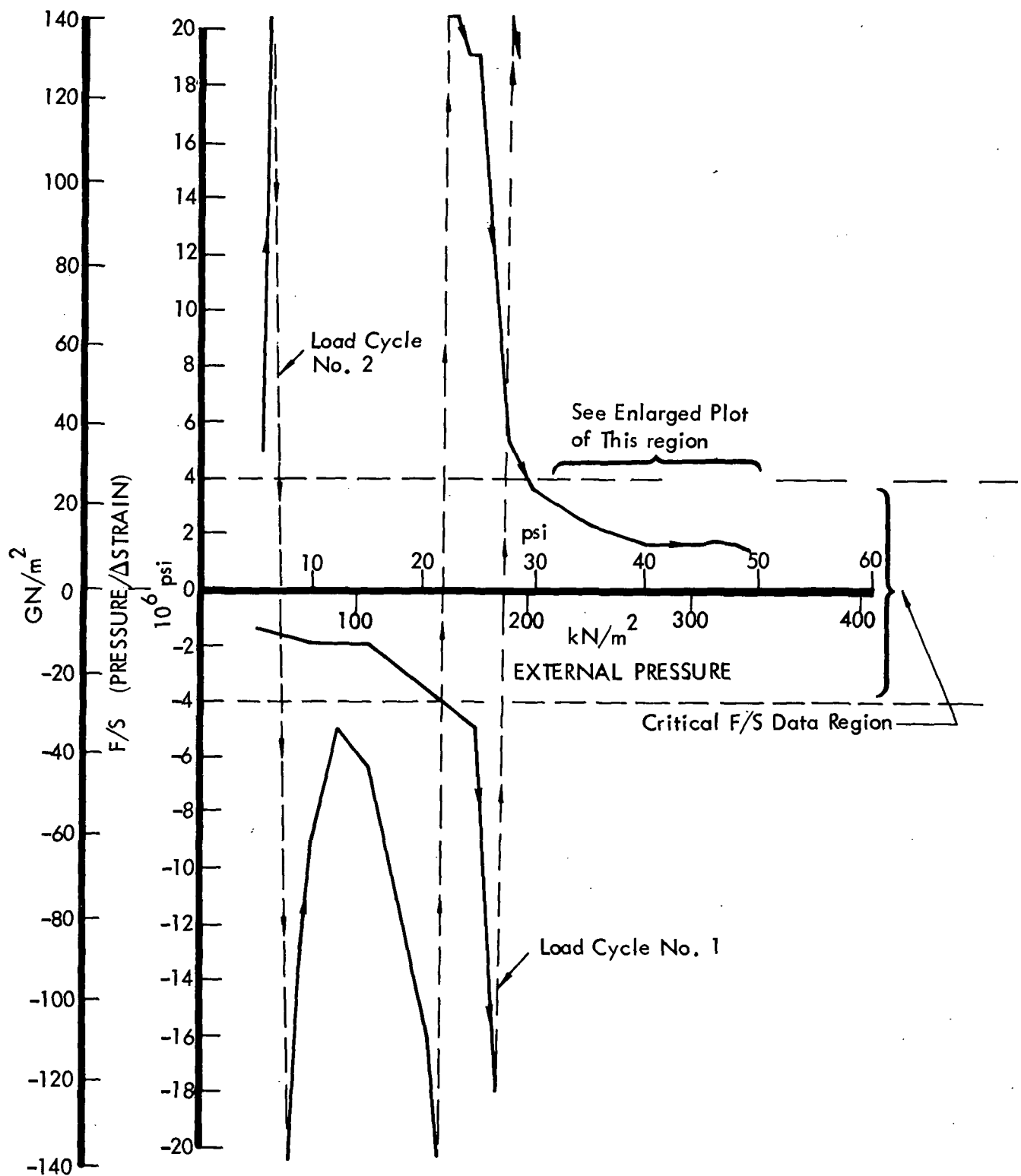


Figure 4-63: F/S vs. EXTERNAL PRESSURE FOR STRAIN GAGE SET 8.

From previous test data and an examination of the individual gage values, the F/S data from 0-30 psi (0 - 207 kN/m²) were not critical although there were large changes in the values. At approximately 30 psi (207 kN/m²) the F/S data began to show a definite trend and the values were in the critical region of ± 400 . This region of the data is plotted on an enlarged scale in Figure 4-64. Not all of the data shown were plotted during the test. The in-test prediction of "monitor" displayed data predicted a buckling pressure of 46-47 psi (318-324 kN/m²) when the external pressure load was 32-36 psi (221-249 kN/m²). The sharp downward turn in the F/S data at 48 psia (332 kN/m²) would normally call for stopping the test. In this case loading continued to measure the buckling load. The shape of the F/S plot from 38-48 psia (262-332 kN/m²) has been observed in other buckling tests. That is, there is often a linear portion of the F/S curve (32-36 psi) (221-249 kN/m²) which predicts the ultimate load. This is followed by an apparently stable portion of the curve (36-48 psi) (249-332 kN/m²) which is followed by a sudden and final drop in the load capacity.

Figures 4-65 and 4-66 are photographs of the failed head. The failure began at the bonded joint between gages S8 and S3. The inside doubler for this joint is clearly visible in Figure 4-66b. Since gage set 8 was nearest this joint it registered the F/S activity. Figure 4-67 is a sketch of the joint cross section. The eccentricity of the circumferential membrane load caused this lap joint to fail. The design of this joint was more than adequate for the ultimate design pressure of 20.6 psi (142 kN/m²).

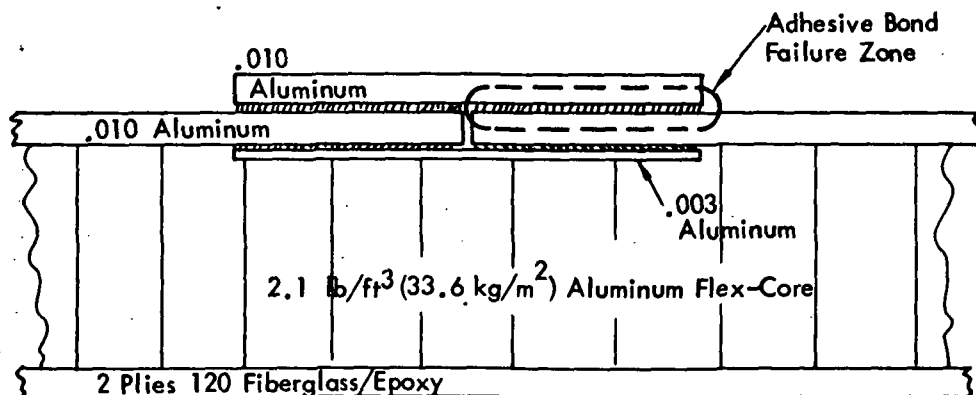


Figure 4-67: FAILURE ZONE OF 45 INCH (1.14m) DIAMETER HEAD

This failure mode introduced the possibility of a local instability mode of failure that should be considered in future F/S tests. This can be handled by plotting F/ϵ vs F for the shell. " ϵ " is the axial or circumferential strain. This was done for gages 8A and 8B after the test. The results are plotted in Figure 4-68. Prediction of failure on this plot requires a definition of the critical local buckling strain. Normally this is accomplished experimentally by testing short

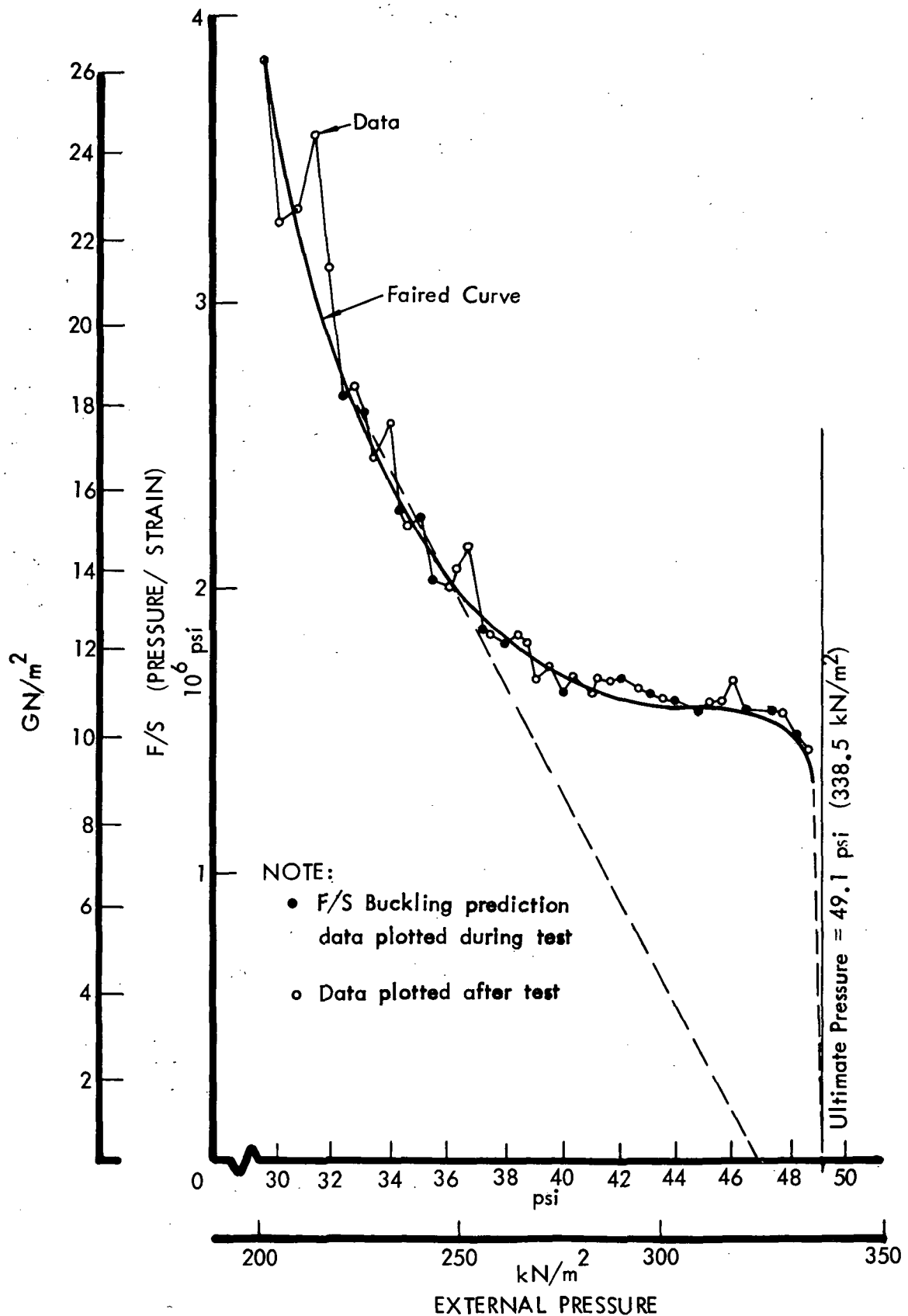


Figure 4-64: ENLARGED PLOT OF THE 30-50 psi REGION OF THE F/S VS. EXTERNAL PRESSURE FOR STRAIN GAGE SET 8.

