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CRYOGENIC TANK SUPPORT EVALUATION

By

B. R. Bullard

LOCKHEED MISSILES & SPACE COMPANY

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA LEWIS RESEARCH CENTER

CONTRACT NAS 3-7979

James R. Barber, Project Manager

FINAL
REPORT

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EVALUATION

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Sunnyvale, California

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CONTRACT NAS 3-7979

NASA Lewis Research Center
Cleveland, Ohio
James R. Barber, Project Manager
Liquid Rocket Technology Branch

FOREWORD

The Lockheed Missiles & Space Company is submitting this final report in partial completion of the requirements of Contract NAS 3-7979, Cryogenic Tank Support Evaluation, dated May 14, 1967. The total scope of work, data, results, and conclusions covering this program are presented in two volumes. The first volume, "Interim Report, Cryogenic Tank Support Evaluation," dated 15 April 1969, covers the conceptual design and analysis of 12 different tank support systems accomplished under Task I. This report covers the experimental evaluation of six of those systems accomplished under Task II of the program.

ABSTRACT

Thermal performance was determined, experimentally, on thermal models simulating six different tank support concepts. These concepts included a thermal disconnect, a heat block (stacked stainless washers), a cooled strut, a filament wound fiberglass strut, a fiberglass cone segment, and a honeycomb segment. Liquid nitrogen and hydrogen were used for cold boundary temperatures and warm boundary temperatures were nominally 440°R and 540°R. The results were used to reevaluate system weights for LH₂ and CH₄ propellant tanks up to 15 feet in diameter.

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SUMMARY

The information contained in this report is the result of the Task II activity under Contract NAS 3-7979. The objective of this activity was to evaluate experimentally the thermal performance of six tank support concepts that had been selected from twelve that were analytically evaluated in Task I. The six selected concepts were:

- Fiberglass strut support (low-heat-leak material)
- Titanium strut with stacked washers (low-heat-leak material)
- Thermal disconnect titanium strut
- Cooled titanium strut
- Fiberglass semimonocoque cone
- Fiberglass honeycomb cone

Thermal models were fabricated for each of these concepts. Tests were then conducted on each of these models at several different boundary conditions to determine the experimental heat rates. Both liquid nitrogen and liquid hydrogen were used for the cold boundary temperature and the outer end temperatures were nominally 440°R and 540°R.

The experimental results were compared to analytical models from Task I, and the results of Task I were modified accordingly. In general, the experimental performance of the fiberglass strut, the cooled strut, and the two continuous support concepts was slightly superior to that predicted in Task I, although the net change in effective system weight was less than 10 percent. The experimental heat rates through the ball and clamp disconnect concept were approximately three times the analytical predictions with associated weight increases as much as 50 percent. The experimental data on the stacked washers was not totally susceptible to correlation, but the trend did indicate washer interface resistances as much as ten times lower than those used in Task I.

The support system that appears to offer the most potential from the considerations of weight, thermal performance, simplicity, and versatility is the filament-wound fiberglass strut.

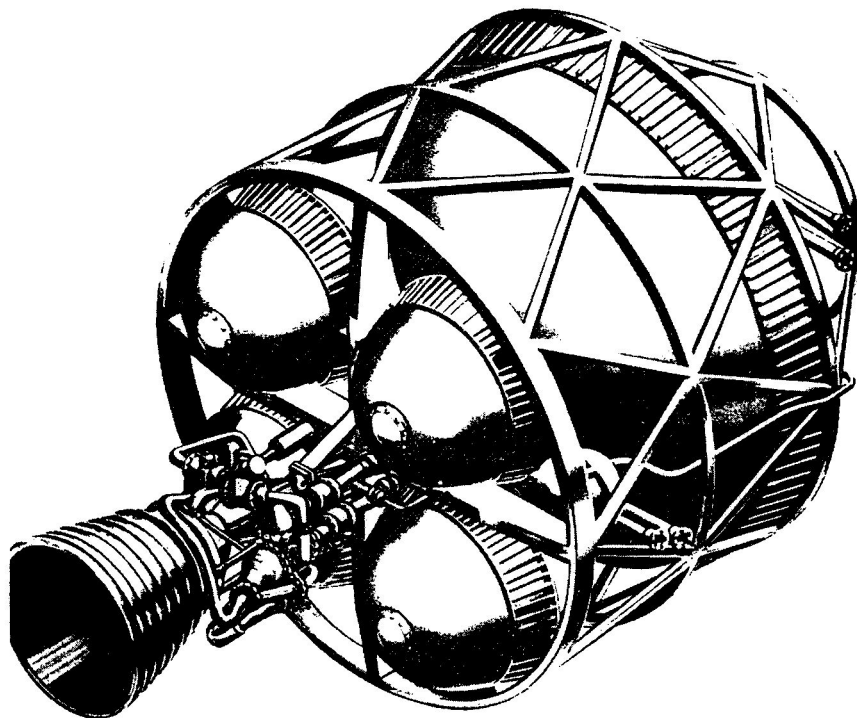
INTRODUCTION

Cryogenic liquids already play a significant role in the nation's space program as propellants in the various Saturn stages. It has long been recognized that the key to extending this role is increasing the storability in a space environment. To this end, the major emphasis has been on development of high-performance insulations. However, with the advancing insulation technology comes the realization that conventional metallic supports and other penetrations contribute a significant portion of the total heat into the propellant.

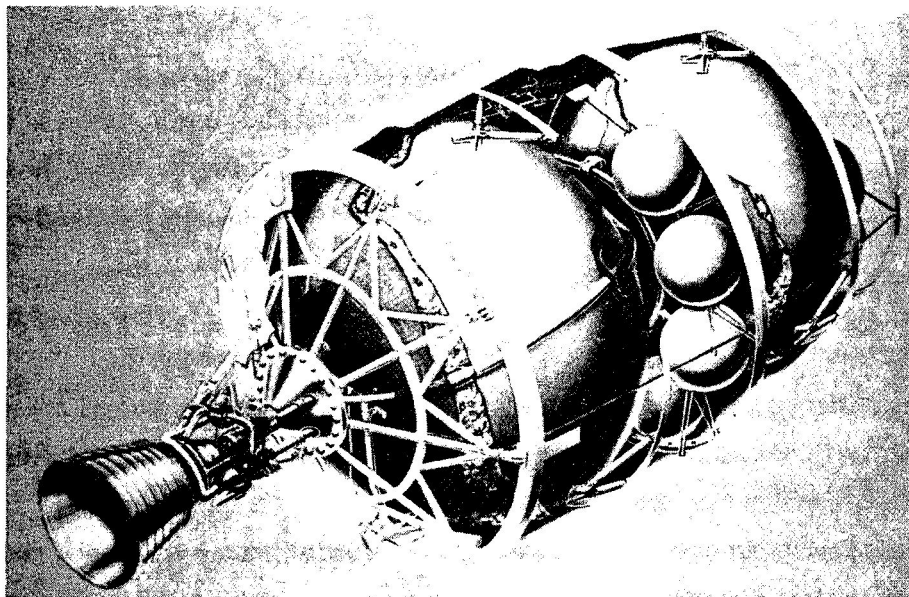
Tank support investigations are often relegated to a minor part of a more extensive study, as was the case for the program reported in Ref. 1. These investigations are then limited to a comparison, for a specific application, of a few candidate support systems that employ conventional designs and materials.

The intent of the program, with which this report is concerned, was to evolve a number of unique designs and evaluate their potential application to future cryogenic vehicles. The program objective is the definition of cryogenic propellant tank support systems that are lightweight, thermally efficient, and provide a wide spectrum of usefulness in terms of tank size and mission duration.

The program is specifically concerned with liquid hydrogen and liquid methane tanks. However, the cryogenic oxidizers, such as oxygen and fluorine, most commonly considered for these fuels will have unit loads (i. e. , load per strut or per inch of tank circumference) and temperatures that fall within the range being investigated. For example, Fig. 1 shows two cryogenic vehicles for which the fuel tank size falls within the range of investigation. For the liquid oxygen tank on the kick stage, the weight is approximately 5,500 lb, which is approximately equal to a nine-ft CH_4 tank.



HYDROGEN-OXYGEN KICKSTAGE



HYDROGEN-FLUORINE SPACE MANEUVERING STAGE

Fig. 1 Representative Cryogenic Vehicles

Consequently, favorable systems for the fuel tanks can also be used for oxidizer tanks. Thus, the implications arising from this study may be judiciously extended to other cryogenics.

The program was comprised of two tasks as follows:

Task I – Conceptual Design and Evaluation for Cryogenic Fuel Tank Supports

Task II – Support System Tests

In Task I, twelve conceptual designs were formulated that potentially result in low heat leak to the propellant. They included both continuous and point supports and the following three different techniques for achieving low heat leak:

- Use of retractable or detachable devices to interrupt the conduction heat paths during storage (i. e. , thermal disconnect)
- Use of low-heat-leak materials (e.g. , stacked washers, glass-fiber blocks)
- Cooling of the support (e.g. , by subliming solids, vaporizing liquid, or the escaping vent gases)

The twelve conceptual designs are categorized in Table 1.

These supports were screened, and a parametric analysis was performed on most of them for the range of independent variables listed in Tables 2 and 3.

A stress analysis using the loads given in Table 4 was performed on each concept to determine the material thicknesses and the structural weights. In conducting this analysis, it was conservatively assumed that the vibration loads were superimposed upon the longitudinal load. This resulted in design conditions of 8.5 and 3.5 g's for tension and compression, respectively.

Table 1

SUMMARY OF CANDIDATE SUPPORT SYSTEM CONCEPTS

Method of Minimizing Heat Leak	Support Type	
	Continuous	Point
Thermal Disconnect	2. Continuous Uncoupling Cone 3. Pyrotechnically Detached Cone	1. Torque Tube Support 4. Ball and Clamp Strut 5. Retracting Wedge Strut
Low-Heat-Leak Material	6. Honeycomb Cone 7. Monocoque Fiberglass Cone	8. Compression-Loaded Washers 9. Fiberglass Strut
Cooled Support	10. Cooled Semi-monocoque Cone	11. Internally Cooled Strut 12. Subliming-Solid Cooled Strut

Table 2
SUPPORT SYSTEM VARIABLES

Fluid and Cold Side Temperature:	Liquid Hydrogen (37°R) Liquid Methane (200°R)	
Hot Side Temperature:	400°R to 520°R	
Storage Time:	10, 100, 1000, 5000, 10,000 hr	
Length of Support:	1 to 3 ft	
Tanks:	√2 Oblate Spheroid	
Tank Major Diameter (ft)	Total Tank and Propellant Mass (lb)	
	CH ₄	LH ₂
5	1,150	250
9	6,600	1,200
12	15,500	2,900
15	30,500	5,600

Table 3
ACCELERATIONS TO BE APPLIED TO TANKS FOR SUPPORTING DESIGN

Launch Loads:	
Longitudinal:	+ 5.7g, max
Lateral:	0.3g, max, occurs when longitudinal acceleration is 2.5g
Vibration:	6-g vibratory longitudinal load between 20 and 150 cps when longitudinal loading is 2.5g's
Loads during Space Maneuvering:	
Lateral:	10 ⁻³ g

Table 4

SUPPORT STRUCTURE DESIGN LOADS

Tank Diameter (ft)	Condition	Conical Support (lb/in.)		Three Point (lb/strut $\times 10^{-3}$)	
		CH ₄	LH ₂ tank	CH ₄	LH ₂ Tank
15	Tension	498	87	66.3	11.6
	Buckling	205	36	27.3	4.8
12	Tension	319	55	34	5.9
	Buckling	131	23	14	2.4
9	Tension	179	31	14.3	2.4
	Buckling	74	13	5.9	1.0
5	Tension	57	12	2.5	0.5
	Buckling	24	5	1.0	0.2

The constituent sizes and materials gages obtained from the stress analyses on each concept were input to the thermal analysis, and the boiloff and insulation weights were determined as a function of time for each tank. The electrical analog technique wherein the support is divided into discrete resistances was used as a thermal model for those systems employing composite materials or thermal disconnects. The results of this Task I activity were described in detail in Ref. 2.

From the results of Task I, the six concepts illustrated in Figs. 2 through 7 were selected for thermal testing in Task II. Test articles were fabricated to provide thermal simulation within the size range covered in Task I. The test articles were installed on a cryostat and the heat rates measured under a variety of boundary conditions. Thermal guards were provided to eliminate heating along the lateral surfaces of the test article. In the case of the ball and clamp and the stacked washer concepts, unique thermal designs were required to match the discrete thermal resistances in these test articles. These guards were made in sections, using different materials, to simulate as nearly as possible, the relative resistances down the test article. Then a heater and a cooling coil was added to each section in such a manner that would allow adjustment of the heat flow in that section to match the temperatures over a like section of the test article. The details of this Task II activity are described in the following sections of this report.

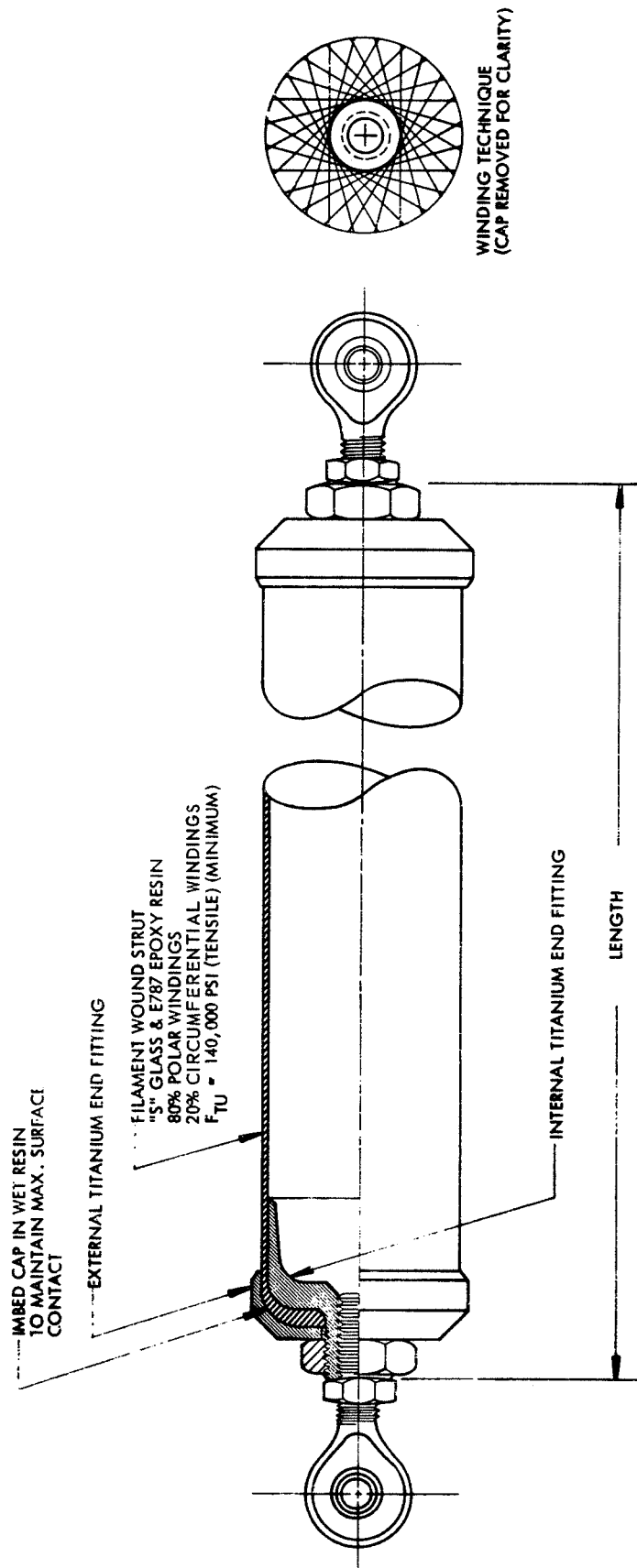


Fig. 2 Filament Fiberglass Strut Concept

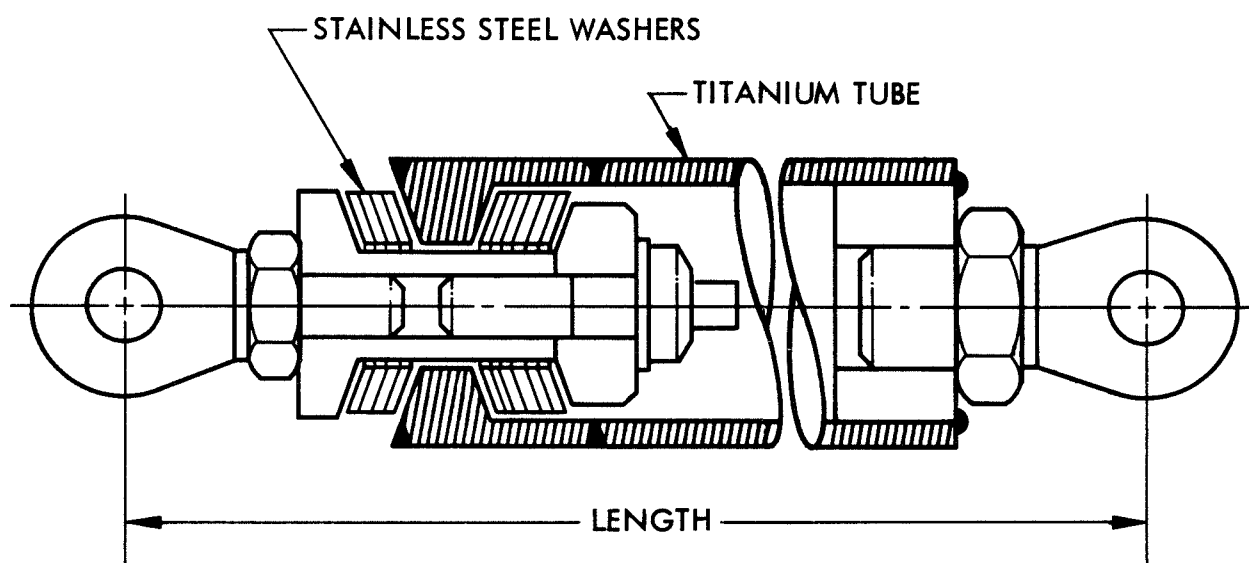


Fig. 3 Stacked Washer Concept

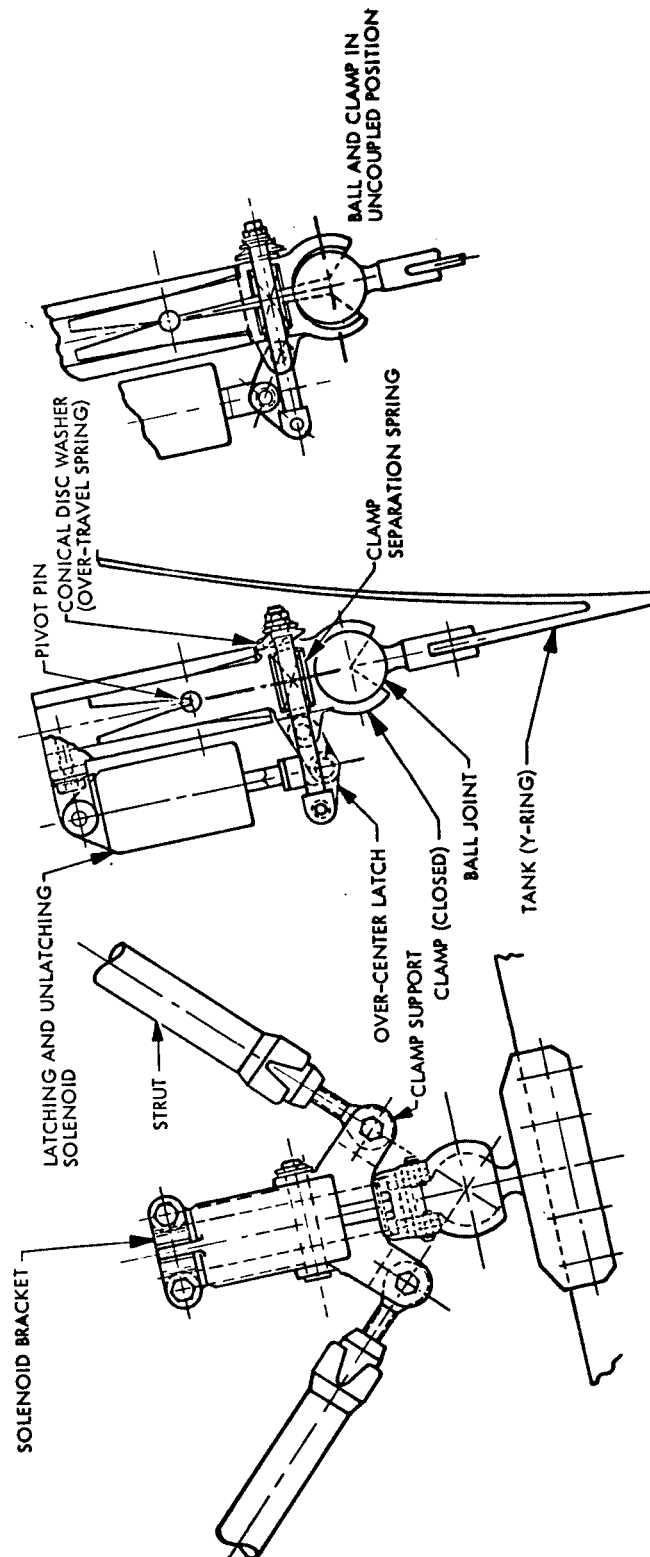


Fig. 4 Ball and Clamp Strut Disconnect Concept

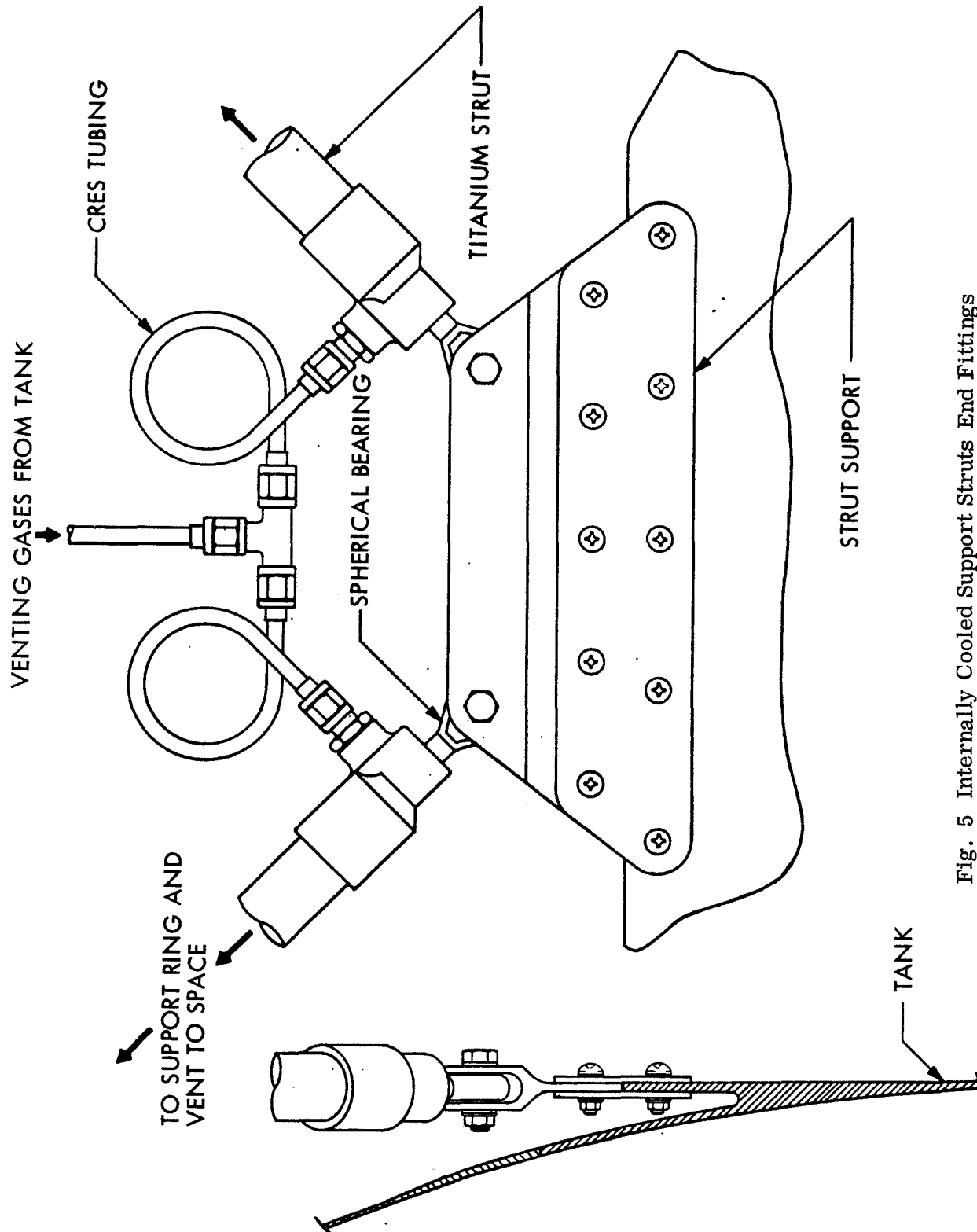


Fig. 5 Internally Cooled Support Struts End Fittings

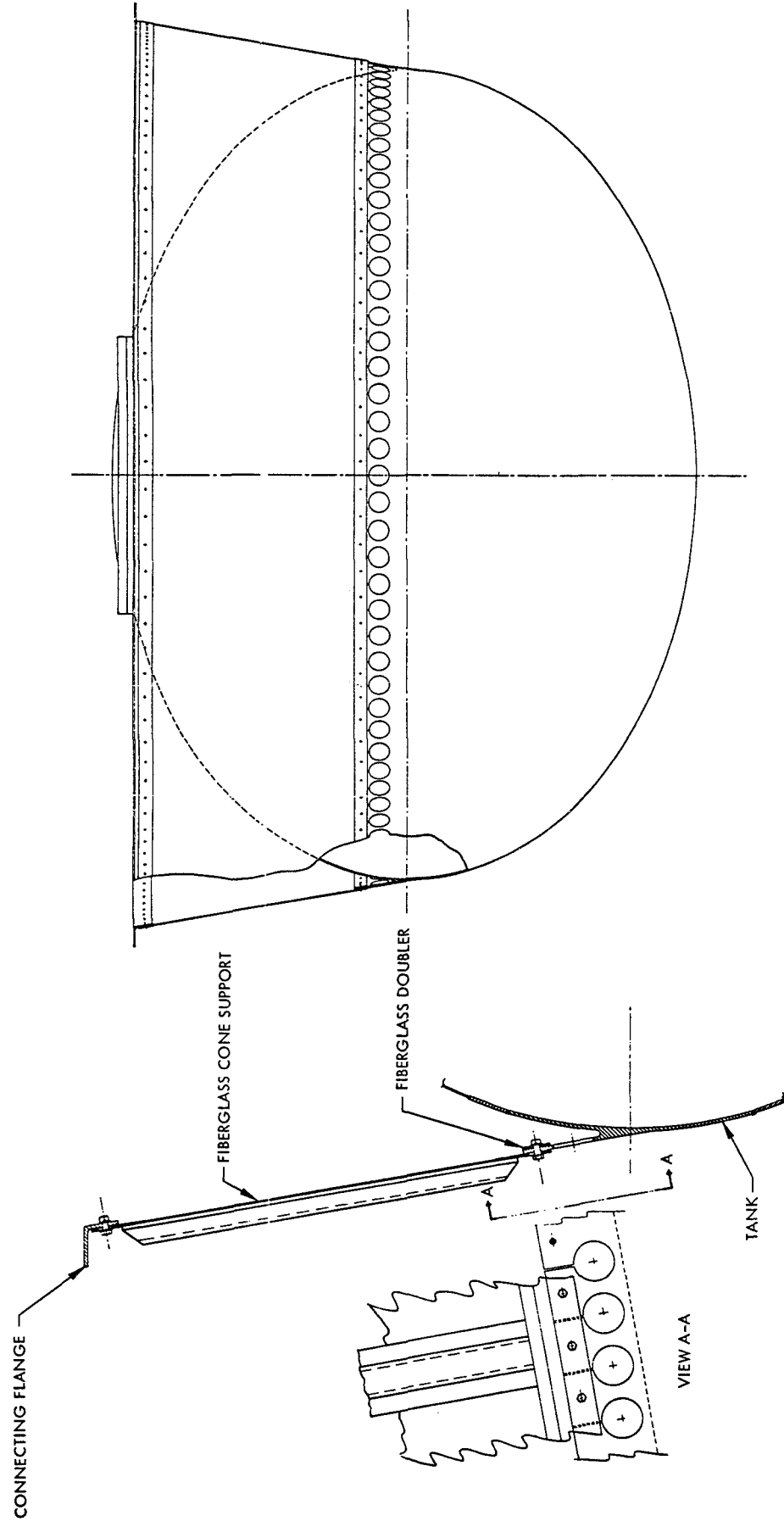


Fig. 6 Semimonocoque Fiberglass Cone Concept

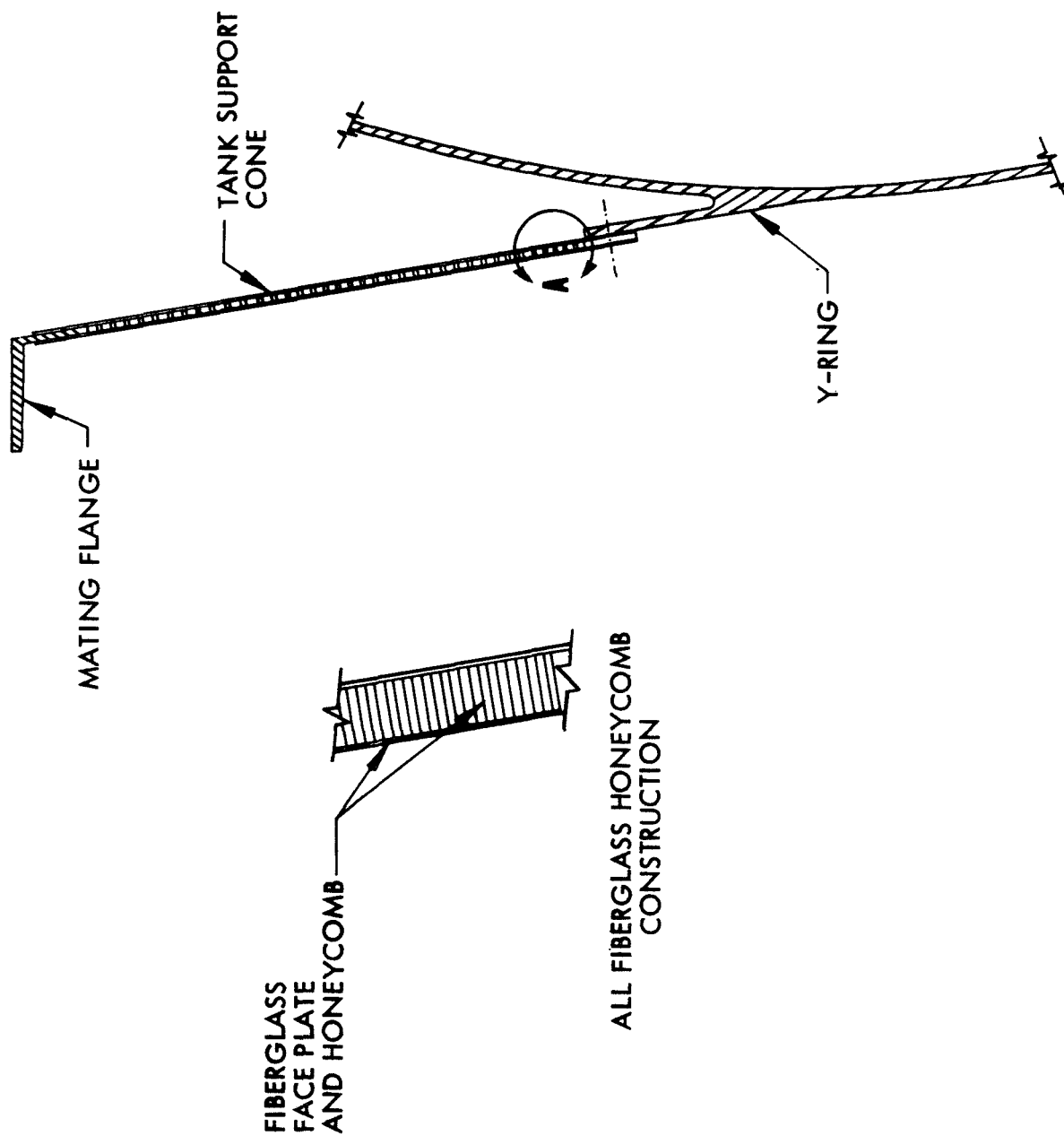


Fig. 7 Honeycomb Conical Support

TASK II – SUPPORT SYSTEM TESTS

The propellant heating that can be attributed to the support system for a cryogenic propellant tank is a consequence of two distinct mechanisms: (1) Direct transfer from the warm end of the support; and (2) interaction between the support and the tank insulation. The latter is dependent upon the particular installation design. Consequently, the Task II test program was designed to measure the effective resistance for each of the isolated support systems, which allows one to evaluate the direct heat transfer. The test apparatus, instrumentation, procedures, and test results are described in this section.

FILAMENT WOUND FIBERGLASS STRUT CONCEPT

TEST FACILITY DESCRIPTION

The tests on the filament wound fiberglass strut were conducted on the penetration cryostat which was previously developed by Lockheed and used in investigations of the penetration (or strut) interaction with multilayer insulation. Figure 8 is a photograph of the penetration cryostat mounted on the door of the 8-ft-diameter vacuum chamber. The essential features of the apparatus are shown schematically in Fig. 9. Its full utility is realized when the test article interacts with the multilayer insulation. In this case, the power input to the two heaters and the temperature drop across the thermal link is sufficient to determine the total heat contribution from a given penetration as well as the percentage that is due to the warm outer boundary and that due to interaction with the insulation.