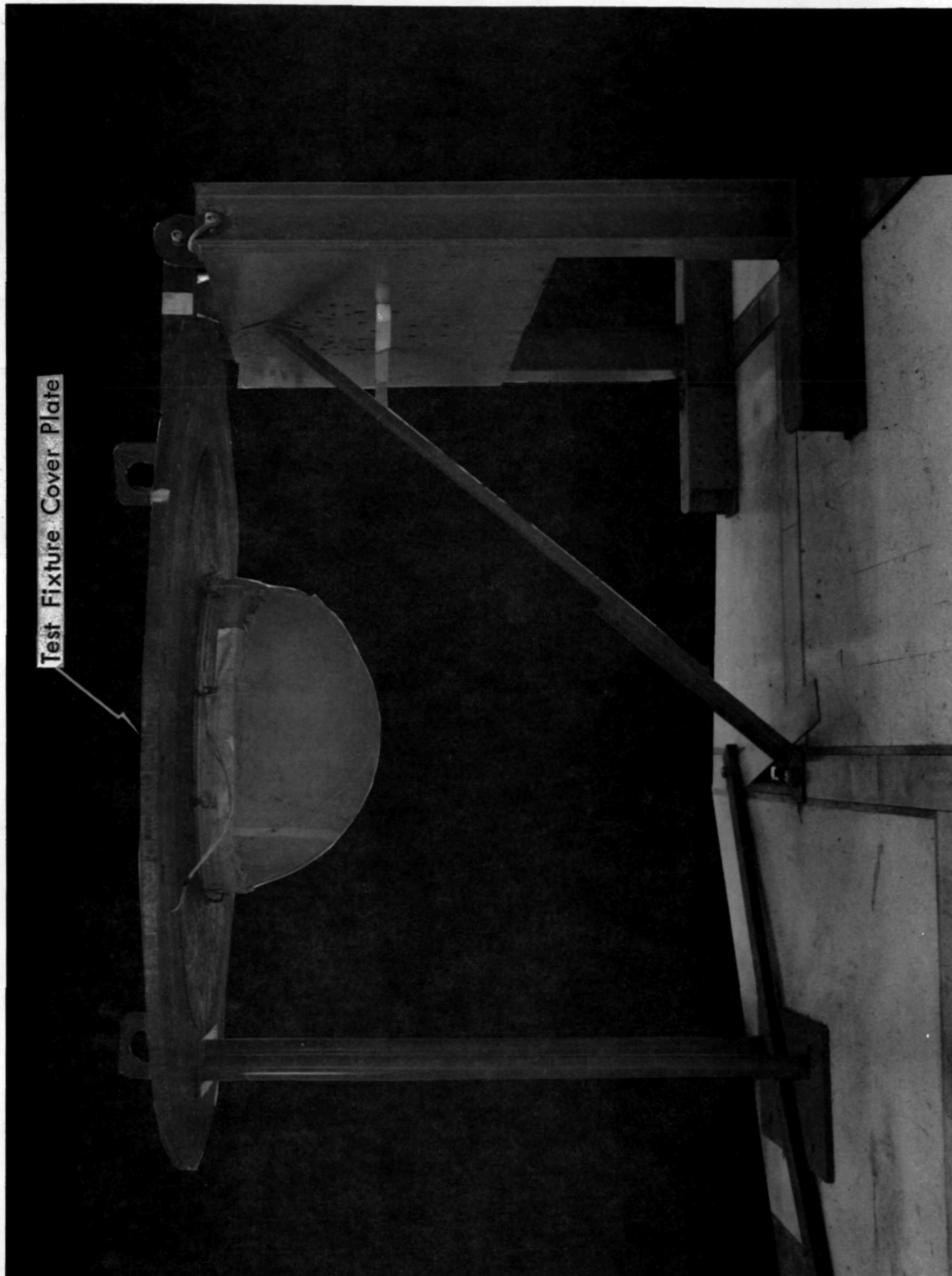
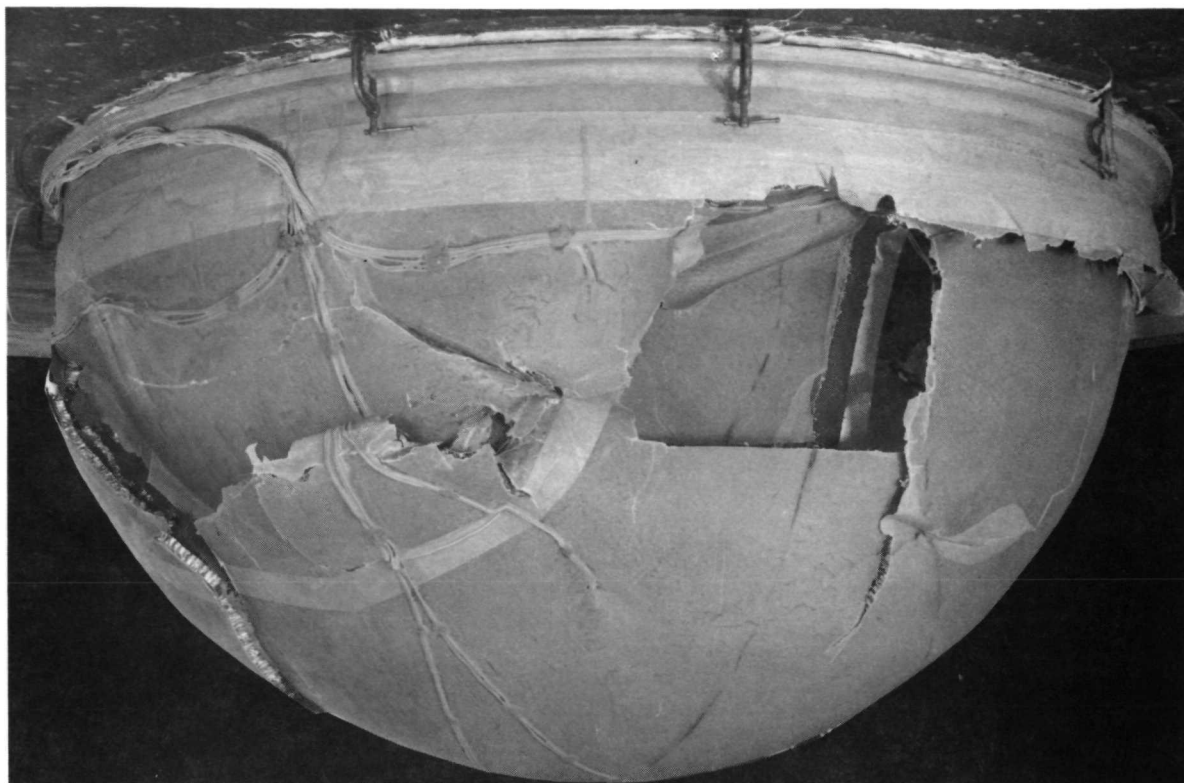
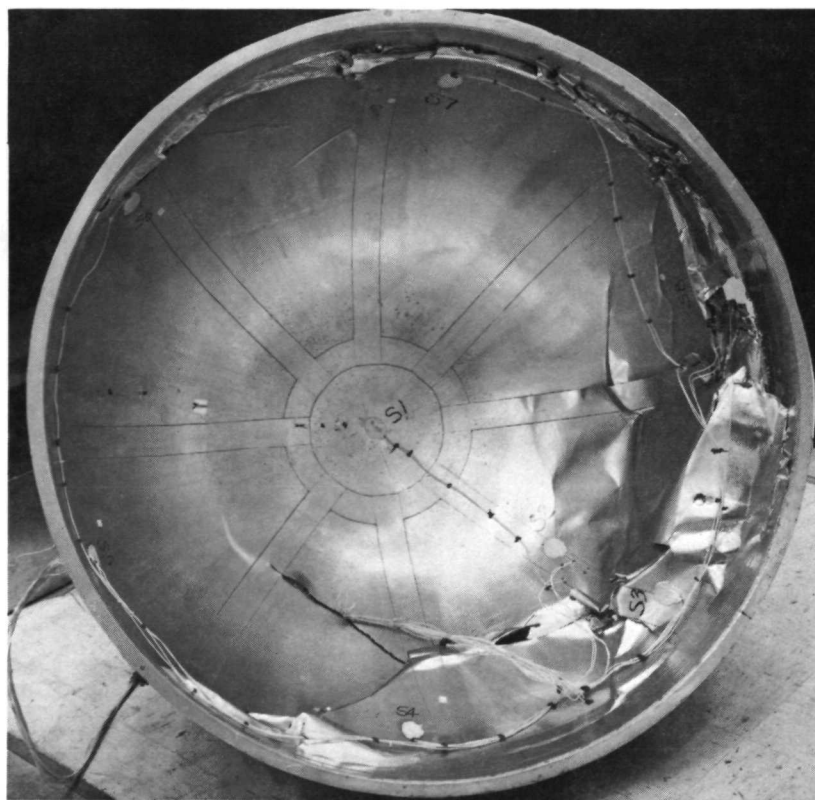


Figure 4-65: 45 INCH (1.14 m) DIAMETER HEAD ASSEMBLY AFTER EXTERNAL PRESSURE TEST



a: View on outer surface



b: View on inner surface

Figure 4-66: 45 INCH (1.14 m) DIAMETER HEAD ASSEMBLY AFTER EXTERNAL PRESSURE TEST

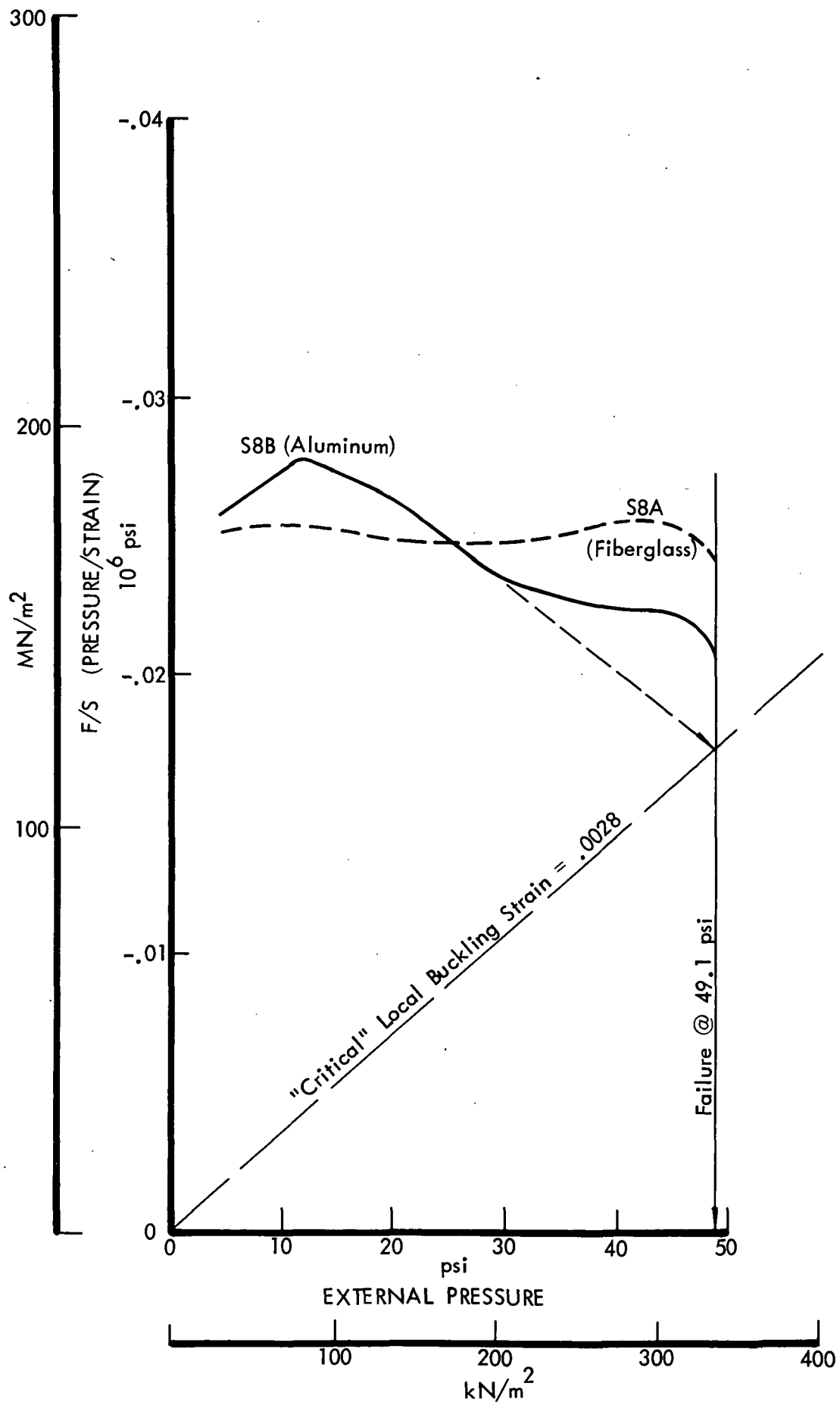


Figure 4-68: F/ϵ VS F FOR SG8A and B

compression specimens. For the 45 in. (1.14 m) head this was "estimated" after the test by connecting F/ϵ prediction line with the failure pressure line. Note in Figure 4-68 that the F/ϵ curve for S8A is fairly level with increasing pressure while the S8B curve falls sharply from 10-30 psi (69-207 kN/m²). This indicates that the aluminum face of the sandwich is the critical one and would provide a good buckling prediction when used with the correct critical strain line.

5.0 DATA EVALUATION

One of the objectives of the external pressure tests was to determine knock-down factors representative of the sandwich construction suitable for vacuum jackets. It should be noted that knock-down factors are identified with a specific analysis method. Knock-down factors for one analysis method (e.g., OPTRAN) would not apply to another method (e.g. BOSOR3). For this reason, the factors for OPTRAN and BOSOR3 are considered separately.

5.1 Knock-Down Factors for OPTRAN

The 8 ft. (2.44 m) diameter head was designed using OPTRAN with a knock-down factor of 0.41 for a 50 percent probability of the design not failing. The analysis treated the ellipsoidal head as a hemisphere with a radius of 65 in. (1.63 m). Using the OPTRAN buckling equations with the room temperature material properties, the theoretical buckling strength was 54.5 psi (376 kN/m²) external pressure. The first test experimental pressure was 14.8 psi (102 kN/m²), plus 1.7 psi (12 kN/m²) hydrostatic pressure so that the knock-down factor was calculated as,

$$\begin{aligned}\gamma_D &= \text{experimental pressure/theoretical pressure} \\ \gamma_D &= 16.5/54.5 = 0.30\end{aligned}$$

However, the test head contained a large dent and this knock-down factor must be identified with that type of imperfection.

The second test was performed on the repaired and reinforced 8 ft. (2.44 m) head. The reinforcement consisted of adding two additional plies of Style 120 fiberglass/epoxy prepreg to the polar cap. As a result, the theoretical strength, using the OPTRAN method, increased to 72.6 psi (501 kN/m²). The experimental pressure for the second test was 23.8 psi (164 kN/m²), plus 1.7 psi (12 kN/m²) hydrostatic pressure so that the knock-down factor was calculated as,

$$\gamma_D = 26.5/72.6 = 0.37$$

This increase in the experimental knock-down factor is due to the removal of the dent from the test head. The reinforcement was accounted for in the increased theoretical strength.

The second test knock-down factor agrees with the data reported by Sullins, et al, in Reference 10. Their recommended knock-down factor was 0.35. The knock-down factor used was 0.41 based on Boeing statistical data. It appears

that the sandwich construction used on the 8 ft. (2.44 m) diameter head is near the quality level predicted by the Boeing statistical data and the Sullins reported sandwich data.

The 45 in. (1.14 m) diameter head was designed with OPTRAN using very conservative knock-down factors for a 99 percent probability of not failing. The 50 percent probability Boeing knock-down factor for that design was 0.47. The knock-down factor recommended by Sullins et al was 0.2. The room temperature, OPTRAN theoretical strength was 148 psi (1040 kN/m²) external pressure. The experimental external pressure was 49.1 psi (337 kN/m²). Since the head did not fail by general instability, the knock-down factor for OPTRAN is greater than

$$\gamma_D > 49.1 \text{ psi} / 148 \text{ psi} > 0.33$$

The experimental knock-down factors for these three tests are plotted with the data of Reference 10 in Figure 5-1. The experimental knock-down factors for the OPTRAN analysis are less than the Boeing 50 percent probability prediction values and about equal to the design values recommended by Sullins, et al in Reference 10. It cannot be determined, on the basis of the small amount of test data, if this is a statistically significant difference.

A general conclusion can be drawn from these results concerning the applicability of the OPTRAN analysis method, and the Boeing statistical basis knock-down factors used in the Shell Trade Studies.

The 45 in. (1.14 m) diameter head was designed for a 20.6 psi (142 kN/m²) external pressure at 350°F (450°K) with a design probability of 99 percent. The head was tested at 70°F (294°K). The equivalent design pressure at that temperature was 24.2 psi (166 kN/m²). The test head failed at 49.1 psi (337 kN/m²) or 203 percent of the design ultimate pressure. This was a conservative but standard design procedure for a vacuum jacket. In statistical terms the test value is well within the range of expected values and there is no reason to doubt the applicability of the statistical factors based on one test.

The 8 ft. (2.44 m) diameter head was designed for a 20.6 psi (142 kN/m²) external pressure at 70°F (294°K) with a design probability of 50 percent. The head was tested at 70°F (294°K) to 24.8 psi (170 kN/m²) or 120 percent of the design ultimate pressure. The test value is well within the range of expected values and there is no reason to doubt the statistical validity of the knock-down factors.

The conclusion is that the OPTRAN analysis method coupled with the Boeing statistical based knock-down factors produce acceptable vacuum jacket designs within the limitations of the assumptions used in the analysis.

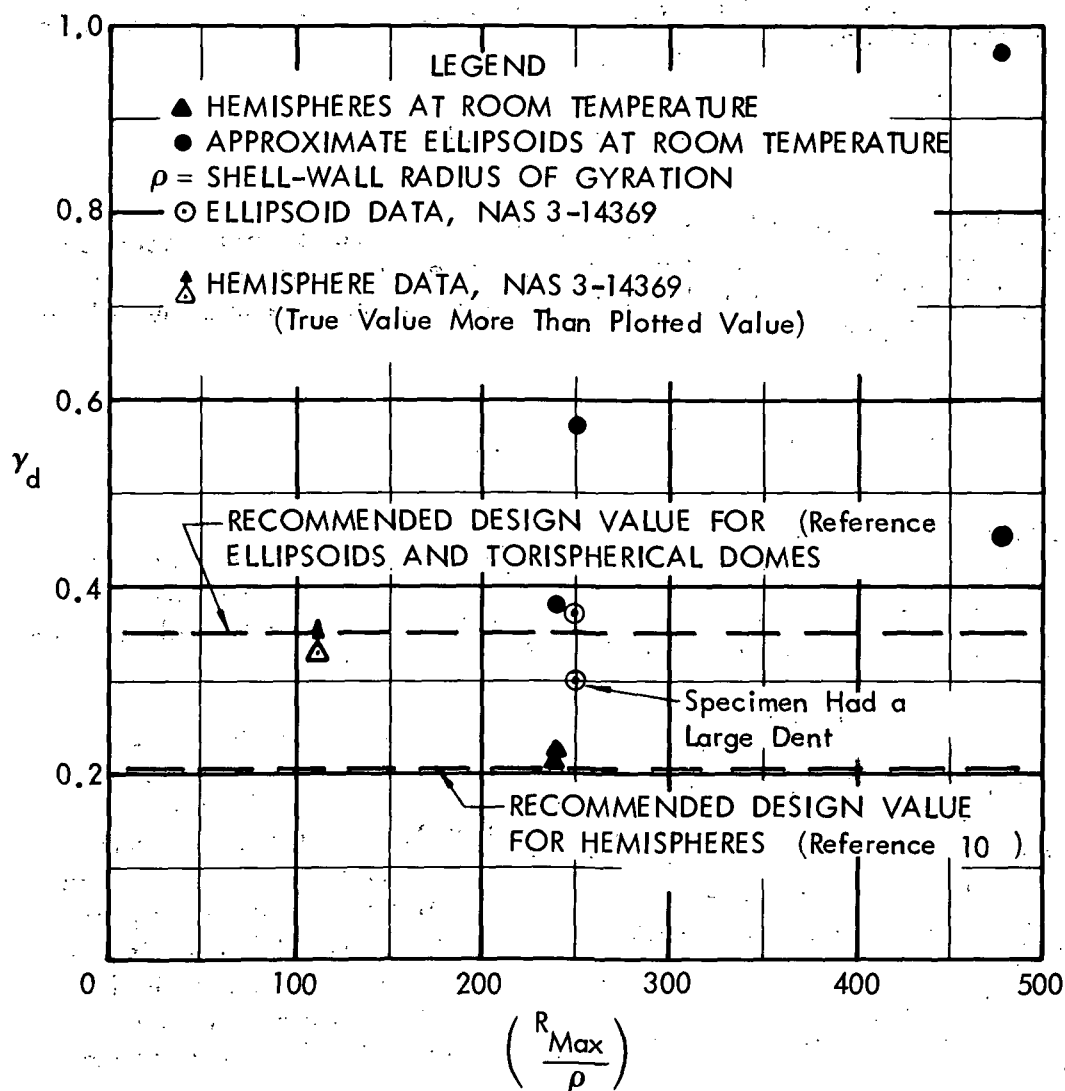


Figure 5-1: KNOCK-DOWN FACTOR γ_d FOR SANDWICH DOMES
SUBJECTED TO UNIFORM EXTERNAL PRESSURE
(Reference 10)

5.2 Knock-Down Factors for BOSOR3

The 8 ft. (2.44 m) diameter head was analyzed with the BOSOR3 method to determine the effect of the ellipsoidal shape and the edge fixity of the test fixture on the shell buckling strength. The predicted theoretical strength was 77.7 psi (534 kN/m²). The experimental strength of the first test was 14.8 psi (102 kN/m²) + 1.7 psi (12 kN/m²) hydrostatic pressure so that the knock-down factor for the dented head was

$$\gamma_D = 16.5 \text{ psi} / 77.7 \text{ psi} = 0.21$$

The head was repaired and reinforced so that the theoretical strength increased to 95.7 psi (688 kN/m²). The experimental pressure for the second test was 24.8 psi (170 kN/m²) + 1.7 psi (12 kN/m²) hydrostatic pressure and the calculated knock-down factor was

$$\gamma_D = 26.5 \text{ psi} / 95.7 \text{ psi} = 0.28$$

Both tests had the same failure mode, axisymmetric buckling, as predicted by BOSOR3. The failures occurred in the critically stressed region predicted by BOSOR3. Therefore, these knock-down factors should be identified with axisymmetric buckling modes predicted by the BOSOR3 analysis method.

The 0.21 factor represents a rather large imperfection which was easily identifiable prior to the test. The 0.28 factor represents the imperfections present, but not readily identifiable before the start of the second test. There is no correlation between these factors and those for OPTRAN even though the same experimental head was being analyzed. The OPTRAN and BOSOR3 methods use different assumptions about the test head and reach different conclusions regarding the failure pressure and failure mode. Of the two methods BOSOR3 is more realistic for the test specimens; OPTRAN is simpler and better suited to shell trade studies at the present time. As long as the correct knock-down factor is used, both methods can reach the same conclusion within the limitations of the theoretical assumptions.

The 45 in. (1.14 m) diameter head was analyzed with the BOSOR3 method. The theoretical strength was 188 psi (1290 kN/m²). The predicted mode was 8 circumferential waves. The experimental pressure was 49.1 psi (337 kN/m²) with a local failure mode involving the adhesive bonded joints of the inner skin. Since the head did not fail in general instability, the knock-down factor for BOSOR3 is greater than

$$\gamma_D > 49.1 \text{ psi} / 188 \text{ psi} > 0.26$$

Since these are the only knock-down factor data available for BOSOR3 a general conclusion cannot be made concerning the buckling prediction method. Based on the result for the second test of the 8 ft. (2.44 m) head, it is recommended that knock-down factor of 0.28 be used with BOSOR3 until more data or a better rationale are available. This factor is a "best estimate" and should be considered to represent a 50 percent design probability. The 0.21 knock-down may be conservative unless dents of the size like those in the first test head are present in the vacuum jacket.

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6.0 CONCLUSIONS

This investigation determined and optimized the key evacuated MLI system elements which have the strongest impact on producing a reliable lightweight system for the Space Shuttle vehicle. A proposed nondestructive proof test method for use on externally pressurized shells was substantiated.

The three system elements most influential to the weight of the MLI system were determined to be

- (1) the vacuum level in the insulation annulus;
- (2) the vacuum jacket, including the girth ring(s), and
- (3) the manner of storing the cryogen during the mission - whether vented or non-vented.

The thermal performance of a MLI system relies in part on the maintenance of a vacuum level of 5×10^{-5} torr (66 mN/m^2). Three conclusions relating to the vacuum integrity of the system were drawn from the results of this program:

- (1) Organic materials, because of their high outgassing rates should not be exposed to the vacuum annulus. This leads to a further conclusion that the inner face skin of a sandwich vacuum shell should be metallic.
- (2) For a closed vacuum system (i.e., no on-board vacuum pumping or venting overboard) only extremely small quantities of the cryogen gas can be permitted to leak into the vacuum annulus.
- (3) Air leakage through the vacuum jacket can be substantially higher than H_2 gas leakage in the LH_2 system due to the cryopumping capability of the -423°F (20.4°K) pressure vessel wall. It should be recognized, however, that a procedure for pumping the cryopumped constituents out of the MLI is necessary during ground turnaround in order to prevent gradual degradation, with time, of the system's thermal performance.

The development of a lightweight vacuum jacket design has to assure structural integrity, vacuum sealing capability and manufacturing feasibility. Two conclusions regarding vacuum jacket design can be drawn from this study.

- (1) For the low and medium L/D tanks, sandwich shell construction with an aluminum alloy core is the most efficient method. The semi-rigid cylindrical shell is competitive for the high L/D tanks.

- (2) The bonded gore inner face skin offers the most promise as a cost effective, repetitive manufacturing process which will produce shells to the contour accuracy and face skin thickness requirements. The vacuum integrity of this construction method appears to rely heavily on processing and helium leak checking procedures.

The conclusion from the results of the system optimization study is that non-vented storage produces the least system weight for both cryogens for the 7, 15 and 20 day mission.

The conclusions from the results of the three external pressure tests on the sandwich shells and the evaluation of the F/S buckling prediction method are:

- (1) Strain gages placed at strategic locations are required for the F/S predictions. The use of deflection data (EDI's) to measure stiffness was inadequate for predicting strength.
- (2) Caution must be exercised when interpreting the F/S data from pairs of strain gages since small differences in strain may lead to large variations in the F/S value. This condition can be identified by noting the strain values used to calculate the F/S value.
- (3) Repair and reinforcement procedures can be used on locally damaged heads to increase the buckling strength. Part of the strength increase is due to local removal of the imperfections and part is due to the reinforcement which makes the structure less sensitive to imperfections.
- (4) The location of the F/S instrumentation is critical to the successful prediction of the critical load. Known areas of structural imperfection and highly stressed areas should be adequately instrumented to detect and predict all anticipated modes of failure. Critical strain values for local instability modes of failure should be determined in advance of the F/S test to correctly interpret the test data.
- (5) Based on the design and analysis of three sandwich head tests the F/S buckling prediction method has been shown to be a reliable method of predicting the shell buckling strength. The F/S predictions were within -6, -3, and -4 percent of the experimental buckling values.

It can be concluded from evaluation of the shell buckling test data that the OPTRAN analysis methods for sandwich heads coupled with the knock-down

factors will produce acceptable vacuum jacket designs within the limitations of the assumptions used in the analysis. The 45.0 in. (1.14 m) diameter sandwich head designed with the 99 percent probability knock-down factor failed at 203 percent of the design ultimate load. The 8 ft. (2.44 m) diameter head designed with the 50 percent probability knockdown factor, but containing a known serious defect, failed at 72 percent of the design ultimate load. Following repair and selective reinforcement the head failed at 119 percent of the design ultimate load. A design factor study showed that the weight of vacuum jacket designs is directly related to the design probability. Therefore, it is generally concluded that the approach to achieve reliable and lightweight vacuum jackets is to design them using lower than 99 probability knockdown factors and to use the nondestructive proof test technique with the F/S buckling prediction method to assure that each jacket has the required strength.

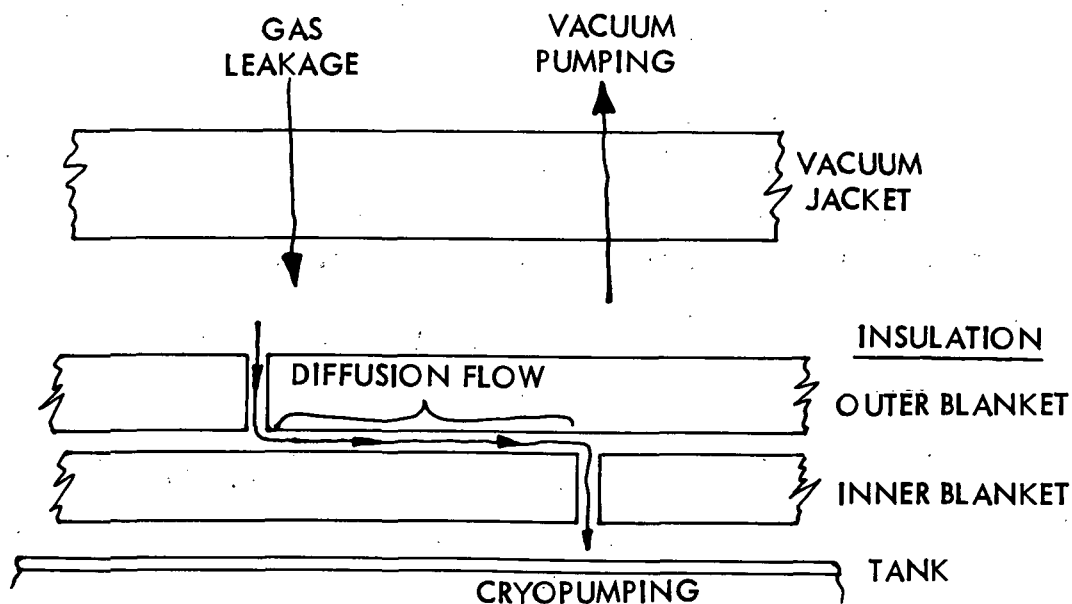
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APPENDIX A

EQUATIONS FOR CONDENSIBLE GAS FLOW SUBROUTINE

ASSUMPTIONS

1. The insulation is a two-blanket (equal thickness) layup with staggered butt joints.
2. The only resistance to gas flow is in the joint overlap; the butt joint gaps do not resist flow.
3. Gas leakage rates and pumping speed are constant.
4. Gas pressure is in the diffusion flow regime
5. Only one gas is present.



EFFECT OF TEMPERATURE GRADIENT ON DIFFUSION FLOW

The mass flow at low pressure is given by

$$m = -A \frac{\partial}{\partial x} (D\rho)$$

where

- A = cross-section area
- x = distance along flow path
- D = Diffusion coefficient
- ρ = gas density

For a channel at very low pressure

$$D = \frac{d_e}{3} \sqrt{\frac{8RT}{\pi}}$$

where d_e = effective channel diameter

R = gas constant

T = temperature

Let

$$D = D_0 \left(\frac{T}{T_0} \right)^{1/2}$$

where D_0 is measured at a reference temperature, T_0

Then

$$\dot{m} = - \frac{D_0 A}{T_0^{1/2}} \frac{\partial}{\partial x} (\rho T^{1/2})$$

$$\dot{m} = - \frac{D_0 A}{RT_0^{1/2}} \frac{\partial}{\partial x} \left(\frac{P}{T^{1/2}} \right)$$

Flow in the insulation system is based on this equation.

For volumes with no flow,

$$\frac{\partial}{\partial x} \left(\frac{P}{T^{1/2}} \right) = 0, \text{ or}$$

$$\frac{P}{T^{1/2}} = \text{constant}$$

$$M = \int_V \rho dV = A \int_x \frac{P}{RT} dx = \frac{A}{R} \left(\frac{P}{T^{1/2}} \right) \int_x \frac{dx}{T^{1/2}}$$

Solving for the constant,

$$\left(\frac{P}{T^{1/2}} \right) = \frac{MR}{A \int_x \frac{dx}{T^{1/2}}}$$

This term will be calculated for two volumes: (1) the inner blanket (subscript T) and (2) the outer blanket and unfilled annulus (subscript A). It is also used to determine gas conductivity since low pressure gas conduction is directly proportional to it.