Heater-1 is utilized to bias the cryogen boiloff rate to the same constant value, both before and after installation of the test strut. This procedure allows the change in heat input to the cryogen to be determined from changes in electrical heater power as well as maintaining constant incidental heat leaks (cryostat support, etc.). Heater-2

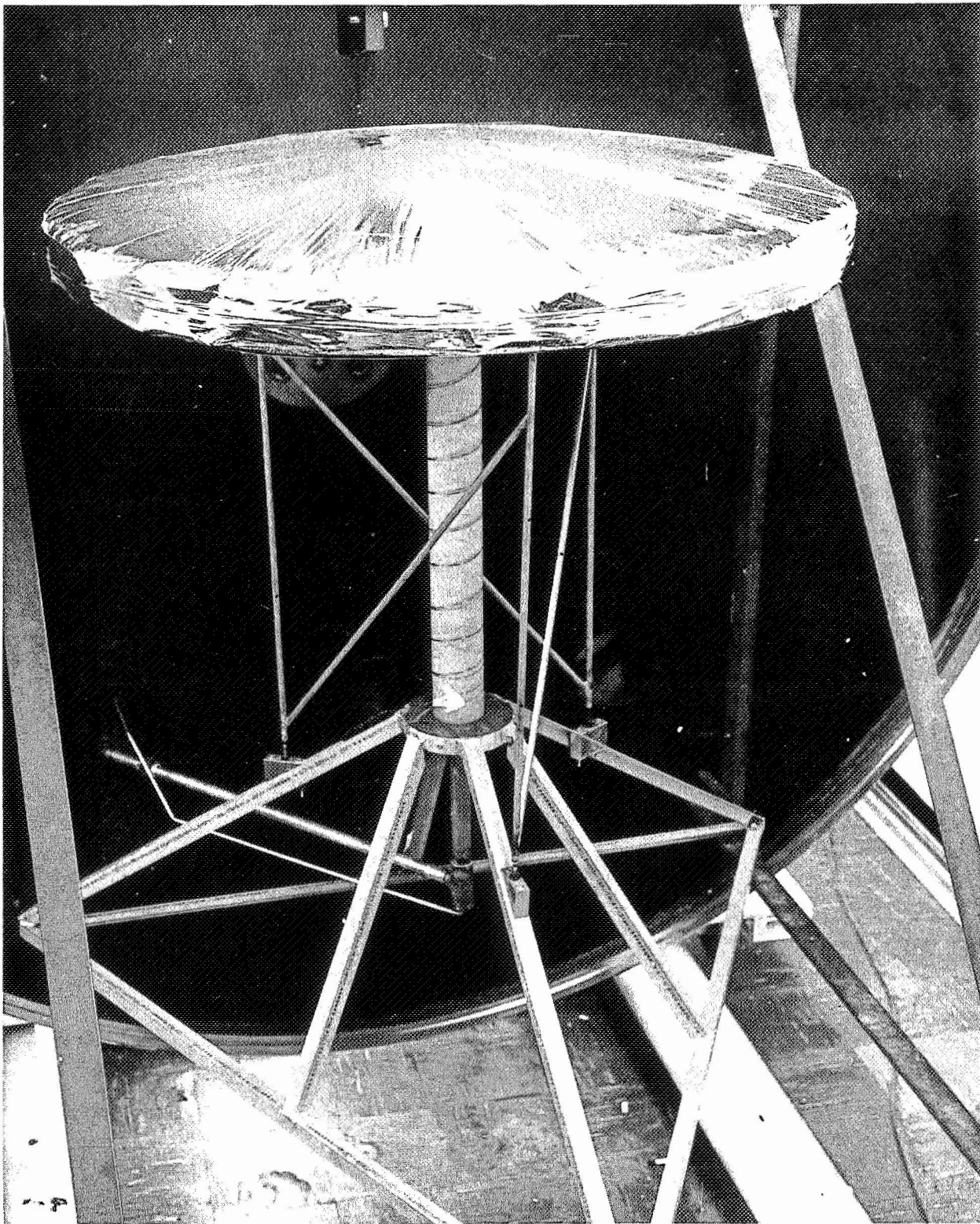


Fig. 8 Penetration Cryostat on Chamber Door

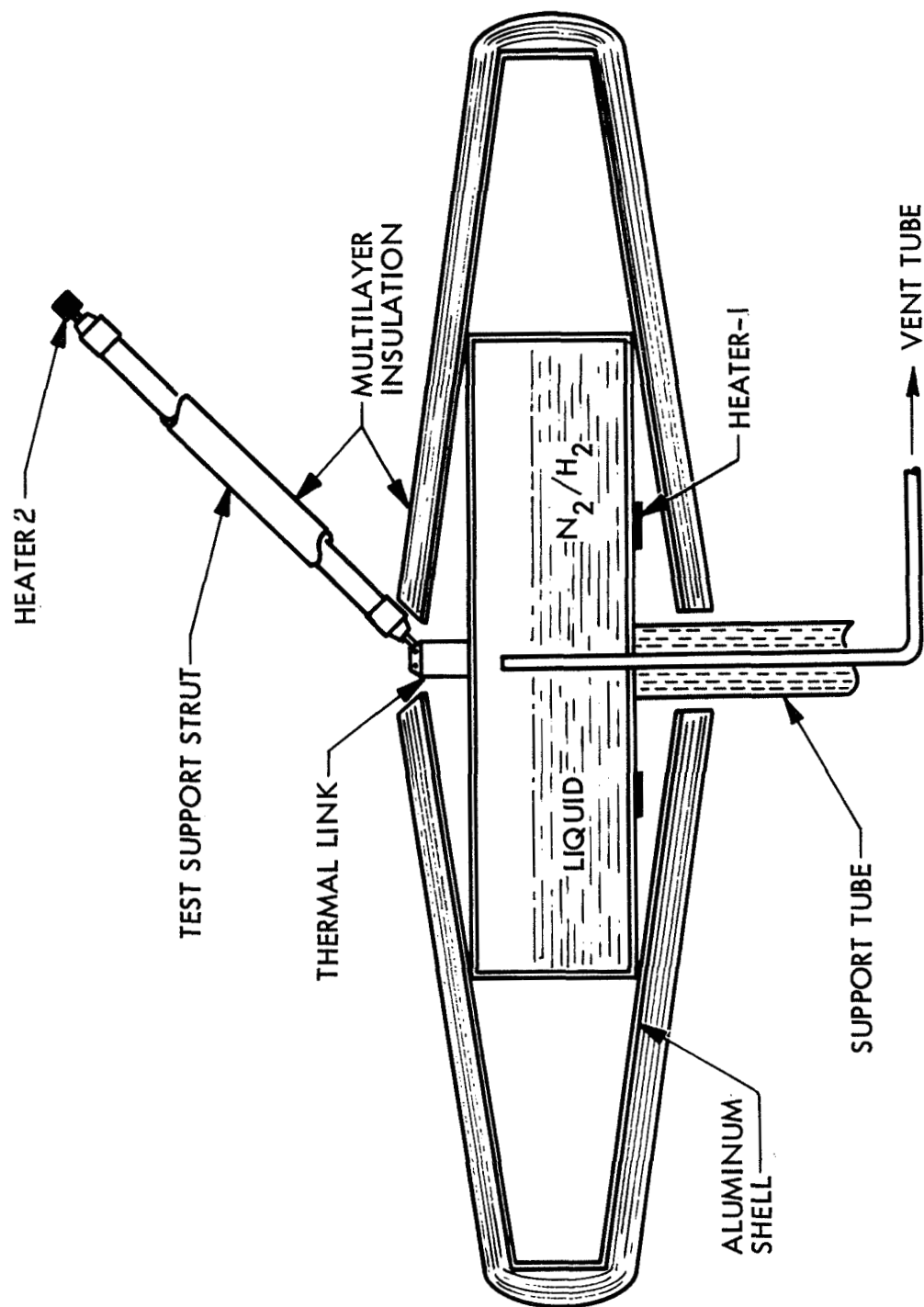


Fig. 9 Penetration Cryostat Schematic

is used to maintain a fixed boundary temperature on the outboard end of the strut. The thermal link is a known thermal resistance which has been precalibrated to give the heat flow, between the bottom of the penetration and the cryogen, as a function of the measured temperature drop. For those cases where the penetration is completely isolated, either the thermal link measurement or the power into heater-2 is all that is required. Figure 10 shows the calibration data for the thermal link that was used for all the point support concepts. This cryostat is insulated with 50 layers of multilayer insulation, which has been applied one layer at a time, with each top layer being tied into its corresponding bottom layer at the outer circumference, thereby providing essentially a continuous wrap. The 1/4-in. -thick aluminum shell supporting the multilayer insulation is thermally shorted to the 3/8-in. -thick aluminum cryogen container. Previous measurements of the temperature distribution on the cryostat indicate that, under the maximum heat loads experienced in this program temperature gradients within the shell are less than 0.3°R .

The maximum diameter of the penetration cryostat assembly is 36 in. , although the cryogen container diameter is only 12 in. and holds approximately one-third cu ft (10 liters) of cryogen. The cryostat is supported from a frame on the chamber door and is elevated approximately 60 inches above the bottom of the chamber. Consequently, the maximum support length that can be conveniently evaluated in this facility is approximately 30 in. The filament wound fiberglass test strut was 15 in.

TEST ARTICLE DESCRIPTION

The filament wound fiberglass strut design that was evaluated in this program is shown in Fig. 11. The total length of the test article, with the rod ends installed, is approximately 15 in. between attachment points, with the fiberglass body comprising 10.8 in. However, 2 in. of the fiberglass is shorted out thermally by the internal titanium end fittings. The glass was wound over the internal end fittings and a salt mandrel to make an integral structure. The external caps were attached, and the assembly was then cured for 12 hours in an air oven. After the cure, the salt mandrel was washed out. Figure 12 is a photograph of the finished product with and without the compression caps. The finished test article had eleven layers of windings: eight longitudinal

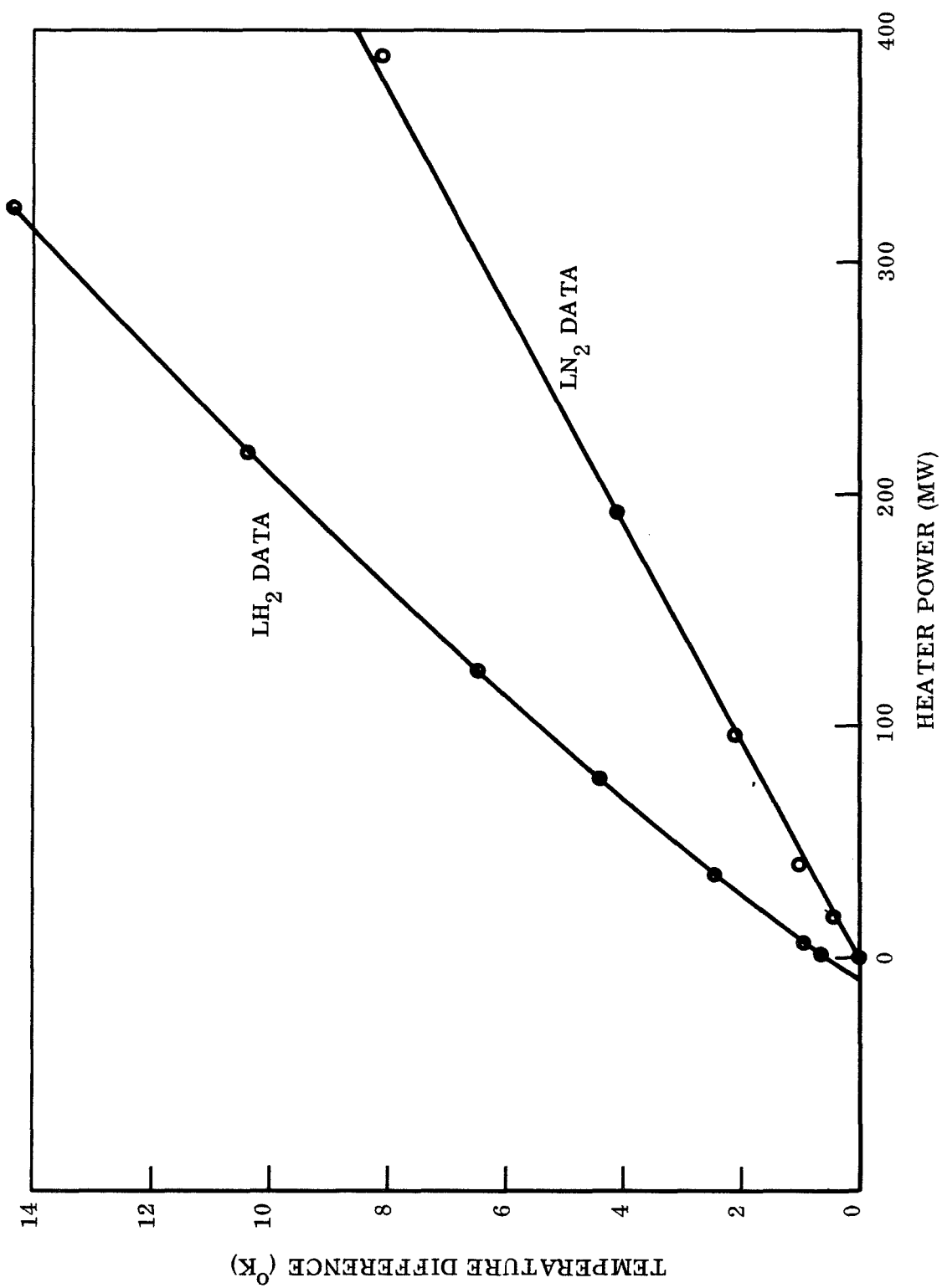
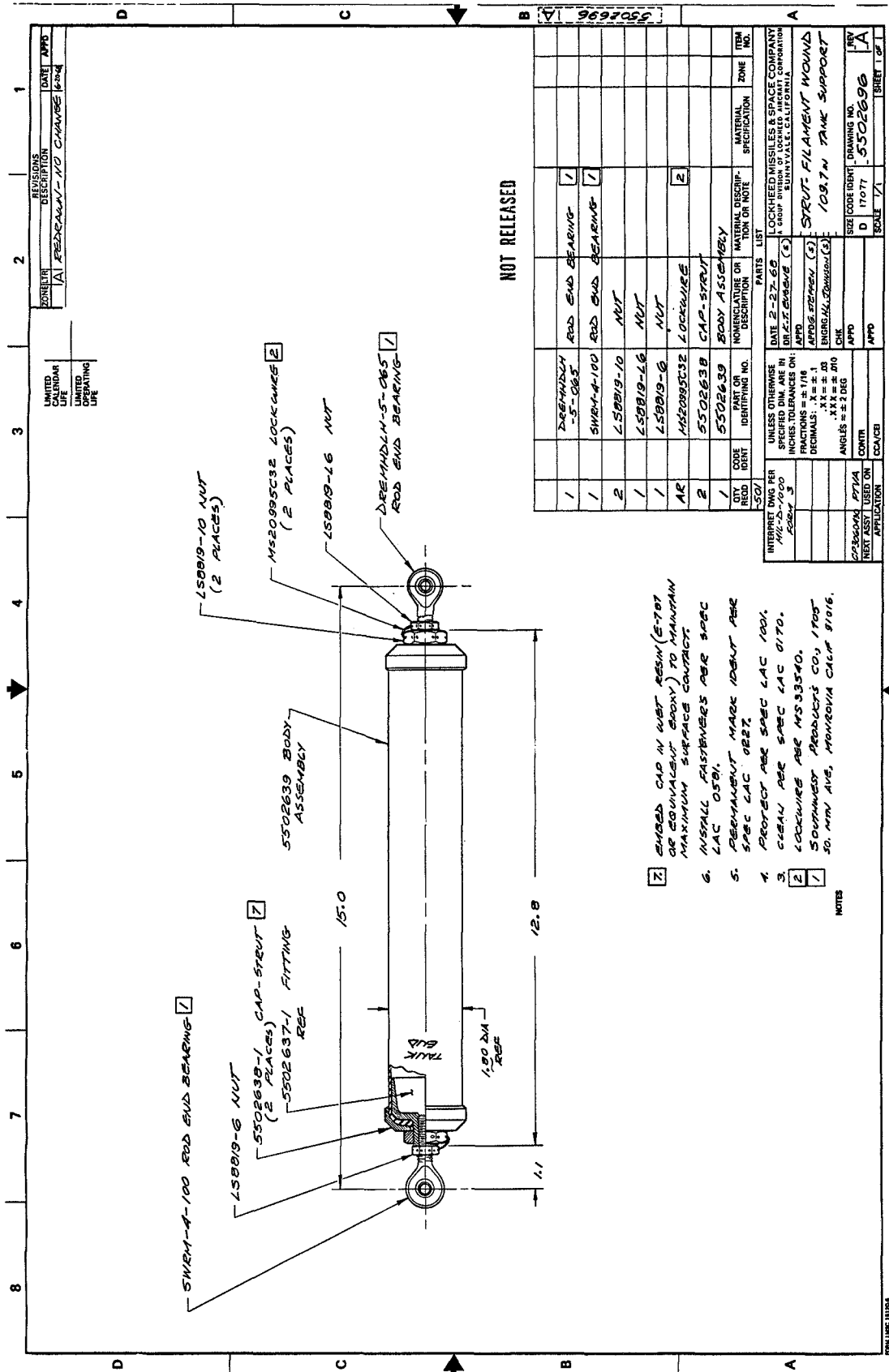


Fig. 10 Thermal Link Calculation



NOT RELEASED

2. ENDED CAP IN RUST RESIN (5-70T OR EQUIVALENT BODY) TO MAINTAIN MAXIMUM SURFACE CONTACT.
6. INSTALL FASTENERS PER SPEC LAC 0501.
5. PERMANENT MARK IDENT PER SPEC LAC 0827.
4. PROTECT PER SPEC LAC 1001.
3. CLEAN PER SPEC LAC 0170.
2. LOCKWIRE PER MS3350.
1. SOURCES: PRODUCTS CO, 1700 50.000 AVE, MONROVIA CALIF 91016.

NOTES

INTERPRET DWG PER MIL-D-1000 FIG 4 3		50/	UNLESS OTHERWISE SPECIFIED DIM. ARE IN INCHES TOLERANCES ON: FRACTIONS: $\pm .118$ DECIMALS: $\pm .010$ X.XX $\pm .010$ X.XXX $\pm .010$ ANGLES $\pm .2$ DEG		DATE 2-27-68 BY: J. E. G. (S) CHECKED BY: J. E. G. (S) ENGINEER: J. E. G. (S) DRAFTSMAN: J. E. G. (S) CHK	PARTS LIST	REVISIONS OR MODIFICATIONS OR REVISIONS OR MODIFICATIONS
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Fig. 12 Photo-Filament Wound Fiberglass Support Strut

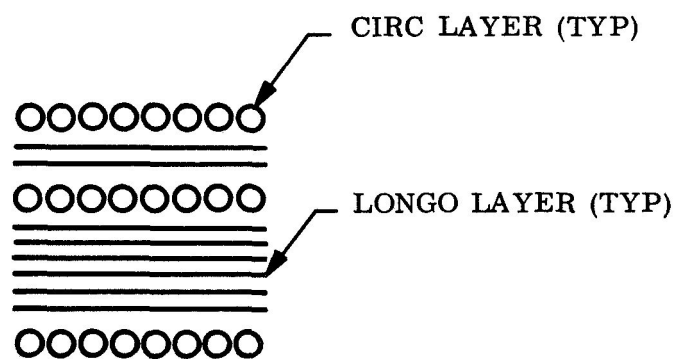


Fig. 13 Distribution of Longitudinal and Circumferential Winding for the Filament Wound Fiberglass Strut

and three circumferential layers distributed as illustrated in Fig. 13. To eliminate internal radiation, the inside of the strut was filled with 12 grams of finely chopped Dexiglas giving a mean density of 2.4 lb/ft³. The glass used in the winding process was three end roving high strength S-901 glass developed by Owens-Corning Fiberglass Corporation. Two resin systems were evaluated. The first was the U.S. Polymeric E-787 system, consisting of 50 parts by weight (pbw) Epon resin 828, 50 pbw Epon resin 1031, 90 pbw NMA curing agent, and 0.55 pbw BDMA accelerator.

This system, was too viscous, and not satisfactory for the wet layup used in winding the fiberglass strut. It resulted in excess resin and when the outer circumferential windings were put on, the resin was squeezed out leaving slack in the longitudinals causing them to be loose at the ends. The one strut that was wound with this resin system had a cured weight distribution of 41.7 percent resin and 58.3 percent glass.

The aforementioned problem was alleviated by removing the heavy Epon resin 1031 and using only 828. Thus, the second resin system consisted of 100 pbw Epon resin 828, 90 pbw NMA, and 0.55 pbw BDMA accelerator. This resulted in a much better fit between fiberglass and titanium at the strut ends (see Fig. 14), as well as improving the glass to resin ratio. The finished test article had 20.5 percent by weight resin, and 79.5 percent by weight glass, and a wall thickness of 0.060 in.

TEST ASSEMBLY

The test assembly used for the fiberglass strut is shown schematically in Fig. 15. The test article is held firmly in place with a female clevis adapter which was fixed to the top of the thermal link. This clevis was machined of leaded brass to provide for good thermal contact. An Evanohm heater assembly was adapted to the compression cap at the warm end of the strut to serve as a heat source for controlling the outer boundary temperature.

The support strut was surrounded with a thermal guard to intercept any heating that would otherwise occur along the lateral surface of the strut. The thermal guard is 2-5/8 in. in diameter and is made of three parts to simulate various resistance

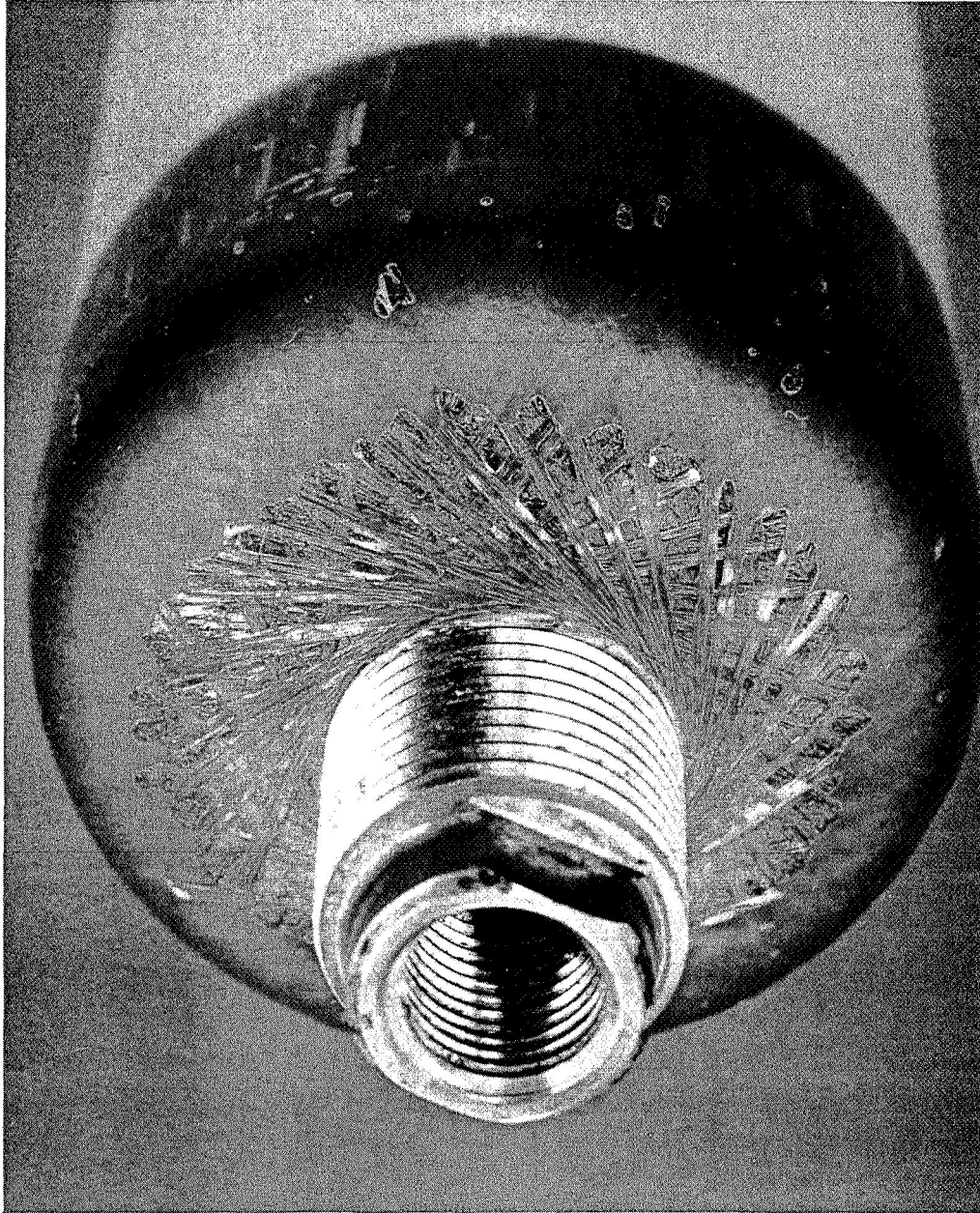


Fig. 14 Photograph Showing the Manner in Which the Fiberglass Windings Are Placed Over the End Fittings

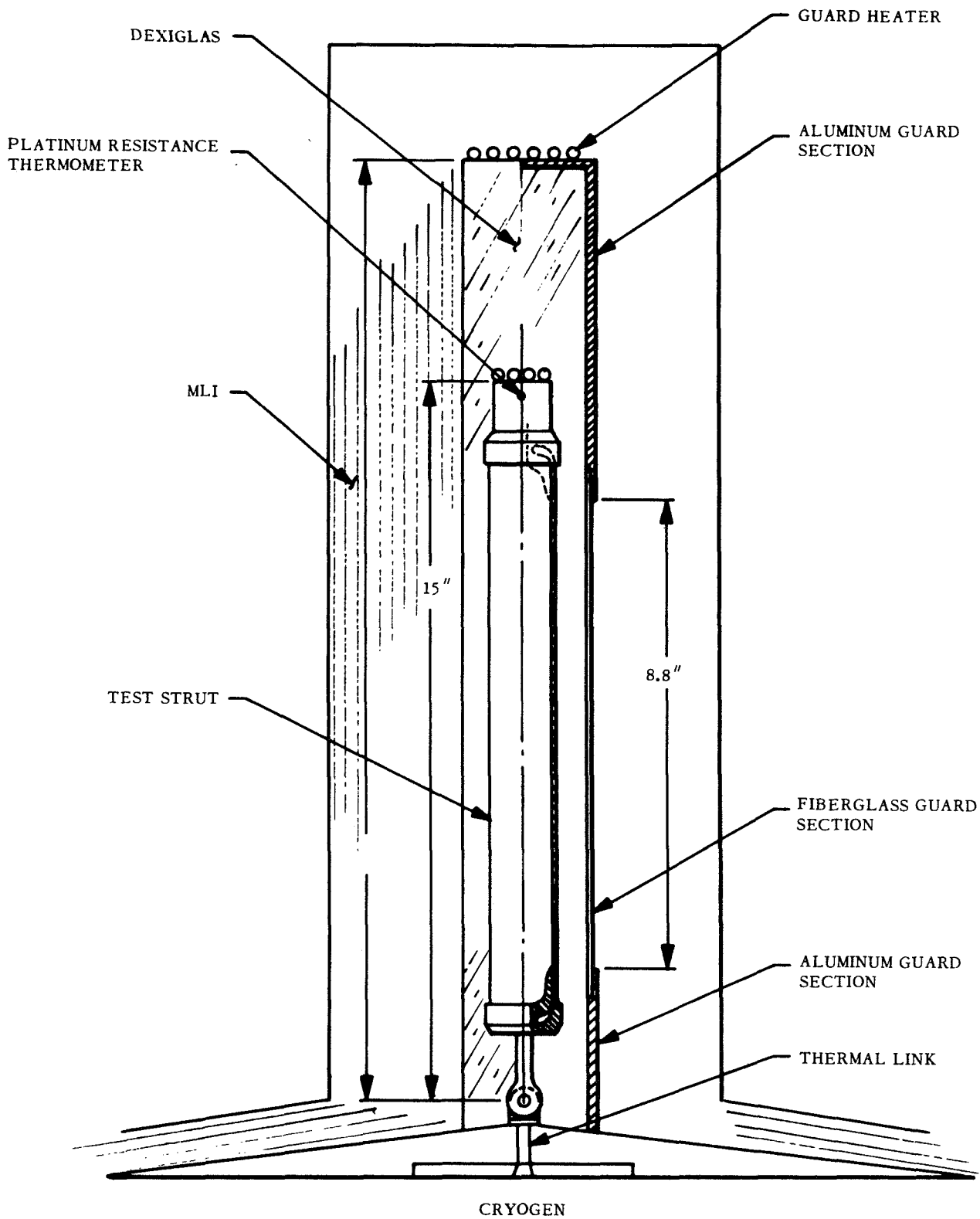


Fig. 15 Test Installation Schematic - Fiberglass Strut

elements of the test article. The center section was made from a fiberglass tube, 0.032-in. thick and cut to the same length as the unshorted fiberglass portion of the fiberglass strut (8.8 in.). An aluminum section, 1/8-in. thick and 6 in. long was bonded to each end of the fiberglass tube. The cold end was then bolted to the isothermal aluminum shield on the penetration cryostat. The 6-in. length of aluminum is equal to the combined length of the attachment clevis, thermal link, and rod end so that the elevations of the fiberglass guard and the strut were the same. The annulus between strut and guard was filled with Dexiglas to prevent radiation tunnelling. A second Evanohm heater assembly was attached to the aluminum section on the warm end of the thermal guard to provide a means for controlling its temperature and matching it to that of the test article. Finally, after the strut and guard were installed, the guard was insulated with 120 layers of multilayer insulation to maintain a constant heat flux down the guard. Figure 16 is a photograph of the test strut in place, and the thermal guard just before installation. The final test assembly is shown in Fig. 17 just before going into the 8-ft-diameter vacuum chamber.

INSTRUMENTATION

This test assembly was instrumented to provide temperature profiles in both the test strut and the thermal guard, as well as the heat flow into the strut at the top and out at the bottom. A platinum resistance thermometer (PRT) was attached to the heater adapter to indicate the warm boundary temperature. Seven chromel-constantan, differential thermocouples were connected in series along the strut and referenced to the PRT. Six of these were equally spaced over the unshorted length of fiberglass (1.75 in. spacing) and the seventh one was actually on the brass clevis. Seven more differential thermocouples were attached to the thermal guard at the same elevations as those on the test strut, and another was used to measure the difference between strut and guard at the warm end. The heater on the guard was used to maintain this difference within $\pm 0.2^{\circ}\text{R}$. Figure 18 shows the location of all temperature measurements.

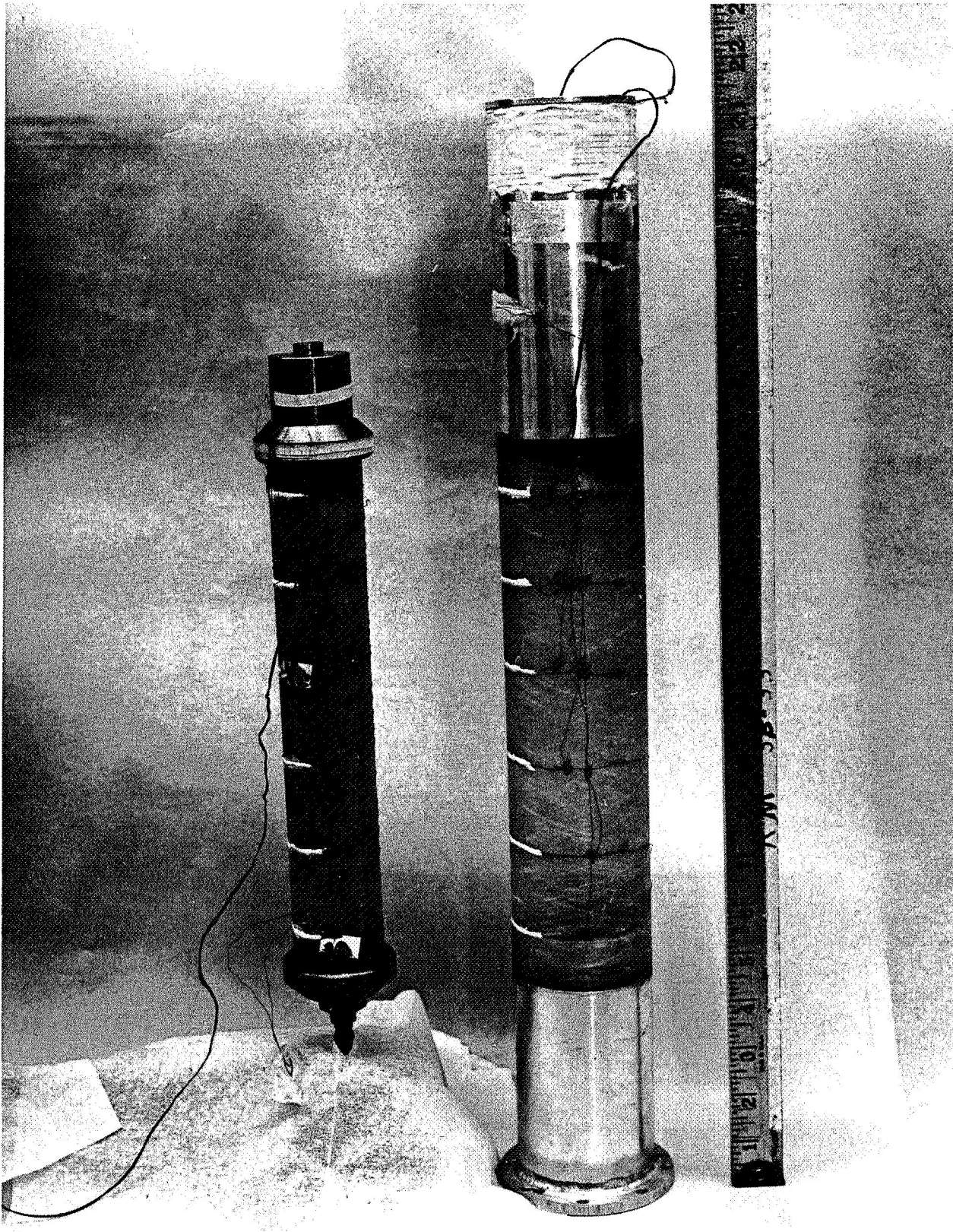
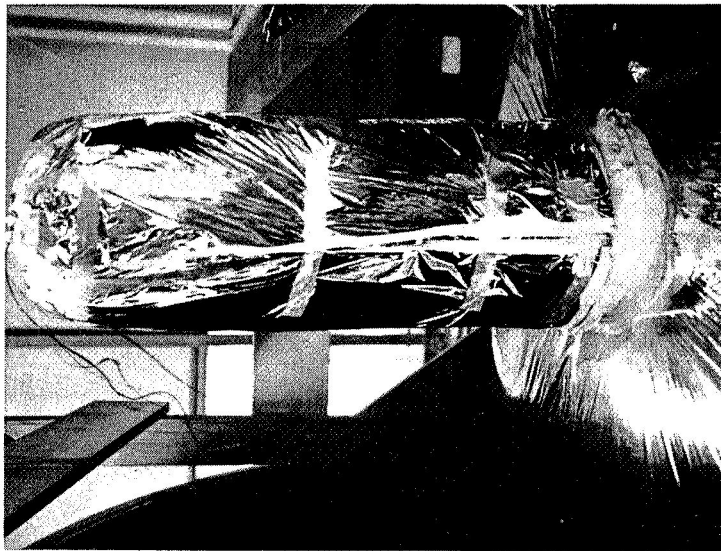
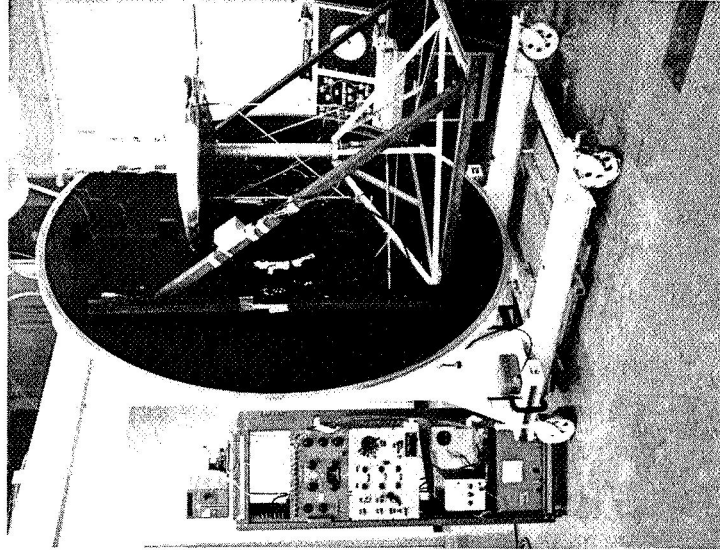


Fig. 16 Photograph of Partially Assembled Fiberglass Strut Test Installation



(a) Insulated Test Section



(b) Test Apparatus Installed on Chamber Door

Fig. 17 Photograph of the Test Installation for the Filament Wound Fiberglass Strut

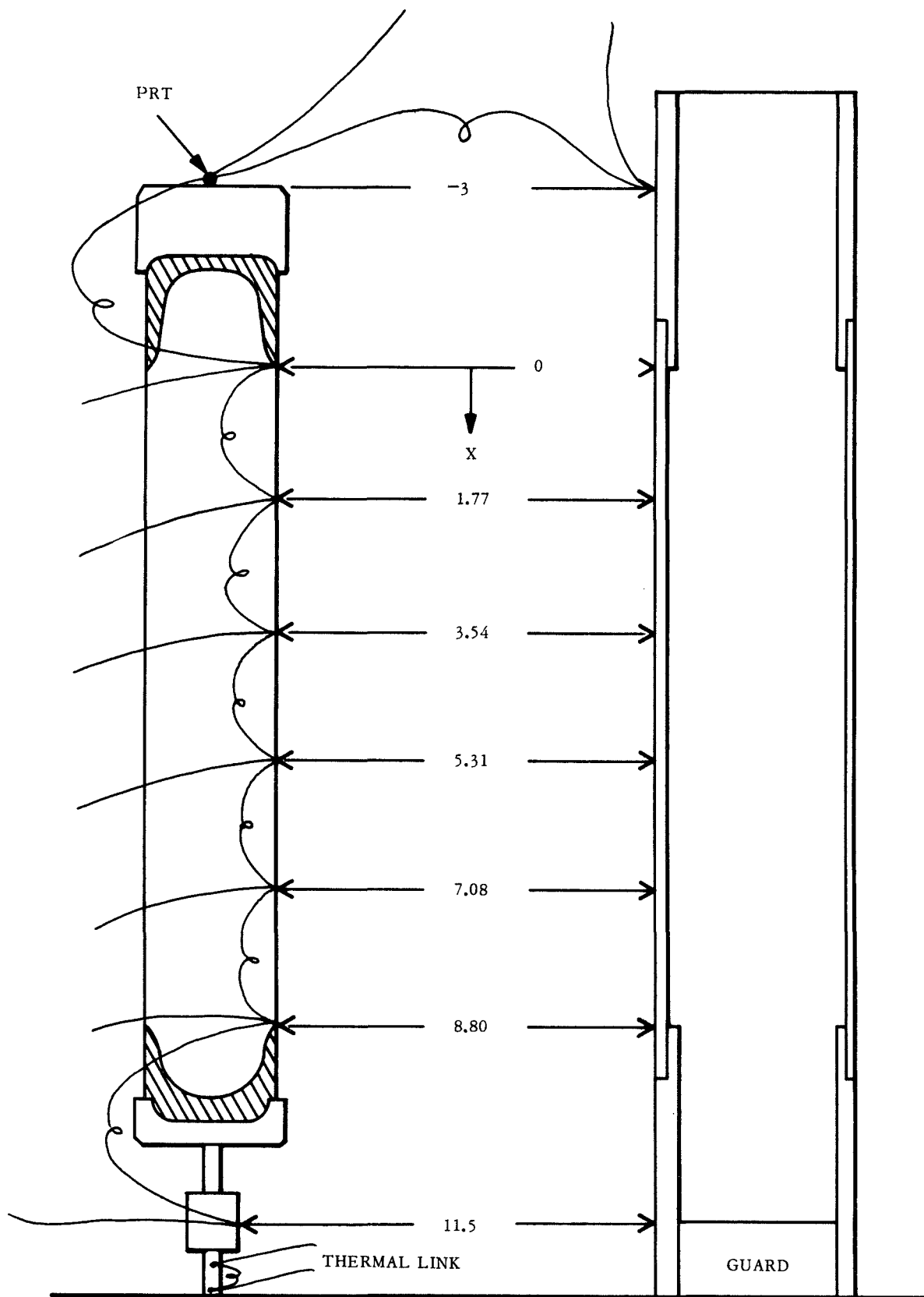


Fig. 18 Instrumentation Schematic - Fiberglass Strut

TEST PROGRAM

Four different boundary conditions were evaluated for the fiberglass strut. These are identified, in sequence, in Table 5.

Table 5

TEST CONDITIONS – FIBERGLASS STRUT

Run No.	Warm End Temperature	Cryogen
I-1	450°R	LN ₂
I-2	530°R	LN ₂
I-3	530°R	LH ₂
I-4	450°R	LH ₂

Prior to closing the assembled test apparatus in the vacuum chamber, all instrumentation and heater wires were checked for electrical continuity. The chamber was then locked up and evacuated to less than 10^{-5} Torr. This condition was held for approximately 24 hr to remove any moisture that might be in the test apparatus.

At the end of the evacuation and hold period, LN₂ was transferred to the penetration cryostat and the system was allowed to cool down to an equilibrium profile at the first test condition. During this period, the outer end temperature, the heat rate, and the total temperature drop across the strut (indicated by the emf across TC No. 8) was tracked. As the outer end temperature approached the test value, current was supplied to heater-1 to prevent further cooling. After the heater adjustment was made, the data was tracked for approximately 24 additional hours to lend assurance of equilibrium at the desired condition.

At the conclusion of the first test, the current through heater-1 was increased to bring the outer end up to 530°R for the second test. Fine adjustments were frequently made in the heater current, as the rest of the strut warmed up, until equilibrium was again established. The second test condition reached equilibrium in approximately 30 hr.

The liquid nitrogen remaining in the cryostat was boiled off by introducing ambient nitrogen gas through the fill port. This was followed with a gaseous helium purge and finally the cryostat was filled with liquid hydrogen for runs I-3 and I-4. Since run I-3 had an ambient outer boundary, the stabilization period was rather short (approximately 19 hr) compared to that for run I-4. The latter required about 50 hr to reach equilibrium.