DIFFUSION THROUGH INSULATION JOINTS

$$\begin{aligned}\dot{M}_{\text{DIFF}} &= - \frac{DA_i}{RT_0^{1/2}} \frac{\partial}{\partial x} \left(\frac{P}{T^{1/2}} \right) \\ &= \frac{DA_i}{RT_0^{1/2}} \frac{(P/T^{1/2})_A - (P/T^{1/2})_T}{\ell}\end{aligned}$$

where A_i = Total cross-section area of flow channels
in joint overlaps
 ℓ = joint overlap distance

$$(P/T^{1/2}) = \frac{MR}{A \int_x \frac{dx}{T^{1/2}}} = \frac{MR}{A \sum_n \frac{\Delta x_n}{T_n^{1/2}}}$$

The series form for $(P/T^{1/2})$ is used to correspond with the thermal analyzer nodal network.

$$\begin{aligned}(P/T^{1/2})_A &= M_A R/A \left[(L_A - L_1) / T_1^{1/2} + \frac{L_1}{10} \sum_{n=2}^6 \frac{1}{T_n^{1/2}} \right] \\ (P/T^{1/2})_T &= M_T R / \frac{AL_1}{10} \sum_{n=7}^{11} \frac{1}{T_n^{1/2}}\end{aligned}$$

EXTERNAL PUMPING

Characterize the pump by $\dot{V} = S = \text{constant}$. That is, S is the pumping speed usually given in liters/second.

$$P_A S = \dot{M}_{\text{PUMP}} \frac{RT}{A}$$

$$\dot{M}_{\text{PUMP}} = \frac{P_A S}{RT}$$

CRYOPUMPING

From Barron, CRYOGENIC SYSTEMS, p. 561-567,

$$\dot{M}_{\text{CRYO}} = \frac{(2g/\pi R)^{1/2} (P/T_1^{1/2})(P-P_U)fA_c}{\left\{P-P_U \left[1-(T_1/T_2)^{1/2}\right]\right\} + (1-f)(T_1/T_2)^{1/2} \left\{P+P_U \left[(T_2/T_1)^{1/2}-1\right]\right\}}$$

where T_1 = temperature of gas source
 T_2 = temperature of cryopumping surface
 f = sticking coefficient
 P_U = ultimate pressure of the cryopump
 A = cryopumping area

$$P_U = \frac{1}{2} P_V \left[1 + \left(\frac{T_1}{T_2}\right)^{1/2}\right]$$

where P_V = vapor pressure of the condensate at temperature T_2

Setting $f = 1$ and changing to thermal analyzer node notation,

$$\dot{M}_{\text{CRYO}} = \frac{(2g/\pi R)^{1/2} (P/T^{1/2})_T (P_T - P_U)A_c}{\left\{P_T - \frac{1}{2} P_U \left[1 - T_7/T_{12}\right]\right\}}$$

where $P_T = (P/T^{1/2})_T (T_7)$ and $P_U = \frac{1}{2} P_V \left[1 + \left(\frac{T_7}{T_{12}}\right)^{1/2}\right]$

ANNULUS AND OUTER BLANKET MASS FLOW

$$\begin{aligned}\dot{M}_A &= \dot{M}_{LEAK} - \dot{M}_{PUMP} - \dot{M}_{DIFF} \\ &= \dot{M}_{LEAK} - \frac{P_A S}{RT_1} - \frac{DA_i}{RT_0} \left[\frac{(P/T^{1/2})_A - (P/T^{1/2})_T}{\ell} \right] \\ M_A &= M_{A_{t=0}} + \int_0^t \dot{M}_A dt\end{aligned}$$

INNER BLANKET MASS FLOW

$$\begin{aligned}\dot{M}_T &= \dot{M}_{DIFF} - \dot{M}_{CRYO} \\ &= \frac{DA_i}{RT_0} \left[\frac{(P/T^{1/2})_A - (P/T^{1/2})_T}{\ell} \right] - \frac{(2g/\pi R)^{1/2} (P/T^{1/2})_T (P_T - P_U) A_c}{\left\{ P_T - \frac{1}{2} P_V [1 - T_7/T_{12}] \right\}} \\ M_T &= M_{T_{t=0}} + \int_0^t \dot{M}_T dt\end{aligned}$$

The equations in $\dot{M}_A$ and $\dot{M}_T$ are interdependent and must be solved simultaneously. To facilitate this, make the following substitutions:

$$\begin{aligned}a_1 &= \frac{DA_i}{\ell RT_0} \\ a_2 &= (P/T^{1/2})_A \\ a_3 &= (2g/\pi R)^{1/2} \left(\frac{P_T - P_U}{P_T - a_8} \right) A_c\end{aligned}$$

$$\text{where } a_8 = \frac{1}{2} P_V (1 - T_7/T_{12})$$

Note that a_3 is not strictly a constant, since P_T , P_U and a_8 can vary with time. However, this form is required to linearize the differential equations.

$$\alpha_4 = (P/T^{1/2})_T$$

$$\alpha_5 = T_1^{1/2}$$

$$\alpha_6 = T_7^{1/2}$$

$$\alpha_7 = \frac{S}{RT} \cdot \alpha_2 \cdot \alpha_5$$

Substituting,

$$(P/T^{1/2})_A = \alpha_2 M_A$$

$$P_A = T_1^{1/2} (P/T^{1/2})_A = \alpha_2 \alpha_5 M_A$$

$$(P/T^{1/2})_T = \alpha_4 M_T$$

$$P_T = T_7^{1/2} (P/T^{1/2})_T = \alpha_4 \alpha_6 M_T$$

$$\dot{M}_{\text{CRYO}} = \alpha_3 \alpha_4 M_T$$

$$\dot{M}_{\text{DIFF}} = \alpha_1 [\alpha_2 M_A - \alpha_4 M_T]$$

The differential equations become

$$\dot{M}_A = \dot{M}_{\text{LEAK}} - \alpha_7 M_A - \alpha_1 [\alpha_2 M_A - \alpha_4 M_T]$$

$$\dot{M}_T = \alpha_1 [\alpha_2 M_A - \alpha_4 M_T] - \alpha_3 \alpha_4 M_T$$

Rearranging and changing to operator form:

$$\left[D + (a_1 a_2 + a_7) \right] M_A - a_1 a_4 M_T = \dot{M}_{LEAK}$$

$$-a_1 a_2 M_A + \left[D + a_4 (a_1 + a_3) \right] M_T = 0$$

Eliminating M_A ,

$$D^2 M_T + \left[(a_1 a_2 + a_7) + a_4 (a_1 + a_3) \right] D M_T + \left[a_4 (a_1 a_2 a_3 + a_1 a_7 + a_3 a_7) \right] M_T = a_1 a_2 \dot{M}_{LEAK}$$

Let $b_1 = a_1 a_2 + a_7 + a_1 a_4 + a_3 a_4$

$b_2 = a_4 (a_1 a_2 a_3 + a_1 a_7 + a_3 a_7)$

$b_3 = a_1 a_2 \dot{M}_{LEAK}$

$$\frac{d^2 M_T}{dx^2} + b_1 \frac{dM_T}{dx} + b_2 M_T = b_3$$

Solving,

$$M_T = c_1 e^{r_1 t} + c_2 e^{r_2 t} + \frac{b_3}{b_2}$$

$$\dot{M}_T = r_1 c_1 e^{r_1 t} + r_2 c_2 e^{r_2 t}$$

$$\text{where } r_1 = \frac{-b_1 + \sqrt{b_1^2 - 4b_2}}{2}, \quad r_2 = \frac{-b_1 - \sqrt{b_1^2 - 4b_2}}{2}$$

Initial conditions

at $t=0$ $M_T = M_{T0}$

$\dot{M}_T = \dot{M}_{T0}$

Substituting and solving for constants

$$c_1 = \frac{1}{r_1 - r_2} \left[r_2 \left(M_{T0} - \frac{b_3}{b_2} \right) - \dot{M}_{T0} \right]$$

$$c_2 = \frac{\dot{M}_{T0} - r_1 c_1}{r_2}$$

M_A is obtained by solving the $\dot{M}_A$ differential equation (bottom of page 224)

$$\dot{M}_A = \frac{1}{a_1 a_2} \left[\dot{M}_T + a_4 (a_1 + a_3) M_T \right]$$

The amount cryopumped is obtained from

$$\begin{aligned} M_{\text{CRYO}} &= (M_{\text{CRYO}})_0 + \int \dot{M}_{\text{CRYO}} dt \\ &= (M_{\text{CRYO}})_0 + a_3 a_4 \left[\frac{c_1}{r_1} (e^{r_1 t} - 1) + \frac{c_2}{r_2} (e^{r_2 t} - 1) + \frac{b_3}{b_2} t \right] \end{aligned}$$

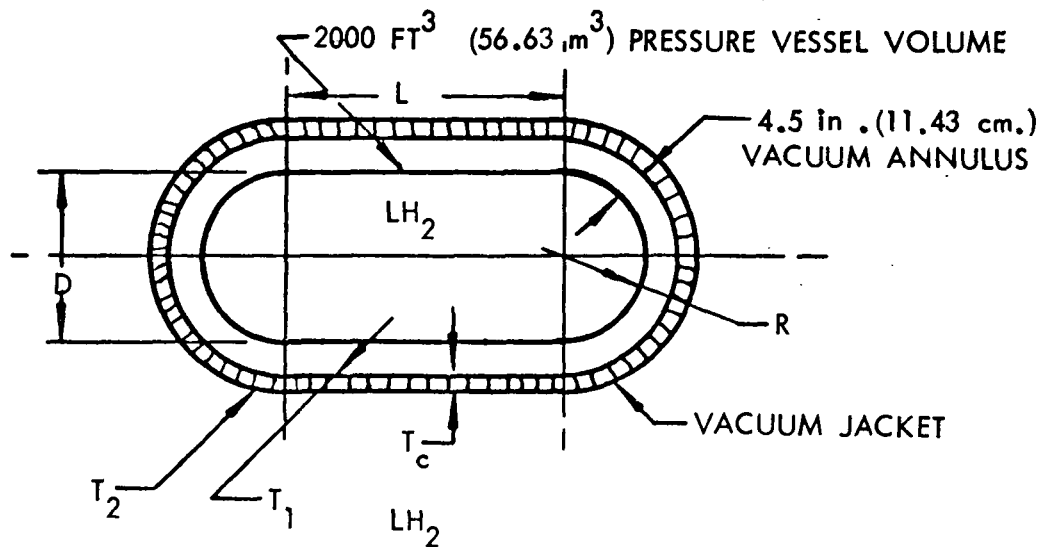
The subroutine to the BETA thermal analyzer program solves the above equations for M_A and M_T , and determines the value of $(P/T^{1/2})_A$ and $(P/T^{1/2})_T$ which are used in the conductivity equation. Analyses in this report used the above equations as shown except for the "variable" constant a_3 . As P_T approaches P_U , a_3 changes rapidly, and a stable solution could not be obtained. To overcome this difficulty, $(P_T - P_U)/(P_T - a_8)$ was set equal to zero. This is effectively saying that the vapor pressure is well below the actual pressure. As shown by the analysis results, this is not always true. However, the actual pressure in this case is low enough so that the insulation conductivity is probably not affected.

The equations are re-evaluated by the subroutine during each iteration of the main thermal analyzer program. The initial conditions are the values determined in the previous iteration.

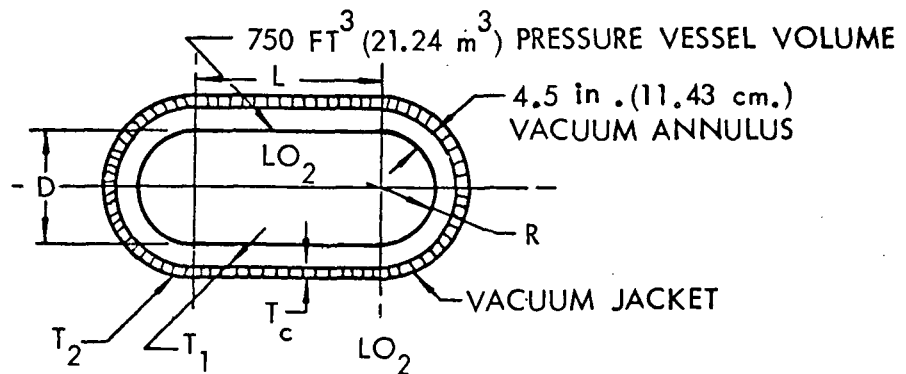
APPENDIX B

RESULTS FROM SHELL TRADE STUDIES

The detail data for the LH_2 and LO_2 tank vacuum jacket sandwich shell trades discussed in Section 3.4 are tabulated in this appendix. Figure B-1 shows the vacuum jacket geometries studied to obtain the different L/D ratios. The data are tabulated in Tables B-1 through B-7. The odd numbered tables are in inch-lb units, the even numbered tables are in SI units.



R		L		L/D
in	cm	in	cm	
36	91.44	801	2034.54	11.1
48	121.92	414	1051.56	4.31
63	160.02	193	490.22	1.53
90	228.60	16	40.64	0.09



R		L		L/D
in	cm	in	cm	
24	60.96	684	1737.36	14.3
30	76.20	418	1061.72	6.97
42	106.68	178	452.12	2.12
60	152.40	35	88.90	0.29

Figure B-1 : VACUUM JACKET GEOMETRIES FOR SANDWICH SHELL TRADE STUDIES

Table B-1: OPTIMUM SANDWICH SHELL DESIGNS
 LH2, 2000 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										CYLINDER										TOTAL JACKET WEIGHT* LB	
		FACE SKINS					CORE					FACE SKINS					CORE						
		T ₁		T ₂		T _c	CELL SIZE	WEIGHT	T ₁		T ₂		T _c	CELL SIZE	WEIGHT	T ₁		T ₂		T _c	CELL SIZE		WEIGHT
		THK	STRESS	THK	STRESS				In	ksi	In	ksi				THK	STRESS	THK	STRESS				
T ₁ ALUMINUM	11.1	.010	-19.5	.012	-20.3	0.82	1/4	3.5	.99	4.7	.020	-12.6	.040	-15.9	2.67	3/8	2.2	2012	9.52	2111			
T ₂ ALUMINUM	4.3	.012	-20.2	.015	-21.1	1.10	3/8	3.2	196	5.54	.024	-15.9	.038	-19.8	2.79	1/4	3.5	1886	12.02	1882			
T ₁ BORON/EPOXY	1.53	.014	-21.8	.018	-22.7	1.52	3/8	3.2	395	6.75	.019	-23.4	.034	-29.5	2.41	1/4	3.5	870	10.42	1265			
T ₂ BORON/EPOXY	0.09	.018	-20.5	.029	-21.4	2.06	3/8	3.2	1033	9.0	.022	-24.4	.047	-31.0	1.71	3/16	4.0	103	10.73	1136			
T ₁ TITANIUM	11.1	.011	-19.6	.017	-12.8	0.53	3/8	3.2	107	5.14	.014	-18.3	.043	-12.9	2.59	3/8	2.2	2071	9.81	2178			
T ₂ TITANIUM	4.3	.011	-27.0	.014	-17.5	0.94	3/8	3.2	203	5.78	.016	-28.9	.030	-22.9	3.0	3/8	3.2	1607	11.46	1810			
T ₁ BORON/EPOXY	1.53	.012	-27.5	.022	-17.8	1.24	3/8	3.2	406	6.96	.015	-33.9	.035	-27.2	2.17	1/4	3.5	869	10.43	1275			
T ₂ BORON/EPOXY	0.09	.010	-40.9	.022	-26.4	1.93	3/16	4.0	1021	8.91	.029	-27.2	.053	-22.2	1.12	3/16	4.0	117	12.33	1138			
T ₁ ALUMINUM	11.1	.014	-15.5	.014	-15.5	0.74	3/8	2.2	103	4.91	.027	-15.4	.032	-15.3	2.88	3/8	2.2	2263	10.70	2366			
T ₂ ALUMINUM	4.3	.018	-15.5	.018	-15.5	0.96	3/8	2.2	212	6.01	.035	-15.1	.041	-15.0	3.00	3/8	2.2	1776	12.63	1988			
T ₁ ALUMINUM	1.53	.016	-22.5	.016	-22.1	1.75	3/8	3.2	451	7.67	.028	-25.5	.030	-24.8	2.72	3/8	3.2	1004	12.00	1455			
T ₂ ALUMINUM	0.09	.030	-15.5	.034	-15.5	1.73	3/8	2.2	1129	9.86	.043	-23.1	.043	-22.9	1.48	3/16	4.0	126	13.22	1255			
T ₁ TITANIUM	11.1	.010	-21.4	.010	-21.2	0.66	3/8	3.2	118	5.63	.015	-30.0	.015	-28.8	2.70	3/8	3.2	2326	11.04	2444			
T ₂ TITANIUM	4.3	.011	-26.5	.011	-26.1	1.05	3/8	3.2	231	6.53	.019	-40.8	.019	-40.0	2.13	3/16	4.0	986	11.83	1446			
T ₁ TITANIUM	1.53	.011	-30.0	.013	-29.5	1.51	3/8	3.2	460	7.86	.016	-40.8	.019	-40.0	2.13	3/16	4.0	986	11.83	1446			
T ₂ TITANIUM	0.09	.012	-40.9	.013	-40.1	2.05	3/16	4.0	1138	9.92	.040	-29.9	.027	-29.5	2.16	3/8	3.2	152	15.90	1290			
T ₁ ALUMINUM	11.1	.013	-27.8	.012	-6.5	1.19	1/4	3.5	123	5.78	.025	-22.5	.057	-5.8	3.00	3/8	3.2	2838	13.36	2961			
T ₂ GLASS/ POLYIMIDE	4.3	.016	-28.0	.018	-6.5	1.29	3/16	4.0	252	7.10	.036	-19.2	.091	-5.0	3.00	3/8	3.2	2384	16.90	2636			
T ₁ GLASS/ POLYIMIDE	1.53	.021	-28.0	.022	-6.5	1.94	1/4	3.5	520	8.83	.038	-27.9	.053	-7.3	2.80	1/4	3.5	1211	14.46	1731			
T ₂ GLASS/ POLYIMIDE	0.09	.028	-28.0	.030	-6.5	2.75	1/4	3.5	1371	11.88	.075	-24.0	.033	-6.3	2.28	3/16	4.0	155	16.29	1526			
T ₁ TITANIUM	11.1	.010	-37.4	.012	-5.4	1.00	3/16	4.0	125	5.95	.022	-29.8	.046	-4.9	2.96	3/8	3.2	2873	13.60	2998			
T ₂ TITANIUM	4.3	.012	-40.6	.017	-5.8	1.29	3/16	4.0	256	7.23	.025	-30.6	.075	-5.1	3.00	1/4	3.5	2345	16.66	2501			
T ₁ GLASS/ POLYIMIDE	1.53	.015	-40.4	.022	-5.8	1.64	3/16	4.0	525	8.96	.029	-59.5	.038	-6.8	2.66	3/16	4.0	1232	14.77	1757			
T ₂ GLASS/ POLYIMIDE	0.09	.021	-40.6	.031	-5.8	2.30	3/16	4.0	1384	12.06	.048	-37.3	.034	-6.3	2.27	3/16	4.0	158	16.52	1542			
T ₁ ALUMINUM	11.1	.013	-28.0	.014	-5.1	1.06	3/16	4.0	127	6.00	.028	-20.9	.071	-4.2	3.00	3/8	3.2	3126	14.70	3253			
T ₂ ALUMINUM	4.3	.017	-28.0	.020	-5.1	1.54	1/4	3.5	265	7.43	.058	-15.1	.086	-3.1	3.00	3/8	3.2	2629	18.71	2894			
T ₁ GLASS/EPOXY	1.53	.022	-28.0	.025	-5.1	1.76	3/16	4.0	540	9.19	.040	-28.0	.057	-5.8	2.99	1/4	3.5	1290	15.38	1630			
T ₂ GLASS/EPOXY	0.09	.031	-28.0	.031	-5.1	2.88	1/4	3.5	1430	12.38	.077	-23.5	.038	-4.9	2.78	1/4	3.5	165	17.29	1595			
T ₁ TITANIUM	11.1	.010	-38.1	.013	-4.3	1.05	3/16	4.0	130	6.16	.023	-29.7	.058	-3.9	3.00	3/8	3.2	3077	14.53	3207			
T ₂ TITANIUM	4.3	.012	-40.6	.019	-4.6	1.33	3/16	4.0	268	7.54	.030	-27.4	.089	-3.6	3.00	3/8	3.2	2527	17.95	2795			
T ₁ GLASS/EPOXY	1.53	.015	-40.6	.025	-4.6	1.70	3/16	4.0	549	9.36	.030	-40.5	.046	-5.3	2.70	3/16	4.0	1296	15.52	1845			
T ₂ GLASS/EPOXY	0.09	.022	-40.6	.031	-4.6	2.48	3/16	4.0	1453	12.64	.049	-37.3	.035	-4.9	2.41	3/16	4.0	163	17.10	1616			

• INCLUDES 0.0006 lb/in² ADHESIVE WEIGHT

• EXCLUDING END RINGS

Table B-2: OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 56.63 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT ^o OF TWO HEADS						CYLINDER						TOTAL JACKET WEIGHT* ^o kg
		FACE SKINS					CORE					FACE SKINS					CORE					CYL. WEIGHT ^o		
		T ₁		T ₂			T _c	CELL SIZE: cm	WEIGHT kg/m ³	T ₁		T ₂			T _c	CELL SIZE: cm	WEIGHT kg/m ³	CYL. WEIGHT ^o						
		THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²	THK cm				STRESS MN/m ²	THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²				kg	kg/m ²	kg	kg/m ²			
T ₁	11.1	.025	-134	.030	-140	2.08	.64	56.0	.051	-87	.102	-109	.95	35.2	913.4	6.69	958.4							
ALUMINUM	4.3	.030	-140	.038	-145	2.79	.95	51.2	.061	-109	.097	-137	.64	56.0	765.4	8.45	854.4							
T ₂	1.53	.036	-150	.046	-157	3.86	1	179.3	.048	-161	.086	-203	.64	56.0	395	7.33	574.3							
BORON/EPOXY	0.09	.046	-141	.074	-146	5.23	1	469	.056	-168	.119	-213	.48	64.0	46.8	7.54	515.7							
T ₁	11.1	.028	-135	.043	-88	1.35	1	48.6	.036	-126	.109	-103	.95	35.2	940.2	6.90	988.8							
TITANIUM	4.3	.028	-186	.036	-121	2.39	1	92.2	.041	-199	.076	-158	.95	51.2	729.6	8.06	821.7							
T ₂	1.53	.030	-190	.056	-123	3.15	.95	51.2	.038	-234	.090	-188	.64	56.0	394.5	7.33	578.9							
BORON/EPOXY	0.09	.025	-282	.056	-182	4.90	.48	463.5	.074	-188	.135	-153	.48	64.0	53.1	8.67	516.7							
T ₁	11.1	.036	-107	.036	-107	1.89	.95	35.2	.069	-106	.081	-105	.95	35.2	1027.4	7.52	1074.2							
ALUMINUM	4.3	.046	-107	.046	-107	2.44	1	35.2	.089	-104	.104	-103	.95	35.2	8063	8.88	902.6							
T ₂	1.53	.041	-155	.041	-152	4.44	1	51.2	.071	-176	.076	-171	.95	51.2	455.8	7.84	660.6							
ALUMINUM	0.09	.076	-107	.086	-107	4.39	1	35.2	.109	-159	.109	-158	.48	64	57.2	9.30	569.8							
T ₁	11.1	.025	-148	.025	-146	1.68	1	53.4	.038	-207	.038	-198	.95	51.2	1056	7.76	1109.6							
TITANIUM	4.3	.028	-183	.028	-180	2.67	1	104.9	.048	-207	.048	-200	.95	51.2	821.3	9.06	926.2							
T ₂	1.53	.028	-207	.033	-203	3.84	.95	51.2	.041	-281	.048	-276	.48	64	447.6	8.32	656.5							
TITANIUM	0.09	.030	-282	.033	-276	5.21	.48	64	.102	-206	.069	-203	.95	51.2	69.0	11.12	585.7							
T ₁	11.1	.033	-192	.030	-45	3.02	.64	56	.064	-155	.145	-40	.95	51.2	1288.5	9.39	1344.3							
ALUMINUM	4.3	.041	-193	.046	-45	3.28	.48	64	.091	-132	.231	-34	.95	51.2	1082.3	11.88	1196.7							
T ₂ GLASS/ POLYIMIDE	1.53	.053	-193	.056	-45	4.93	.64	56	.097	-192	.135	-50	.64	56	549.8	10.17	785.9							
0.09	0.09	.071	-193	.076	-45	6.99	.64	56	.191	-165	.084	-43	.95	51.2	1304.3	9.56	1361.1							
T ₁	11.1	.025	-257	.030	-37	2.54	.48	64	.056	-205	.117	-34	.95	51.2	70.4	11.45	692.8							
TITANIUM	4.3	.030	-280	.043	-40	3.28	.48	64	.064	-211	.191	-35	.95	51.2	1064.6	11.71	1180.9							
T ₂ GLASS/ POLYIMIDE	1.53	.038	-279	.056	-40	4.17	.48	64	.074	-410	.097	-47	.64	64	559.3	10.38	797.7							
0.09	0.09	.053	-280	.079	-40	5.84	.48	64	.122	-257	.086	-43	.64	64	71.7	11.62	700							
T ₁	11.1	.033	-193	.036	-35	2.69	.48	64	.071	-144	.180	-29	.95	51.2	1419.2	10.34	1475.9							
ALUMINUM	4.3	.043	-193	.051	-35	3.91	.64	56	.147	-104	.218	-21	.95	51.2	1193.6	13.15	1313.9							
T ₂ GLASS/ EPOXY	1.53	.056	-193	.064	-35	4.47	.48	64	.102	-193	.145	-40	.64	56	585.7	10.81	830.8							
0.09	0.09	.079	-193	.079	-35	7.31	.64	56	.196	-162	.097	-34	.64	56	74.9	12.16	724.1							
T ₁	11.1	.025	-263	.033	-29	2.68	.48	64	.058	-205	.147	-27	.95	51.2	1397	10.22	1456							
TITANIUM	4.3	.030	-280	.048	-32	3.38	.48	64	.076	-226	.226	-25	.95	51.2	1147.3	12.62	1269							
T ₂	1.53	.038	-280	.064	-32	4.32	.48	64	.076	-280	.117	-37	.64	64	588.4	10.91	837.6							
GLASS/EPOXY	0.09	.056	-280	.079	-32	6.30	.48	64	.124	-257	.089	-34	.48	64	74	12.02	733.7							

^o INCLUDES 0.42 kg/m² ADHESIVE WEIGHT

* EXCLUDING END RINGS

Table B-3: OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 2000 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										CYLINDER										TOTAL JACKET WEIGHT LB																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																			
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		T ₁		T ₂			T _c	T ₁		T ₂			T _c	T ₁		T ₂			T _c	CYL WEIGHT [•]																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																					
		THK	STRESS	THK	STRESS	THK		STRESS	THK	STRESS	THK	STRESS		THK	STRESS	THK	STRESS	THK		STRESS	CYL WEIGHT [•]		CYL WEIGHT [•]																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																		
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T ₁	11.1	.021	-14.0	.010	-13.3	.615	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022	-12.4	2.66	.040	-15.6	.022