TEST RESULTS AND ANALYSIS

The results obtained for each test condition included the steady-state heat input at the top of the strut; that flowing into the cryogen, and the temperature profiles in both the thermal guard and the test article. The two heat rates are compared in Table 6.

Table 6

FIBERGLASS STRUT TEST RESULTS

Run No.	Cryogen	Warm End Temperature	Measured Heat Rate	
			Heater Power	Thermal Link
I-1	LN ₂	430°R	0.201 Btu/hr	0.248 Btu/hr
I-2	LN ₂	520°R	0.334 Btu/hr	0.341 Btu/hr
I-3	LH ₂	520°R	0.353 Btu/hr	0.367 Btu/hr
I-4	LH ₂	430°R	0.251 Btu/hr	0.250 Btu/hr

The heater power is that supplied to the top of the strut to maintain it in thermal equilibrium with the top of the thermal guard. The heat rate indicated by the thermal link is that determined from the measured emf across this calibrated resistance between the end of the strut and the cryogen. If there is no interaction between the guard and the test strut then the heat coming in at the top should be equal to that going out at the bottom. Three of the four test points show agreement within 4 percent between the heater

power and the thermal link. However, the first run (I-1) indicates approximately 25 percent more heat leaving at the bottom than is being put in at the top. No explanation for this discrepancy is readily apparent.

The temperature distribution in the fiberglass portion of both the strut and guard are shown in Figs. 19 through 22. It can be seen that the thermal guard was significantly warmer than the strut over the lower half. However, the region between the strut and guard was packed with Dexiglas to eliminate radiation. This was apparently sufficient to damp out any heat transfer from the guard to the test article, as indicated by the good agreement between heater power and thermal link measurements. In this regard, it is interesting to note that run I-1 which showed the maximum variance in heat rate measurements, shows the least variance between strut and guard temperature profiles. This may indicate that the strut was still in a transient condition near the cold end which would produce a high value for the thermal link measurement. The thermal conductivity of the filament wound fiberglass was determined as a function of temperature, using the profiles from Figs. 19 through 22 and the heat flow rates given in Table 6 (heater values). The results are shown on Fig. 23. Twenty of the twenty-four data points fall within ± 10 percent around the straight line which has been drawn through the data. Also shown on Fig. 23 is the curve used in the calculations performed in Task I. It can be seen that the experimental data gives significantly higher values of conductivity above 300°R and somewhat lower values below 300°R . The net effect is to change the previously calculated heat rates as indicated below:

520 $^{\circ}\text{R}$ to CH_4 - 12% increase
400 $^{\circ}\text{R}$ to CH_4 - 4% increase
520 $^{\circ}\text{R}$ to LH_2 - No change
400 $^{\circ}\text{R}$ to LH_2 - 10% decrease

The open symbols shown in Fig. 23 are based upon temperature gradients that were graphically determined at each instrument location. The filled symbols represent average values determined from total length and total temperature drop. In view of the apparent linearity between conductivity and temperature, it is not surprising that the average values fall within the locally determined data.

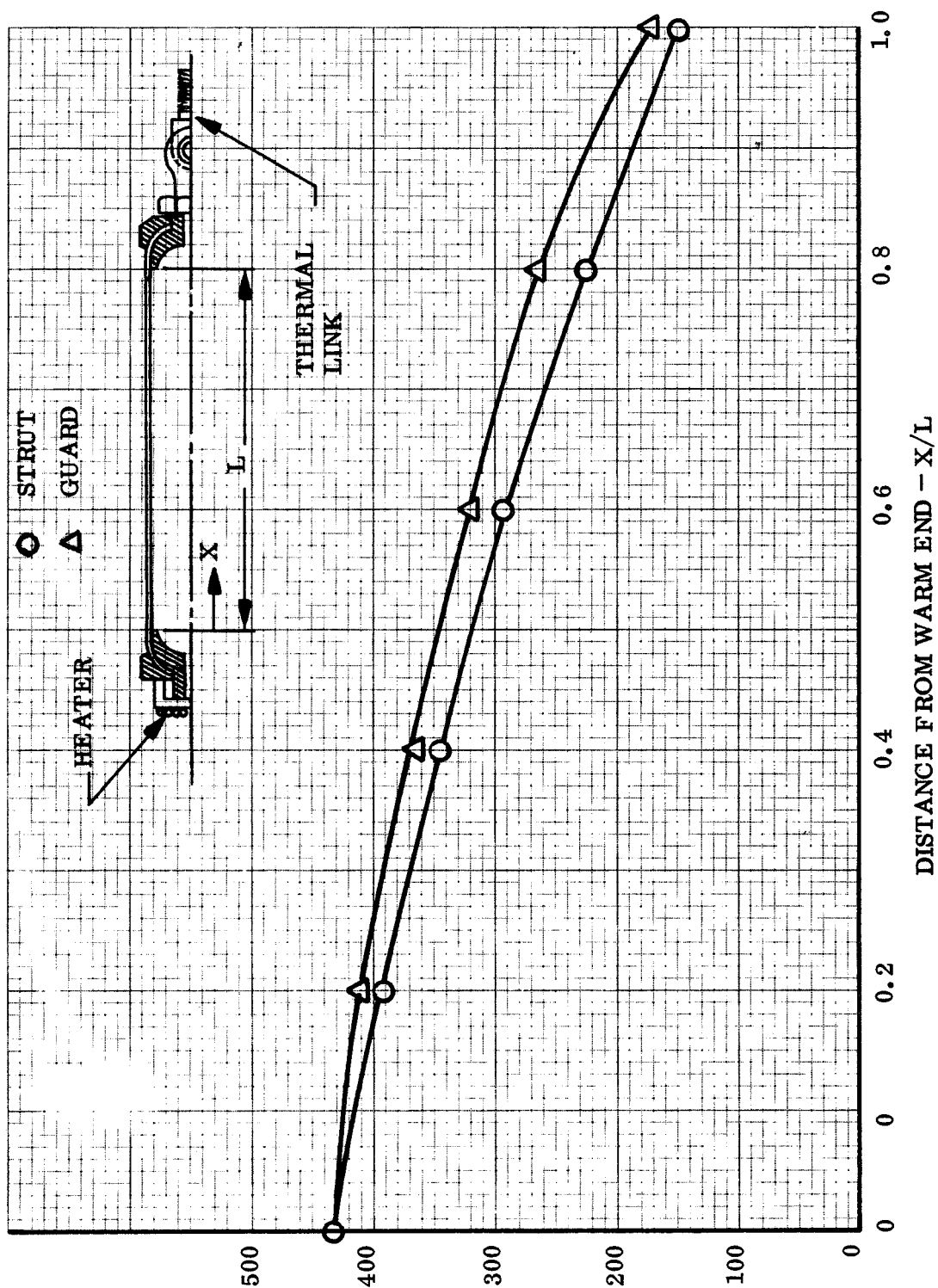


Fig. 19 Temperature Distribution With LN_2 (Run I-1)

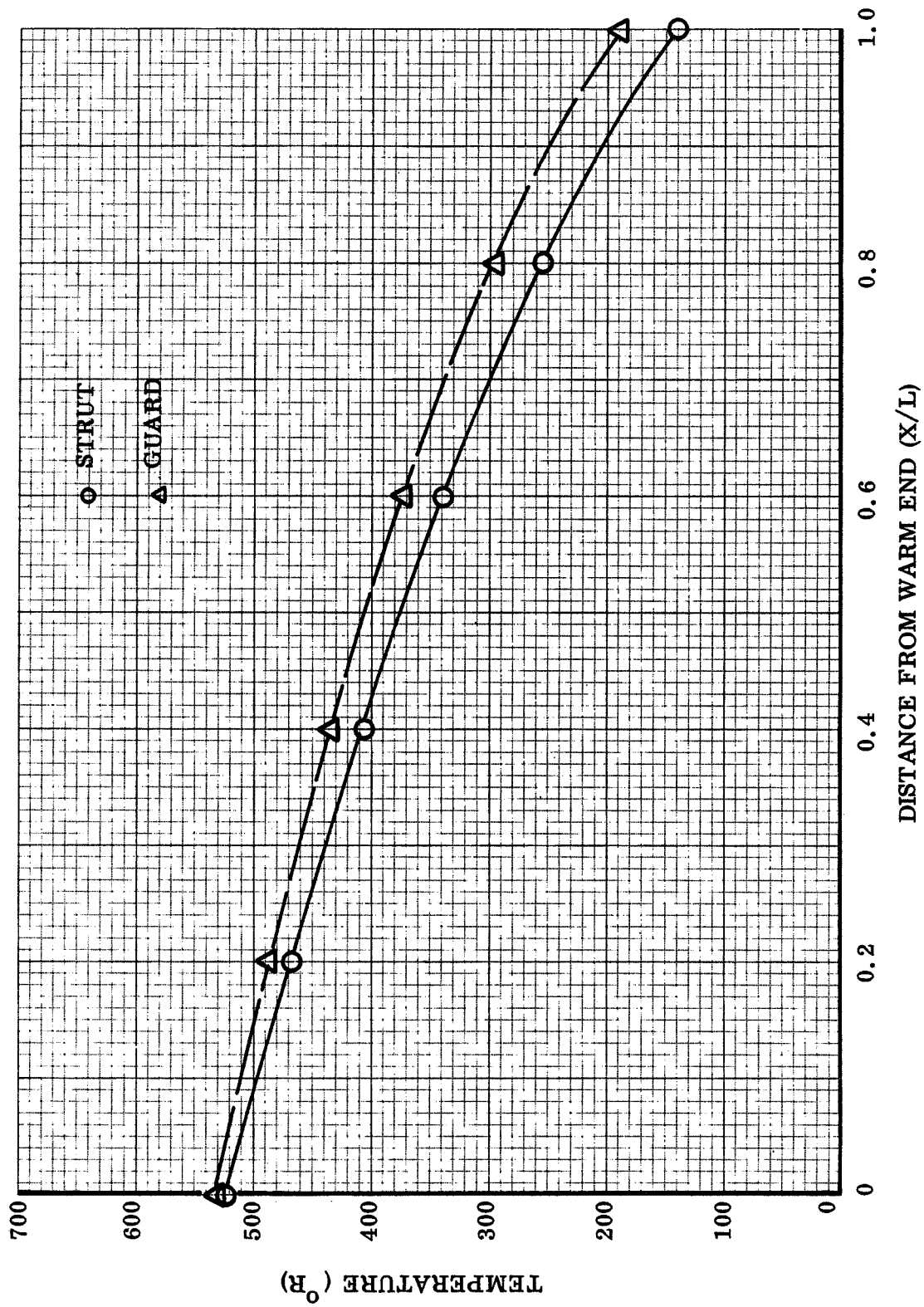


Fig. 20 Temperature Distribution in Fiberglass With LN₂ (Run I-2)

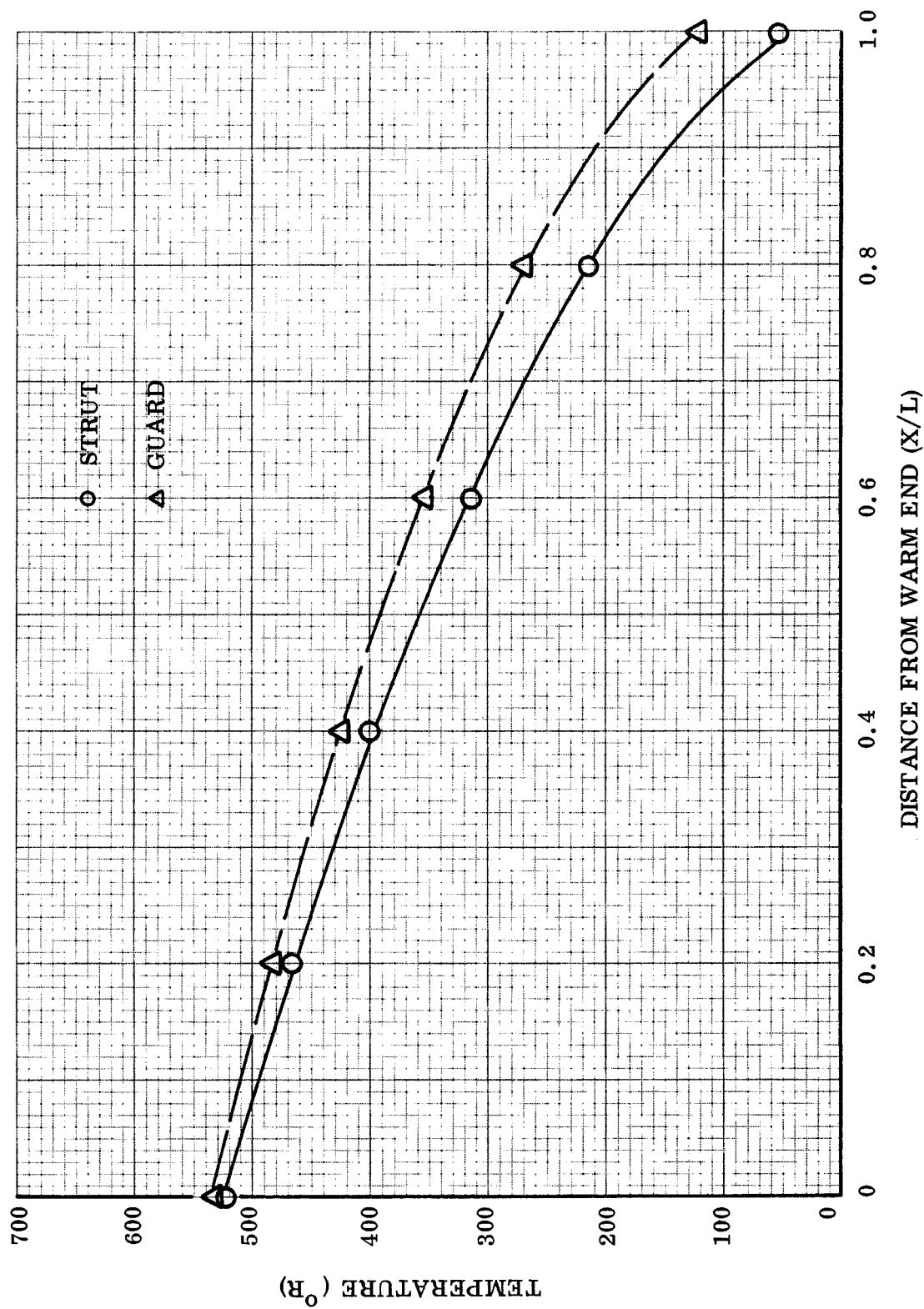


Fig. 21 Temperature Distribution In Fiberglass With LH_2 (Run I-3)

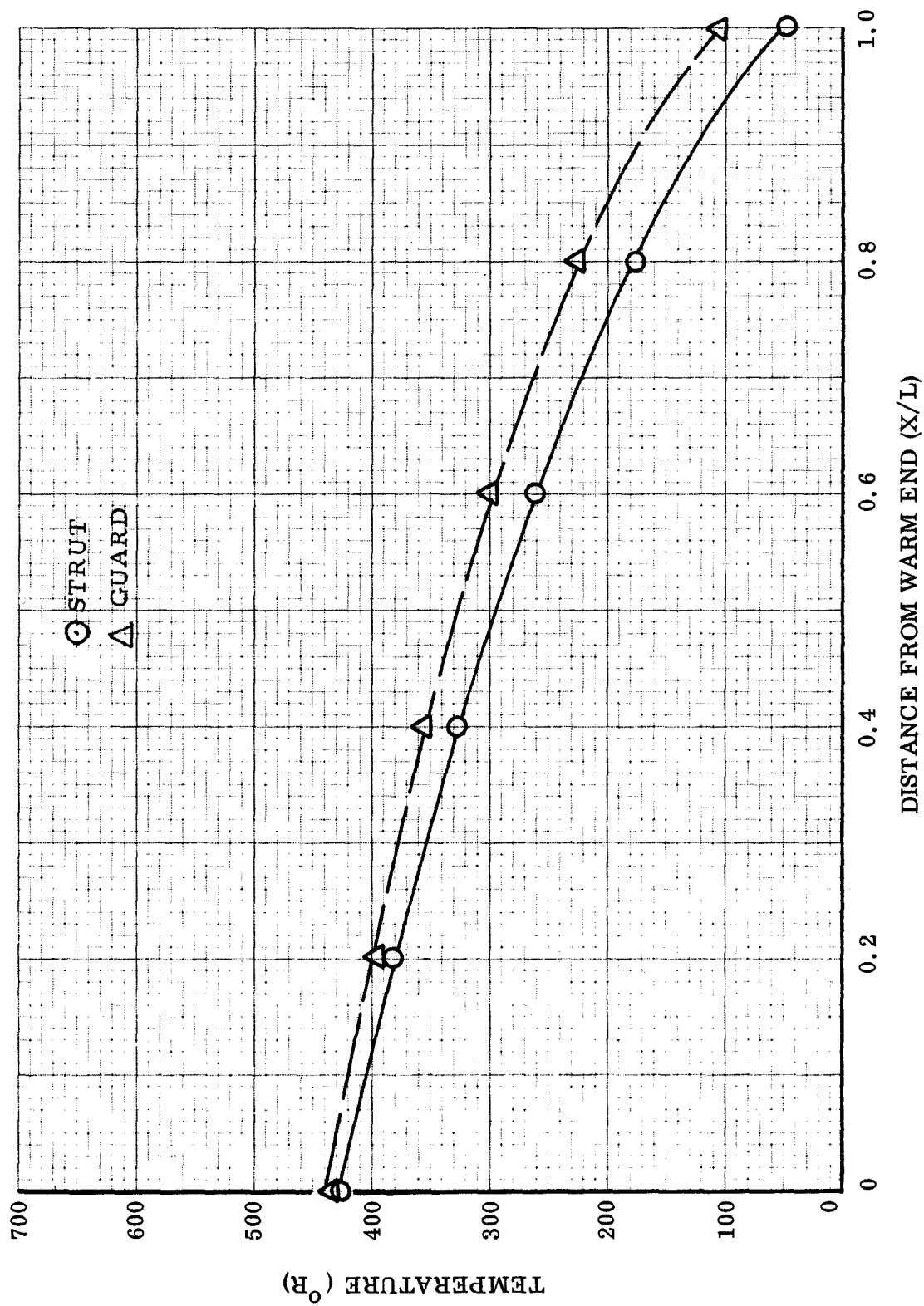


Fig. 22 Temperature Distribution In Fiberglass With LH₂ (Run I-4)

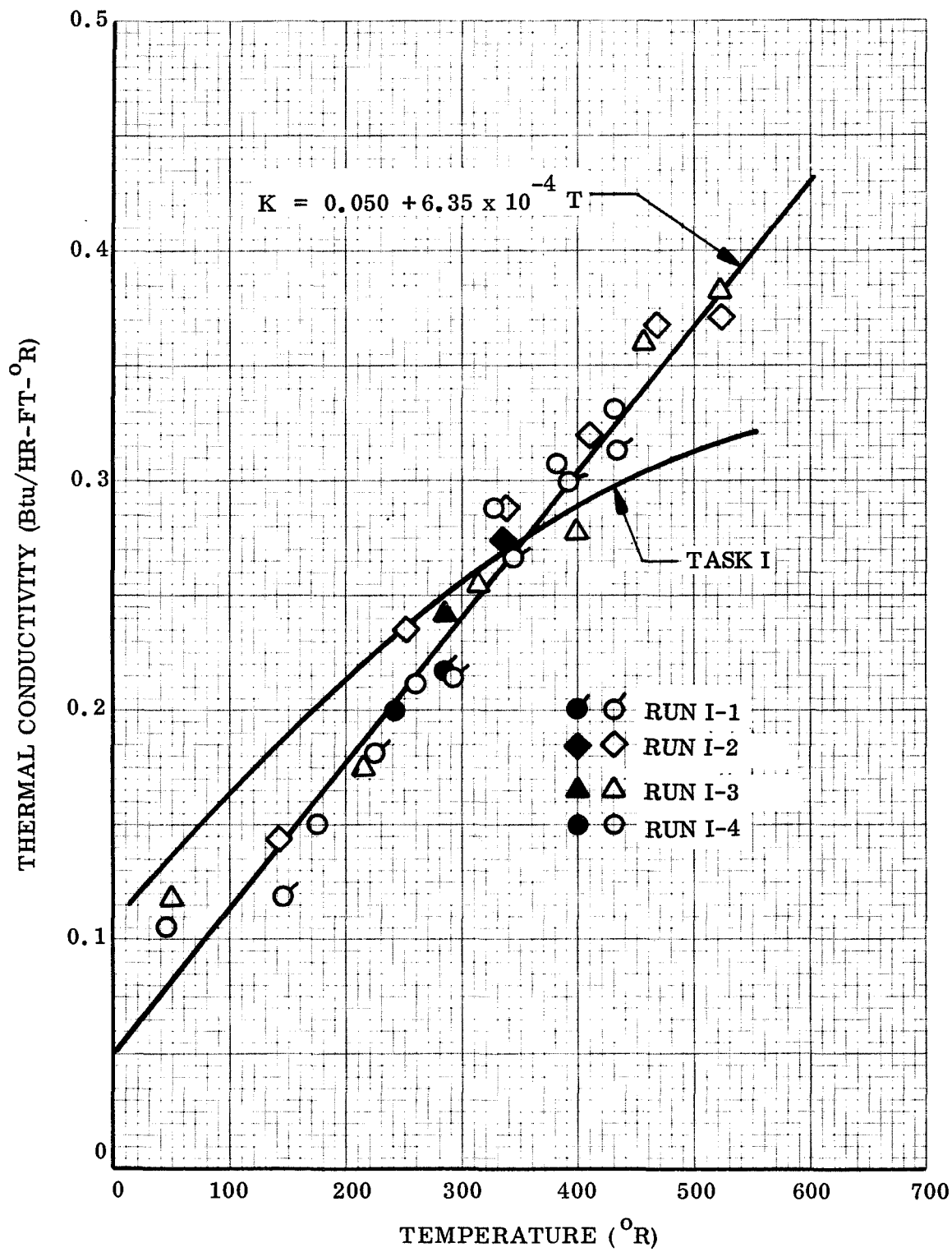


Fig. 23 Thermal Conductivity for Fiberglass Strut

STACKED WASHER CONCEPT

TEST FACILITY DESCRIPTION

The penetration cryostat was replaced with a cryogen container made from a 4-in. section of 12-in. -diameter aluminum pipe. The capacity was approximately 7 liters. A 3/8-in. -thick aluminum plate was centered and welded on the top of this cryostat. This plate was designed with a center hole, drilled and tapped to fit the thermal link, and eight additional holes on a 4-in. -diameter circle, drilled and tapped for attaching thermal guards. Thus, the test articles were assembled in the same manner as with the penetration cryostat.

The revised test facility used a cryogen dewar as a vacuum chamber, (see Fig. 26) with internal dimensions of 15-in. -diameter by 36-in. deep. This was a double wall dewar with multilayer insulation in the annulus. The dewar was equipped with a vacuum tight lid to which all the internal hardware was attached. The cryostat was suspended from the lid with three 1/2-in. -diameter stainless steel tubes. Two of these served as fill and vent lines, and the third was capped off at the cryostat.

The third component in this revised test facility was an auxiliary cryogen container. This pot was used to provide localized guard cooling, as required for tests on both the stacked washer and ball and clamp concepts, or to provide the necessary coolant flow for the cooled strut concept. The auxiliary cooling pot was made from a 14-inch section of 3-in. -diameter aluminum pipe. It was suspended from the dewar lid with a 1/2-inch stainless steel tube which also served as a fill line. An internal heater was put inside the pot to vary the boiloff rate as required for the cooled strut tests. Both the auxiliary and primary cryostats were insulated with 50 layers of double aluminized Mylar and Dexiglas spacers.

TEST ARTICLE DESCRIPTION

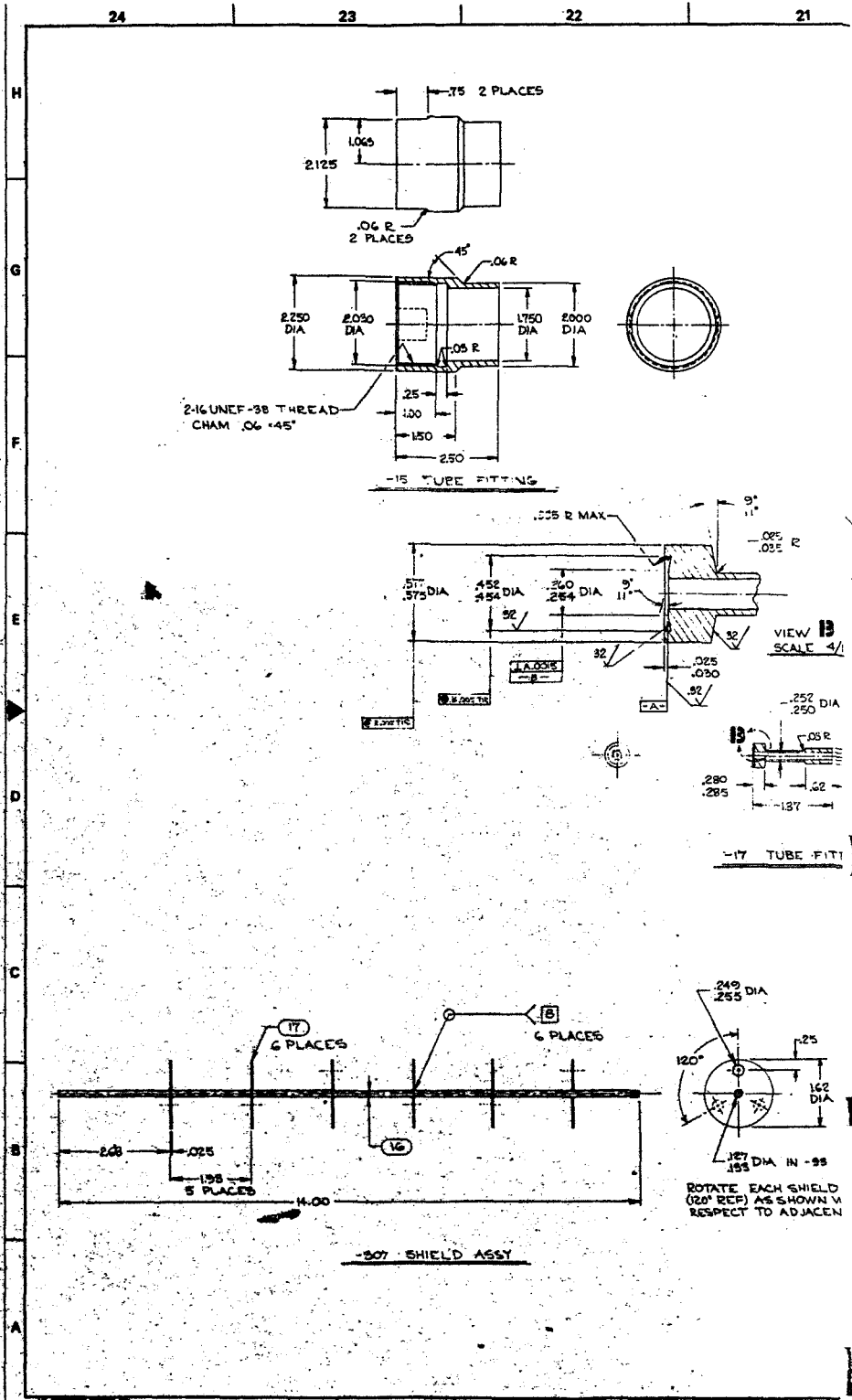
The -501 assembly on Fig. 24 shows the design features of the test article that was used for evaluating the stacked washer concept. This design provides very close thermal simulation to that of an 18-in. , flight-type strut for a 15-ft-diameter

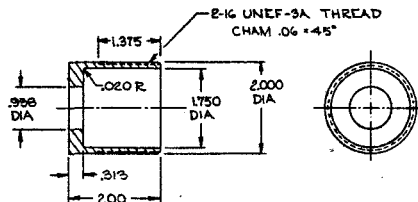
liquid hydrogen tank. The calculated thermal resistance of the controlling element (i.e., the washer stacks) was 10 percent larger for the test article. However, an oversized titanium tube was available and was used for the tube body. This resulted in the net thermal resistance of the test article being 6 percent smaller than that of the flight strut.

Several features are included in the test article that represent deviations from the flight concept; some dictated by economic considerations and some dictated by test requirements. Most noticeable change is the use of flat washers for testing, although the flight concept employs conical washers for self alignment. In the test article, alignment was achieved with the aid of three circular "feeler" gages which were placed around the periphery, during assembly, to hold the washers away from the wall until the locknut was torqued sufficiently to hold the washers in place. The feeler gages were then removed and the entire washer assembly was screwed into the threaded tube body. In the flight concept the washer assembly would be welded to the tube but the threaded joint was selected for the test article because it permitted the same tube body to be used with the ball and clamp concept without compromising the test results for either.

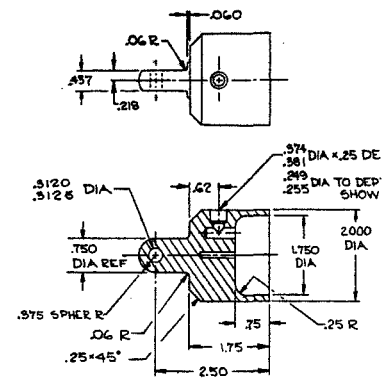
The contact resistance between washers is known to be dependent upon the bearing pressure. The test article shown in Fig. 24 incorporates a feature which allowed the loading to be changed remotely, while the strut was under test. The forward washer which compresses the washer stacks, is fitted with a pin, perpendicular to the strut axis, which goes through slots in the hollow center body. A small wire was then attached to this pin and brought out through the center of the tube, over a pulley system, and attached to calibrated weights. With the center body connected directly to the cryostat, it would remain stationary, and the tension in the wire would cause the forward loading washer to slide along the center body until the calibrated load is transmitted to the washer assembly. Figure 25 is a photograph of the finished parts prior to assembly.

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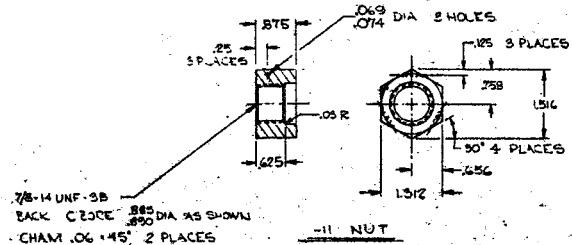
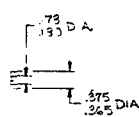




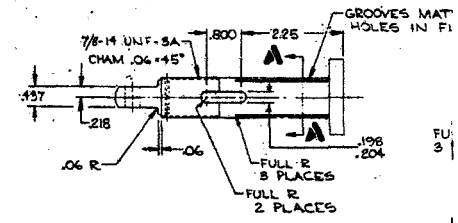
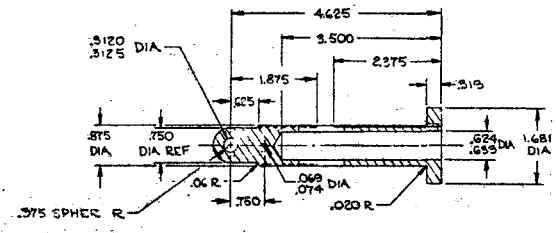
-9 TUBE FITTING



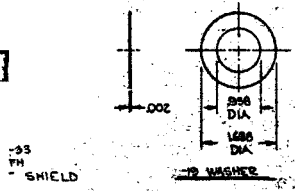
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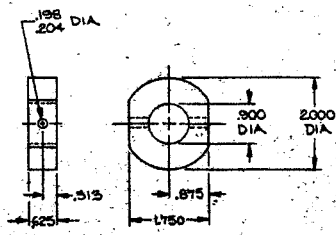
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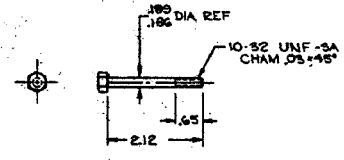
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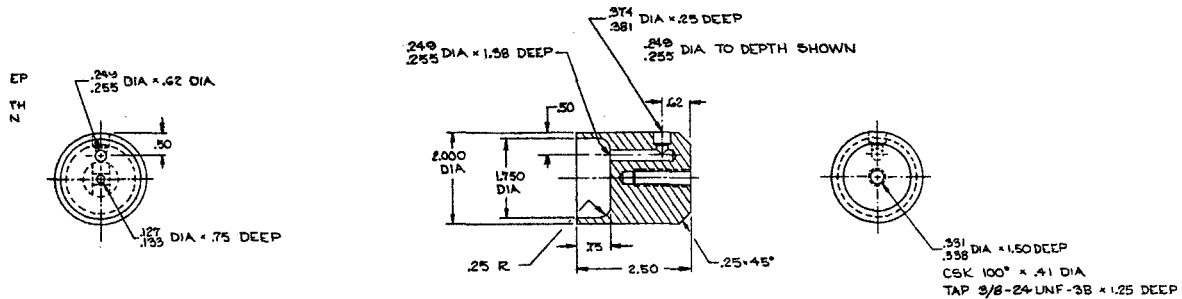
-12 WASHER



-13 WASHER

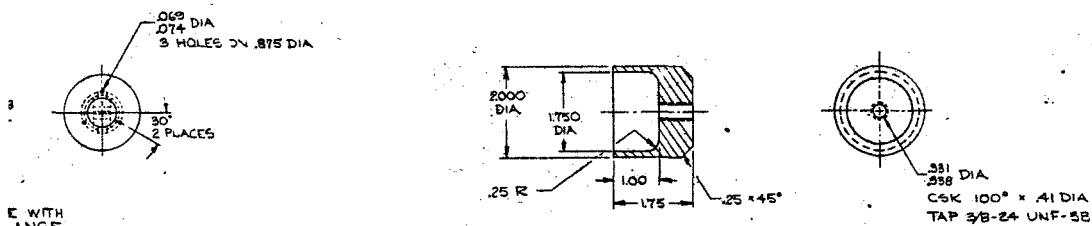


-25 BOLT



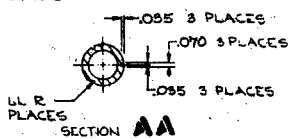
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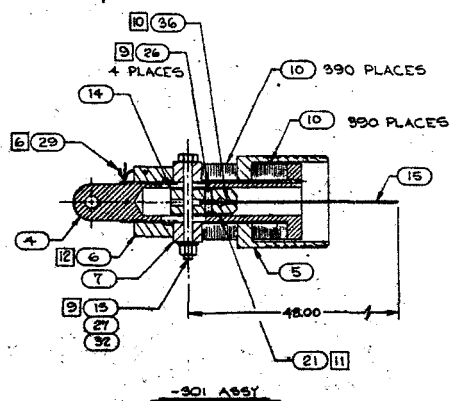
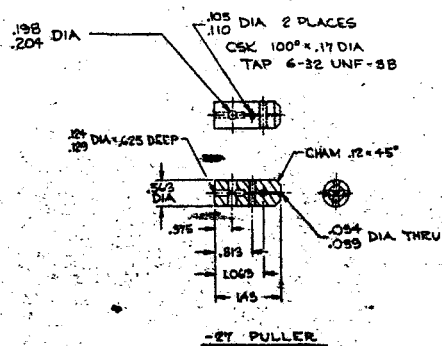


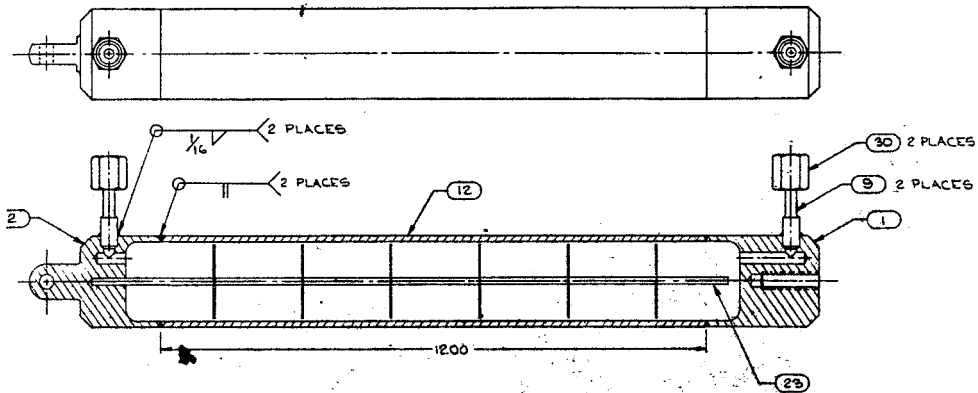
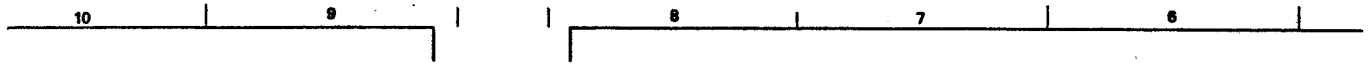
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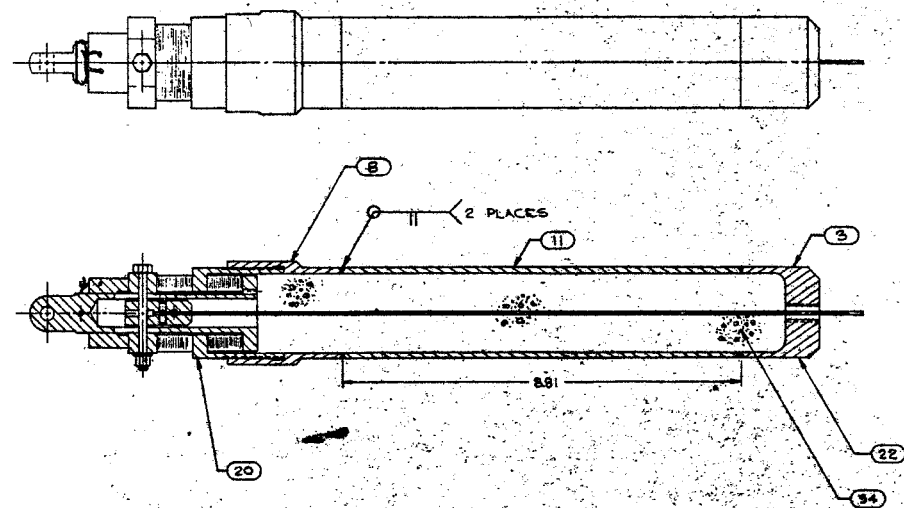


-5 END FITTING





-503 ASSY



-501 ASSY

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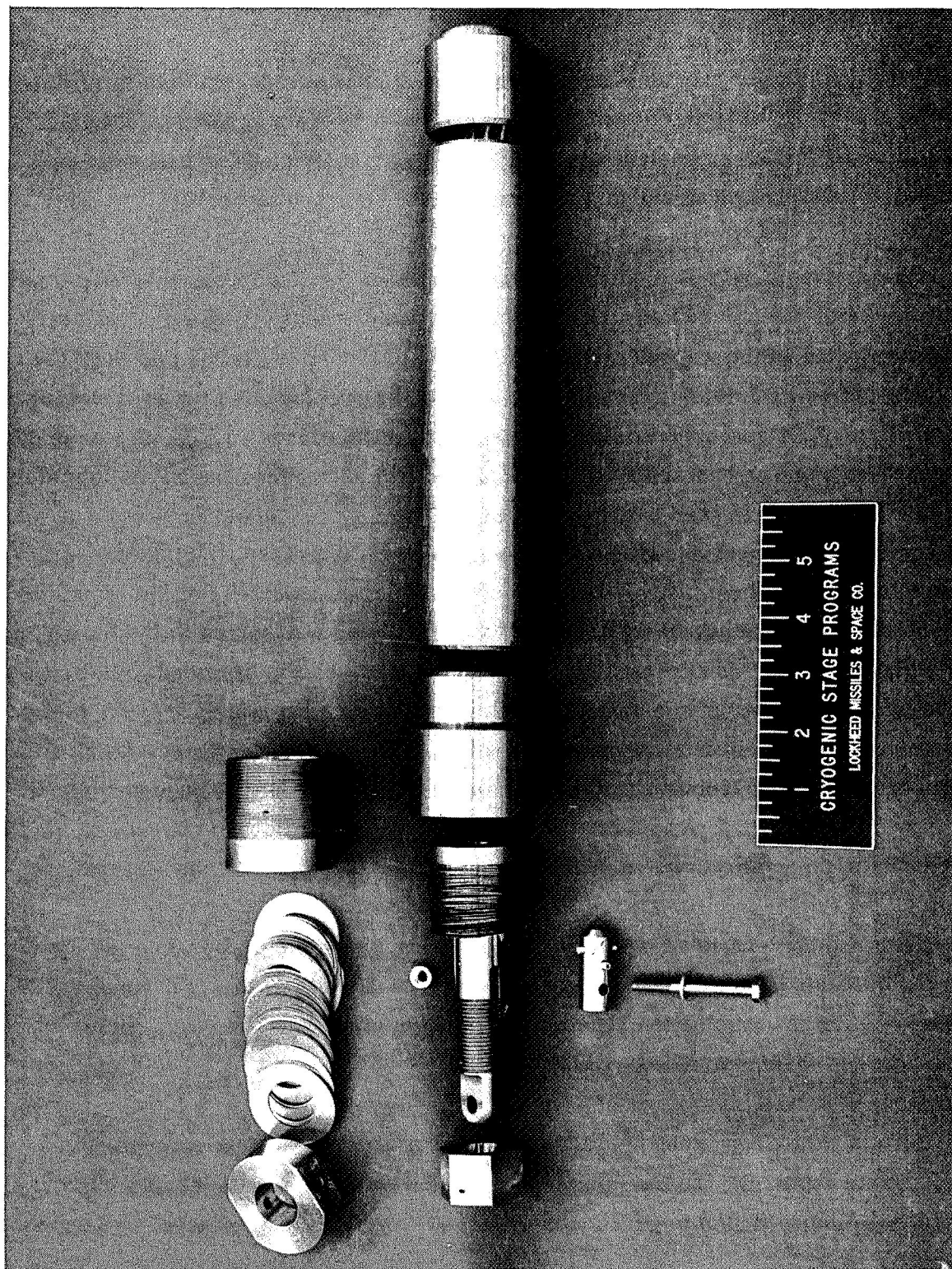


Fig. 25 Stacked Washer Concept Prior to Final Assembly

TEST ASSEMBLY

The test assembly used for the stacked washer strut is shown schematically in Fig. 26. The strut was attached to the top of the thermal link with the same clevis attachment previously used for the fiberglass strut. The same Evanohm heater assembly was bolted to the outer end to serve as a heat source and temperature controller.