• INCLUDES 0.0006 LB/IN² ADHESIVE WEIGHT

• EXCLUDING RING

Table B-4 : OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 56.63 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT* OF TWO HEADS				CYLINDER						TOTAL* JACKET WEIGHT kg		
		FACE SKINS					CORE									CORE			CYL. WEIGHT*					
		T ₁		T ₂			T _c		T ₂							T _c		WEIGHT						
		THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS	THK	STRESS		THK	STRESS
		cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²	cm	MN/m ²		cm	MN/m ²
	11.1	.053	-96.5	.025	-91.7	1.56	35.2	.95	35.2	3.19	3.19	.102	-108	.056	-85.5	6.76	.95	35.2	922.1	6.79	965.2			
BORON/EPOXY	4.31	.025	-198	.025	-183	2.78	48	.95	48	87.6	87.6	.137	-108	.061	-84.8	7.04	.95	35.2	706.9	7.78	788.6			
	1.53	.046	-156	.038	-144	3.78	51.2	.64	51.2	180.2	180.2	.084	-201	.053	-151	5.94	.64	56.0	396.8	7.37	577.0			
ALUMINIUM	0.09	.132	-107	.036	-102	4.75	61.5	.95	51.2	451.7	451.7	.107	-200	.086	-154	3.12	.48	64.0	45.4	7.38	497.1			
	11.1	.038	-97.9	.025	-148	1.48	51.2	.95	51.2	47.2	47.2	.051	-179	.033	-205	7.06	.64	56.0	990.6	7.26	1037.8			
BORON/EPOXY	4.31	.033	-137	.025	-205	2.64	64	.64	64	94.0	94.0	.081	-153	.041	-179	7.47	.95	51.2	740.0	8.16	834.0			
	1.53	.028	-186	.025	-279	3.48	64	.48	64	181.1	181.1	.107	-173	.033	-205	5.64	.95	51.2	395.4	7.35	576.6			
TITANIUM	0.09	.051	-188	.028	-282	4.93	48	.48	48	464.9	464.9	.111	-230	.030	-276	4.14	.48	64.0	44.0	7.16	508.9			
GLASS/ POLYIMIDE	11.1	.028	-46.9	.033	-193	3.00	64	.64	64	113.5	113.5	.142	-43.4	.061	-153	7.62	.95	35.2	1264.3	9.30	1320.2			
	4.31	.038	-46.9	.041	-193	3.40	64	.48	64	235.7	235.7	.229	-29	.130	-105	7.62	.95	35.2	1052.8	11.63	1166.3			
	1.53	.041	-43.4	.061	-179	5.49	51.2	.95	51.2	611.5	611.5	.074	-46.9	.183	-171	7.11	.64	56	535.3	9.95	773.6			
ALUMINIUM	0.09	.066	-46.9	.076	-193	7.04	64	.64	64	56	56	.074	-46.9	.183	-171	4.83	.48	64	65	676.5	676.5			
	11.1	.028	-37.8	.025	-253	2.49	48	.48	64	56.3	4.13	.117	-42.7	.046	-234	7.54	.64	56	1286.2	9.43	1342.5			
GLASS/ POLYIMIDE	4.31	.043	-42	.028	-284	3.23	48	.64	64	114.9	4.99	.137	-22.8	.132	-128	7.62	.95	51.2	1110.9	12.16	1225.8			
	1.53	.051	-42	.036	-283	4.24	48	.48	64	235.2	6.19	.109	-41.4	.089	-234	6.33	.64	56	566.6	10.48	801.8			
TITANIUM	0.09	.066	-35.2	.066	-235	6.60	64	.64	56	653.8	8.73	.069	-48.3	.114	-274	5.18	.48	64	65.8	10.62	719.6			
	11.1	.033	-36.5	.033	-193	3.04	64	.64	56	56.8	4.13	.167	-31	.076	-139	7.62	.95	51.2	1385.6	10.19	1442.4			
GLASS/ EPOXY	4.31	.041	-36.5	.043	-193	3.99	64	.64	56	117.6	5.10	.226	-26.2	.117	-119	7.62	.64	56	1168.1	12.91	1285.7			
	1.53	.051	-36.5	.056	-190	5.21	64	.64	56	242.4	6.34	.127	-39.3	.114	-179	7.62	.95	51.2	567.5	10.47	809.9			
ALUMINIUM	0.09	.074	-36.5	.076	-193	7.24	64	.64	56	636.5	8.50	.076	-37.2	.188	-170	6.60	.95	51.2	68.1	10.97	704.6			
	11.1	.030	-30.3	.025	-259	2.64	48	.64	64	58.6	4.27	.140	-29.6	.056	-208	7.62	.95	51.2	1344.7	10.00	1423.3			
GLASS/ EPOXY	4.31	.048	-33.1	.028	-284	3.35	48	.64	64	119.4	5.20	.208	-26.2	.079	-185	7.62	.95	51.2	1125.0	12.42	1244.4			
	1.53	.053	-33.1	.038	-280	4.42	48	.64	64	246.1	6.46	.099	-33.1	.094	-234	7.16	.64	56	596.6	10.99	842.6			
TITANIUM	0.09	.084	-33.1	.051	-284	6.05	48	.64	64	647.4	8.69	.074	-37.2	.119	-270	5.26	.48	64	68.1	10.97	715.5			

• INCLUDES 0.42 kg/m² ADHESIVE WEIGHT • EXCLUDING END RINGS

Table B-5 : OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 2000 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										CYLINDER										TOTAL JACKET WEIGHT ^a Lb
		FACE SKINS					CORE					FACE SKINS					CORE					
		T ₁		T ₂			T _c	CELL SIZE	WEIGHT Lb./Ft. ³	T ₁		T ₂			T _c	CELL SIZE	WEIGHT Lb./Ft. ³	CYL WEIGHT ^a x 10 ⁻³ Lb./IN. ²				
		THK	STRESS	THK	STRESS	THK				STRESS	THK	STRESS	THK	STRESS								
		In	ksi	In	ksi	In				ksi	In	ksi	In	ksi				In	ksi			
T ₁	11.1	.010	-20.8	.010	-21.8	.010	.622	.30	2.1	77	3.69	.015	-23.7	.019	-29.0	3.00	.30	2.1	1629	7.71	1706	
ALUMINUM	4.31	.011	-25.9	.010	-27.1	.010	.925	.30	2.1	146	4.15	.022	-19.5	.030	-24.1	3.00	.30	2.1	1291	9.20	1437	
T ₂	1.53	.011	-28.0	.013	-29.2	.013	1.26	.30	2.1	283	4.86	.016	-27.1	.029	-34.1	2.43	.30	2.1	661	7.92	944	
BORON/EPOXY	0.09	.013	-28.0	.022	-29.2	.013	1.81	.30	2.1	720	6.28	.013	-27.0	.048	-34.4	1.84	.30	2.1	78.4	8.18	798	
T ₁	11.1	.010	-25.8	.010	-16.8	.010	.557	.30	2.1	88	4.21	.011	-38.4	.016	-30.0	2.90	.30	2.1	1604	7.61	1692	
TITANIUM	4.31	.010	-33.6	.010	-21.8	.010	.830	.30	2.1	159	4.54	.014	-32.2	.027	-24.4	3.00	.30	2.1	1272	9.07	1431	
T ₂	1.53	.011	-40.6	.010	-26.3	.010	1.21	.30	2.1	301	5.18	.013	-42.7	.026	-34.2	2.22	.30	2.1	655	7.86	956	
BORON/EPOXY	0.09	.011	-42.6	.019	-27.5	.019	1.71	.30	2.1	737	6.46	.020	-35.7	.043	-29.0	1.06	.30	2.1	97	10.06	834	
T ₁	11.1	.010	-21.5	.010	-21.2	.010	.683	.30	2.1	85	4.03	.020	-22.4	.022	-21.3	3.00	.30	2.1	1894	8.95	1979	
ALUMINUM	4.31	.011	-26.4	.011	-26.0	.011	1.02	.30	2.1	161	4.56	.031	-18.8	.031	-18.1	3.00	.30	2.1	1549	11.03	1710	
T ₂	1.53	.013	-28.0	.012	-29.5	.012	1.37	.30	2.1	318	5.44	.026	-27.8	.027	-27.0	2.69	.30	2.1	815	9.75	1133	
ALUMINUM	0.09	.019	-28.0	.018	-27.5	.018	1.92	.30	2.1	818	7.14	.035	-25.5	.043	-25.2	1.52	.30	2.1	104	10.85	922	
T ₁	11.1	.010	-21.4	.010	-21.1	.010	.539	.30	2.1	105	5.06	.011	-42.1	.011	-39.9	3.00	.30	2.1	1764	8.34	1869	
TITANIUM	4.31	.010	-27.8	.010	-27.5	.010	.794	.30	2.1	189	5.37	.016	-34.2	.018	-32.9	3.00	.30	2.1	1446	10.30	1635	
T ₂	1.53	.010	-35.9	.010	-35.4	.010	1.20	.30	2.1	341	5.85	.017	-42.6	.017	-41.7	2.10	.30	2.1	765	9.19	1106	
TITANIUM	0.09	.011	-42.7	.012	-42.0	.012	1.88	.30	2.1	834	7.28	.026	-40.4	.023	-39.9	1.59	.30	2.1	105	11.01	939	
T ₁	11.1	.013	-28.0	.011	-6.5	.011	1.00	.30	2.1	95	4.49	.021	-26.6	.051	-6.8	3.00	.30	2.1	2245	10.56	2340	
ALUMINUM	4.31	.017	-28.0	.014	-6.5	.014	1.30	.30	2.1	193	5.47	.034	-21.3	.076	-5.5	3.00	.30	2.1	1931	13.69	2124	
T ₂ GLASS/ POLYIMIDE	1.53	.022	-27.9	.017	-6.5	.017	1.67	.30	2.1	392	6.70	.042	-27.9	.039	-7.3	2.92	.30	2.1	978	11.71	1370	
0.09	.032	-27.9	.021	-6.5	2.50	.30	2.1	1022	8.91	.077	-24.0	.024	-6.4	2.20	.30	2.1	126	13.27	1148			
T ₁	11.1	.010	-36.9	.010	-5.3	.010	.953	.30	2.1	100	4.24	.014	-42.0	.046	-6.9	3.00	.30	2.1	2203	10.38	2303	
TITANIUM	4.31	.011	-42.8	.014	-6.1	.014	1.22	.30	3.1	219	6.18	.022	-34.1	.072	-5.6	3.00	.30	2.1	1900	13.49	2119	
T ₂ GLASS/ POLYIMIDE	1.53	.015	-41.4	.016	-5.9	.016	1.69	.30	2.1	401	6.85	.028	-41.6	.038	-7.0	2.56	.30	2.1	966	11.59	1367	
0.09	.022	-41.3	.016	-5.9	2.68	.30	2.1	1059	9.24	.048	-37.5	.029	-6.3	2.08	.30	2.1	128	13.49	1187			
T ₁	11.1	.014	-28.0	.011	-5.1	.011	1.07	.30	2.1	99	4.67	.025	-24.1	.061	-4.9	3.00	.30	2.1	2495	11.73	2594	
ALUMINUM	4.31	.018	-28.0	.014	-5.1	.014	1.43	.30	2.1	202	5.69	.039	-19.9	.092	-4.1	3.00	.30	2.1	2165	15.34	2367	
T ₂	1.53	.023	-23.0	.019	-5.1	.019	1.78	.30	2.1	409	6.98	.042	-28.0	.047	-5.8	3.00	.30	2.1	1043	12.45	1452	
GLASS/EPOXY	0.09	.032	-27.9	.025	-5.1	.025	2.56	.30	2.1	1069	9.31	.079	-23.8	.024	-5.0	2.40	.30	2.1	131	13.73	1200	
T ₁	11.1	.010	-37.6	.010	-4.2	.010	1.04	.30	2.1	103	4.88	.016	-38.4	.055	-4.9	3.00	.30	2.1	2421	11.40	2524	
TITANIUM	4.31	.012	-41.4	.013	-4.7	.013	1.40	.30	2.1	206	5.83	.025	-31.7	.084	-4.1	3.00	.30	2.1	2102	14.92	2308	
T ₂	1.53	.016	-40.9	.018	-4.6	.018	1.76	.30	2.1	420	7.18	.030	-41.4	.036	-5.4	2.89	.30	2.1	1013	12.14	1433	
GLASS/EPOXY	0.09	.022	-41.3	.022	-4.6	.022	2.63	.30	2.1	1095	9.54	.049	-37.3	.033	-4.9	2.00	.30	2.1	143	14.99	1238	

* INCLUDES 0.0006 lb/in² ADHESIVE WEIGHT

* EXCLUDING RING WEIGHT

Table B-6: OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 56.63 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT OF TWO HEADS					CYLINDER						TOTAL JACKET WEIGHT* kg	
		FACE SKINS					CORE					FACE SKINS					CORE							
		T ₁		T ₂		T _c	CELL SIZE	WEIGHT	T ₁		T ₂		T _c	CELL SIZE	WEIGHT	T ₁		T ₂		T _c	CELL SIZE	WEIGHT		
		THK	STRESS	THK	STRESS				THK	STRESS	THK	STRESS				THK	STRESS	THK	STRESS					THK
		cm	MN/m ²	cm	MN/m ²	cm	cm	kg/m ³	cm	cm	kg	kg/m ²	cm	cm	kg	kg/m ²	cm	cm	kg	kg/m ²	cm	cm	kg	kg/m ²
ALUMINUM	11.1	.025	-143	.025	-150	1.58	.76	33.6			34.9	2.59	.038	.048	-200	7.62	.76	33.6	739.6	5.42	774.5			
	4.31	.028	-179	.025	-187	2.36				66.3	2.92	.056	.076	-166	7.62			586.1	6.47	652.4				
	1.53	.028	-193	.033	-201	3.20				128.5	3.42	.041	-187	.074				300	5.57	428.6				
BORON/EPOXY	0.09	.033	-193	.056	-201	4.60				326.9	4.42	.033	-186	.122	-237	4.67			35.6	5.75	362.3			
	11.1	.025	-178	.025	-116	1.41				39.9	2.96	.028	-265	.041	-207	7.37			728.2	5.35	768.2			
	4.31	.025	-232	.025	-150	2.11				72.2	3.19	.036	-222	.069	-168	7.62			577.5	6.38	649.6			
BORON/EPOXY	1.53	.028	-279	.025	-181	3.07				136.7	3.64	.033	-294	.066	-236	5.64			297.4	5.53	434			
	0.09	.028	-294	.048	-189	4.34				334.6	4.54	.051	-248	.109	-200	2.69			44	7.07	378.6			
	11.1	.025	-145	.025	-146	1.73				38.6	2.83	.051	-154	.056	-147	7.62			859.8	6.29	898.5			
ALUMINUM	4.31	.028	-182	.028	-179	2.59				73.1	3.21	.079	-130	.079	-125	7.62			703.3	7.76	776.3			
	1.53	.033	-193	.030	-203	3.48				144.5	3.82	.066	-191	.069	-186	6.83			370.0	6.86	514.4			
	0.09	.048	-193	.046	-190	4.88				371.4	5.02	.089	-176	.109	-174	3.86			47.2	7.63	418.6			
TITANIUM	11.1	.025	-148	.025	-145	1.35				47.7	3.56	.028	-290	.028	-275	7.62			800.8	5.86	848.5			
	4.31	.025	-192	.025	-189	2.02				85.8	3.78	.041	-236	.046	-227	7.62			656.5	7.24	742.3			
	1.53	.025	-248	.025	-244	3.05				154.8	4.11	.043	-293	.043	-288	5.33			347.3	6.46	502.1			
TITANIUM	0.09	.028	-294	.030	-290	4.78				378.6	5.12	.066	-279	.058	-275	4.04			47.7	7.74	426.3			
	11.1	.033	-193	.028	-44.8	2.54				43.1	3.16	.053	-183	.129	-47	7.62			1019.2	7.42	1062.4			
	4.31	.043	-193	.036	-44.8	3.30				87.6	3.85	.086	-147	.193	-38	7.62			876.7	9.63	964.3			
ALUMINUM T ₂ GLASS/ POLYIMIDE	1.53	.056	-192	.043	-44.8	4.24				177.9	4.71	.107	-192	.099	-50	7.42			444	8.23	622			
	0.09	.081	-192	.053	-44.8	6.35				46.4	6.26	.196	-165	.061	-44	5.99			57.2	9.33	521.2			
	11.1	.025	-254	.025	-36.5	2.42				45.4	2.98	.036	-290	.117	-48	7.62			1000.2	7.30	1045.6			
TITANIUM T ₂ GLASS/ POLYIMIDE	4.31	.028	-295	.028	-42.1	3.10				99.4	4.35	.056	-235	.183	-39	7.62			862.6	9.48	962			
	1.53	.038	-285	.041	-40.7	4.29				180.1	4.82	.071	-287	.097	-48	6.50			438.6	8.15	620.6			
	0.09	.056	-284	.041	-40.7	6.81				480.8	6.50	.122	-258	.074	-43	5.28			57.7	9.48	538.9			
ALUMINUM	11.1	.036	-193	.028	-35	2.72				44.9	3.28	.064	-166	.155	-34	7.62			1132.7	8.25	1177.7			
	4.31	.046	-193	.036	-35	3.63				91.7	4.00	.099	-137	.234	-28	7.62			982.9	10.79	1074.6			
	1.53	.058	-193	.048	-35	4.53				185.7	4.91	.107	-193	.119	-40	7.62			473.5	8.75	659.2			
GLASS/EPOXY	0.09	.081	-192	.064	-35	6.50				485.3	6.55	.200	-164	.061	-35	6.09			59.5	9.65	544.8			
	11.1	.025	-259	.025	-29	2.64				46.7	3.43	.041	-265	.140	-34	7.62			1099.1	9.02	1145.9			
	4.31	.030	-285	.033	-32	3.55				93.5	4.10	.064	-219	.213	-28	7.62			954.3	10.49	1047.8			
TITANIUM	1.53	.041	-282	.046	-31.7	4.47				190.7	5.05	.076	-285	.091	-37	7.34			459.9	8.54	650.6			
	0.09	.056	-285	.056	-31.7	6.68				497.1	6.71	.124	-257	.084	-34	5.28			64.9	10.54	562.1			