The ideal thermal guard has the same temperature distribution as the test article. In view of the fact that the stacked washer test strut has the major portion of its resistance concentrated in the washer stacks, and a smaller portion distributed over the body, it is difficult to match the various gradients by proper selection of material properties alone. Consequently, the five-part guard illustrated in Fig. 26 was used in conjunction with two heaters (Nos. 2 and 3) and two sets of cooling wires to match the gradients in the tube body and the washers. The top portion of the guard was made to be isothermal, using 3-in.-diameter copper tubing. A 2-1/8-in.-diameter stainless steel tube was joined to the copper tube. This stainless steel was the guard for the titanium tube. Another short section of copper tubing was joined to the lower edge of the stainless steel to be concentric with the relatively thick, threaded joint between the tube body and the washer assembly. It also provided a base for attaching cooling wires which would minimize the potential for uneven cooling. The thermal guard for the washers was made from a short section of teflon tubing. This was joined to the short copper tubing and to an aluminum lower guard. The latter was the guard for the structure between the cryostat and the washers (i. e., thermal link, attachment clevis).

The heaters and coolers were provided for adjusting the heat flow between the upper and lower portions of the guard to make up for any mismatch in thermal resistances. It further minimizes interaction between the thermal guard and the test article; the annulus between them was filled with Dexiglas.

Figure 27 is a photograph showing the support strut and the lower aluminum guard attached to the cryostat, and the remainder of the thermal gradient just before it is installed and insulated. This guard was also insulated with 120 layers of mylar and Dexiglas.

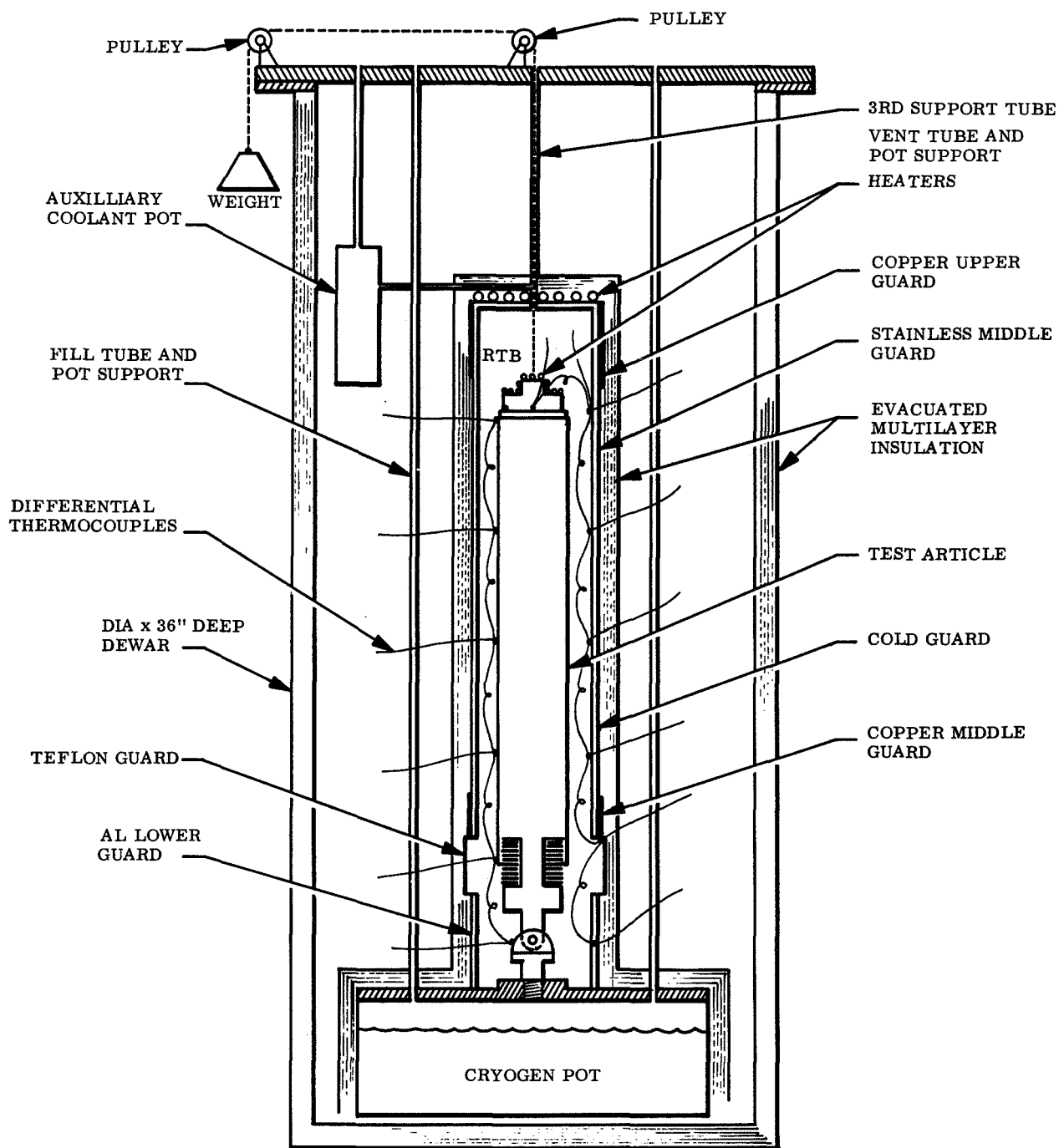


Fig. 26 Test Assembly for Stacked Washer Concept

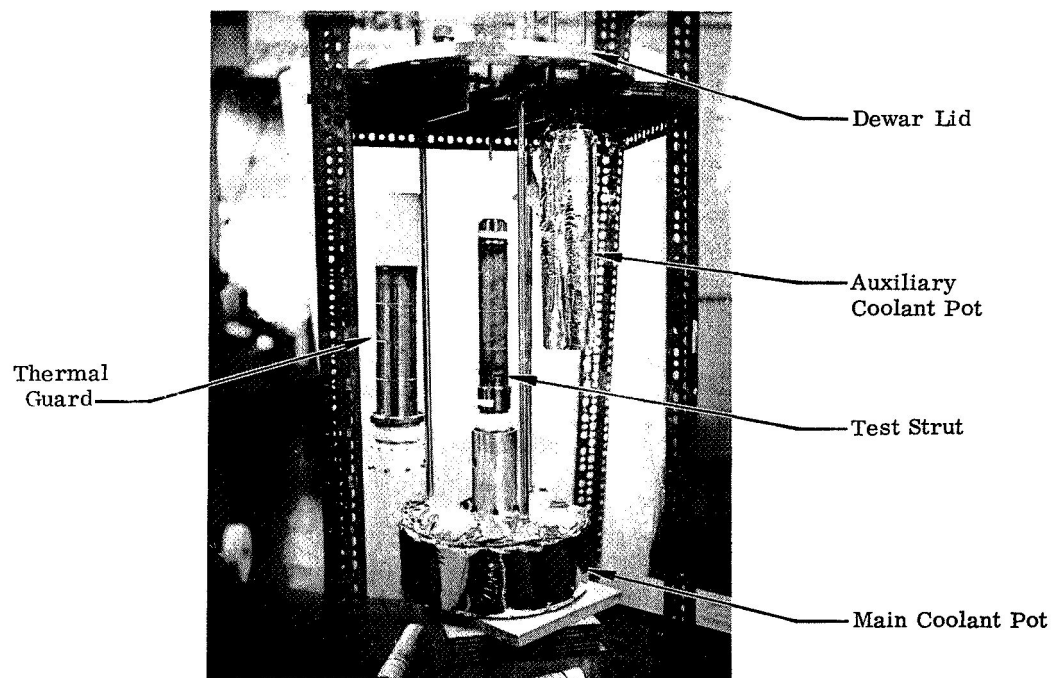


Fig. 27 Stacked Washer Support Concept and Test Facility Components Prior to Installation of Thermal Guard

INSTRUMENTATION

The instrumentation was similar to that used for the fiberglass strut. Figure 28 shows the number and location of the measurements. A PRT was attached to the heater assembly to measure the warm boundary temperature. Seven chromel-constantan, differential thermocouples were placed on the strut and an eighth on the cryostat. These were wired in series and referenced to the PRT, to give the temperature distribution in the titanium tube body and the temperature drop across the stack of washers. Eight differential thermocouples were placed on the thermal guard — one at the top and the other seven placed at the same elevations as those on the test article. A ninth one was on the cryostat. These were also in series and referenced to the PRT.

TEST PROGRAM

The objective of this test program was to determine the effectiveness of the contact resistance obtained from the stacked washer concept, under different loading pressures and temperatures. A complete summary of the test conducted on this support concept is presented in Table 7.

Table 7

STACKED WASHER STRUT TEST CONDITIONS

Run	Boundary Temperature		Load (lb)	Washer Bearing Pressure (psi)	Midguard Bias * (°R)
	Warm (°R)	Cold			
II-1	446	LN ₂	20	13	0
II-2	440	LN ₂	20	13	5.4
II-3	448	LN ₂	0	0	9.0
II-4	536	LN ₂	0	0	6.3
II-5	536	LN ₂	20	13	7.0
II-6	536	LN ₂	20	13	22.6
II-7	536	LN ₂	80	52	22.6
II-8	539	LH ₂	20	13	7.0
II-9	445	LH ₂	20	13	7.0

*Midguard bias is the ΔT between the strut and middle guard.

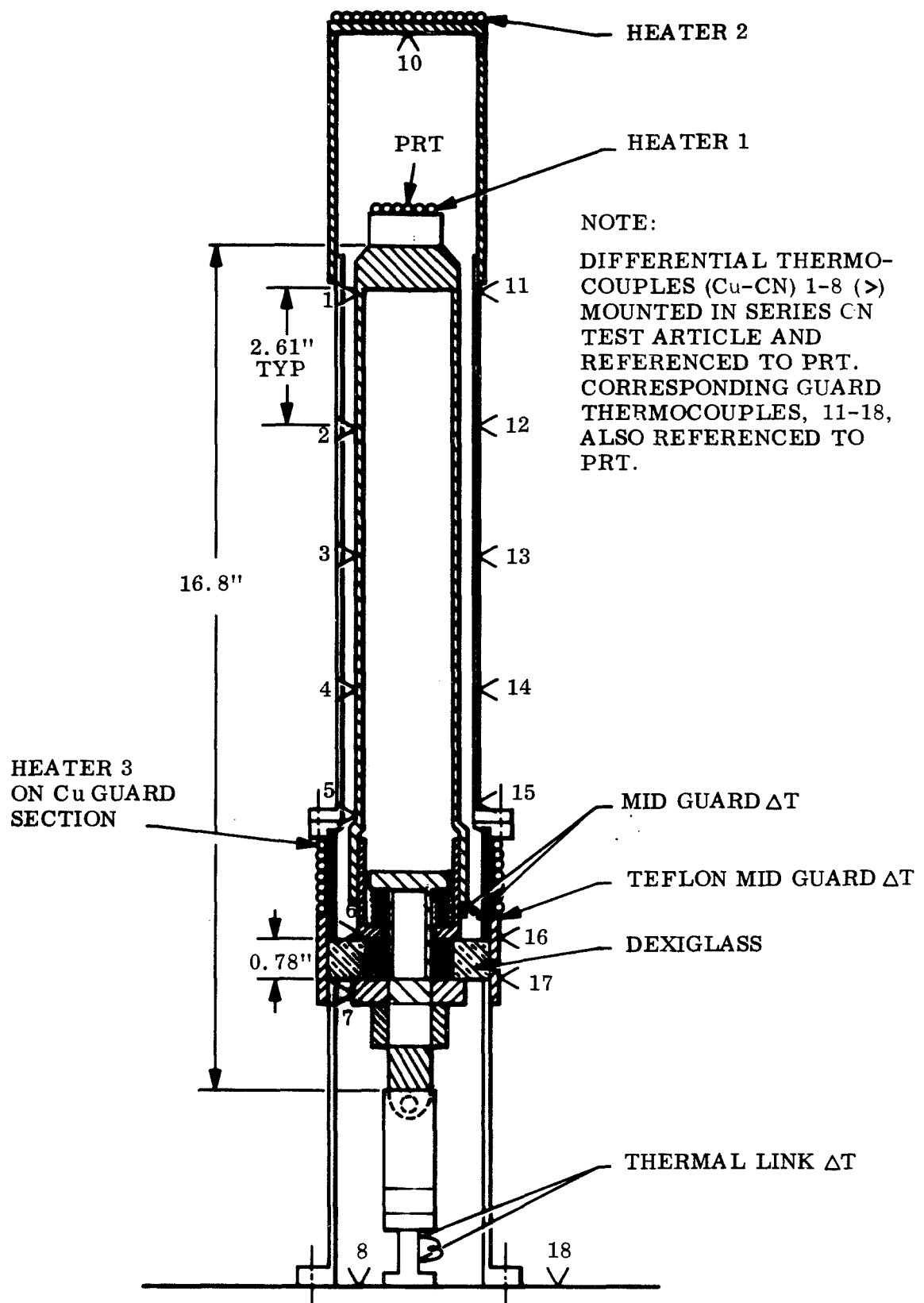


Fig. 28 Instrumentation Schematic - Stacked Washer Concept

The stacked washer strut was the first to be tested in the thermal conductivity test apparatus, and it was necessary to know the approximate storage time for the cryogen in both the main cryostat and the auxiliary cooling pot. Therefore, a boiloff test was conducted on both before the actual strut tests were initiated. First, the dewar was evacuated to 1×10^{-6} Torr and the auxiliary pot was filled with LN_2 . The boiloff rate quickly leveled off at approximately 32 standard liters per hour of GN_2 which represented approximately 20 hr storage time for the 1-liter container. The main cryostat was then filled and the flow rate was monitored. The initial boiloff rate indicated that the LN_2 would last for 23 hr, and after 7 hr the boiloff rate had dropped off approximately 50 percent. These results indicated that the cooling pot limited the time between fills and, further, that it would be necessary to refill the pots twice each 24-hr day to avoid disturbing a test through the loss of the heat sink. For the liquid hydrogen tests, the refill frequency would be four times per day.

During these boiloff tests the temperature of the outer end of the strut dropped to 380°R . However, the intent was to run all the support systems with approximately the same boundary temperatures as used for the fiberglass strut. Therefore, current was supplied to heater No. 1 to bring the temperature up for the first test. This took approximately four hours, and during that time heaters Nos. 2 and 3 were being monitored and adjusted to keep the top and middle guard temperatures in a condition of quasi-equilibrium with the strut. A 20-lb weight was then loaded to compress the washers, and the heat rates, total temperature differentials, and outer end temperature were tracked for 48 hr, with no significant changes occurring in the last 16 hr.

It had been planned to increase the loading on the washers to 200 lb for the second test and then come back down to the 20-lb loading to see if permanent set would occur which adversely affects the contact resistance. However, when the load reached 140 lb (it was being applied with 20-lb weights), the O-ring seal in the top of the dewar broke loose, causing a loss of vacuum. When the load was removed, the vacuum would return. Inspection indicated that the central pulley was slightly off-center and thus, the tension wire which penetrated the lid through the seal actually applied a side load which distorted the O-ring sufficiently to break the vacuum. The pulley was adjusted but the decision was made to defer the high loads rather than risk having to dismantle the test to replace a seal.

A preliminary evaluation was made on the data from the first test which indicated that the temperatures in the teflon midguard were too low when the temperatures at the top and bottom of the upper (stainless steel guard) were matched to the corresponding ones in the titanium tube on the test article. This was primarily because the resistance in the washer stack was lower than anticipated. Further, the test results indicated some heat loss from the test strut to the thermal guard. Therefore, the decision was made to raise the midguard temperature approximately 5°R and repeat the test. Also, it was decided that this could best be accomplished by eliminating the coolers and readjusting the power input to heaters 2 and 3. This was done by blowing the nitrogen from the cooling pot and replacing it with ambient helium which actually contributed a constant heat input and reduced heater power requirements. The procedures for test No. 2 were the same as for test No. 1 with one exception — in the first test, the current to heater 3 was regulated to give a zero emf across the midguard bias thermocouple (i. e., same temperature for the strut and the copper guard) but in the second test it was adjusted to maintain a 120-microvolt output which corresponds to approximately 5 deg elevation in guard temperature above that of the strut. This test condition was held for 48 hr with an approximate net reduction of 3 percent in heat input across the thermal link and 10 percent from heater No. 1.

At the conclusion of test No. 2, the 20-lb weight was removed for test No. 3. A preliminary evaluation of the data indicated that the heater power and thermal link were still not in agreement, although the discrepancy had decreased. Therefore, the midguard bias was increased to 240 microvolts ($\sim 10^{\circ}\text{R}$) and the test procedures were repeated until equilibrium was indicated. These test conditions were held for 48 hr.

At the conclusion of the third test the current to heater No. 1 was increased to bring the strut warm end temperature up to 536°R . When the PRT approached the new temperature setting, heater Nos. 2 and 3 were adjusted to bring the guard back into the desired state of equilibrium with the strut. For heater No. 3 this meant maintaining a 240 microvolt midguard bias.

Test No. 5 was run at the same conditions as test No. 4, with the exception of the compressive load. This was again returned to 20 lb to provide the same comparison with load at 536°R that runs 2 and 3 provide at 440°R.

For tests 6 and 7, the midguard bias was increased to 730 microvolts in an attempt to further reduce the difference between indicated heat flows at the top and bottom of the strut. For the seventh test the loading was increased to 80 lb to provide a direct comparison over a load range of four to one. When applying the weights, a vacuum dump occurred at 60 lb. The O-ring was reseated, however, without dismantling the apparatus and the vacuum was quickly regained without further mishap.

The two tests with LH₂ were conducted to check the effect of cold boundary temperature. Only one compressive loading was to be tested with the hydrogen; 20 lb was selected because most of the nitrogen data was taken at that point. The midguard bias was held at 260 microvolts for both tests, thus making these two tests comparable to runs 3 and 5.

TEST RESULTS AND ANALYSIS

The performance criteria by which the support concept must be judged is the heat rate to the cryogen. Other data, such as temperature profiles only serve to help in understanding the mechanism by which the heat rate occurs. Table 8 contains a summary of the test results for the stacked washer concept.

Table 8
TEST RESULTS - STACKED WASHER CONCEPT

Run No.	Warm (°R)	Cold	Load (lbs)	Midguard Bias (°R)	No. 1 Heater Power (Btu/hr)	Thermal Link Power (Btu/hr)	Washer Interface Conductance (Btu/hr-ft ² °R)
II-1	446	LN ₂	20	0	0.983	0.614	55
II-2	440	LN ₂	20	5.4	0.853	0.586	
II-3	448	LN ₂	0	9.0	0.764	0.645	38
II-4	536	LN ₂	0	6.3	1.44	1.04	53
II-5	536	LN ₂	20	7.0	1.53	1.06	69
II-6	536	LN ₂	20	22.6	1.12	1.18	74
II-7	536	LN ₂	80	22.6	1.52	1.53	119
IV-8	539	LH ₂	20	7.0	1.63	1.35	74
III-9	445	LH ₂	20	7.0	1.03	0.87	56

Two heat rates are presented in Table 8. The heater power is that supplied to the top of the strut to maintain it in thermal equilibrium with the top of the guard. The thermal link power is that determined from the measured emf across the calibrated resistance. Except for runs II-6 and II-7, the thermal link power is significantly less than that put out by the heater which indicates a heat loss from the test article. From Table 8, it is noted that both of these runs have much higher midguard bias than the others. This is a temperature elevation of the copper guard section above that of the strut (see Fig. 28), which was impressed across the midguard differential thermocouple and used as a control for heater 3.

The temperature profiles are shown in Figs. 29 - 36. These indicate that in the upper part of the guard, where radiation might be a problem, the guard and strut follow reasonably well. However, it is noted that for the last two nitrogen tests the upper guard is warmer than the titanium tube while it is cooler for the rest of the runs. Although it would appear that the heat loss occurs by radiation to the upper guard, no correlation could be obtained between the power loss and integrated fourth power temperature differences.

The temperature agreement between the test article and thermal guard is poor in the region of the stacked washers. However, the temperature drop in the teflon thermal guard is approximately as expected; but that across the washers is less than expected. This is a consequence of two things: (1) the washer conductance is higher than that used in pre-test calculations, and (2) the test article had a high thermal resistance between the washers and the cryogen which had not been anticipated. This latter conclusion is based upon the consistently high temperature difference between T_7 and T_8 , which may be caused by poor contact between the strut and the clevis adapter at the thermal link.

The performance of the washers can be deduced from the schematic and the thermal analog circuit illustrated in Fig. 37. The resistance for each stack of washers is given by:

$$R_w = \frac{2(T_6 - T_7)}{Q_T}$$

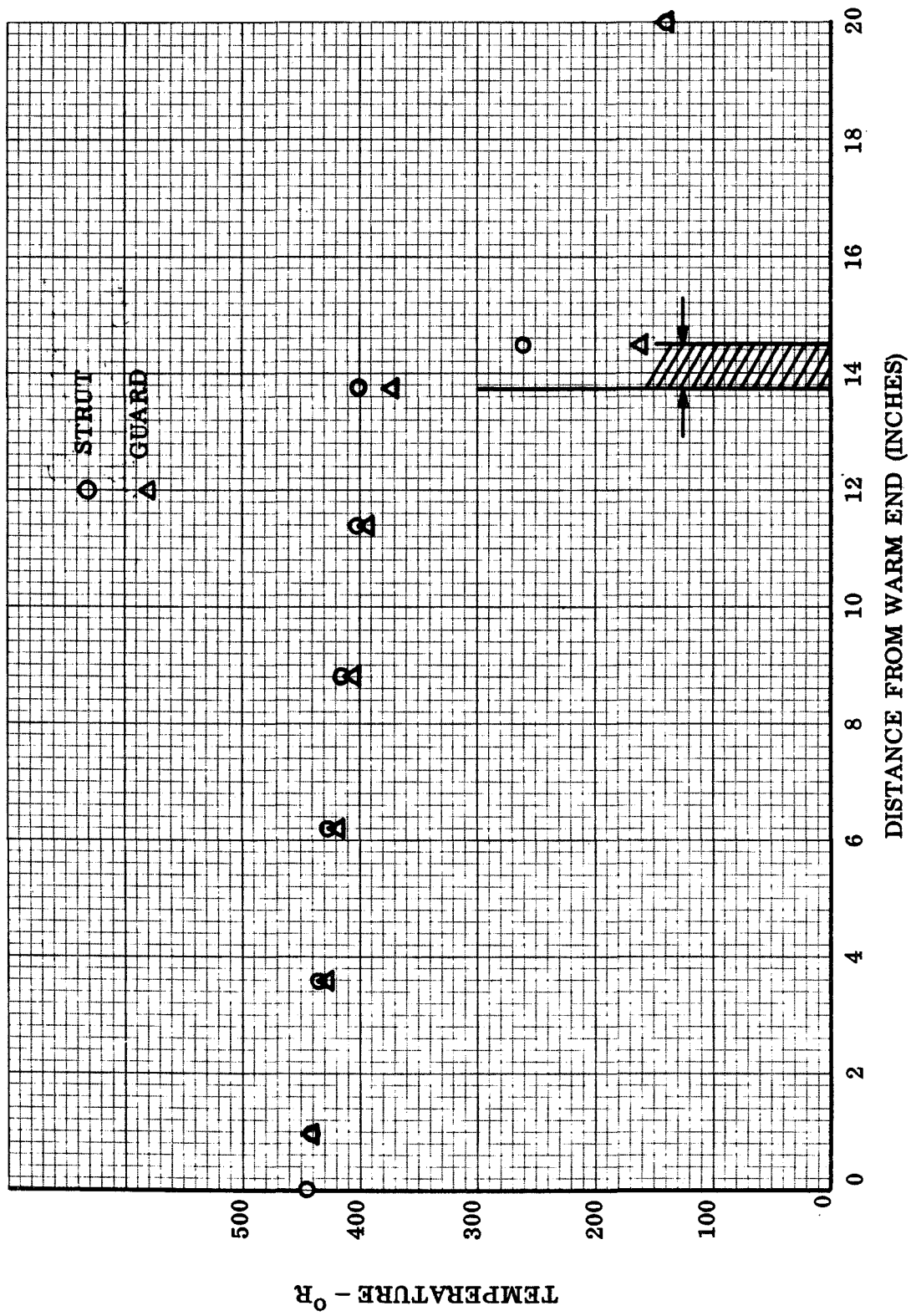


Fig. 29 - Temperature Profile - Run 31-3

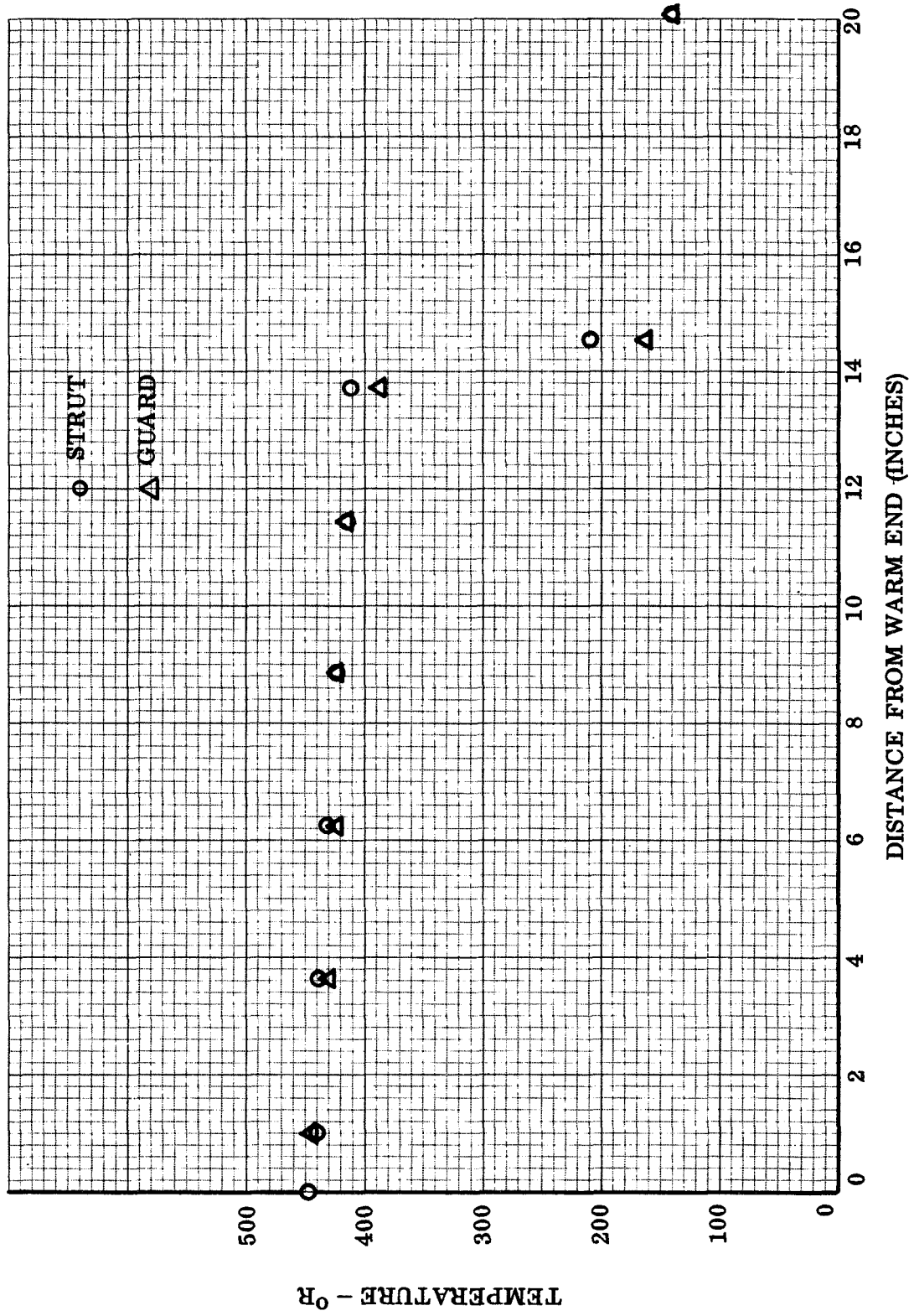


Fig. 30 - Temperature Profile - Run II-3

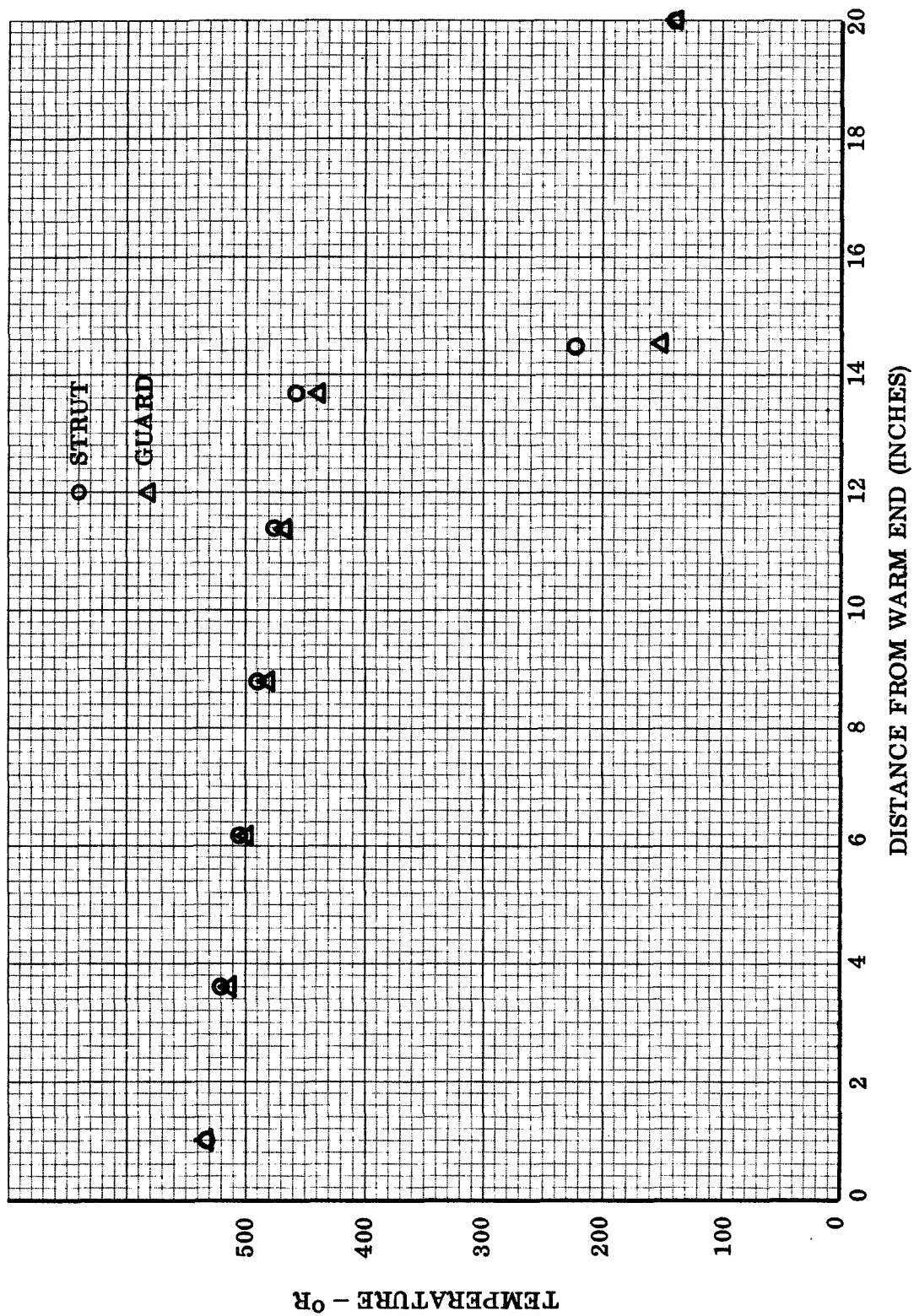


Fig. 31 Temperature Profile - Run II-4

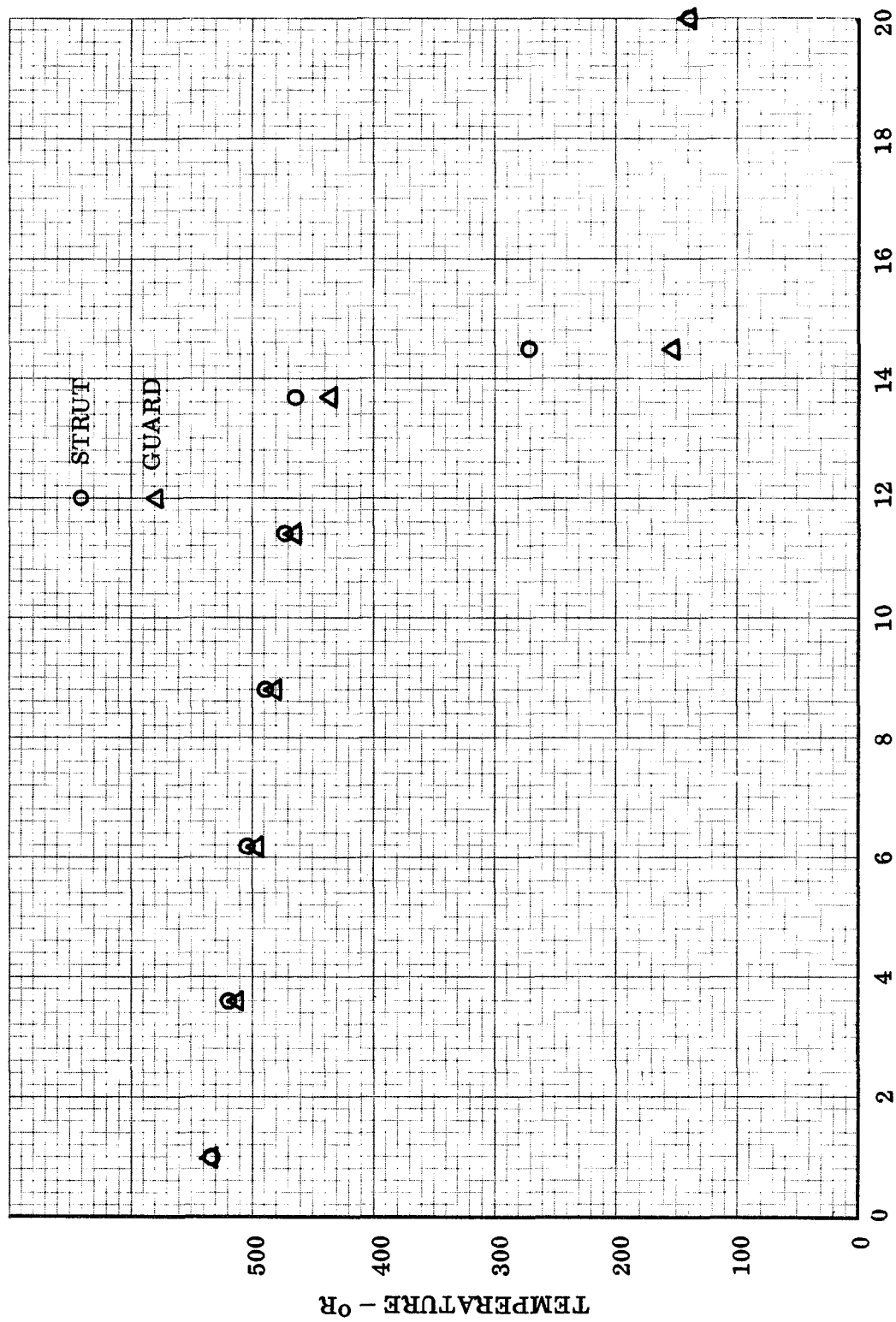


Fig 32 -- Temperature Profile -- Run II-5

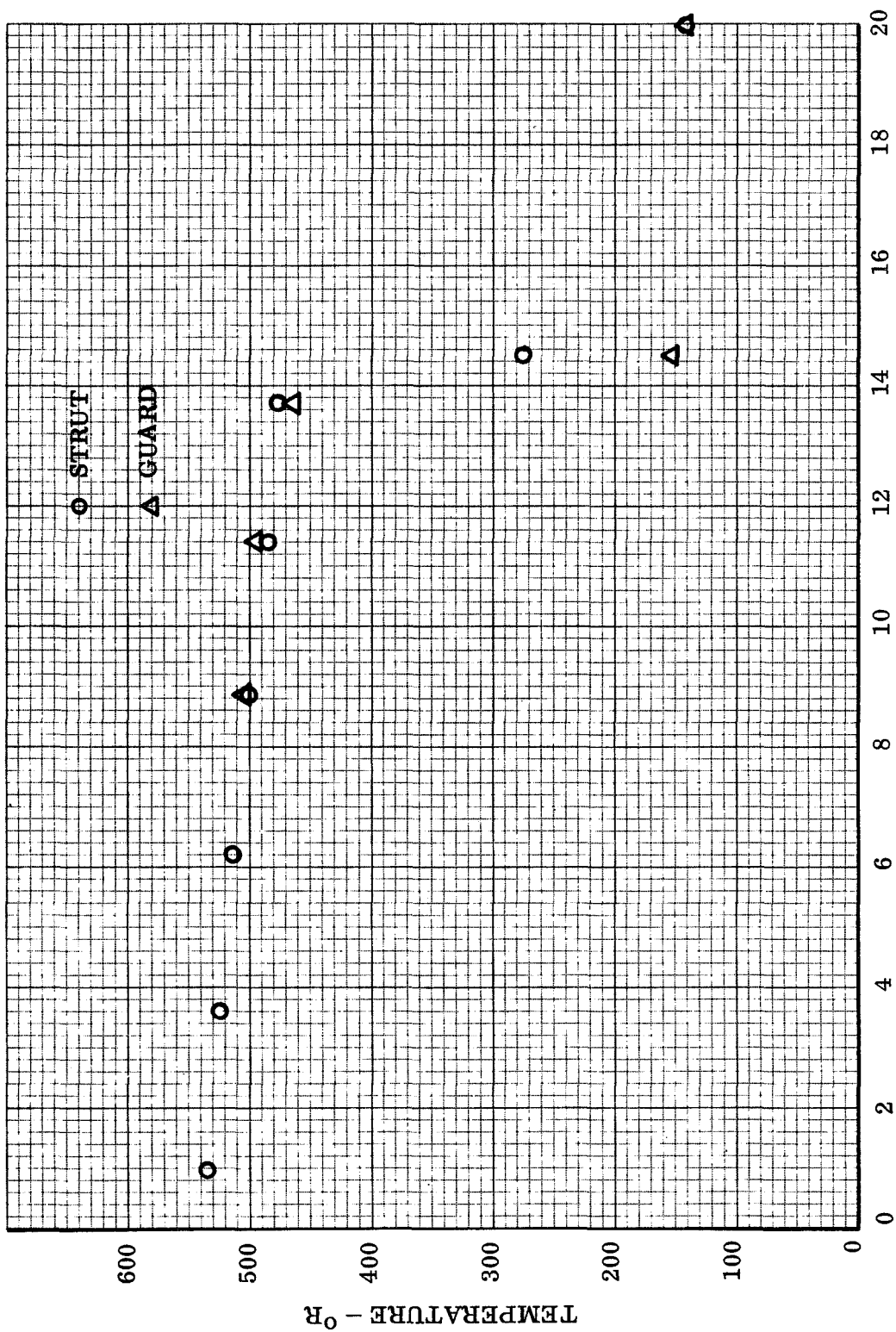


Fig. 33 - Temperature Profile - Run II-6

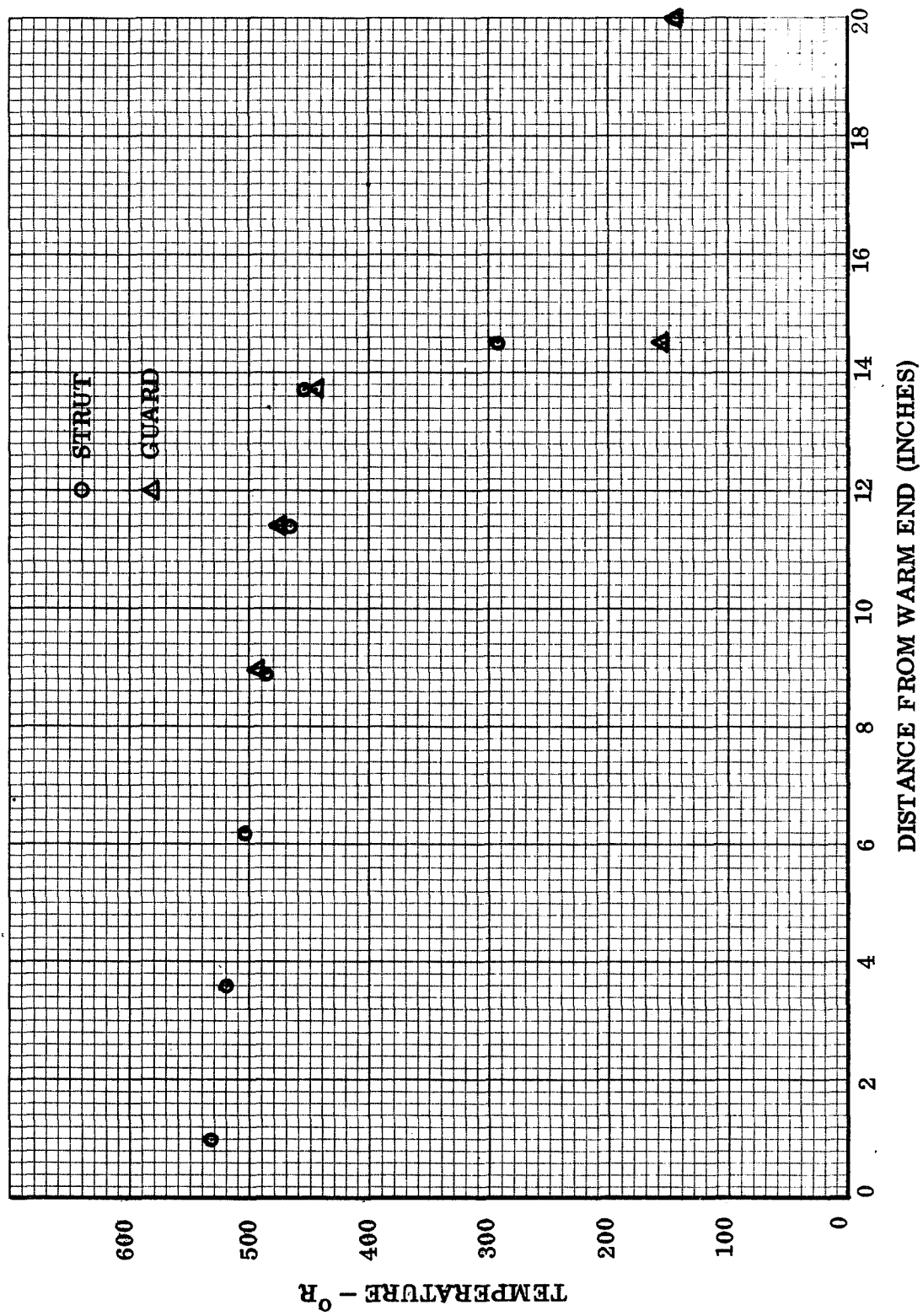


Fig. 34 Temperature Profile -- Run II-7

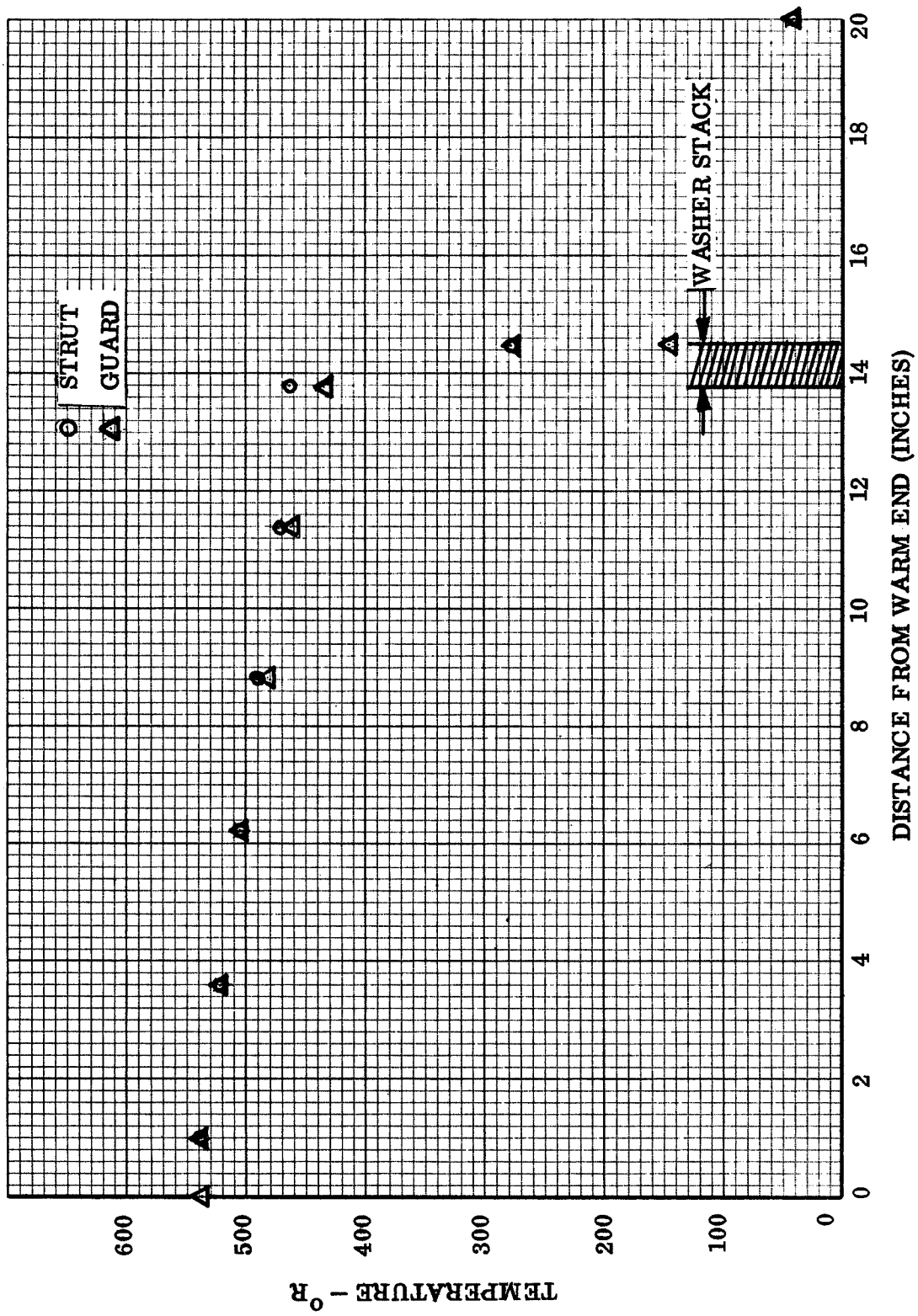


Fig. 35 Temperature Profile - Run II-8

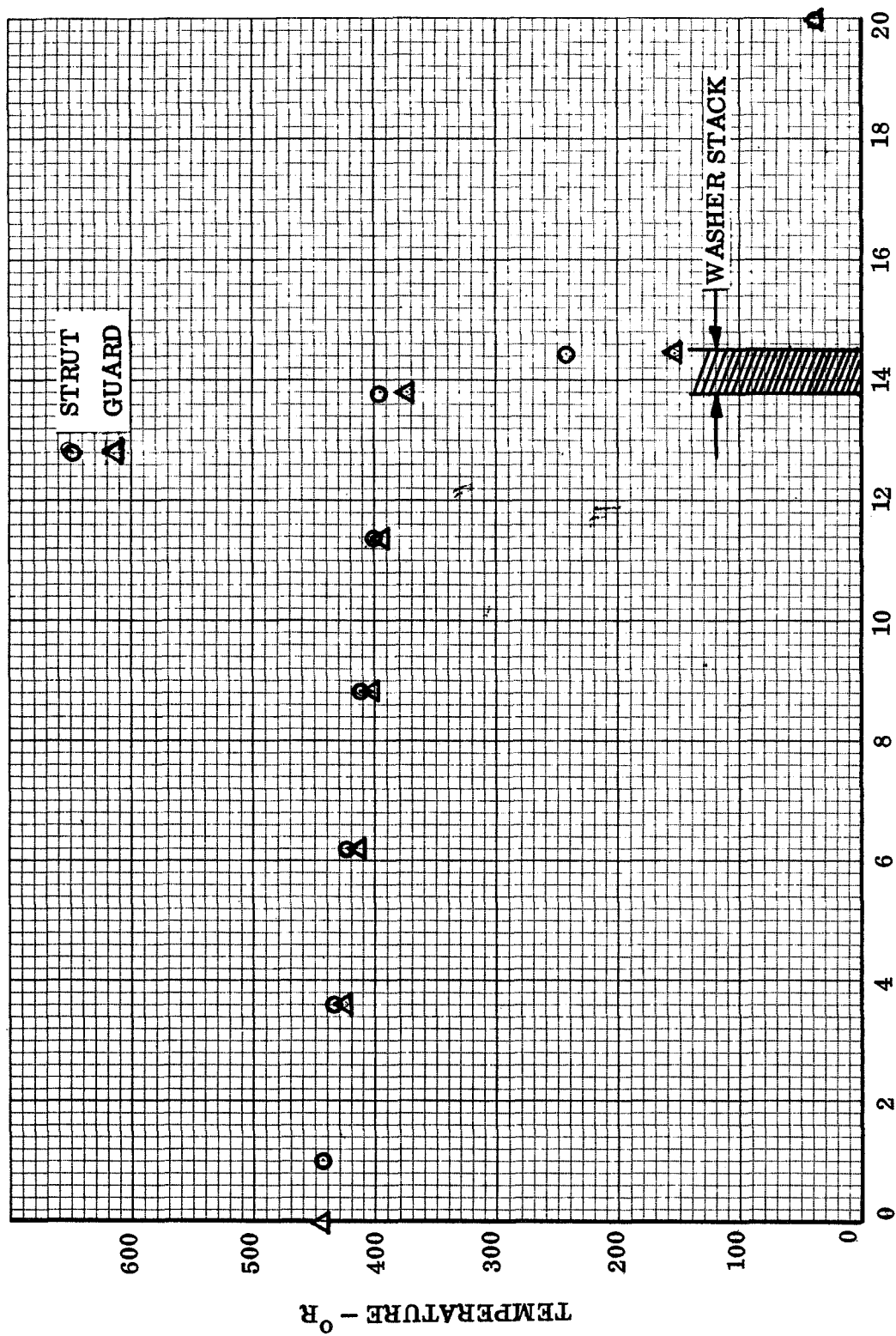
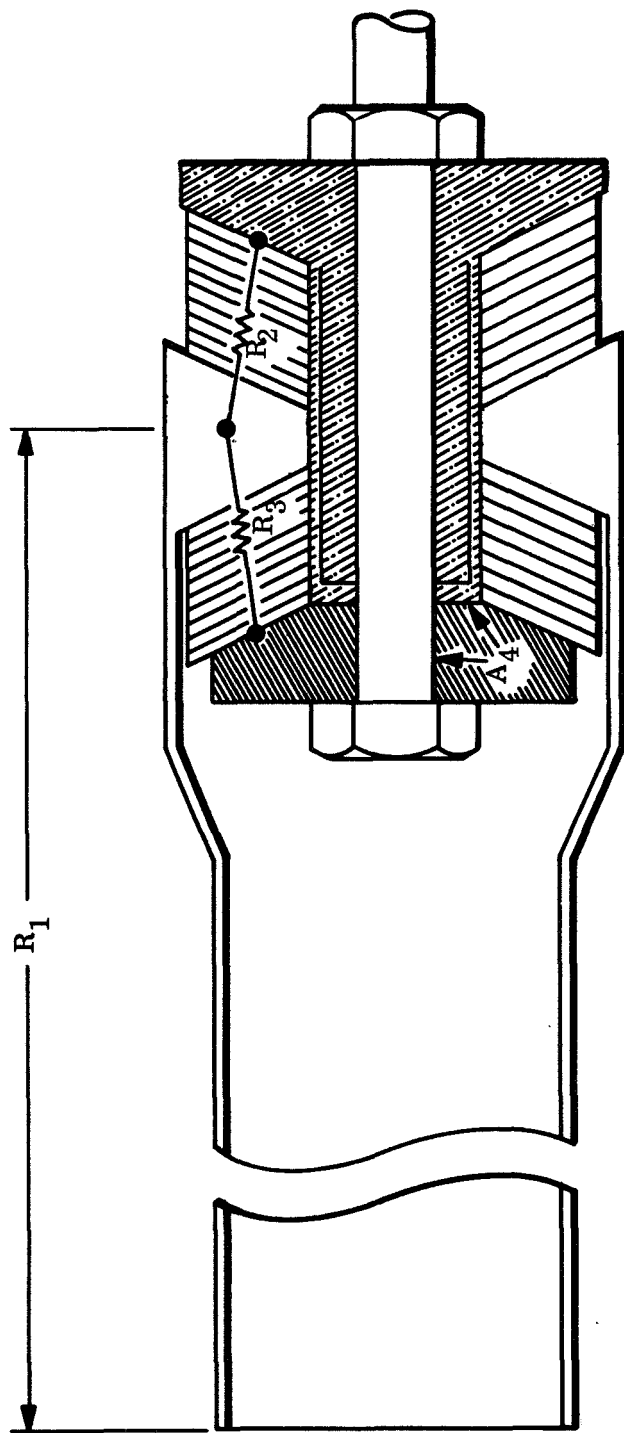


Fig. 36 Temperature Profile - Run II-9

Fig. 36 Temperature Profile - Run II-9



CONICAL WASHER SUPPORT DIAGRAM

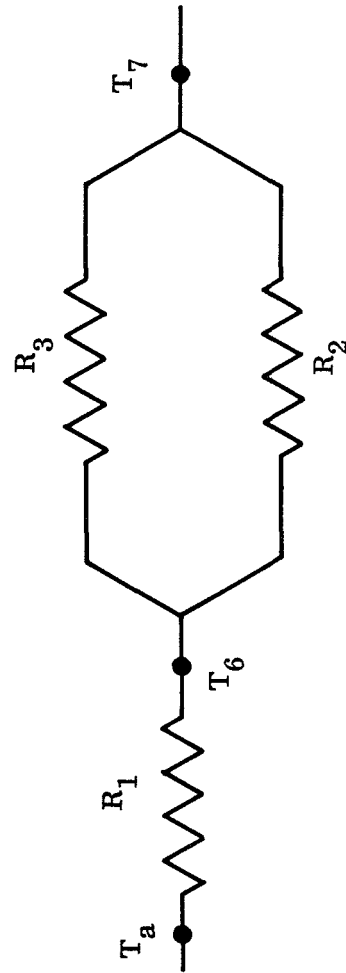


Fig. 37 Equivalent Thermal Resistance Network - Stacked Washer Concept

Also, the defining relationship between the interface conductance and the total resistance is:

$$C = \frac{N}{A_i R_w} = \frac{N Q_T}{2A_i (T_6 - T_7)}$$

where:

- A_i = contact interface area - ft²
- C = interface conductance - Btu/hr-ft² - °R
- Q_T = measured heat flow - Btu/hr
- N = number of interfaces in each stack

When the experimentally determined heat rates and temperatures are combined with the washer area (1.55 sq in.) and the number of washers (390), the conductance values indicated in Table 8 are obtained. These are considerably higher than the 10 Btu/hr-ft² - °R used in design. Apparently, this is partly due to the relatively high temperature of the washers which, in turn, is a consequence of the unexpected resistance between the washers and the cryogen. The data from Thomas and Probert (Ref. 3), which was obtained with small ΔT 's shows two orders of magnitude increase when the tests are conducted at room temperature, as compared to LN₂ temperature. The data of Mikesell and Scott (Ref. 4) was obtained with large temperature differences and falls inside that of Ref. 3, but still shows a large temperature effect.

One of the test variables in this program was the compressive load on the washers. Figure 38 shows the experimental values of conductance as a function of this pressure. For comparison, the extrapolated correlations from Ref. 4 are also shown.

All of the present data fall inside the bounds established from the data of Mikesell and Scott, and, it is apparent that the trends are very similar with regard to the effect of load. Also, comparison of the hydrogen and nitrogen data indicates consistency within the present data with regard to the effect of warm-boundary temperature. These facts

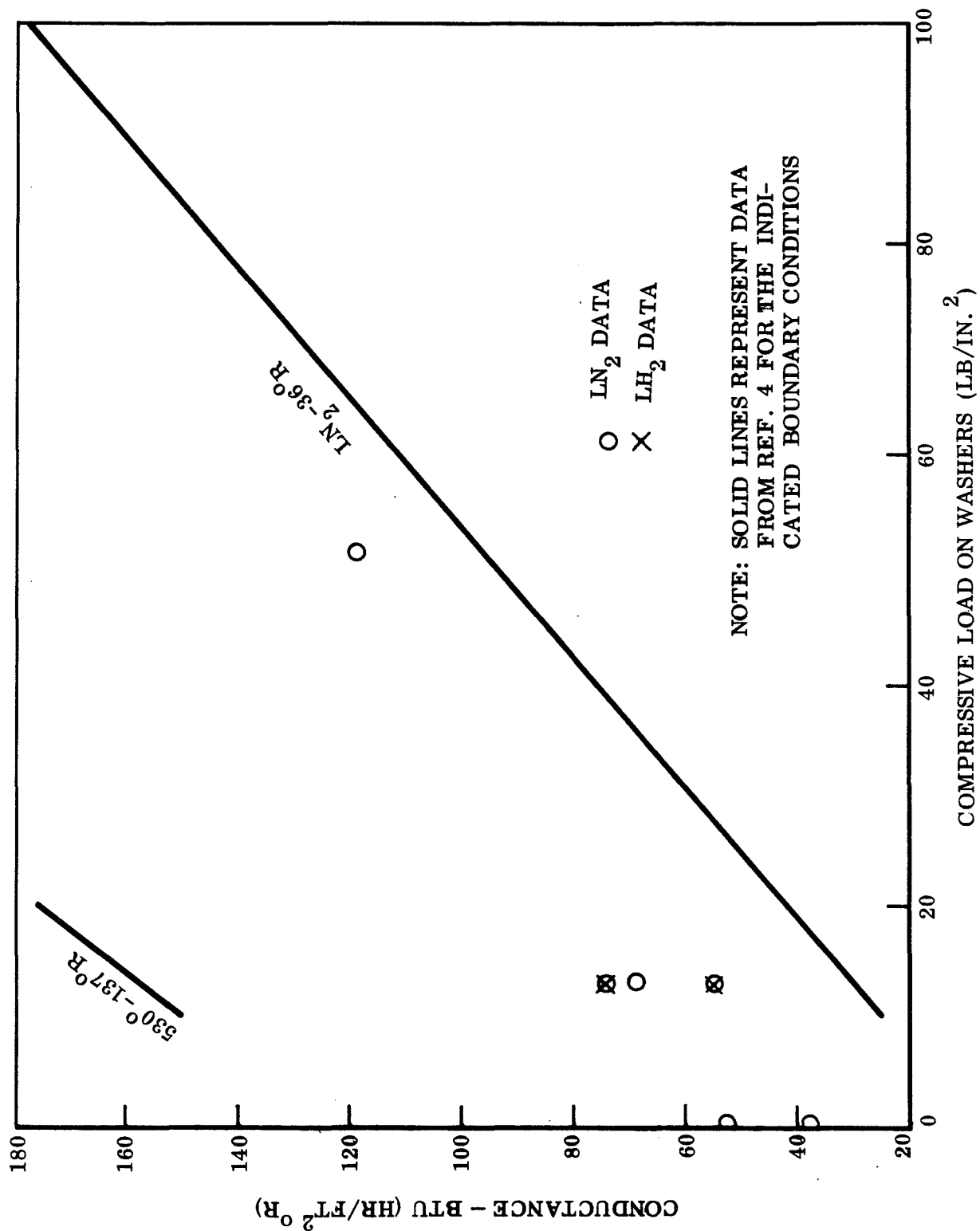


Fig. 38 Effect of Pressure on Washer Interface Conductance

tend to support the conclusion that the present test results on the stacked washers are generally consistent with other data. One might expect these test data to fall closer to the upper Mikesell and Scott temperature line (i. e. , higher conductance) in view of the experimental temperatures. However, the reference lines were established using stainless steel washers that were 0.0008-in. thick, whereas, those in the present test program were 0.002-in. thick. This probably contributes somewhat to the low conductance values as indicated by the fact that Mikesell and Scott reported approximately 50 percent lower conductance values for 0.004-in. washers. It is to be expected that a given number of thick washers will have a higher resistance (lower conductance) than the same number of thinner washers, but the complex nature of the heat transfer mechanism makes it impossible to predict the exact effect. However, it is clear from the test data that the washer conductance used for design (10 Btu/hr-ft²-°R) is inconsistent with the use of stainless steel washers. Therefore, further tests using washers made from tool steel should be investigated because the data of Ref. 4 indicates nearly an order of magnitude improvement over that obtained with stainless steel washers.

BALL AND CLAMP DISCONNECT CONCEPT

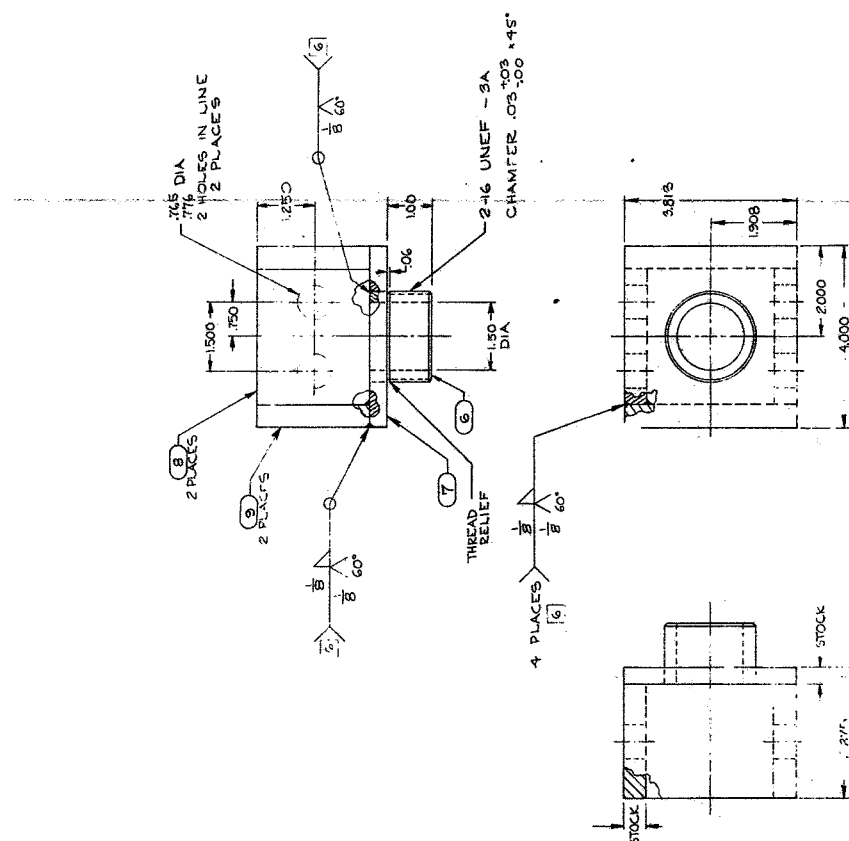
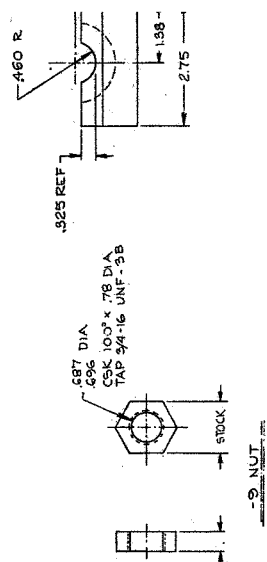
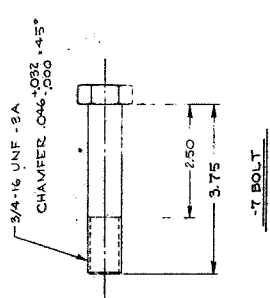
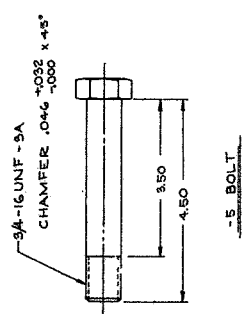
TEST ARTICLE DESCRIPTION

The test article that was used to evaluate the thermal disconnect concept is shown in Fig. 39. This evaluation was conducted in the thermal conductivity apparatus used for the stacked washer tests.

The important feature of this concept is a mechanical joint which can be partially separated to interrupt the highly conducting path between the vehicle skin and the propellant tank. The joint consists of a titanium ball, which is connected to the tank, and a clamp which can be locked tightly around the ball during high load periods and uncoupled from the ball when the loads are removed. In this disconnected position, the clamp makes only line contact with the ball and thus the primary mode of heat transfer is by radiation between the ball and clamp. Conventional titanium struts are connected to lugs which are attached, through pins, to the outboard end of the two halves of the split cylindrical clamp.

The tests conducted in this program were designed to evaluate the thermal performance of this disconnect concept, rather than the mechanical features. Therefore, the test hardware did not incorporate the actuation mechanism. A washer was used for maintaining the joint in the uncoupled position throughout the tests.

The major thermal resistance in this test article (i. e. , the ball and clamp joint) was sized to simulate that in the support strut on a 15-ft-diameter CH₄ tank. The remaining components were made thicker than would normally be required on the flight strut to minimize the fabrication costs of the test article. The tube body is the same one used with the stacked washer assembly. The clamps were made simply by machining the spherical ball seat into flat plate stock. In the flight configuration, these clamps would be contoured to minimize weight. The calculated effect of these compromises was to reduce the total strut resistance by 3 percent compared to that of a flight configuration. Figure 40 is a photograph of the various components before they were assembled into the test article.



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			3502699	

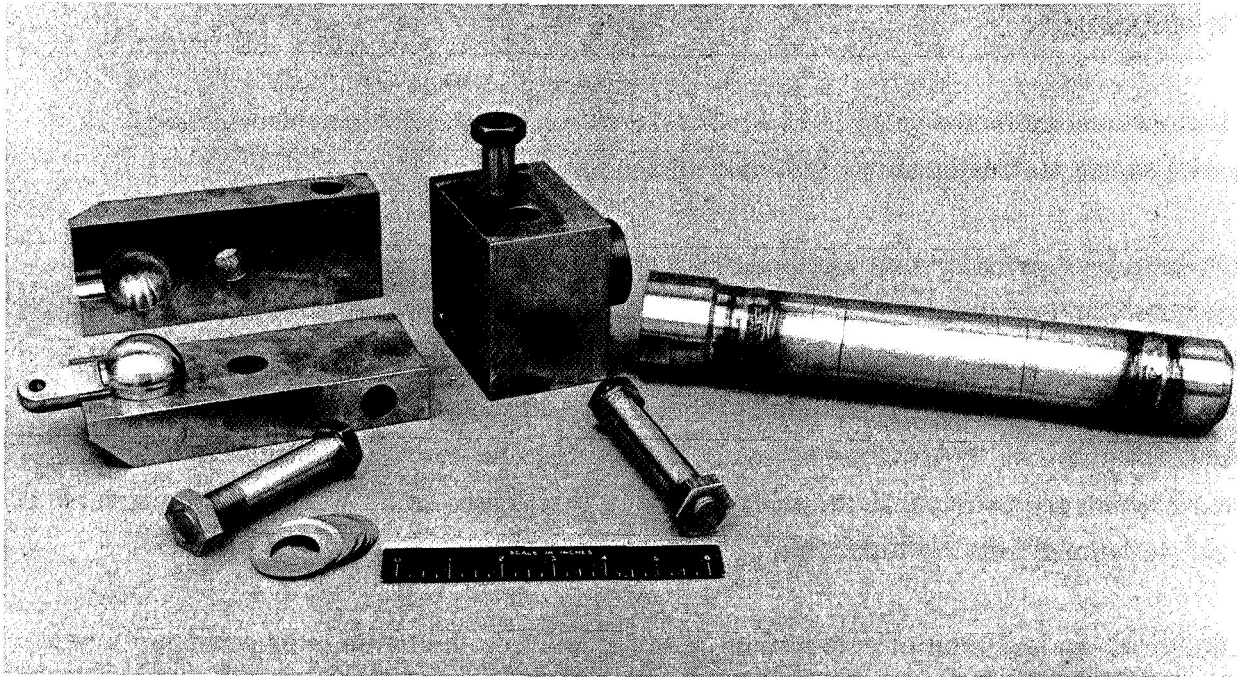


Fig. 40 Ball and Clamp Point Support Prior to Assembly

TEST ASSEMBLY

The test assembly shown in Fig. 41 is used to evaluate this disconnect concept; it is very similar to that used for the stacked washer.

A four-part thermal guard was used to match the temperature drops across specific segments of the test strut. The resistance of the titanium tube is matched by that of the upper stainless guard; the low resistance of the clamp assembly is simulated with the thick copper middle guard made from 6-in. -diameter pipe. The thermal link, with strut attachment, is provided a nearly isothermal enclosure with the aluminum lower guard which is grounded to the cryogen pot. This guard is shown in the photograph of Fig. 42, along with the test article.

The stainless and aluminum portions of the guard are the same ones previously used with the stacked washer assembly.

The thermal resistance of the radiation joint is simulated with a gap between the copper middle guard and the aluminum lower guard. The teflon sleeve serves as a low conductance support for the heavy thermal guard. The region between the guard and test strut was filled with Dexiglass, and 120 layers of MLI were applied to the outside of the thermal guard.

The disconnect between the ball and the clamps was maintained with a thick washer inserted between the inner faces of the clamps. A wire attached to the top of the strut was used to pull the clamps up off the ball so that only the lower lips were in contact with the ball. This wire was tied to the top of the thermal guard.