* INCLUDES 0.42 kg/m² ADHESIVE WEIGHT
 * EXCLUDING RING WEIGHT

Table B-7: OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 2000 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										CYLINDER										TOTAL* JACKET WEIGHT LB																																																																																																																																																																																																																																																																																																																																																																																																																					
		FACE SKINS					CORE					FACE SKINS					CORE																																																																																																																																																																																																																																																																																																																																																																																																																										
		T ₁		T ₂			T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS		T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	STRESS	T _c	THK	STRESS	T ₁	THK	ST

* INCLUDES 0.0006 LB/IN² ADHESIVE WEIGHT

* EXCLUDING END RINGS

Table B-8: OPTIMUM SANDWICH SHELL DESIGNS
 LH₂, 56.63 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT OF TWO HEADS				CYLINDER								TOTAL* JACKET WEIGHT kg							
		FACE SKINS					CORE					THK		STRESS		T _c		THK		STRESS		T _c			CELL SIZE		WEIGHT				
		T ₁		T ₂		THK	T _c	CELL SIZE	WEIGHT	kg	kg/m ²	THK	T _c	CELL SIZE	WEIGHT	kg	kg/m ²	THK	T _c	CELL SIZE	WEIGHT	kg	kg/m ²		THK	T _c	CELL SIZE	WEIGHT	kg	kg/m ²	
		THK	STRESS	THK	STRESS																										THK
T ₁	11.1	.025	-152	.025	-141	.025	1.58	.76	33.6	34.9	2.59	.046	-208	.041	-152	.046	.76	33.6	746	5.46	780.1										
BORON/EPOXY T ₂	4.31	.025	-190	.028	-177	.028	2.35			66.3	2.92																				
ALUMINUM T ₁	1.53	.033	-208	.028	-193	.033	3.25			128	3.41	.074	-234	.046	-176	.074	.76	33.6	302	5.61	430.4										
ALUMINUM T ₂	0.09	.053	-208	.033	-193	.033	4.70			326	4.42	.119	-241	.030	-184	.119	.76	33.6	33.6	5.48	360.0										
BORON/EPOXY T ₁	11.1	.025	-117	.025	-176	.025	1.41			40	2.95	.038	-228	.028	-262	.038	.76	33.6	733.6	5.36	773.6										
BORON/EPOXY T ₂	4.31	.025	-152	.025	-228	.025	2.09			72.6	3.19	.066	-183	.038	-213	.066	.76	33.6	582	6.41	654.2										
ALUMINUM T ₁	1.53	.025	-184	.028	-276	.028	3.04			137	3.63	.071	-241	.028	-285	.071	.76	33.6	296	5.51	433.6										
TITANIUM T ₁	0.09	.046	-196	.028	-294	.028	4.39			335	4.53	.109	-239	.028	-288	.109	.76	33.6	34	5.53	369.1										
GLASS/ POLYIMIDE T ₁	11.1	.025	-46.9	.033	-193	.033	2.50			42.7	3.12	.119	-49.6	.056	-172	.119	.76	33.6	1000.6	7.35	1043.3										
GLASS/ POLYIMIDE T ₂	4.31	.036	-46.9	.041	-193	.041	3.20			87.2	3.79										1049.6										
ALUMINUM T ₁	1.53	.038	-46.9	.056	-193	.056	4.34			176.2	4.62	.089	-53.8	.107	-193	.089	.76	33.6	437.7	8.06	613.8										
ALUMINUM T ₂	0.09	.061	-46.9	.076	-193	.076	5.87			457.6	6.14	.058	-46.9	.188	-171	.058			54	8.73	511.7										
GLASS/ POLYIMIDE T ₁	11.1	.025	-37.2	.025	-250	.025	2.33			44.9	3.30	.109	-49.6	.038	-273	.109	.76	985.6	7.24	1030.6											
GLASS/ POLYIMIDE T ₂	4.31	.028	-40.7	.030	-273	.030	3.43			90.8	3.94	.170	-40.7	.058	-225	.170	.76	847.6	9.37	938.4											
TITANIUM T ₁	1.53	.033	-44.8	.036	-301	.036	4.60			178.9	4.68	.076	-51.7	.074	-292	.074		434	7.99	612.9											
TITANIUM T ₂	0.09	.051	-44.8	.051	-301	.051	6.22			463.5	6.20	.056	-51	.109	-290	.056		53.6	8.66	517.1											
GLASS/EPOXY T ₁	11.1	.028	-36.5	.033	-193	.033	2.69			44.5	3.23	.145	-35.8	.066	-159	.145	.76	1105.9	8.14	1150.4											
GLASS/EPOXY T ₂	4.31	.028	-36.5	.046	-193	.046	3.75			90.8	3.94	.213	-29	.107	-129	.213	.76	960.2	10.62	1051.0											
ALUMINUM T ₁	1.53	.043	-36.5	.056	-193	.056	4.62			183.9	4.82	.114	-42.7	.107	-193	.114	.76	468.1	8.63	651.9											
ALUMINUM T ₂	0.09	.058	-36.5	.079	-193	.079	6.53			479	6.41	.051	-37.2	.193	-171	.051		55.8	8.98	534.8											
GLASS/EPOXY T ₁	11.1	.025	-30.3	.025	-256	.025	2.59			46.3	3.39	.137	-37.9	.041	-264	.137	.76	1073.3	7.89	1119.6											
GLASS/EPOXY T ₂	4.31	.030	-35.2	.028	-302	.028	3.68			91.7	3.97	.196	-29.6	.069	-209	.196		934.3	10.32	1026											
TITANIUM T ₁	1.53	.051	-40.7	.030	-344	.030	4.45			205.2	5.40	.089	-42.1	.071	-301	.089	.76	450.8	8.30	656											
TITANIUM T ₂	0.09	.038	-35.2	.051	-301	.051	6.53	.76	33.6	484	6.47	.071	-38.6	.114	-281	.071	.76	33.6	59.9	9.68	543.9										

* INCLUDES 0.42 kg/m² ADHESIVE WEIGHT * EXCLUDING RING WEIGHT

Table B-9 : OPTIMUM SANDWICH SHELL DESIGNS
 LO₂, 750 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT* OF TWO HEADS				CYLINDER										TOTAL* JACKET WEIGHT LB
		FACE SKINS					CORE					x 10 ⁻³ LB/in ²	4b	FACE SKINS				CORE				CYL WEIGHT* x 10 ⁻³ LB/IN ²				
		T ₁		T ₂		T _c	CELL SIZE	WEIGHT LB/FT ³	T ₁		T ₂			T _c	CELL SIZE	WEIGHT LB/FT ³										
		THK	STRESS	THK	STRESS				THK	STRESS	THK						STRESS									
		In	ksi	In	ksi				In	In	In						ksi	In	ksi	In	In		In	ksi		
T ₁	14.3	.013	-13.6	.010	-12.9	.397	3/8	2.2	3.8	38	3.63	.031	-15.5	.012	-12.4	2.02	3/8	2.2	912	7.22	950					
BORON/EPOXY	6.97	.011	-17.6	.011	-16.4	.598	3/8	3.2	4.17	64	4.17	.033	-15.6	.019	-12.4	2.15	3/8	2.2	773	8.28	837					
T ₂	2.12	.025	-14.0	.010	-13.4	.736	3/8	2.2	5.01	138	5.01	.026	-25.9	.017	-19.5	1.85	3/8	3.2	434	8.19	572					
ALUMINUM	0.29	.018	-22.7	.014	-21.0	1.44	3/8	3.2	6.52	349	6.52	.027	-35.0	.015	-26.7	1.07	3/16	4.0	102	7.17	451					
T ₁	14.3	.010	-11.8	.010	-17.9	.352	3/8	3.2	4.19	43	4.19	.015	-22.4	.011	-26.1	1.84	3/8	3.2	949	7.48	992					
BORON/EPOXY	6.97	.011	-13.6	.010	-20.5	.481	3/8	3.2	4.54	69	4.54	.017	-25.1	.012	-29.2	2.31	3/8	3.2	803	8.55	872					
T ₂	2.12	.011	-18.9	.010	-28.4	.885	1/4	3.5	5.40	150	5.40	.025	-25.2	.012	-29.9	1.71	3/8	3.2	431	8.14	581					
TITANIUM	0.29	.017	-19.8	.012	-29.7	1.27	3/8	3.2	6.67	356	6.67	.027	-34.0	.010	-40.7	1.05	3/16	4.0	103	7.28	459					
T ₁	14.3	.010	-5.8	.010	-24.3	.650	3/16	4.0	4.43	46	4.43	.028	-7.4	.017	-25.6	2.43	3/8	3.2	1200	9.40	1246					
GLASS/POLYIMIDE	6.97	.010	-6.8	.011	-28.0	.992	1/4	3.5	5.00	77	5.00	.036	-7.4	.020	-25.7	2.82	3/8	3.2	1033	10.96	1110					
T ₂	2.12	.015	-6.8	.014	-28.0	1.14	3/16	4.0	6.33	177	6.33	.036	-7.2	.029	-25.7	2.29	3/8	3.2	583	10.93	760					
ALUMINUM	0.29	.020	-6.7	.020	-28.0	1.60	3/16	4.0	8.32	447	8.32	.022	-7.7	.043	-28.0	1.56	1/4	3.5	145	10.18	592					
T ₁	14.3	.010	-3.9	.010	-26.1	.581	3/16	4.0	4.87	51	4.87	.030	-5.5	.015	-30.2	2.12	3/8	3.2	1241	9.75	1292					
GLASS/POLYIMIDE	6.97	.012	-2.8	.018	-18.5	.780	3/8	2.2	5.92	91	5.92	.035	-6.2	.016	-33.9	2.53	1/4	3.5	1076	11.44	1167					
T ₂	2.12	.014	-6.1	.010	-40.9	1.15	3/16	4.0	6.44	180	6.44	.040	-6.0	.022	-33.8	1.92	1/4	3.5	614	11.54	794					
TITANIUM	0.29	.020	-6.1	.014	-41.2	1.58	3/16	4.0	8.45	454	8.45	.024	-7.1	.029	-40.9	1.33	3/16	4.0	152	10.65	606					
T ₁	14.3	.022	-2.6	.017	-13.9	.497	3/16	4.0	5.70	59	5.70	.032	-5.9	.018	-25.6	2.57	3/8	3.2	1284	10.04	1343					
GLASS/EPOXY	6.97	.011	-5.3	.011	-28.0	.908	3/16	4.0	5.20	80	5.20	.030	-3.6	.042	-15.8	3.11	3/8	2.2	1104	11.55	1184					
T ₂	2.12	.016	-5.3	.015	-28.0	1.20	3/16	4.0	6.59	184	6.59	.040	-5.7	.030	-25.7	2.44	3/8	3.2	621	11.63	805					
ALUMINUM	0.29	.016	-4.9	.024	-25.8	2.13	3/8	3.2	8.67	471	8.67	.025	-6.0	.043	-28.0	1.47	3/16	4.0	152	10.70	623					
T ₁	14.3	.010	-3.2	.010	-27.0	.754	1/4	3.5	5.06	53	5.06	.029	-4.4	.017	-30.2	2.33	3/8	3.2	1317	10.30	1370					
GLASS/EPOXY	6.97	.011	-3.8	.010	-32.3	.824	3/16	4.0	5.49	84	5.49	.041	-4.9	.017	-33.8	2.62	1/4	3.5	1142	12.13	1226					
T ₂	2.12	.016	-4.8	.010	-41.2	1.19	3/16	4.0	6.69	187	6.69	.036	-4.8	.024	-34.0	2.17	1/4	3.5	644	12.07	931					
TITANIUM	0.29	.021	-4.8	.014	-41.1	1.67	3/16	4.0	8.82	475	8.82	.023	-5.6	.030	-41.2	1.40	3/16	4.0	155	10.87	633					