INSTRUMENTATION

The instrumentation used on the titanium tube body is the same as previously used with the stacked washer assembly, i. e. , a PRT at the top and 5 differential thermocouples placed 2.61 in. apart. A sixth differential thermocouple was placed at the bottom of the thick titanium clamps. These six thermocouples, referenced to the PRT gave the

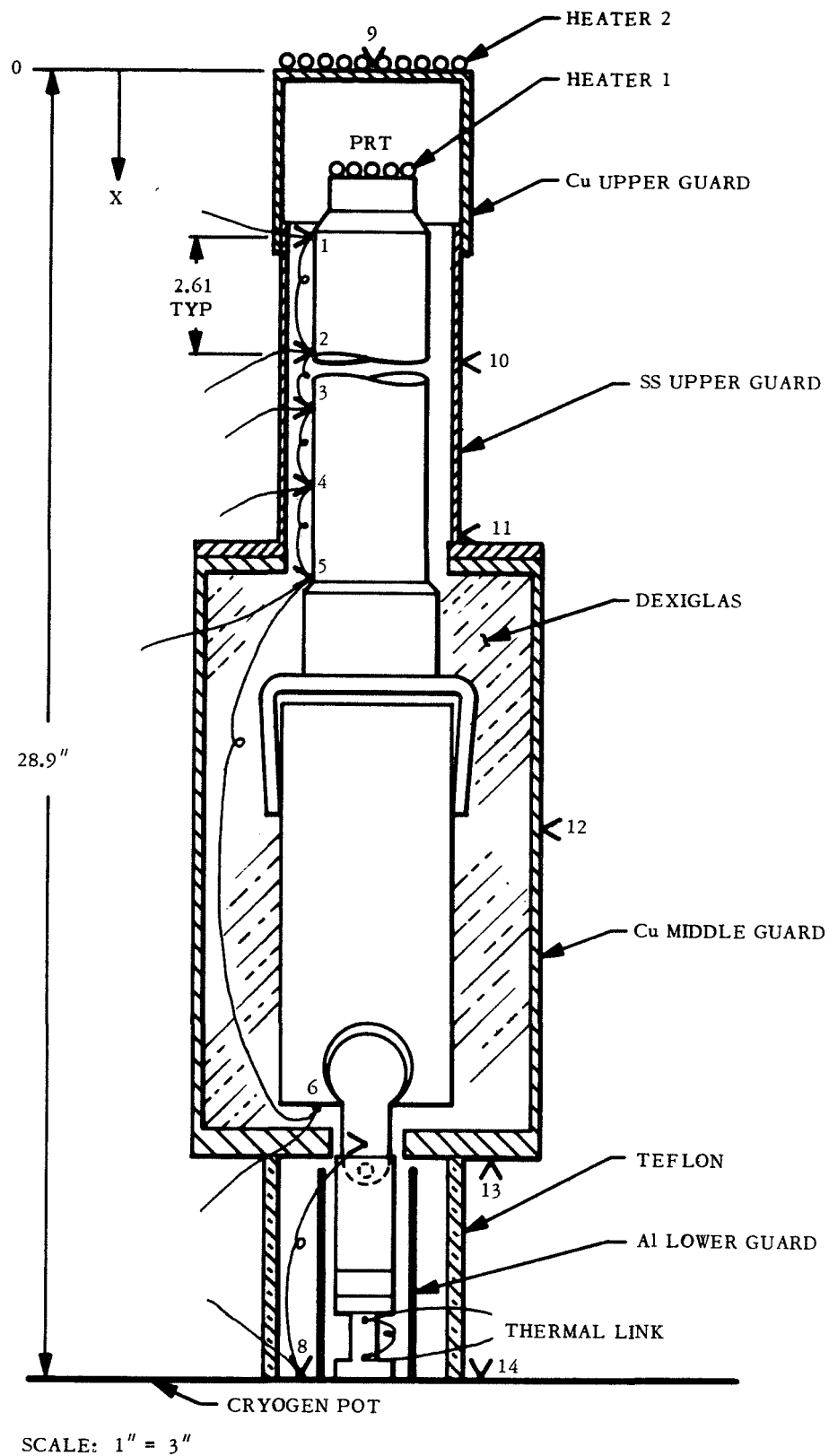


Fig. 41 Installation & Instrumentation Schematic For Ball & Clamp Concept

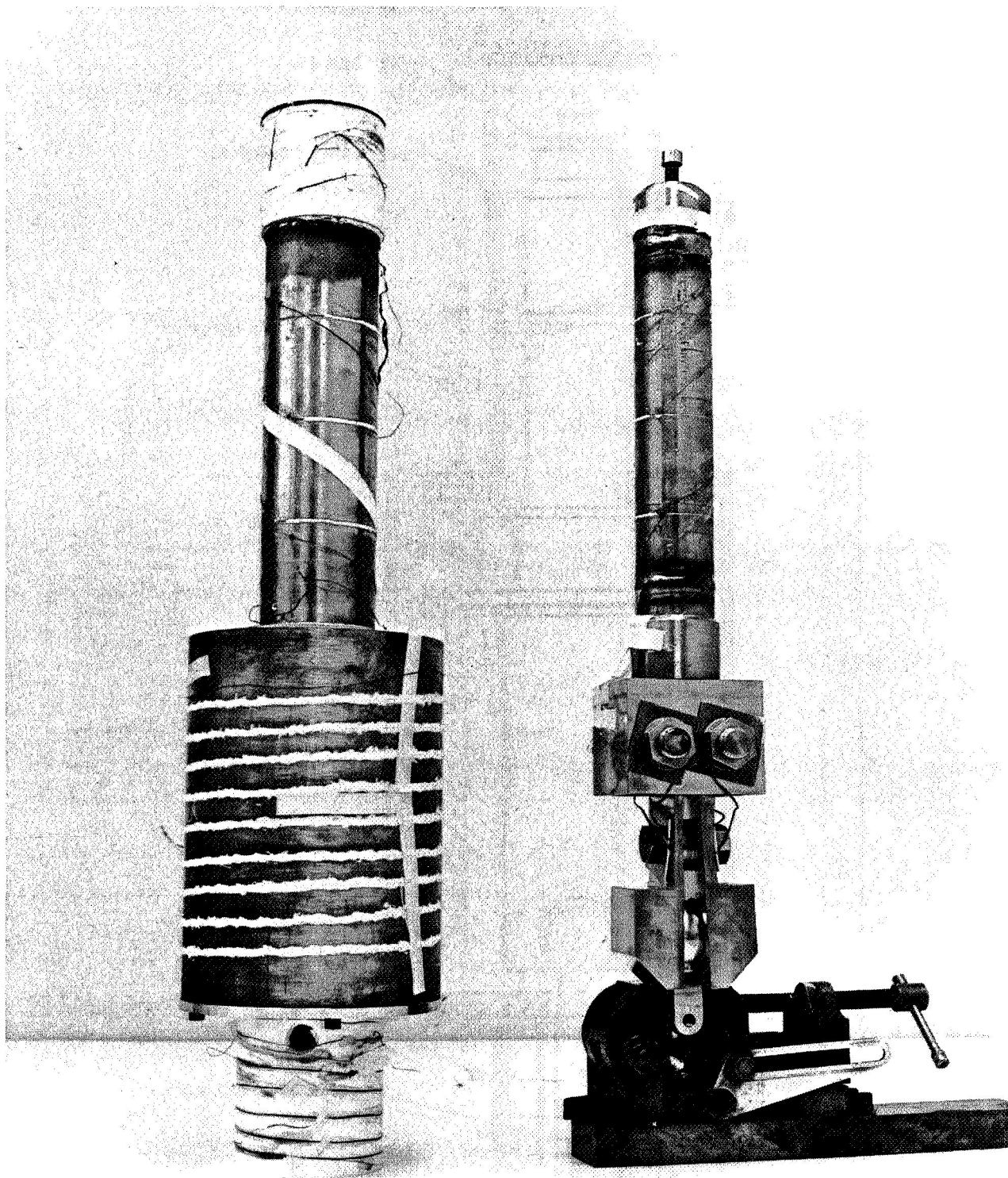


Fig. 42 Photograph of Test Article and Thermal Guard for the Ball and Clamp Concept

temperatures down to the warm side of the radiation joint. A seventh differential thermocouple was placed at the bottom of the titanium ball and referenced to the cryogen pot to give the temperature on the cold side of the disconnect joint.

An additional differential thermocouple was referenced to the PRT and attached to the bottom of the clamps. Thus it measured the total temperature differential from the top of the strut to the warm side of the thermal disconnect joint. This provided a check on the summation of thermocouples 1 – 6 but the primary purpose was to serve as an operational convenience for determining equilibrium.

Six differential thermocouples were used to obtain the temperature distribution in the thermal guard – one on the upper copper guard and two, each, on the stainless steel and middle copper guards. The sixth one was placed on the cryostat at the base of the teflon sleeve. Figure 41 shows the location of all temperature measurements.

TEST PROGRAM

The primary objective of this program was to determine the extent to which the heat transferred down this strut is governed by radiation between the ball and clamp. Three tests were conducted on this thermal disconnect concept – two with liquid nitrogen and one with liquid hydrogen. The conditions are summarized in Table 9.

Table 9

TEST CONDITIONS – BALL AND CLAMP STRUT TEST

<u>Run No.</u>	<u>Warm Boundary Temperature (°R)</u>	<u>Cryogen</u>
III-1	540	LN ₂
III-2	446	LN ₂
III-3	446	LH ₂

The procedures for conducting the tests on this thermal disconnect were the same as those previously employed: the first step being to adjust the current to heater No. 1 to give the desired warm boundary temperature. Then, as this temperature was approached, the current to heaters 2 and 3 was adjusted to match the strut and guard temperatures at the top and at the middle of the copper middle guard.

During the first test, the emf output from both thermocouples 7 and 15 were periodically monitored, as was the thermal link output. For the first 20 hr, all appeared to be in a smooth transient and heater No. 3 was maintaining midguard and strut temperatures within 1°R , although the guard was always warmer. At this point, the temperatures began to diverge and within the next 24 hr the difference reached 4°R . Heater No. 3 was then turned completely off and the data was tracked for an additional 24 hr at which time the test was concluded. During the last 14 hr, the temperature of the clamps dropped 3°R and the indicated heat range dropped off approximately 3 percent.

To start the second test, all heaters were turned off to allow the warm boundary temperature to drop to approximately 450°R . This took about 28 hr. The test was then continued for an additional 40 hr with no control problems. At these lower temperatures, heater 3 maintained the variance between midguard and clamp temperatures within 1°R . At the conclusion of the test, the rate of change in clamp temperatures was approximately 0.3°R per hour. The indicated heat input was dropping at approximately 0.3 percent per hour.

The liquid nitrogen was boiled out of the cryostat by introducing ambient nitrogen. This was followed with a gaseous helium purge prior to filling the cryostat with liquid hydrogen. The hydrogen test lasted for approximately 100 hr. This was due primarily to two problems that developed during the test. The first was a short which occurred in the control circuit to the top guard heater. This short disturbed the equilibrium established between the upper guard and the titanium tube body and resulted in the downward trend of the heater input to be reversed. The second incident occurred when the cryostat went dry because the need for a refill had been overlooked. The test was continued for 24 hr after filling the cryostat. The last five hours of the test, the heat rate dropped approximately 0.25 percent.

TEST RESULTS AND ANALYSIS

The measured heat rates for each of the three test conditions are given in Table 10.

Table 10

TEST RESULTS - BALL AND CLAMP CONCEPT

Test No.	Experimental Q (Btu/Hr)		Calculated Q (Btu/Hr)
	Htr. 1	Thermal Link	
III-1	1.08	1.29	1.15
III-2	0.57	0.72	0.62
III-3	0.72	0.63	0.60

This data indicates that the measured heat rate across the thermal link is less than the heater input for the nitrogen tests, but greater for the hydrogen test. This seems to indicate an interaction between the guard and strut which changes direction with the change in cryogen. However, the temperature profiles, shown in Figs. 43 through 45, do not substantiate this conclusion. They show the same general trend with both fluids, i. e., good agreement between the guard and strut over the body of the support system, with the strut only slightly warmer than the guard. The temperature drop in the guard between thermocouples 13 and 14 really represents that across the teflon support ring. The lower aluminum guard is 1/8-in. thick and should be nearly isothermal at thermocouple 14, in which case the disconnects in both the strut and guard are reasonably well matched.

It is suspected that the discrepancy between thermal link and heater power is a consequence of the long time constant associated with the massive clamps. The large thermal mass in these clamps results in a slow response at the bottom of the system to changes in heater power. Figure 46 shows the transient response of the thermal link power and the clamp temperature. It can be seen that the system was still in a transient even after 72 hr. However, the platinum resistance thermometer, which was used to set the outer boundary temperature was extremely sensitive to changes in

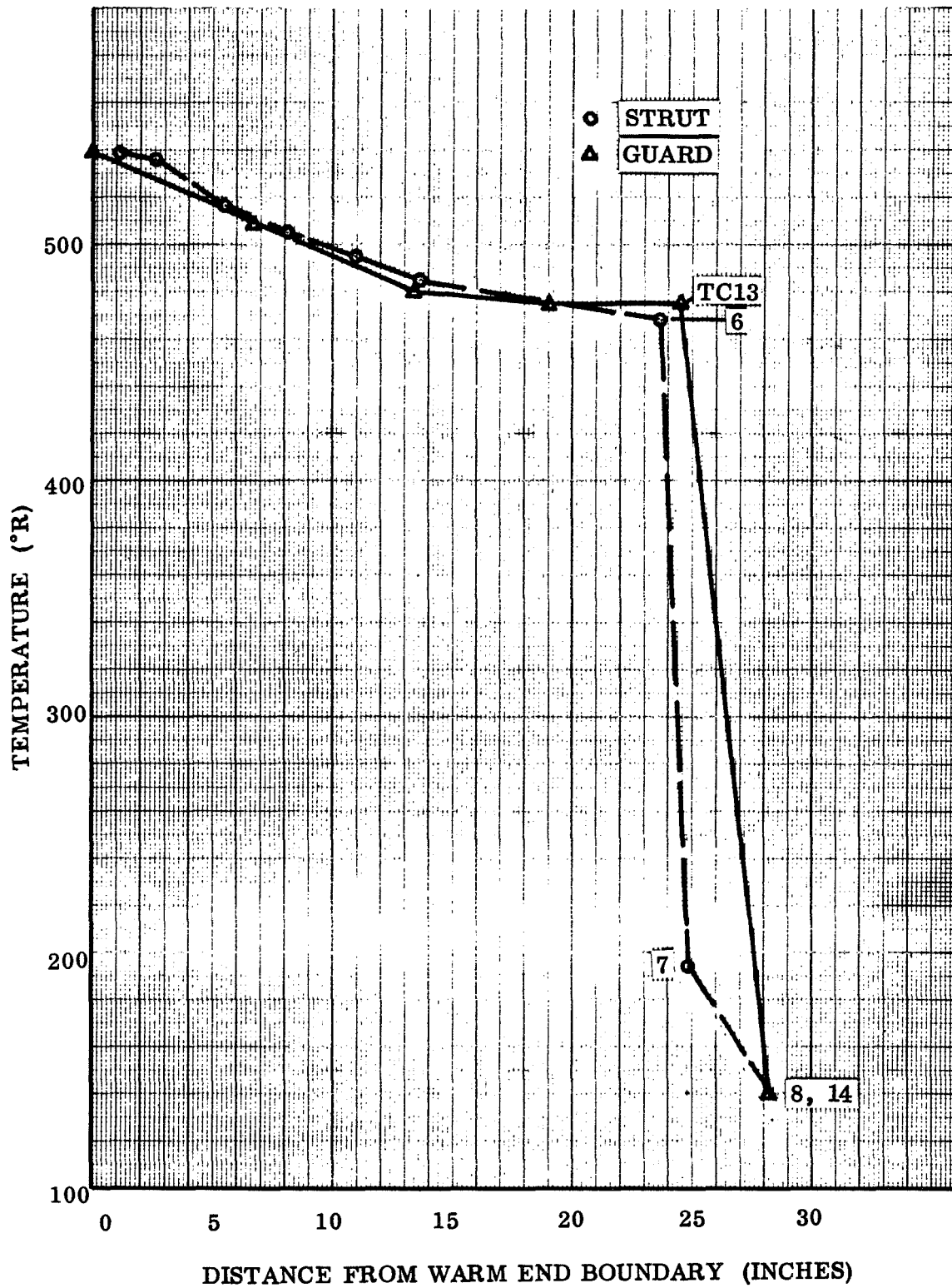


Fig. 43 Temperature Profile - Ball and Clamp, Run III-1

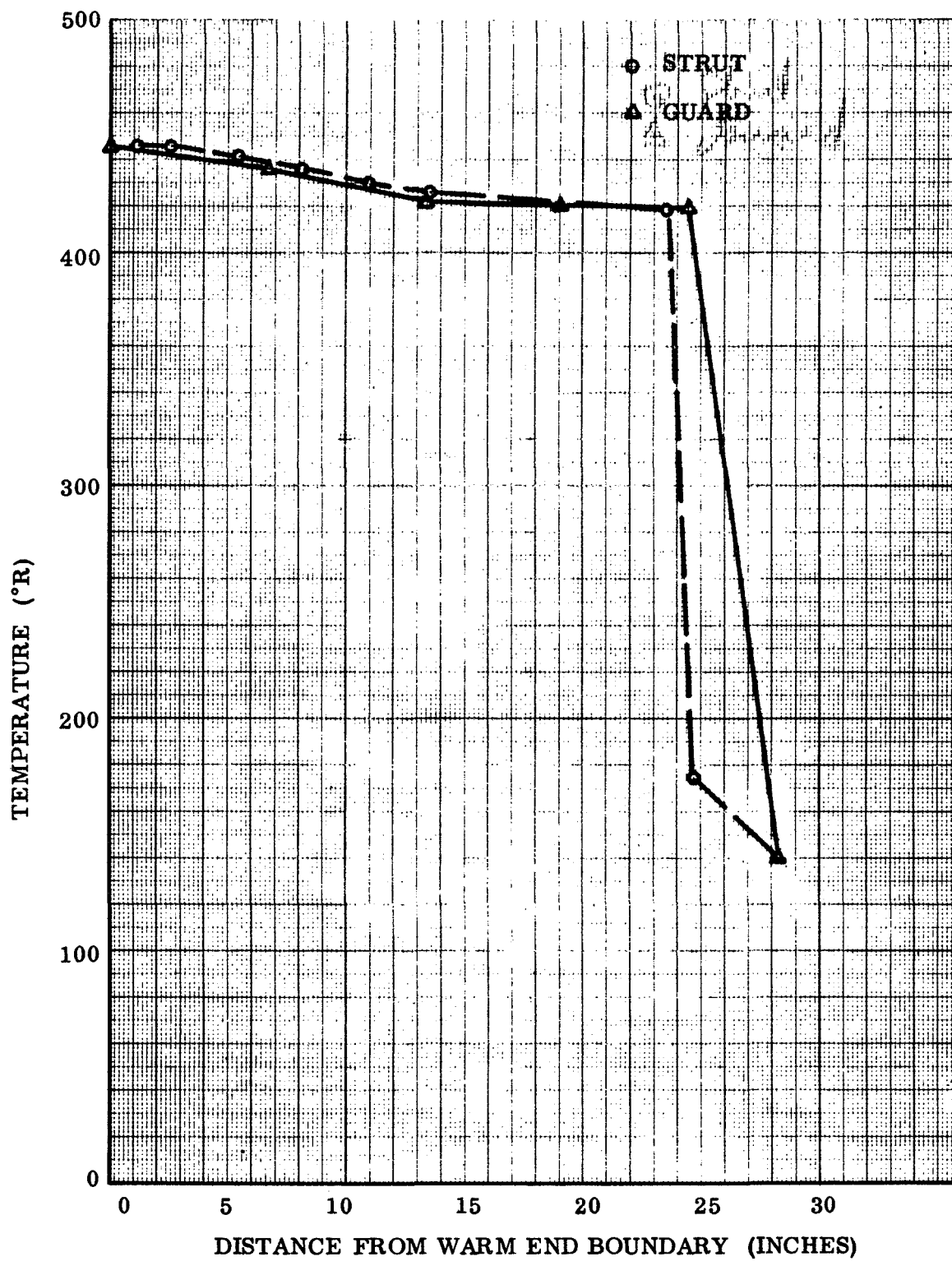


Fig. 44 Temperature Profile — Ball and Clamp Run HHb2

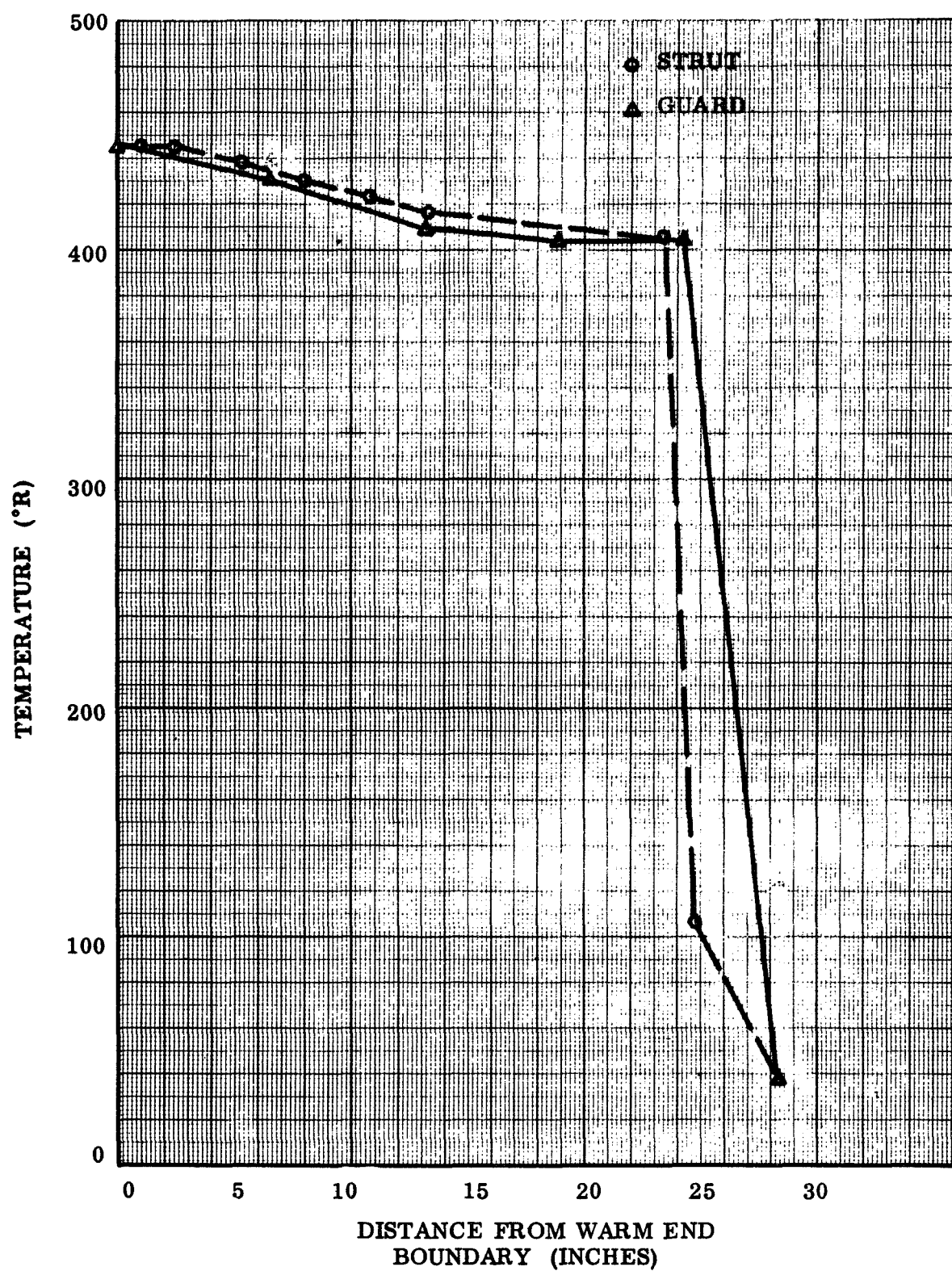


Fig. 45 Temperature Profile — Ball and Clamp Run CH₂3

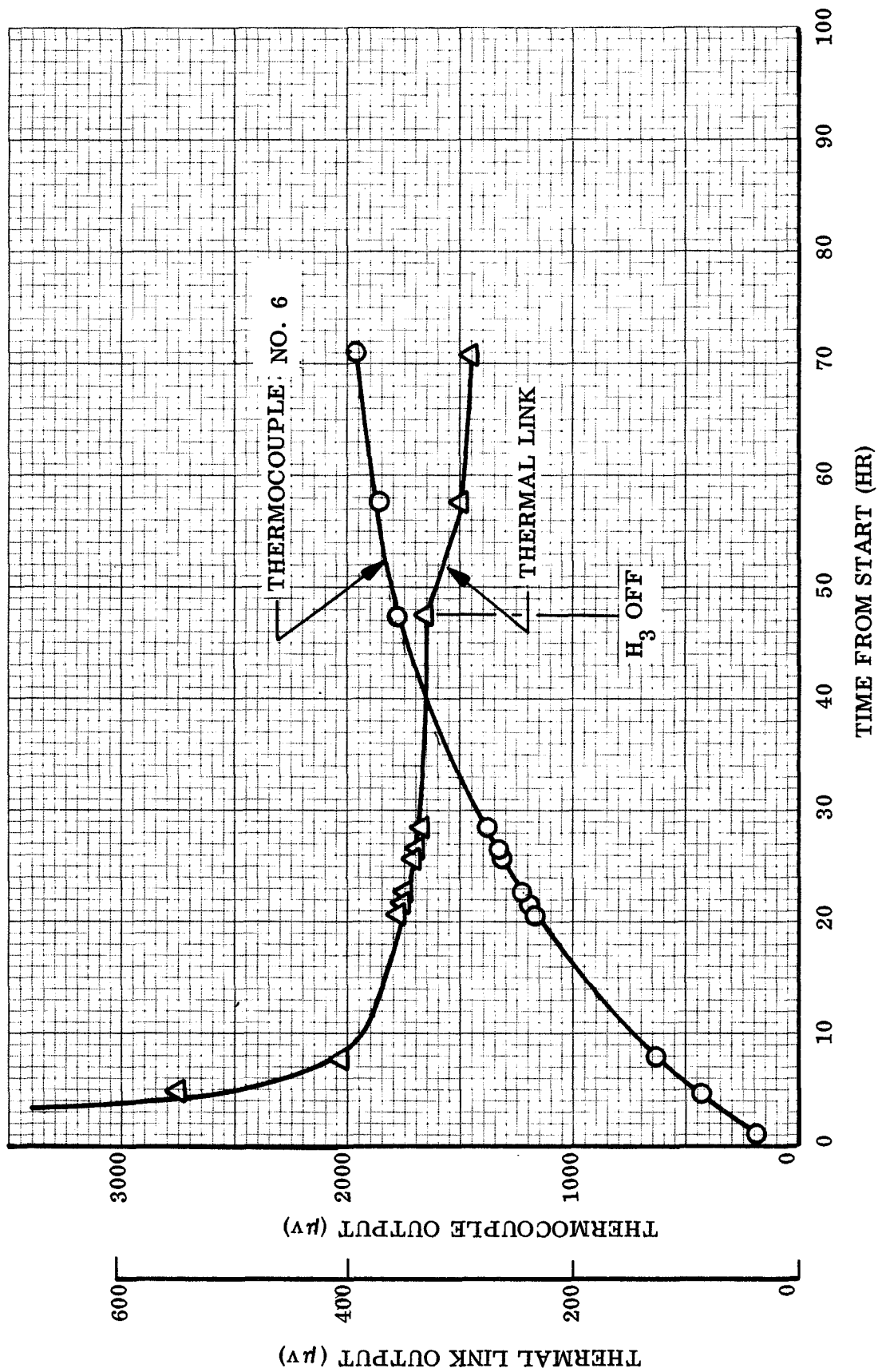


Fig. 46 Thermal Response of Ball and Clamp Concept During Run III-2

heater power. When attempting to reach steady state, the heater had to be frequently adjusted up and down as the warm end would undershoot and overshoot the desired temperature.

The test data indicate that heat transfers across the disconnect joint. For this to be so, the heat transfer is given by:

$$Q = \text{const} \left(T_6^4 - T_7^4 \right)$$

Figure 47 shows the experimental heat rate plotted against the difference in the fourth powers of temperatures T_6 and T_7 . Also shown for comparison are theoretical radiation curves for various emittance values. It can be seen that an emittance value of 0.3 correlates the data within ± 7 percent. The flagged symbols on Fig. 47 were taken over a 25-hr transient period while trying to reach steady state for run III-2. Since the ball has very little thermal mass, these points are quasi-equilibrium data, and agree very well with the steady state data.

Once it was established that the joint resistance was governed by radiation, the steady state heat rate at the actual test conditions could be calculated, using an analogous electrical network representing the tube, joint, and attachment resistance. These calculated heat rates, which are also indicated in Table 10, fall between the experimental values determined from the thermal link and from the heater.

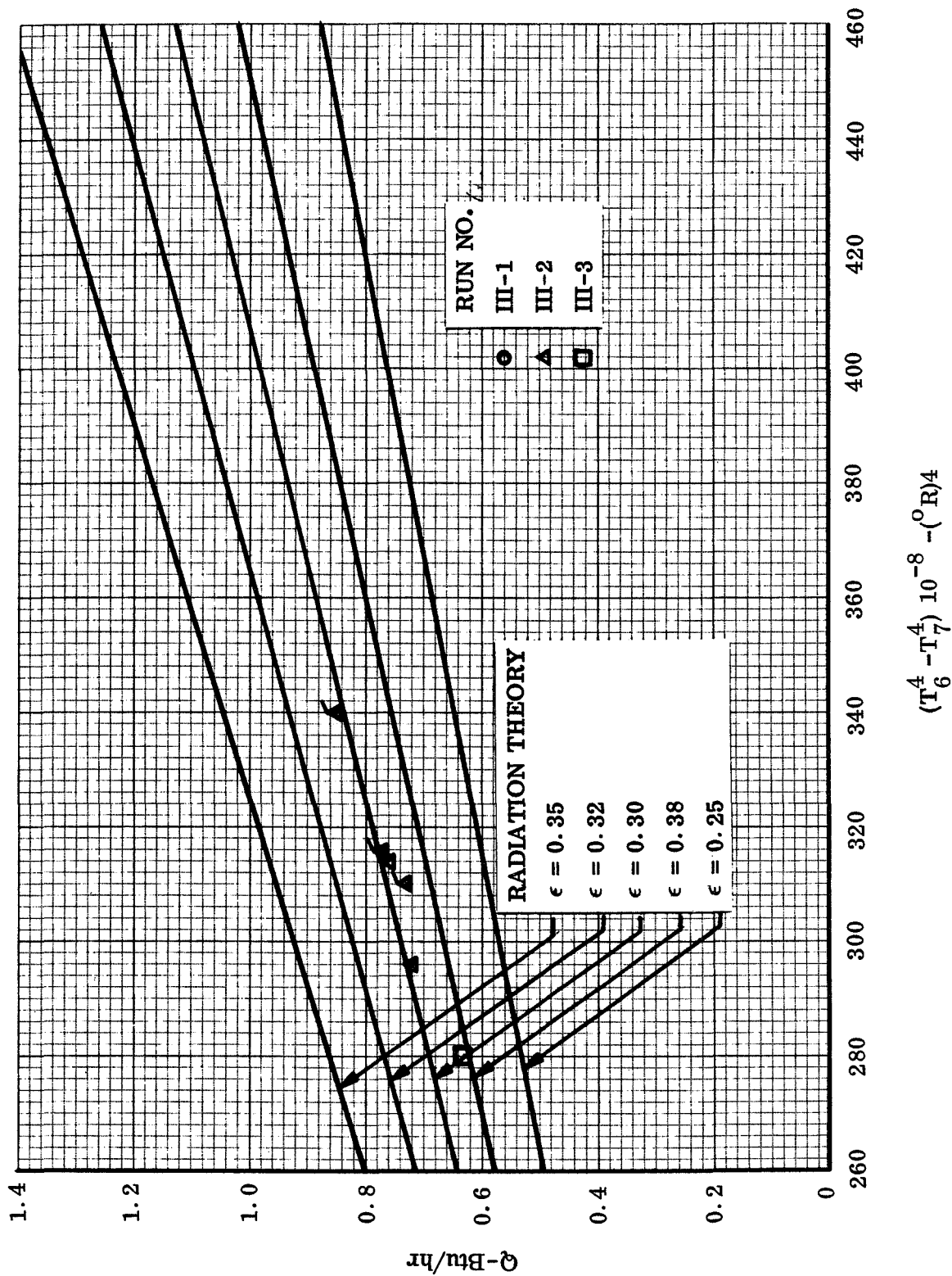


Fig. 47 Comparison of Heat Transfer With Radiation Theory
For Ball & Clamp Disconnect Concept

COOLED TITANIUM STRUT CONCEPT

TEST ARTICLE DESCRIPTION

The test article for the cooled strut concept is shown as the -503 assembly in Fig. 24. It was made from the same tube stock (2 in. O.D. x 0.125 in.) used for the stacked washer and ball and clamp concepts and incorporates the same size end clevis to permit use of a single adapter on the cryostat. Figure 48 is a photograph showing the tube, radiation shields, etc. before the test article was assembled. It was recognized that the tube size exceeds that of even the largest methane tank considered in Task I. However, with this concept, in particular, tube size is meaningful only with regard to a particular fluid, flow rate, and end conditions. The important thing is to evaluate the theory used in the design rather than a particular set of numbers.

Internal radiation shields are used for this cooled concept, whereas the stacked washer concept uses chopped Dexiglas to block internal radiation. These radiation shields require a rigid center body for support which is a relatively high heat leak path that would short-circuit the high resistance washers. However, with hydrogen flowing inside the tube, the center body, as well as the tube walls, can be effectively cooled. With Dexiglas packed in the tube, the danger exists that the flow would tunnel through some preferred path and actually reduce the cooling effectiveness.

TEST ASSEMBLY

The test assembly for this concept is illustrated in Fig. 49. The small cryogen pot that was previously filled with LN_2 for the cooling wires, the ball and clamp, and the stacked washer tests was used for a hydrogen coolant supply in these tests. A 1/8 in. diameter copper tube was connected from this supply pot to the fitting on the lower end of the test article. A similar tube was connected to the discharge fitting at the warm end of the strut, and routed through a water bath to condition the effluent before it passed through the precalibrated flow rater. The coolant flow rate was controlled with a valve located just down stream of the flow rater, and with a heater which was used to raise the pressure in the coolant supply pot. It was decided that only another

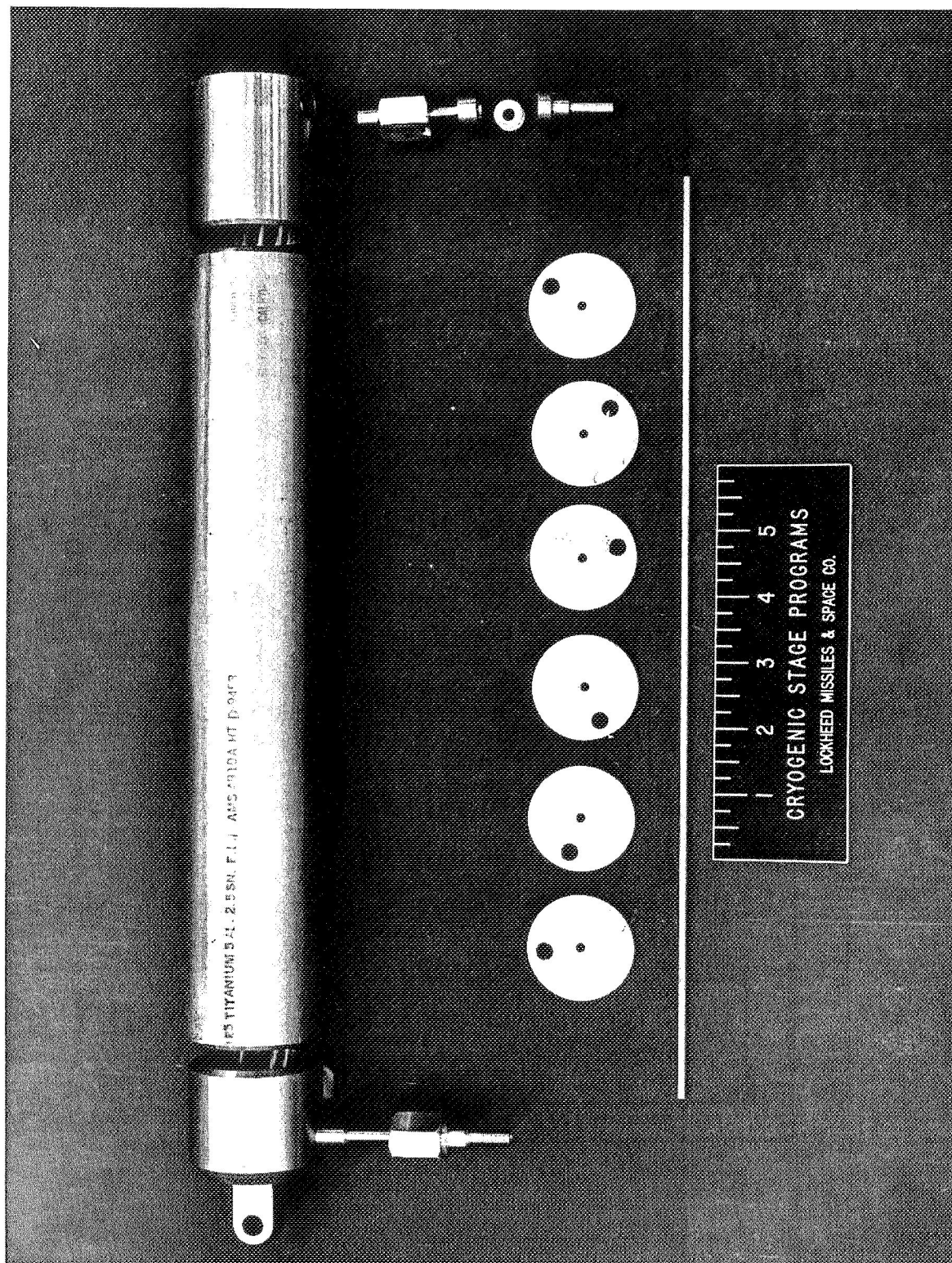


Fig. 48 Cooled Strut Components

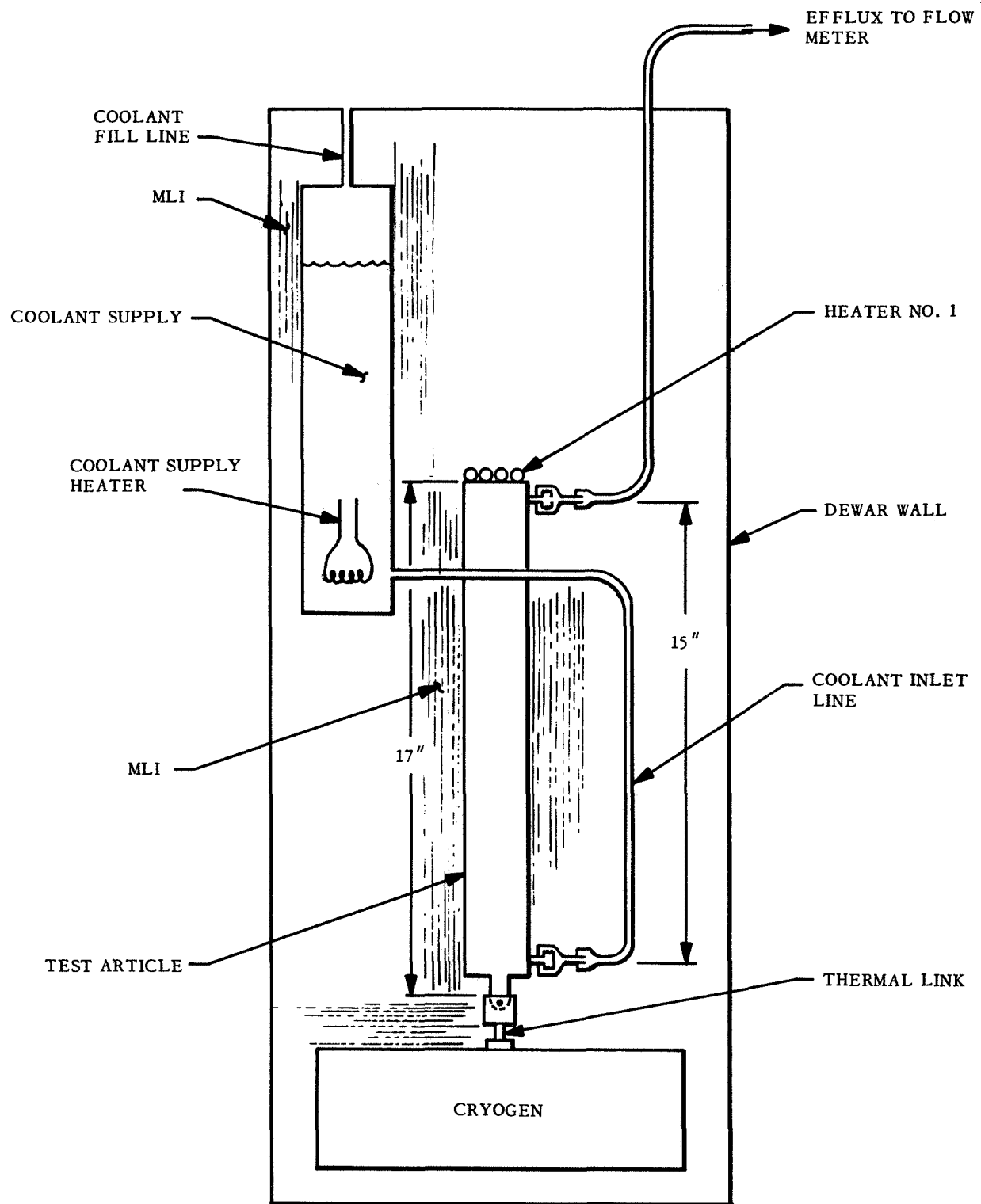


Fig. 49 Installation Schematic for Cooled Strut Tests

cooled tube, larger and with special cooling passages could adequately be used as a thermal guard for this concept. Since this is not too practical, no guard was provided. The protection from the outside environment was obtained by encapsulating the strut with 3 in. of multilayer insulation. Side wall heating was calculated to be inconsequential through this thick layer, compared to that coming in from the top.

INSTRUMENTATION

The instrumentation for this strut is shown in Fig. 50. It consists of six differential thermocouples, referenced to the PRT, and equally spaced along the strut. Coolant flow rate was measured with a volumetric flowrater after the effluent was passed through a controlled temperature water bath.

Two chromel-constantan differential thermocouples were mounted at the inlet and outlet to measure coolant temperatures. The discharge temperature measurement was referenced directly to the PRT. The inlet measurement was wired in series with the thermocouples on the strut, as the seventh instrument in the string.

TEST PROGRAM

The objective of this test program was to evaluate the effectiveness of using vent gas as a means of reducing the heat conducted into a tank through its supports. Eight tests, for which the conditions are summarized in Table 11, were conducted on this cooled strut concept.

Table 11
TEST CONDITIONS – COOLED STRUT CONCEPT

Run No.	Warm Boundary Temperature ($^{\circ}$ R)	Cryogen	Coolant Rate (Lb/Hr)
IV-1	539	LH ₂	0.0375
IV-2	536	LH ₂	0.0193
IV-3	536	LH ₂	0.0103
IV-4	537	LH ₂	0.039
IV-5	445	LH ₂	0.015
IV-6	441	LH ₂	0.022
IV-7	439	LN ₂	0
IV-8	441	LH ₂	0

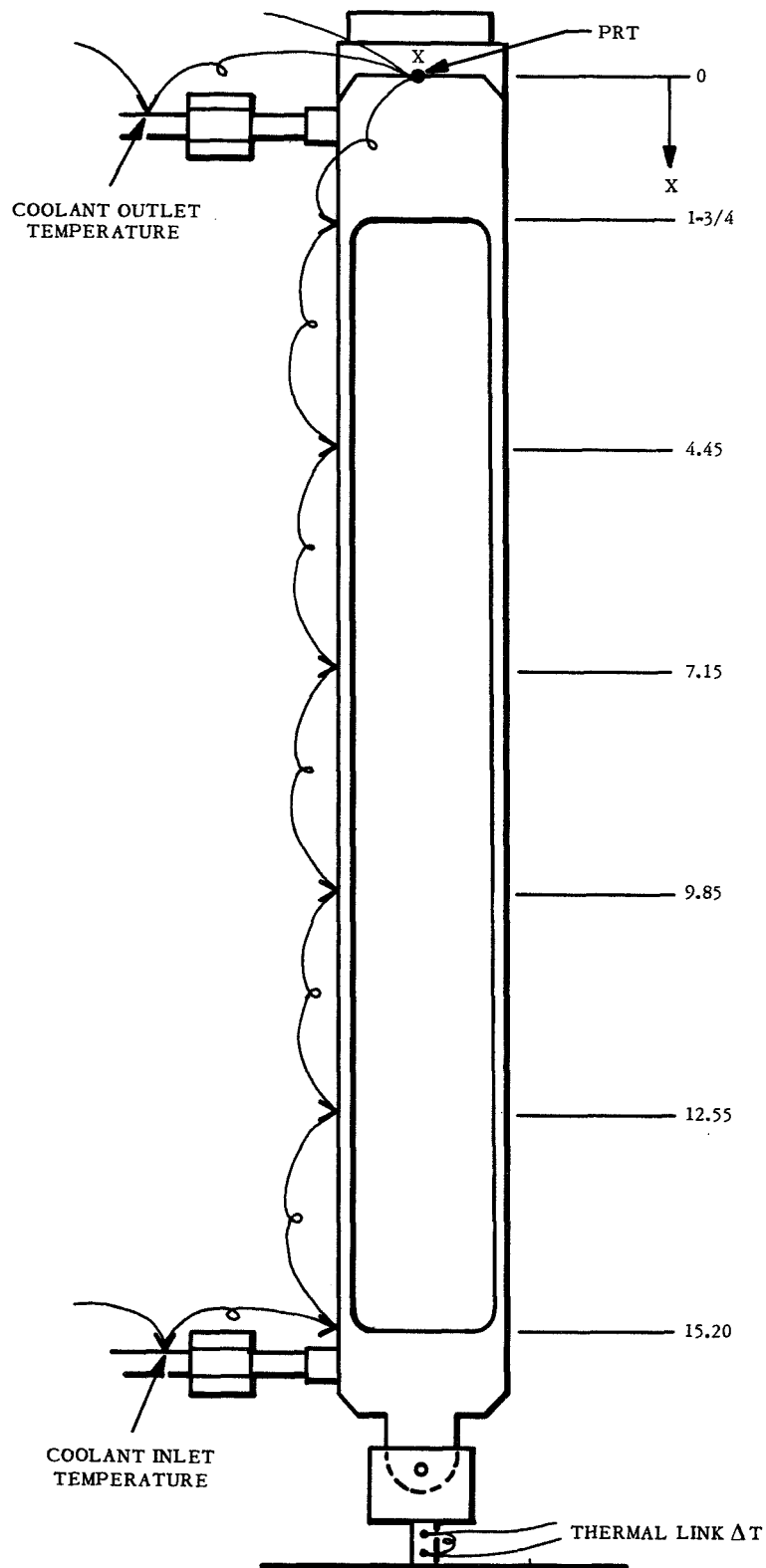


Fig. 50 Instrumentation Schematic – Cooled Strut Test

For the first run (IV-1), the main cryostat was filled and the outer end temperature was brought to approximately 540°R . Then the coolant supply pot was filled and the throttling valve was opened to start the coolant flowing. For the first 4 hr, the flow rate was approximately 0.008 lb/hr, even though the throttling valve was wide open. During this time, the indicated heat input (approximately 6 Btu/hr) dropped only 5 percent. Current was then applied to the heater in the coolant pot to increase the flow rate to 0.038 lb per hr. Within one-half hour, the thermal link output dropped from 1200 to 80 μvolts . It remained constant for the next 2 hr and then started to increase slowly. However, approximately 1 hr before the reversal, the test conditions were upset, when the power to the strut heater dropped off, and simultaneously, the coolant temperature started to increase slowly (approximately 3 deg in 2 hr). Therefore the first test was discontinued two hours after the reversal in heat rate.

Both the cryostat and the coolant pot were filled to start the second test. Within 45 minutes the coolant flow rate had reached the desired level and remained constant throughout the remainder of the test. Within 4 hr the heat rate dropped to 8 percent of the precoolant value and remained steady for the next 2 hr, at which point the test was concluded.

After the pots were again filled, the coolant flow rate was established at the free flow value, i. e., no heater. The current to the strut heater was reduced to maintain the same outer end temperature as for the two previous tests. Within 6 hr the heat rate reached a steady value which lasted until the test was concluded after 16 hr.

Test IV-4 was intended to duplicate, as nearly as possible, the conditions for the first test which was cut short due to instabilities. The experience was very similar during this test. In the first 3 hours, the thermal link readings dropped from 180 to 55 μvolts . The outer boundary temperature also dropped and it took about 3 hours to dampen out the fluctuations in temperature by adjustments to the current in the strut heater. The thermal link reading became reasonably steady between 50 to 60 μvolts for the next 3 hours. Then, once again the heat rate started to rise slowly as in run IV-I. Simultaneously, an increase in the coolant temperature was again noted. Apparently the

high flow rates and associated heat input was causing significant stratification at the lower liquid levels encountered in tests IV-1 and IV-4. Consequently, it was concluded that the coolant supply was too small to achieve steady state at these test conditions.

Tests IV-5 and IV-6 were conducted to determine the effect of lowering the warm end temperature on the performance of the cooled strut. The flow rates were selected to give approximately the same ratio of heat rate to flow rate as for runs IV-1, and IV-2. As with the two previous tests (IV-2, 3) at reduced flows, no operational or control problems were encountered. It took less than 1 hr to achieve steady-state conditions.

For the two no-flow tests, the copper tubes used for coolant inlet and discharge lines were removed to avoid short-circuiting the strut. Otherwise, the test setup and procedures were the same as for runs IV-1 through 6. These tests were intended to provide the heat rate through the titanium strut, which would serve as a comparative reference for all of the strut concepts.

TEST RESULTS AND ANALYSIS

The experimental heat rates obtained for each of the eight tests are summarized in Table 12. It is immediately obvious that cooling is very effective in reducing the heat conducted down the strut.

Table 12

TEST RESULTS - COOLED STRUT

Test No.	Boundary Temperature ($^{\circ}$ R)		Coolant Flow Rate (lb/hr)	Heat Rate (Btu/Hr)
	Warm	Cold		
IV-1	539	LH ₂	0.0375	0.26
IV-2	536	LH ₂	0.0193	0.50
IV-3	536	LH ₂	0.0103	1.46
IV-4	537	LH ₂	0.039	0.46
IV-5	441	LH ₂	0.015	0.92
IV-6	439	LH ₂	0.022	0.35
IV-7	441	LN ₂	0	3.7 (2.7)
IV-8	441	LH ₂	0	4.6 (3.7)

The two heat rates given in parenthesis above for Tests IV-7 and 8 are from the heater measurements, whereas all others were obtained from the thermal link data. Neither of these constitute a true reference for normalizing the heat rates from the cooled tests. The reason for this can be deduced from the temperature profiles shown in Fig. 51. It will be noted that the cold end of the strut is considerably elevated in the tests, a condition which would not exist in an actual installation. Approximately 60 percent of the total temperature potential is dropped across the thin wall portion of the tube, and the remainder across the attachment and thermal link. These latter elements were designed and calibrated for much lower heat conduction rates with corresponding temperature differentials that are relatively insignificant compared to the total, i. e., to have thermal resistance that is small compared to that of the test article. Such is not the case, however, with the highly conducting titanium tube, and the net result is that the connecting resistance imposes a superficial cold boundary temperature on the test article.

The discrepancy between the measured heat rates into and out of the strut seem to indicate some lateral heating. However, it must be noted that the thermal link had been calibrated with heat rates only to 1 Btu/hr and ΔT 's up to 25^oR. Consequently, the thermal link measurements indicated for these two tests were actually obtained by extrapolating over a range three times that for which experimental data exists and one might expect some error. To check this, the theoretical temperature profile over the tubular length of strut was calculated, using the experimental warm end temperature and the indicated heat rates. The resulting temperature gradients, based on thermal link measurements, were larger than those experimentally observed but gave excellent agreement when calculated with the heater measurements.

In view of this and the fact that the thermal link correlation was extrapolated over a range three times that for which experimental data exists, it seems reasonable to conclude that: (1) the indicated heat rate is high; (2) the thermal conductivity of 2.5 Al titanium is in excellent agreement with the test results and can be used to predict, with confidence, the actual heat rate of the uncooled titanium strut. With the cold end at liquid hydrogen temperature, the reference heat rates are 7.2 Btu/hr and 4.9 Btu/hr for a warm end temperature of 540^oR and 440^oR, respectively.

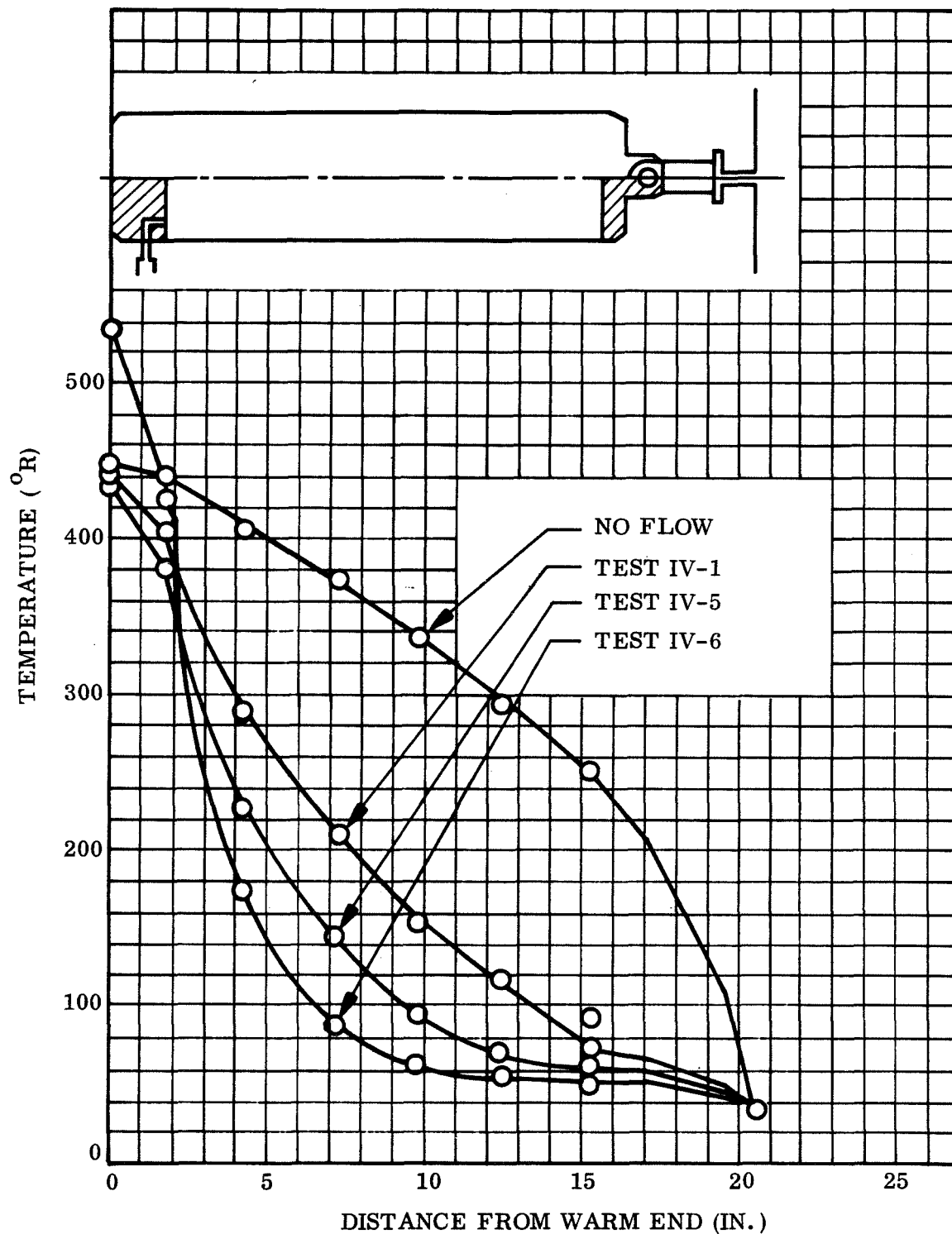


Fig. 51 Temperature Profiles - Cooled Strut

The theoretical analysis of the cooled strut that was presented in Ref. 2, described the cooling effectiveness in terms of the dimensionless parameters γ , and ψ , which are defined by:

$$\gamma = \frac{h P L^2}{k A_c}$$

$$\psi = \frac{h P L}{w C_p}$$

where

- h = heat transfer coefficient
- P = tube wetted perimeter
- L = cooled length of tube
- k = thermal conductivity of tube material
- A_c = conduction area of tube
- w = coolant flow rate
- C_p = specific heat of coolant

These parameters were evaluated for the actual test conditions and are given in Table 13 which compares the experimental heat rates to the theoretical values.

Table 13

COMPARISON OF EXPERIMENTAL AND THEORETICAL
VALUES FOR COOLED STRUT

Test No.	γ	ψ	$(Q/Q_0)^*$	
			Experiment	Theory
IV-1	52	6.7	0.036	0.050
IV-2	20	4.8	0.070	0.180
IV-3	24	10.5	0.20	0.320
IV-4	56	6.6	0.024	0.042
IV-5	22	6.0	0.190	0.280
IV-6	43	8.0	0.07	0.10

*Q/Q₀ ratio of heat rate through the cooled strut to that through the uncooled strut

It can be seen that the experimental data consistently falls below the theory. This is also illustrated in Fig. 52, which indicates that the theoretical predictions should be reduced by approximately 30 percent. This provides a maximum saving of 10 lb, relative to the Task I results (Ref. 2) for the 15 ft CH₄ Tank and 10⁴ hours. For all the other vehicles, the savings would be insignificant.

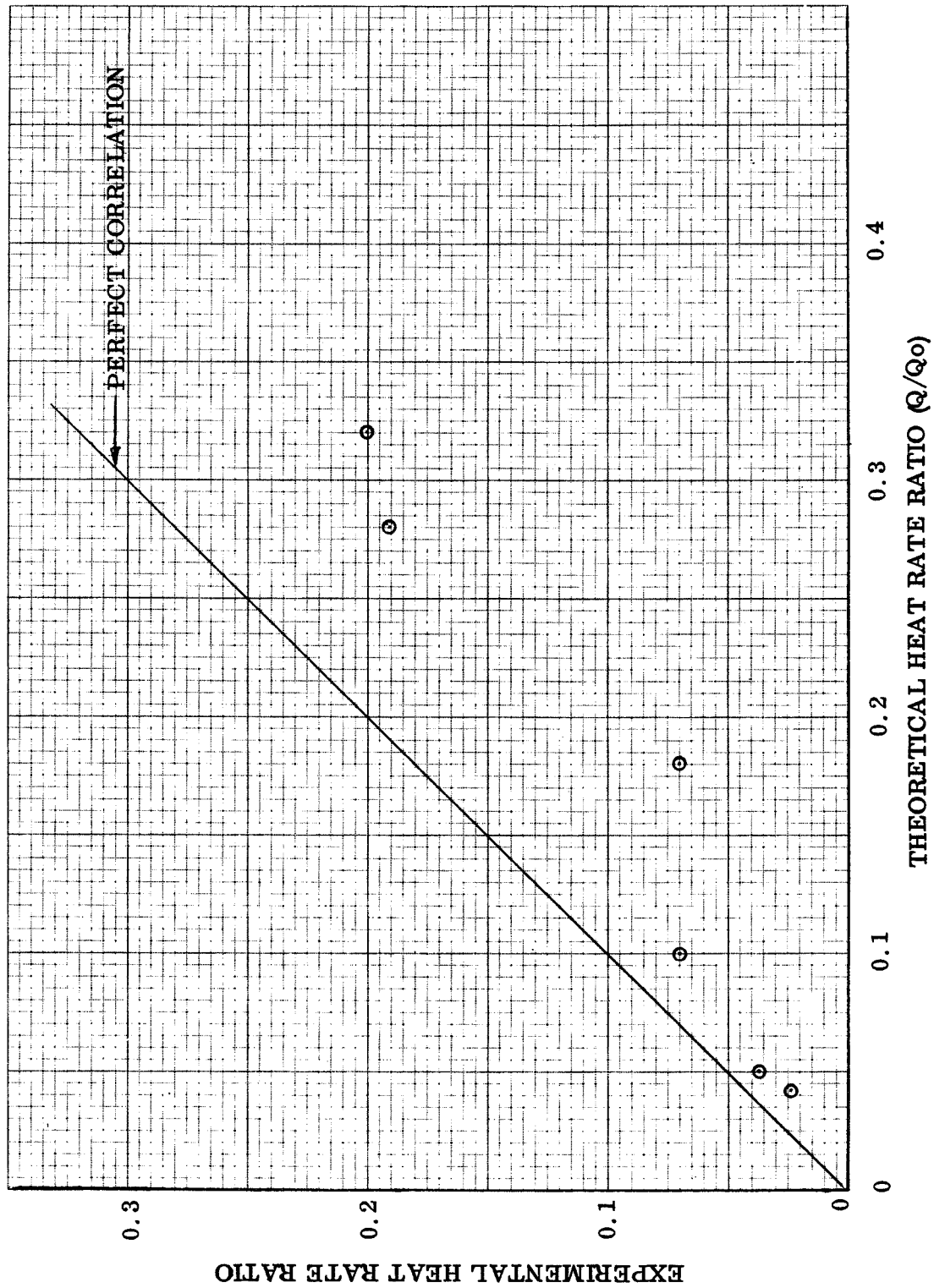


Fig. 52 Comparison of Theory & Experiment - Cooled Strut

FIBERGLASS MONOCOQUE CONE SUPPORT CONCEPT

TEST ARTICLE DESCRIPTION

All the tests conducted in this program were concerned only with evaluating the heat transfer associated with fixed, uniform temperatures. Consequently, test results obtained on an arc segment of a cone are applicable to the entire cone, providing there are no edge effects.

The results previously obtained on the fiberglass strut indicated that with a proper thermal guard and sufficient insulation, lateral heating (hence edge effects) could be eliminated. Therefore, a 10 deg segment of a 30-in.-diameter, monocoque cone was chosen for the test article. The design, shown in Fig. 53, was made of 181 glass fabric to have the same length and thickness as specified in Task I for the 5-ft-diameter methane tank. The test article was made in a wet lay-up using Epon 828 as the resin with triethylene triamine as a catalyst. The wet sheet was vacuum bagged and cured at room temperature for 16 hours. The resin content of the finished product was 28 percent by weight and the average thickness was 0.040 in. rather than the 0.038 specified in Fig. 53.

TEST ASSEMBLY DESCRIPTION

Both the monocoque and the honeycomb conical segments were tested on the same cryostat and in the same dewar as the three metallic strut support concepts. However, the thermal link was replaced with the new attachment fixture shown in Fig. 54. The entire fitting was made of aluminum to provide a near isothermal base for both the test article and the thermal guard, which attach to the vertical ribs. The ribs were contoured to match the test section curvature.

The test assembly is illustrated in Fig. 55. The test article was bonded to the inner face of the thick vertical rib on the support fixture. A thin copper strip, 0.5 in. wide, was bonded to both sides of the upper end of the test article, and then wrapped with 2 mil Evanohm heater wire, to provide uniform temperature control for the warm edge of the test article.

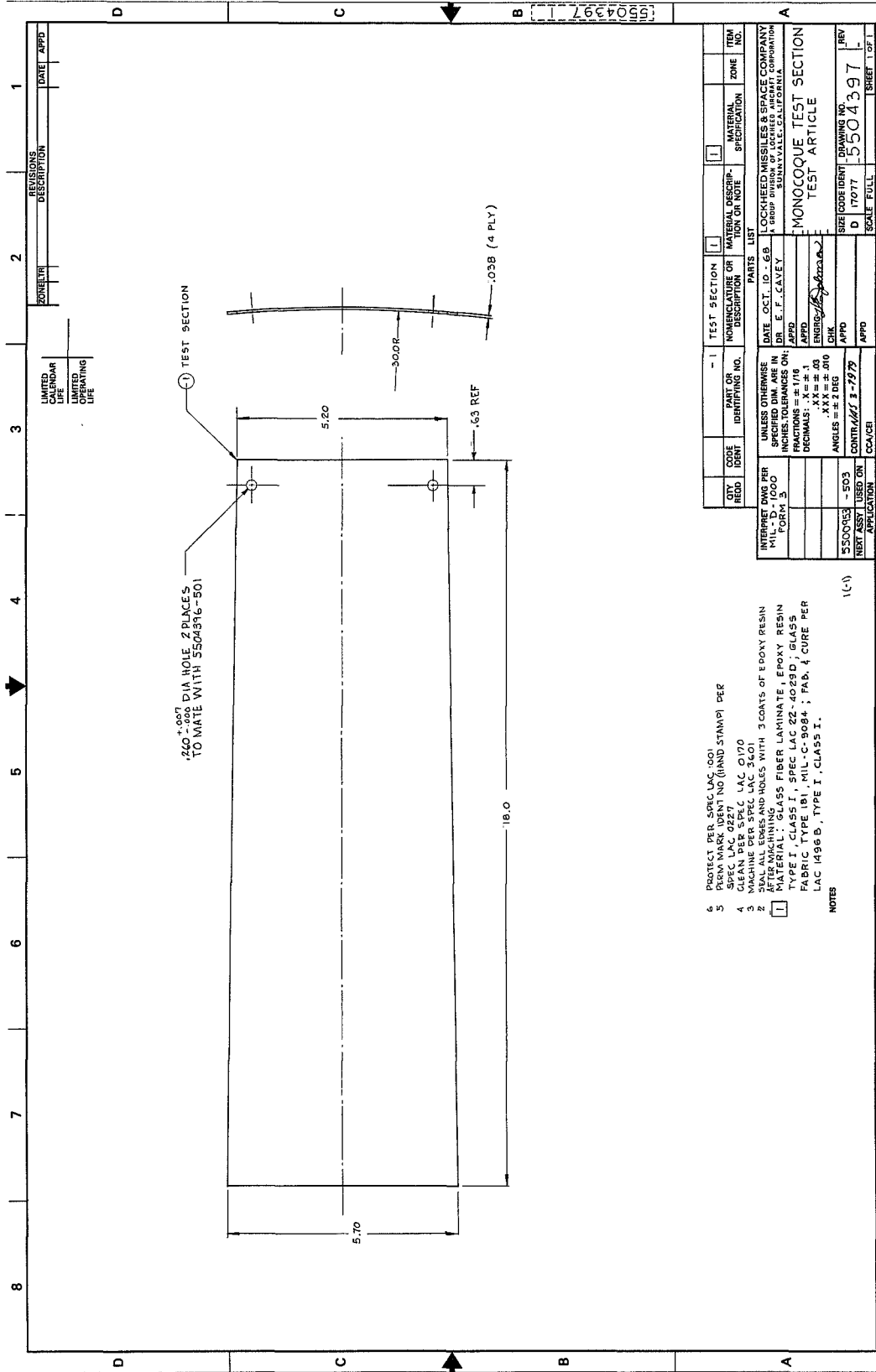


Fig. 53 Monocoque Test Article

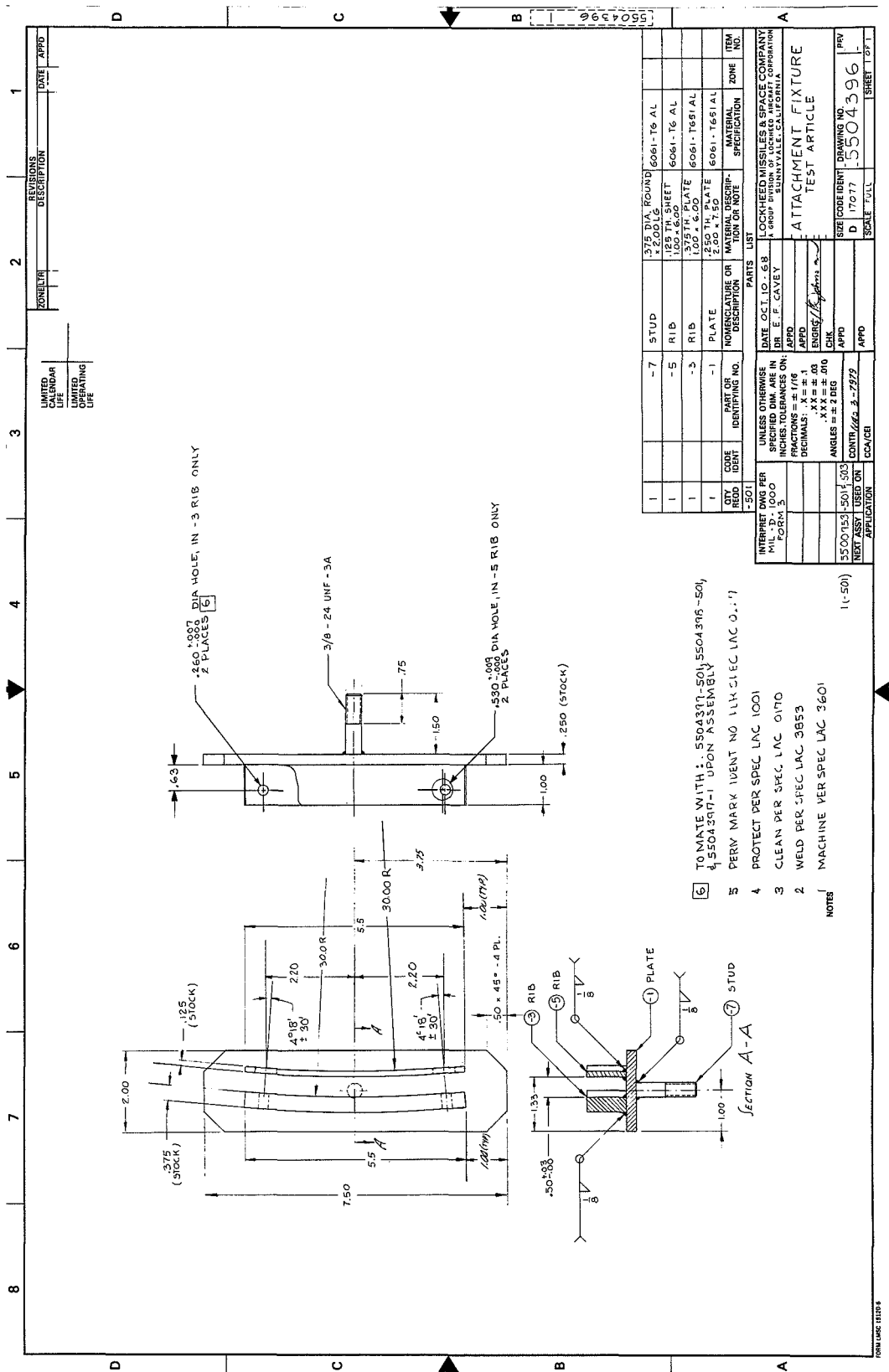


Fig. 54 Attachment Fixture for Cone Tests

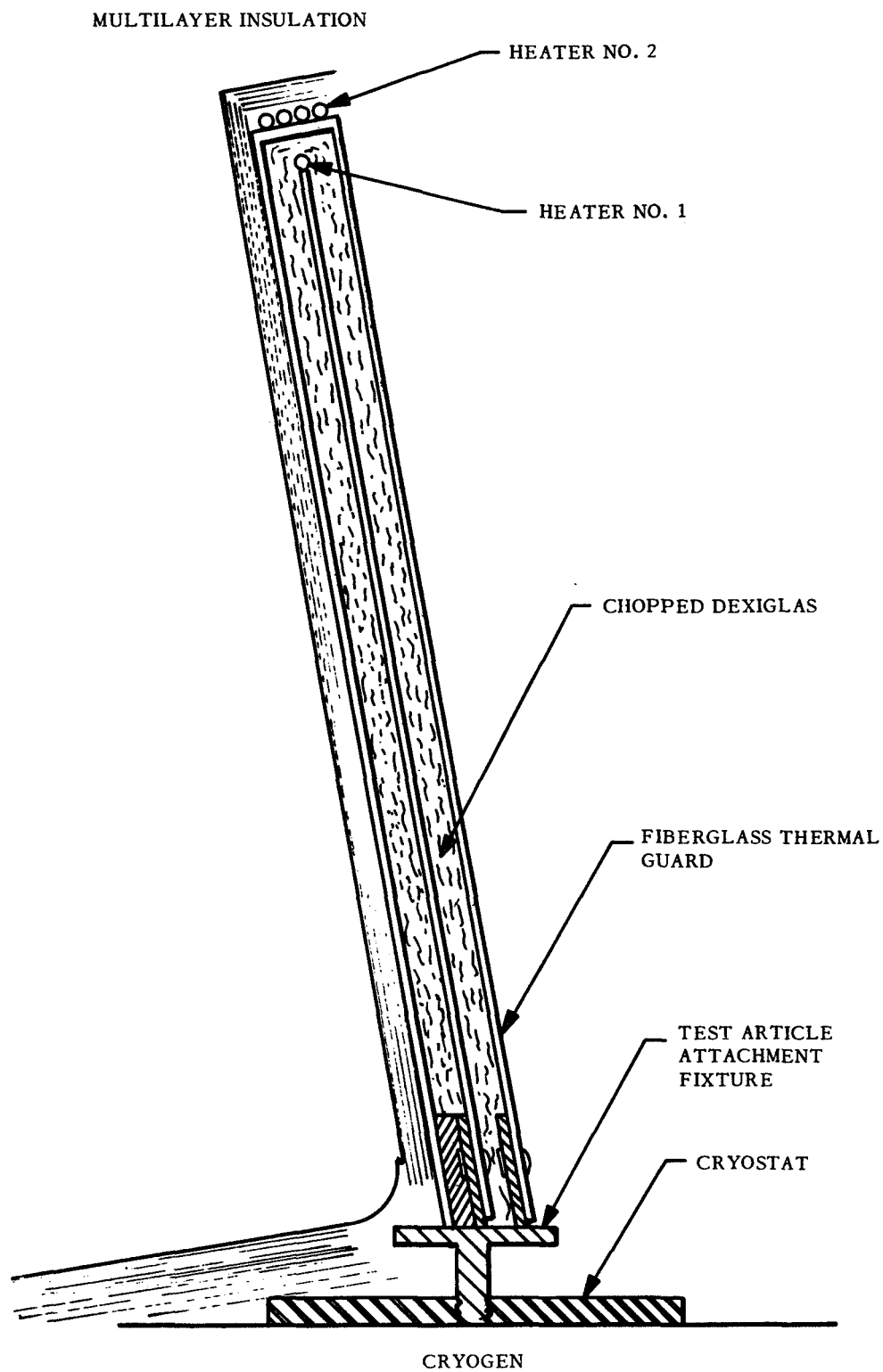


Fig. 55 Schematic of Test Installation for Fiberglass Cone Segment

A fiberglass thermal guard completely surrounded the test article. The guard design is shown in Fig. 56. The wraparound portion was bolted to the outer surface of the thick rib, and the flatter portion bolted to the thin rib to form the enclosure.

A 3-in.-wide copper sheet was bonded to the guard inner surfaces at the upper end, to provide an isothermal enclosure for the end of the test article. A heater assembly made with copper sheet and Evanohm heater wire formed the outer end closure, and was used to match the temperatures in copper clad portions of the guard and test article. The lower edge of the copper cladding was at 0.5 in. higher elevation on the guard, than on the test article, making the unshorted length of fiberglass 17 in. and 16.5 in. respectively.

The 3/8-in. space between the guard and test section was filled with Dexiglas to minimize interaction between the two, and 100 layers of aluminized mylar, with Dexiglas spacers were used to insulate the thermal guard from the environment.

INSTRUMENTATION

The test section was instrumented with eight cu-cn differential thermocouples, wired in series and referenced to a platinum resistance thermometer. The PRT and six of the thermocouples were directly on the fiberglass. The seventh one was on the base of the aluminum support fixture and the eighth was on the surface of the cryostat.

The guard was instrumented with seven differential thermocouples. Four are equally spaced over the fiberglass; two are on the copper cladding at the top of the guard; and the last one was also mounted on the surface of the cryostat. The location of the temperature measurements are shown on Fig. 57.

The heat input was determined by measuring the current to the heater which had a resistance of 178 ohms.