* EXCLUDING END RINGS

• INCLUDES 0.0006 lb/in² ADHESIVE WEIGHT

Table B-10: OPTIMUM SANDWICH SHELL DESIGNS
 LO₂, 21.24 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 HRP CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT* OF TWO HEADS				CYLINDER						TOTAL* JACKET WEIGHT kg
		FACE SKINS					CORE					TOTAL WEIGHT* OF TWO HEADS				CYLINDER						
		T ₁		T ₂			T _c	CELL SIZE cm	WEIGHT kg/m ³	T ₁		T ₂		T _c	CELL SIZE cm	WEIGHT kg/m ³	CYL. WEIGHT*					
		THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²	THK cm				STRESS MN/m ²	THK cm	STRESS MN/m ²	kg				kg/m ²					
																		THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²	
BORON/EPOXY T ₁ T ₂	14.3	.033	-93.8	.025	-88.9	1.01	.95	35.2	2.55	17.3	.079	-107	.030	-85.5	5.13	.95	35.2	414	5.08	431.3		
	6.97	.028	-121	.028	-113.1	1.52	.95	51.2	2.93	29.1	.084	-108	.048	-85.5	5.46	.95	35.2	351	5.82	380		
	2.12	.064	-96.5	.025	-92.4	1.87	.95	35.2	3.52	62.7	.066	-178	.043	-134	4.70	.95	51.2	197	5.76	260		
ALUMINIUM T ₁ T ₂	0.29	.046	-157	.036	-145	3.66	.95	51.2	4.58	158	.069	-241	.038	-184	2.72	.48	64	46.3	5.04	205		
	14.3	.025	-81.4	.025	-123	.89	.95	51.2	2.95	19.5	.038	-154	.028	-180	4.67	.95	51.2	431	5.26	450		
	6.97	.028	-93.8	.025	-141	1.22	.95	51.2	3.19	31.3	.043	-173	.030	-202	5.87	.95	51.2	364.5	6.01	396		
BORON/EPOXY T ₁ T ₂	2.12	.028	-130	.025	-196	2.25	.64	56	3.80	68.1	.064	-174	.030	-206	4.34	.95	51.2	196	5.72	264		
	0.29	.043	-136	.030	-205	3.23	.95	51.2	4.69	162	.069	-234	.025	-281	2.68	.48	64	46.8	5.12	208		
	14.3	.025	-40.0	.025	-168	1.65	.48	64	3.12	20.9	.071	-51.0	.043	-177	6.17	.95	51.2	544.8	6.61	565.7		
GLASS/ POLYIMIDE T ₁ T ₂	6.97	.025	-46.9	.028	-193	2.52	.64	56	3.52	35	.091	-51.0	.051	-177.2	7.16	.95	51.2	469	7.71	504		
	2.12	.038	-46.9	.036	-193	2.90	.48	64	4.45	80.4	.091	-49.6	.074	-177.2	5.82	.95	51.2	265	7.69	345		
	0.29	.051	-46.2	.051	-193	4.06	.48	64	5.85	203	.056	-53.1	.109	-193	3.96	.64	56	65.8	7.16	268.8		
ALUMINIUM T ₁ T ₂	14.3	.025	-26.9	.025	-180	1.48	.48	64	3.42	23.2	.076	-37.9	.038	-208	5.38	.95	51.2	563.4	6.86	586.6		
	6.97	.030	-19.3	.046	-128	1.98	.95	35.2	4.16	41.3	.089	-42.7	.041	-234	6.43	.64	56	488.5	8.04	529.8		
	2.12	.036	-42.1	.025	-282	2.92	.48	64	4.53	81.7	.102	-41.4	.056	-233	4.88	.64	56	279	8.11	360		
TITANIUM T ₁ T ₂	0.29	.051	-42.1	.036	-285	4.01	.48	64	5.94	206	.061	-48.9	.074	-282	3.38	.48	64	69	7.49	275		
	14.3	.036	-17.9	.043	-95.8	1.26	.48	64	4.01	26.8	.081	-40.7	.046	-177	6.40	.95	51.2	583	7.06	609.7		
	6.97	.028	-36.5	.028	-193	2.31	.48	64	3.66	36.3	.076	-24.8	.107	-109	7.95	.95	35.2	501	8.12	537.5		
GLASS/EPOXY T ₁ T ₂	2.12	.041	-36.5	.038	-193	3.05	.48	64	4.63	83.6	.102	-39.3	.076	-177	6.20	.95	51.2	282	8.18	365.5		
	0.29	.041	-33.8	.061	-176	5.41	.95	51.2	6.10	213.8	.064	-41.4	.109	-193	3.80	.48	64	69	7.52	282.8		
	14.3	.025	-22.1	.025	-186	1.91	.64	56	3.56	24.1	.074	-30.3	.043	-208	5.88	.95	51.2	598	7.24	622		
GLASS/EPOXY T ₁ T ₂	6.97	.028	-26.2	.025	-223	2.10	.48	64	3.86	38.1	.104	-33.8	.043	-233	6.65	.64	56	518.5	8.53	556.6		
	2.12	.041	-33.1	.025	-284	3.04	.48	64	4.70	84.9	.091	-33.1	.061	-233	5.50	.64	56	292.4	8.49	377.3		
	0.29	.053	-33.1	.036	-283	4.24	.48	64	6.20	215.7	.058	-38.6	.076	-286	3.56	.48	56	20.4	7.64	286		

*EXCLUDING RING WEIGHT

• INCLUDES 0.42 kg/m² ADHESIVE WEIGHT

Table B-11: OPTIMUM SANDWICH SHELL DESIGNS
 LO₂, 750 FT³ PRESSURE VESSEL, 4.5 IN VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT OF TWO HEADS				CYLINDER						TOTAL* JACKET WEIGHT Lb						
		FACE SKINS					CORE					x 10 ⁻³ Lb/in ²	Lb	FACE SKINS			CORE			CYL WEIGHT*								
		T ₁		T ₂			T _c	CELL SIZE	WEIGHT Lb/Ft ³	T ₁				T ₂			THK	STRESS ksi	THK	STRESS ksi	T _c		CELL SIZE	WEIGHT Lb/Ft ³	Lb	x 10 ⁻³ Lb/in ²		
		THK	STRESS	THK	STRESS	THK				STRESS	THK			STRESS	THK	STRESS											THK	STRESS
T ₁ BORON/EPOXY	14.3	.010	-15.5	.010	-14.4	.368	.30	2.1	3.38	35	3.38	.012	-31.6	.011	-23.1	.011	2.19	.30	2.1	742	5.82	777						
T ₂ BORON/EPOXY	6.97	.010	-18.7	.010	-17.4	.432	.30	3.1	3.71	56	3.71	.015	-31.5	.013	-23.1	.013	2.58	.30	2.1	632	6.70	688						
T ₁ ALUMINUM	2.12	.010	-25.0	.010	-23.2	.770	.30	2.1	3.90	108	3.90	.019	-35.0	.013	-26.1	.013	2.10	.30	2.1	343	6.45	451						
T ₂ ALUMINUM	0.29	.012	-30.1	.011	-28.0	1.21	.30	2.1	4.70	251	4.70	.030	-35.0	.011	-26.7	.011	1.02	.30	2.1	81	5.74	332						
T ₁ BORON/EPOXY	14.3	.010	-11.9	.010	-17.9	.354	.30	2.1	3.97	41	3.97	.010	-28.7	.010	-33.2	.010	1.91	.30	2.1	747	5.87	788						
T ₂ BORON/EPOXY	6.97	.010	-14.4	.010	-21.7	.443	.30	2.1	4.07	62	4.07	.013	-30.4	.010	-35.2	.010	2.35	.30	2.1	626	6.65	688						
T ₁ TITANIUM	2.12	.010	-19.5	.010	-29.3	.681	.30	2.1	4.36	121	4.36	.015	-35.0	.011	-41.2	.011	1.91	.30	2.1	341	6.41	462						
T ₂ TITANIUM	0.29	.010	-25.9	.011	-38.9	1.13	.30	2.1	5.03	268	5.03	.025	-35.0	.011	-42.1	.011	.915	.30	2.1	84	5.93	352						
T ₁ GLASS/POLYIMIDE	14.3	.010	-5.7	.010	-23.6	.580	.30	2.1	3.66	38	3.66	.024	-8.1	.016	-28.0	.016	2.41	.30	2.1	952	7.46	990						
T ₂ GLASS/POLYIMIDE	6.97	.010	-6.5	.011	-27.0	.794	.30	2.1	3.99	61	3.99	.030	-8.1	.019	-28.0	.019	2.84	.30	2.1	819	8.68	880						
T ₁ ALUMINUM	2.12	.010	-6.8	.015	-28.0	1.20	.30	2.1	4.90	137	4.90	.023	-7.9	.030	-28.0	.030	2.59	.30	2.1	480	8.95	617						
T ₂ ALUMINUM	0.29	.015	-6.8	.021	-28.0	1.62	.30	2.1	6.33	341	6.33	.017	-7.7	.044	-27.9	.044	1.44	.30	2.1	123	8.59	464						
T ₁ GLASS/POLYIMIDE	14.3	.010	-3.9	.010	-26.1	.507	.30	2.1	4.14	43	4.14	.019	-8.0	.011	-43.2	.011	2.50	.30	2.1	951	7.41	994						
T ₂ GLASS/POLYIMIDE	6.97	.010	-4.7	.010	-31.6	.704	.30	2.1	4.38	67	4.38	.027	-8.0	.013	-43.2	.013	2.78	.30	2.1	811	8.59	878						
T ₁ TITANIUM	2.12	.011	-5.9	.011	-39.7	1.12	.30	2.1	5.07	142	5.07	.033	-7.6	.018	-42.8	.018	2.08	.30	2.1	474	8.90	616						
T ₂ TITANIUM	0.29	.015	-6.5	.013	-43.7	1.62	.30	2.1	6.40	345	6.40	.017	-10.5	.020	-60.3	.020	1.44	.30	2.1	116	8.12	461						
T ₁ GLASS/EPOXY	14.3	.010	-4.6	.010	-24.3	.636	.30	2.1	3.74	39	3.74	.028	-6.4	.016	-28.0	.016	2.57	.30	2.1	1021	7.98	1060						
T ₂ GLASS/EPOXY	6.97	.010	-5.3	.011	-27.7	.867	.30	2.1	4.11	63	4.11	.029	-6.4	.021	-28.0	.021	3.20	.30	2.1	884	9.31	947						
T ₁ ALUMINUM	2.12	.013	-5.3	.015	-28.0	1.14	.30	2.1	5.71	159	5.71	.029	-6.2	.030	-28.0	.030	2.63	.30	2.1	508	9.48	667						
T ₂ ALUMINUM	0.29	.014	-5.3	.022	-28.0	1.84	.30	2.1	6.61	357	6.61	.017	-6.1	.045	-28.0	.045	1.61	.30	2.1	127	8.87	484						
T ₁ GLASS/EPOXY	14.3	.010	-3.1	.010	-26.8	.555	.30	2.1	4.20	44	4.20	.024	-6.3	.011	-43.4	.011	2.56	.30	2.1	1002	7.82	1046						
T ₂ GLASS/EPOXY	6.97	.010	-3.8	.010	-32.6	.776	.30	2.1	4.47	69	4.47	.028	-6.3	.014	-43.4	.014	3.04	.30	2.1	864	9.11	933						
T ₁ TITANIUM	2.12	.010	-4.8	.011	-40.8	1.28	.30	2.1	5.23	147	5.23	.028	-6.2	.019	-43.6	.019	2.45	.30	2.1	496	9.26	643						
T ₂ TITANIUM	0.29	.015	-5.1	.014	-43.8	1.76	.30	2.1	6.66	360	6.66	.020	-8.3	.020	-60.3	.020	1.48	.30	2.1	120	8.44	480						

• INCLUDES 0.0006 Lb/in² ADHESIVE WEIGHT
 • EXCLUDING RING WEIGHT

Table B-12 : OPTIMUM SANDWICH SHELL DESIGNS
 LO₂, 21.24 m³ PRESSURE VESSEL, 11.43 cm VACUUM ANNULUS
 5056 ALUMINUM FLEX-CORE, 99% PROBABILITY

FACE SKIN MATERIAL	PRESSURE VESSEL CYLINDER L/D	HEMISPHERICAL HEADS										TOTAL WEIGHT OF TWO HEADS				CYLINDER						TOTAL JACKET WEIGHT kg
		FACE SKINS					CORE					TOTAL WEIGHT OF TWO HEADS				CYLINDER						
		T ₁		T ₂		T _c	CELL SIZE cm	WEIGHT kg/m ³	T ₁		T ₂		T _c	CELL SIZE cm	WEIGHT kg/m ³	CYL. WEIGHT						
		THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²				THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²				THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²	THK cm	STRESS MN/m ²	
T ₁	14.3	.025	-106.8	.025	-99.3	.92	.76	33.6	2.38	15.9	15.9	.030	-218	.028	-159	5.56	336.9	4.09	332.8			
BORON/EPOXY	6.97	.025	-129	.025	-120	1.10	.038	25.4	2.61	25.4	25.4	.038	-217	.033	-159	6.58	287	4.71	312.4			
T ₂	2.12	.025	-172	.025	-160	1.95	.048	49	2.74	49	49	.048	-241	.033	-180	5.33	155.7	4.53	204.8			
ALUMINUM	0.29	.030	-207	.028	-193	3.05	.076	114	3.30	114	114	.076	-241	.028	-184	2.56	36.8	4.04	150.7			
T ₁	14.3	.025	-82	.025	-123	.90	.025	18.6	2.79	18.6	18.6	.025	-198	.025	-229	4.85	339	4.13	357.8			
BORON/EPOXY	6.97	.025	-99.3	.025	-149	1.12	.038	28.2	2.86	28.2	28.2	.038	-210	.025	-243	5.97	284.2	4.68	312.4			
T ₂	2.12	.025	-134	.025	-202	1.73	.038	54.9	3.07	54.9	54.9	.038	-241	.028	-284	4.85	154.8	4.51	209.7			
TITANIUM	0.29	.025	-179	.028	-267	2.87	.064	121.6	3.54	121.6	121.6	.064	-241	.028	-290	2.32	38.1	4.17	159.8			
T ₁ GLASS/POLYIMIDE	14.3	.025	-39.3	.025	-163	1.47	.061	17.3	2.57	17.3	17.3	.061	-55.9	.041	-193	6.10	432	5.25	450			
T ₂	6.97	.025	-44.8	.028	-186	2.11	.076	27.7	2.81	27.7	27.7	.076	-55.9	.048	-193	7.21	372	6.10	400			
ALUMINUM	0.29	.025	-47	.038	-193	3.05	.058	62.2	3.45	62.2	62.2	.058	-54.5	.076	-193	6.58	218	6.29	280			
T ₁ GLASS/POLYIMIDE	14.3	.038	-47	.053	-193	4.11	.043	155	4.45	155	155	.043	-53	.111	-192.4	3.66	55.8	6.04	210			
T ₂	6.97	.025	-26.9	.025	-180	1.29	.048	19.5	2.91	19.5	19.5	.048	-55	.028	-298	6.35	43.2	5.21	451			
TITANIUM	0.29	.025	-32.4	.025	-219	1.79	.069	30.4	3.08	30.4	30.4	.069	-55	.033	-298	7.06	368.2	6.04	399			
T ₁	14.3	.028	-40.7	.028	-274	2.84	.084	64.5	3.57	64.5	64.5	.084	-52.4	.046	-295	5.28	214.3	6.26	280			
GLASS/EPOXY	6.97	.038	-44.8	.033	-301	4.11	.043	157	4.50	157	157	.043	-72	.051	-416	3.66	52.7	5.71	209			
T ₂	2.12	.025	-31.7	.025	-168	1.62	.071	17.7	2.63	17.7	17.7	.071	-44.1	.041	-193	6.53	463.5	5.61	481.2			
ALUMINUM	0.29	.033	-36.5	.038	-193	2.90	.074	28.6	2.89	28.6	28.6	.074	-44.1	.053	-193	8.13	401.3	6.55	430			
T ₁	14.3	.036	-36.5	.056	-193	4.67	.043	162	4.65	162	162	.043	-42.0	.114	-193	4.09	231	6.67	303			
GLASS/EPOXY	6.97	.025	-21.4	.025	-185	1.41	.061	20	2.95	20	20	.061	-43.4	.028	-299	6.50	57.7	6.24	220			
T ₂	2.12	.025	-26.2	.025	-225	1.97	.071	31.3	3.14	31.3	31.3	.071	-43.4	.036	-299	7.72	455	5.50	475			
TITANIUM	0.29	.038	-35.2	.036	-302	4.47	.051	66.7	3.68	66.7	66.7	.051	-42.7	.048	-300	6.22	392.3	6.41	423.6			

• INCLUDES 0.42 kg/m² ADHESIVE WEIGHT

• EXCLUDING RING WEIGHT

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