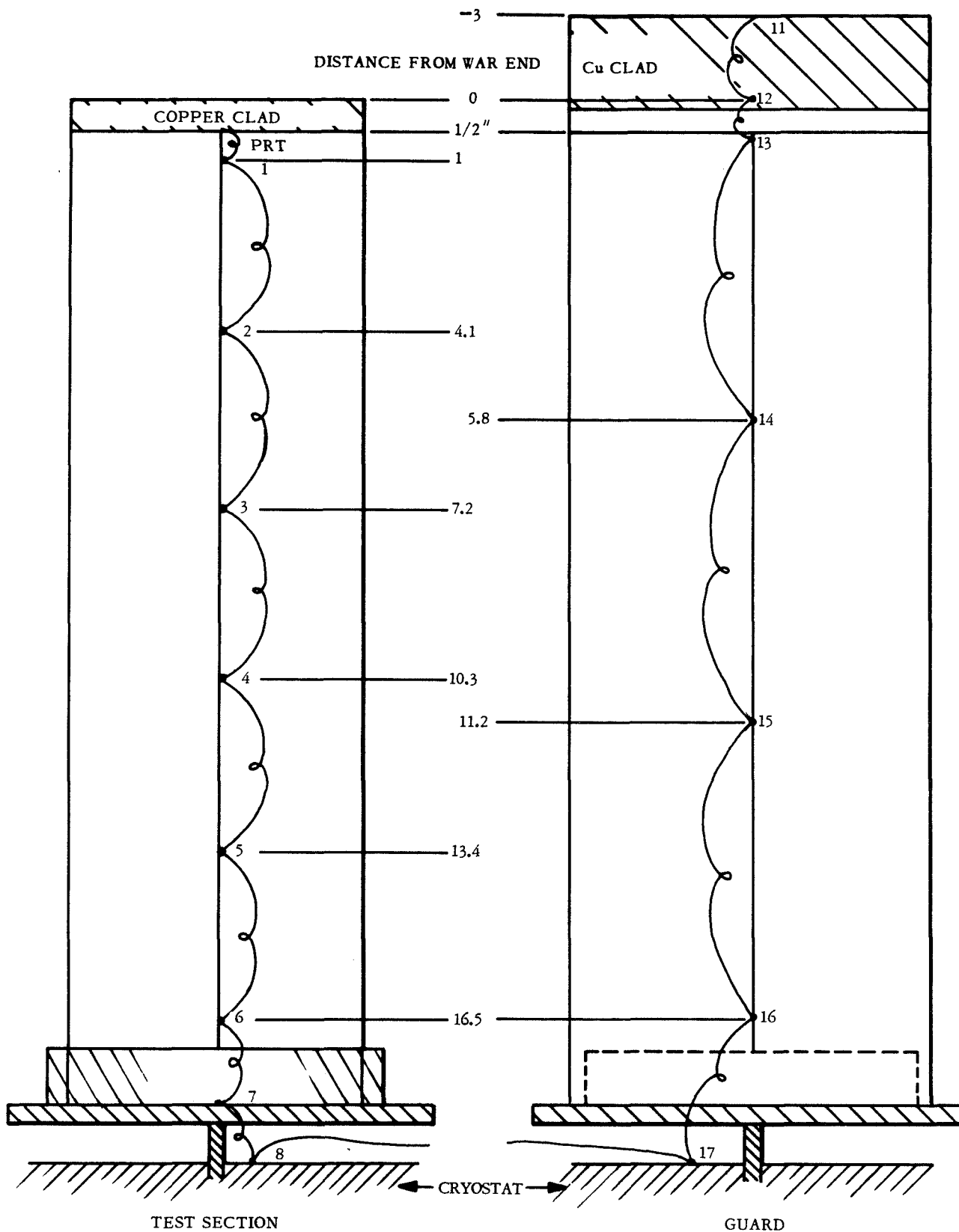


Fig. 57 Instrumentation Schematic - Fiberglass Cone

TEST PROGRAM

The basic objective of this test program was to determine the thermal conductivity of the cone material as a function of temperature. To achieve this objective, the three sets of boundary conditions identified in Table 14 were evaluated.

Table 14

TEST CONDITIONS – FIBERGLASS CONE

Test No.	Warm End Temperature ($^{\circ}$ R)	Cryogen
V-1	440	LN ₂
V-2	440	LH ₂
V-3	540	LH ₂

The test procedures were essentially the same as for the fiberglass strut. However, the heater currents and the temperature profiles were the only indicators of steady-state conditions, since there was no thermal link.

TEST RESULTS AND ANALYSIS

The experimental heat rates as determined from the heater input currents for the tests on the fiberglass cone segment are given in Table 15.

Table 15

TEST RESULTS – FIBERGLASS CONE

Test No.	Boundary Temperatures		Heat Rate Btu/hr
	Warm	Cryogen	
V-1	447 $^{\circ}$ R	LN ₂	0.0704
V-2	448 $^{\circ}$ R	LH ₂	0.0744
V-3	540 $^{\circ}$ R	LH ₂	0.105

There was no thermal link measurement to compare with the heater power as in the previous tests. However, the tests on the fiberglass strut indicated that the heater power was all that is necessary, providing the guard is reasonably well matched to the test article.

Figures 58 through 60 show the temperature profiles in both the guard and the test article for each of the three tests. It can be seen that they agreed much more closely than for the fiberglass strut tests. In fact, the mismatch is hardly noticeable, except for two or three inches at the bottom where the guard tends to be a few degrees cooler.

These tests are only useful if they can be applied to other geometries and boundary conditions. Therefore, the thermal conductivity has been determined as a function of temperature, from the experimental data, using the following steady-state analysis:

$$Q = -k(T) A(x) dT/dx \quad (1)$$

For a cone, the cross sectional area, at any point x, is

$$A(x) = t \left[y_o - \frac{(y_o - y_1)}{L} x \right] \quad (2)$$

Substituting Eq. 2 into Eq. 1, and rearranging, we get

$$\int_{x_i}^{x_{i+1}} \frac{Q dx}{t \left[y_o - \frac{y_o - y_1}{L} x \right]} = \int_{T_i}^{T_{i+1}} -k(T) dT \quad (3)$$



Fig. 58 Temperature Profile in Conical Support – Test V-1

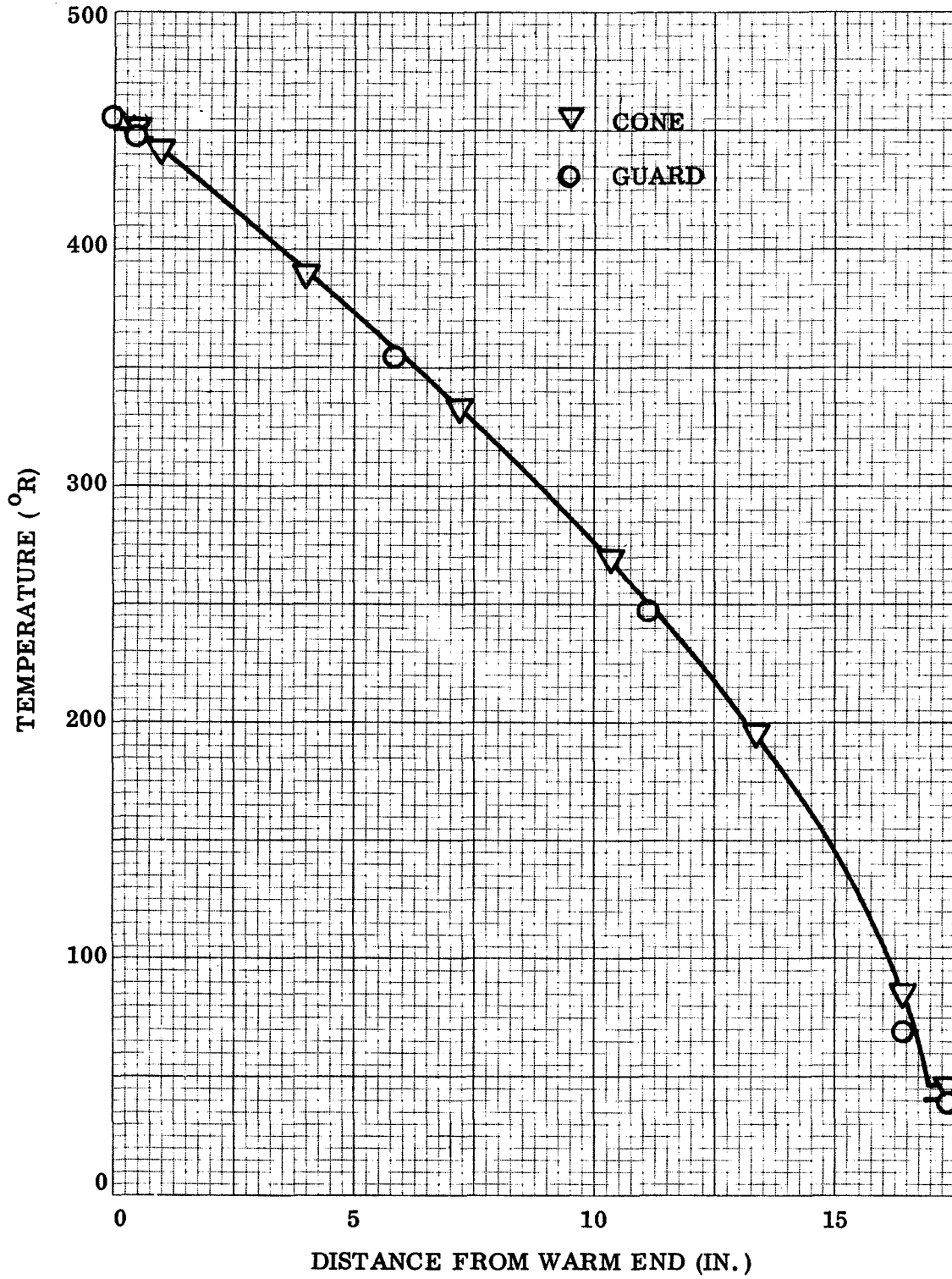


Fig. 59 Temperature Profile in Conical Support – Test V-2

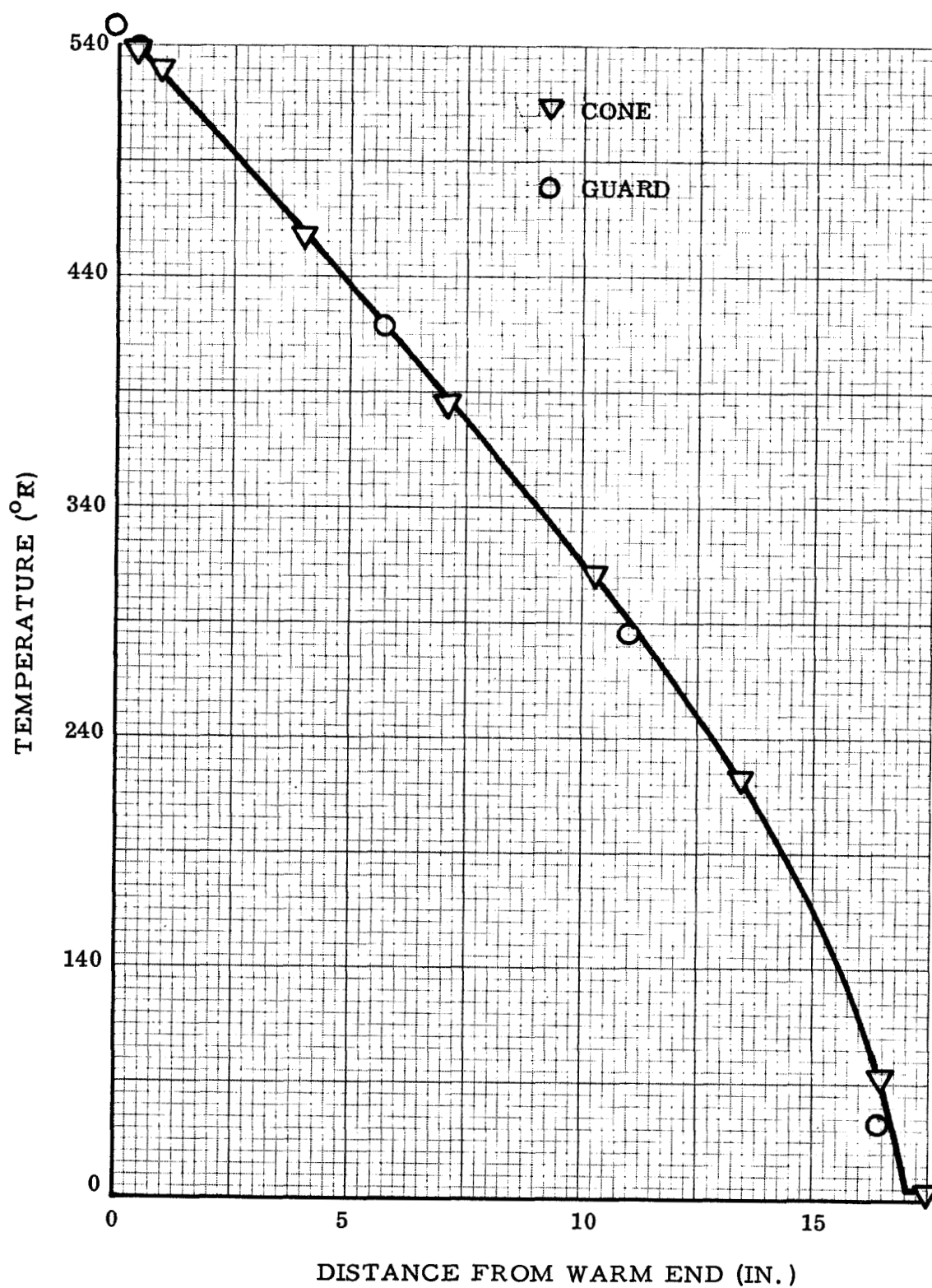


Fig. 60 Temperature Profile in Conical Support - Test V-3

The left side of Eq. 3 can be integrated between any two known points and we get:

$$Q \left[\frac{L}{t (y_o - y_L)} \ln \left(\frac{y_{i+1}}{y_i} \right) \right] = \int_{T_i}^{T_{i+1}} k(T) dT \quad (4)$$

Q = heat rate

y_o = cone circumference at $x = 0$

y_L = circumference at $x = L$

$k(T)$ = thermal conductivity

T = temperature

where, the left side is a function only of the geometry and the heat rate. The mean thermal conductivity between T_i and T_{i+1} is then given by:

$$k(T) = \frac{\left[\frac{QL}{t (y_o - y_L)} \ln \left(\frac{y_{i+1}}{y_i} \right) \right]}{(T_{i+1} - T_i)} \quad (5)$$

Equation 5 was evaluated for each of the five measured temperature increments for each test and the results are shown on Fig. 61. Also shown is the curve that was used for the thermal analysis in Task I.

The present data are very consistent from one test to the other. The nitrogen data are slightly higher than that from either of the hydrogen tests, but the scatter is less than 15 percent.

It is also apparent that the present data consistently fall below the reference curve. The difference effects a net reduction in the calculated heat rates as indicated below:

520°R to CH₄ - 11 percent

400°R to CH₄ - 14 percent

520°R to LH₂ - 17 percent

400°R to LH₂ - 19 percent

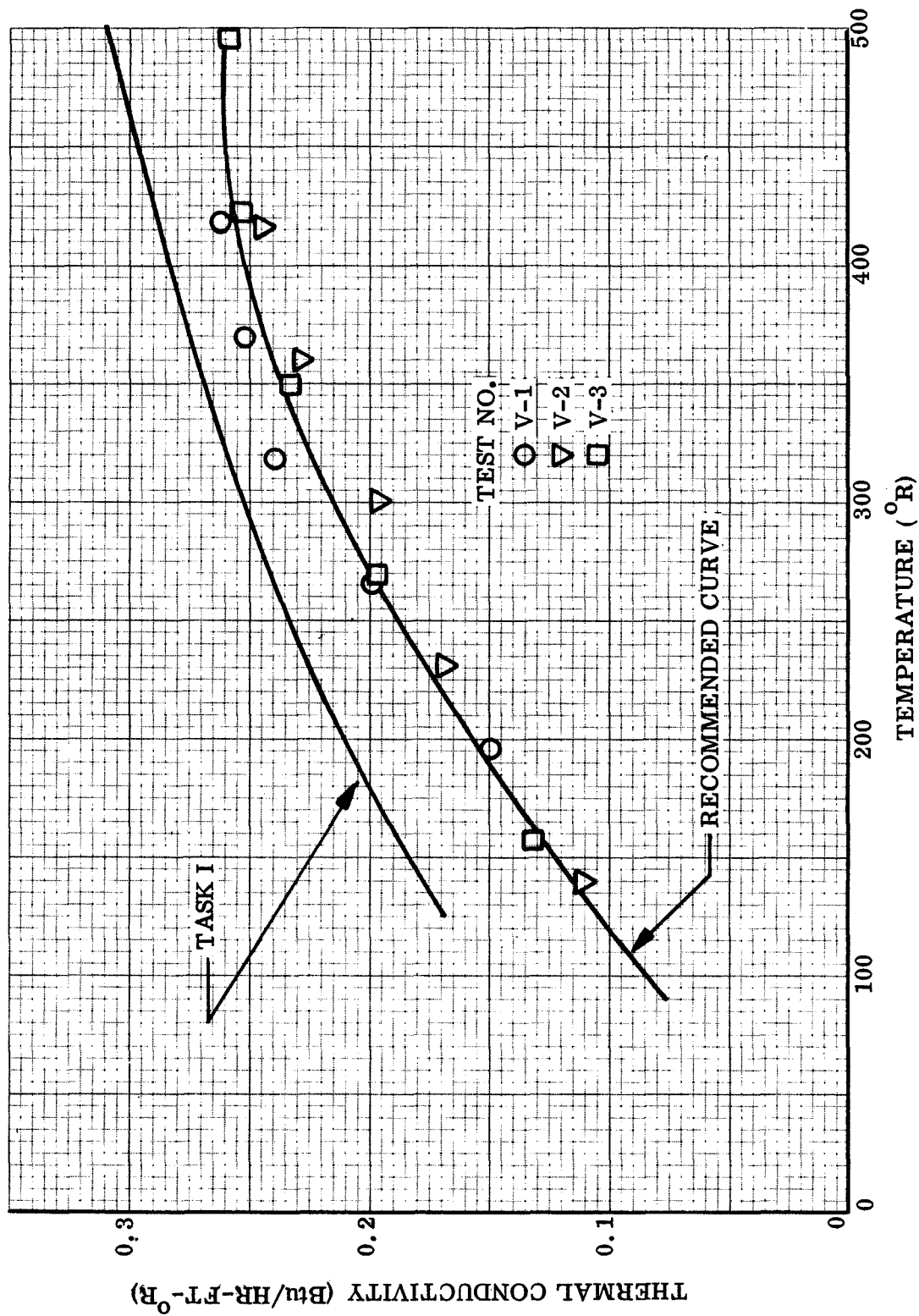


Fig. 61 Experimental Thermal Conductivity - Fiberglass Cone Support

FIBERGLASS HONEYCOMB SUPPORT CONCEPT

TEST ARTICLE DESCRIPTION

The philosophy of testing only a segment of a cone was also applicable to the fiberglass honeycomb continuous support concept. Therefore, the 10-deg-arc segment shown in Fig. 62 was used for evaluating this concept.

The test article was intended to represent a segment of a cone designed for the 60-in. - diameter CH₄ tank, and thus the test results would be directly comparable to those from the fiberglass monocoque cone. Also, it permitted the use of the same test assembly for both continuous support concepts.

The face plates were cut from a two-ply sheet that was made in a wet lay-up that used for the monocoque section.

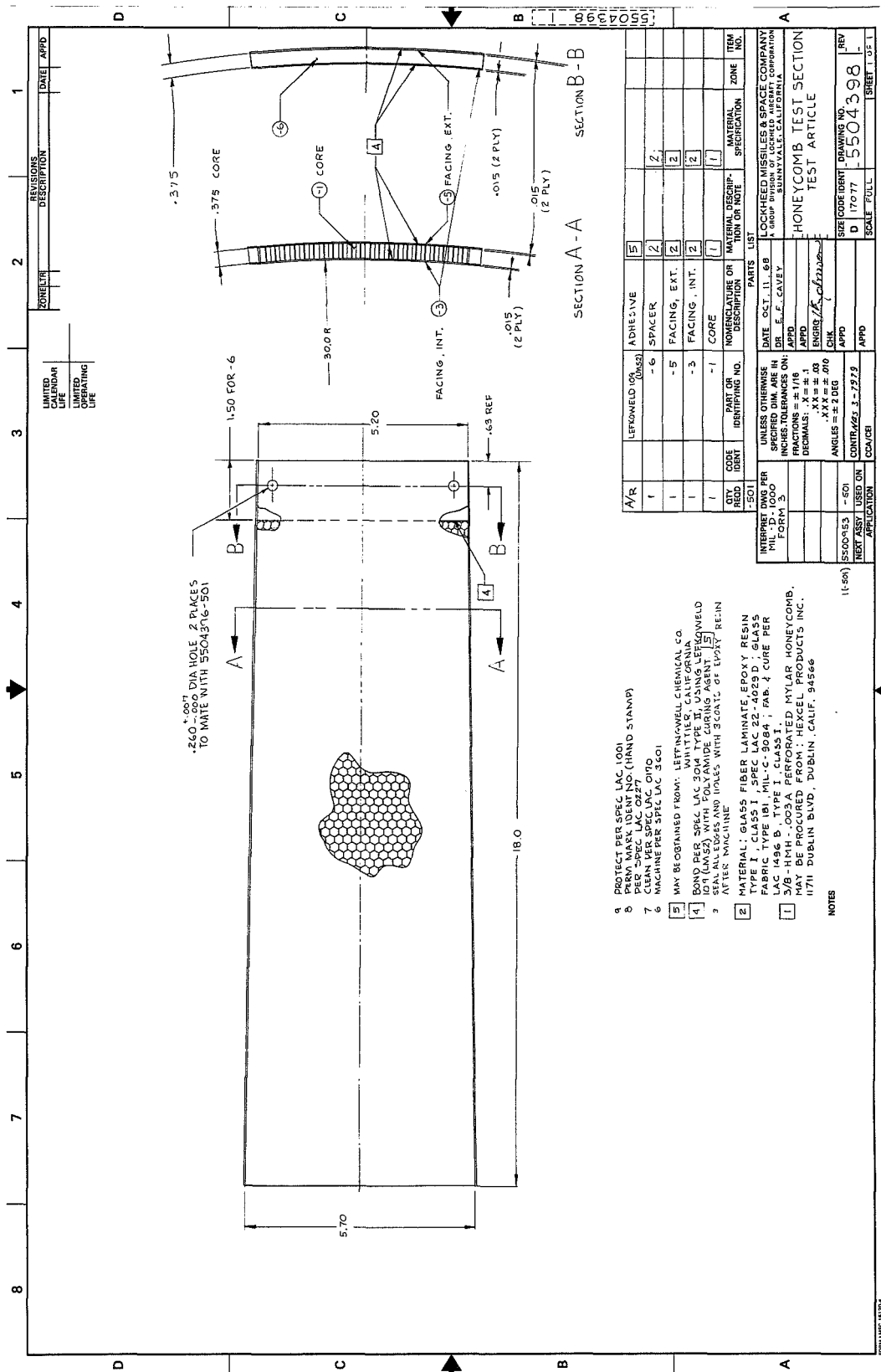
With the wet lay-up approach, the two-ply would not give an apparent homogeneous sheet at the specified thickness. Consequently, the finished product had an average thickness of 0.033 in. - and a resin content of 55 percent.

TEST ASSEMBLY

The test assembly for this conception was the same as that for the monocoque cone except for one minor change in the heater placement on the test article. The heater strip was bonded over the end of the honeycomb, whereas it was on the vertical surface for the monocoque. As a result, the effective length of the test section was 17 in. rather than 16.5 in. There were no other changes made in either the thermal guard, the cryostat, or attachment fixture.

INSTRUMENTATION

The instrumentation schedule was the same as previously described for the monocoque test assembly.



TEST PROGRAM

The basic objective of this program was to obtain sufficient data for a direct comparison between the honeycomb and fiberglass monocoque concepts. The tests on the fiberglass strut showed that the effect of changing cold boundary temperature on heat rate was small compared to the effect of changing warm boundary temperature. This was later substantiated by the test results on the cone. Therefore, only two test conditions were evaluated for the honeycomb test section, both with liquid nitrogen. The two warm boundary temperatures were 539°R and 475°R.

TEST RESULTS AND ANALYSIS

The experimental heat rates for each of the two boundary conditions are given in Table 16.

Table 16

TEST RESULTS – HONEYCOMB CONE SUPPORT

Test No.	Boundary Temperature (°R)	Heat Rate Btu/hr
VI-1	539	0.17
VI-2	475	0.12

For comparison, the heat rates expected just through the face plates were calculated using the thermal conductivity data from the monocoque tests and the temperature profiles shown in Fig. 63. These calculated rates were 0.155 and 0.122 Btu/hr for Tests VI-1 and VI-2, respectively. The difference between these calculated rates and the total measured heat rates is within the thermal conductivity data scatter. It does indicate that the core contributes a very small portion of the total heat rate.

The effective thermal conductivity was determined for the honeycomb. This is defined by the relationship:

$$Q = -k_{\text{eff}} A_{\text{eff}} \frac{dT}{dx}$$

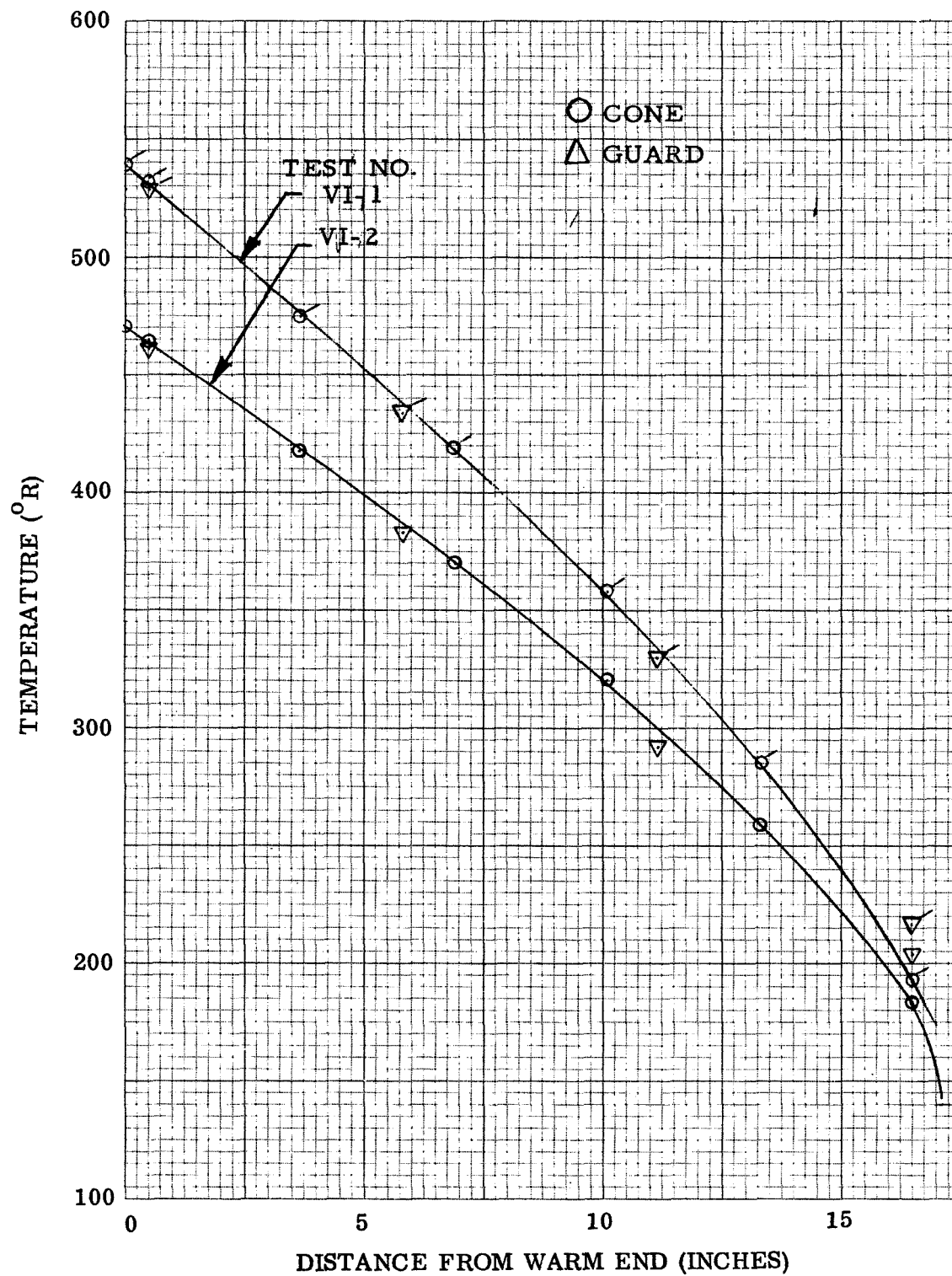


Fig. 63 Temperature Profiles – Honeycomb Cone Tests

For the honeycomb test article, the effective conducting area was approximately 14 percent greater than that of the face plates. Figure 64 shows the effective conductivity as a function of temperature. Also shown is the recommended curve from Fig. 61 for the fiberglass monocoque. The scatter about this curve is within 20 percent, although the trend of the honeycomb data appears to be somewhat steeper. However, it is reasonable to apply the same percentage corrections for the honeycomb support concept as was recommended for the monocoque concept.

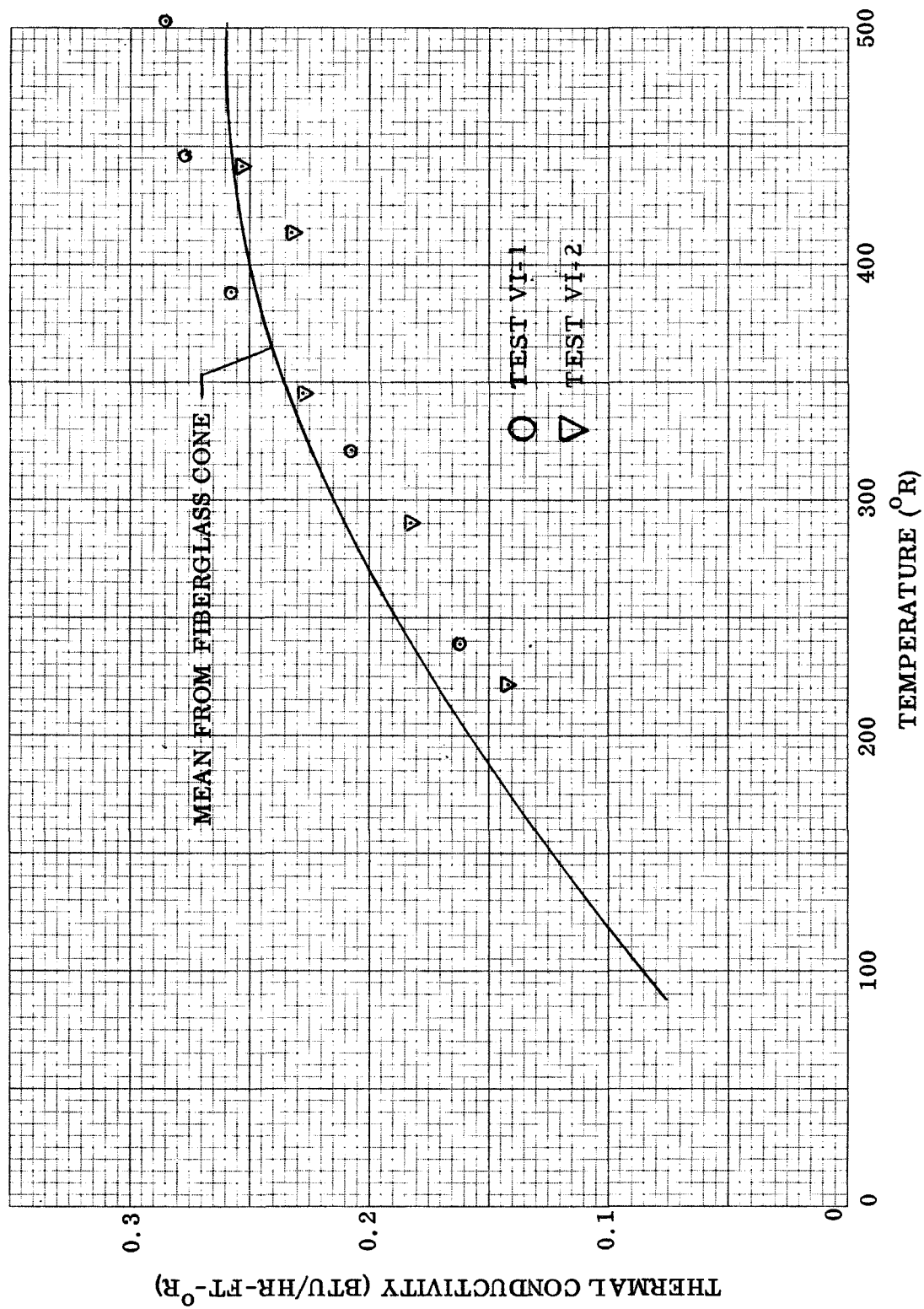


Fig. 64 Experimental Thermal Conductivity - Honeycomb Cone Support

REVISED TASK I RESULTS

The testing in Task II was limited to thermal evaluation of the six concepts that were recommended as a result of the analysis conducted in Task I. Therefore, the revisions presented herein reflect only changes in the predicted thermal performance.

FIBERGLASS STRUT CONCEPT

The heat rates presented in Ref. 2 for this concept are inconsistent with the designated support length because a portion of the length, consisting of metal rod ends, and titanium end fittings, is highly conductive, relative to the fiberglass body. For the test article, these amounted to approximately 6 in., and this will not change significantly for the struts considered in Task I. Consequently, if the total support length is 18 or 36 in., the effective thermal length is 12 and 30 in. respectively. The heat rates presented in Table 17 reflect this revision as well as the thermal conductivity changes recommended in Task II.

Table 17

THERMAL PERFORMANCE – FIBERGLASS POINT SUPPORT CONCEPT

Tank Diameter (ft)	Support Length (in.)	Heat Rate – Btu/hr/strut		System Boiloff Rate (lb/hr) x 10 ⁴	
		LH ₂ tank	CH ₄ tank	LH ₂	CH ₄
5	18	0.032	0.022	10.0	5.2
	36	0.012	0.015	3.9	4.8
9	18	0.032	0.062	10.0	14.7
	36	0.030	0.048	9.2	11.7
12	18	0.054	0.13	17.5	31
	36	0.055	0.10	17.0	25
15	18	0.068	0.22	21.7	52
	36	0.065	0.138	20.2	33.2

The heating rates in Table 17 are for an outer boundary temperature of 400°R . These increase 57 percent and 72 percent for hydrogen and methane respectively, when the outer end temperature is 520°R .

STACKED WASHER STRUT

The Task I results for this concept were based upon the assumption that the washer conductance would not exceed $10 \text{ Btu/hr-ft}^2 \text{ }^{\circ}\text{R}$ per interface. Although there was considerable scatter in the test data on stainless steel washers, it did indicate that $100 \text{ Btu/hr-ft}^2 \text{ }^{\circ}\text{R}$ is more realistic. The revised heating rates shown on Figs. 65 and 66, are based upon this conductance value. However, a much more extensive experimental program must be conducted before sufficient data are available for design.

BALL AND CLAMP SUPPORT STRUT

The tests conducted on this concept confirmed that the major resistance occurs at the thermal disconnect joint, and that it is governed by radiation. However, for the calculations conducted in Task I, a value of 0.1 was used for emissivity of the ball and clamp. The test data were correlated with a nominal value of 0.3. This was used to reevaluate the network resistances for all the tanks considered in Task I. The resulting heat rates are given in Table 18. The effect of these revisions was to increase the effective system weights 15 to 30 percent for methane tanks, and a 10,000 hour mission. The maximum weight increment was 25 lb.

COOLED STRUT CONCEPT

The theory developed in Task I for the cooled strut predicted an insignificant amount of heating with cooled struts. The tests conducted in Task II confirmed these predictions. Consequently, no revisions are needed for this concept.

CONICAL SUPPORT CONCEPTS

The test results from Task II indicated that the thermal conductivity used for calculating the heat rates through both the fiberglass cone and the fiberglass honeycomb continuous

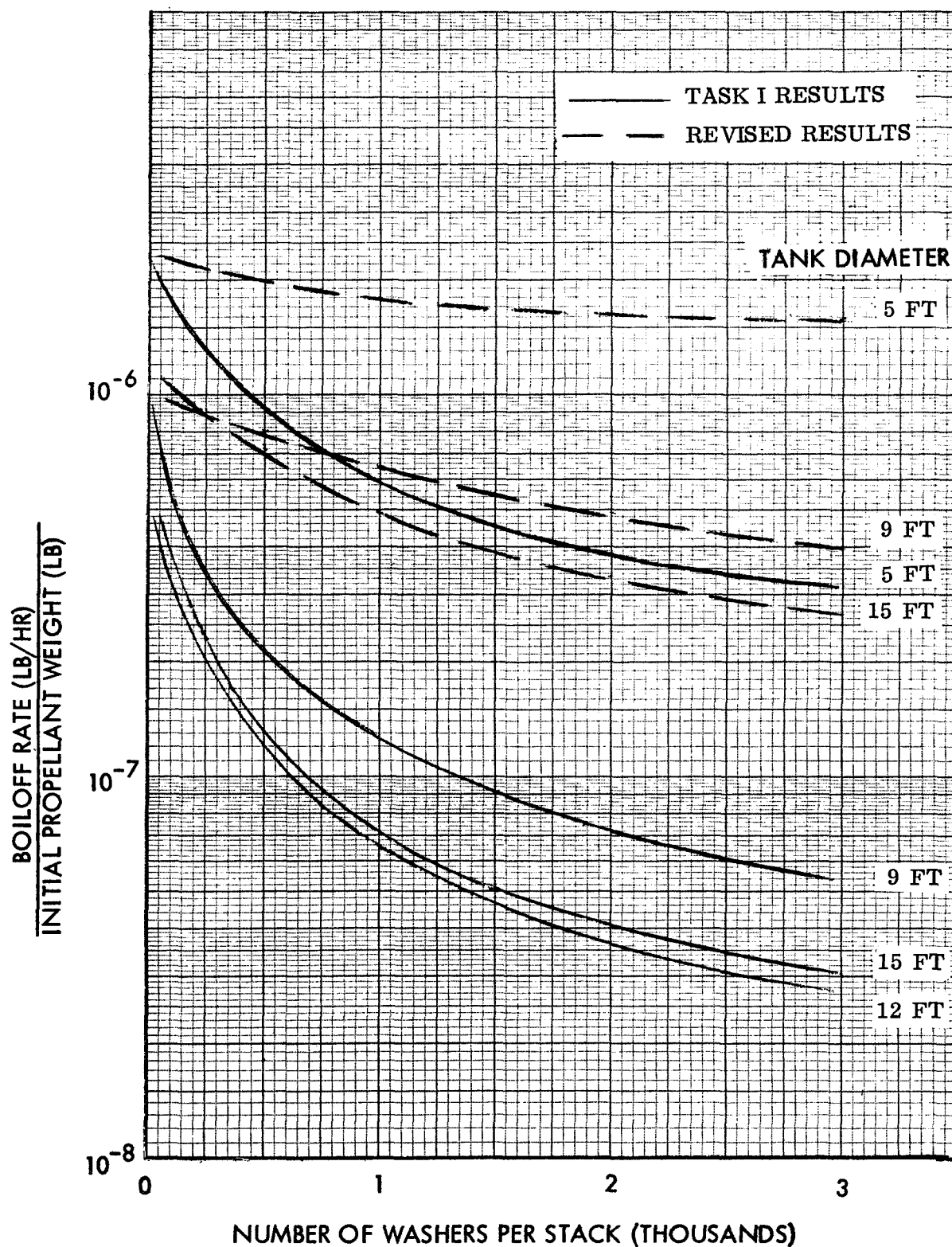


Fig. 65 Methane Boiloff Rates With Stacked Washer Concept

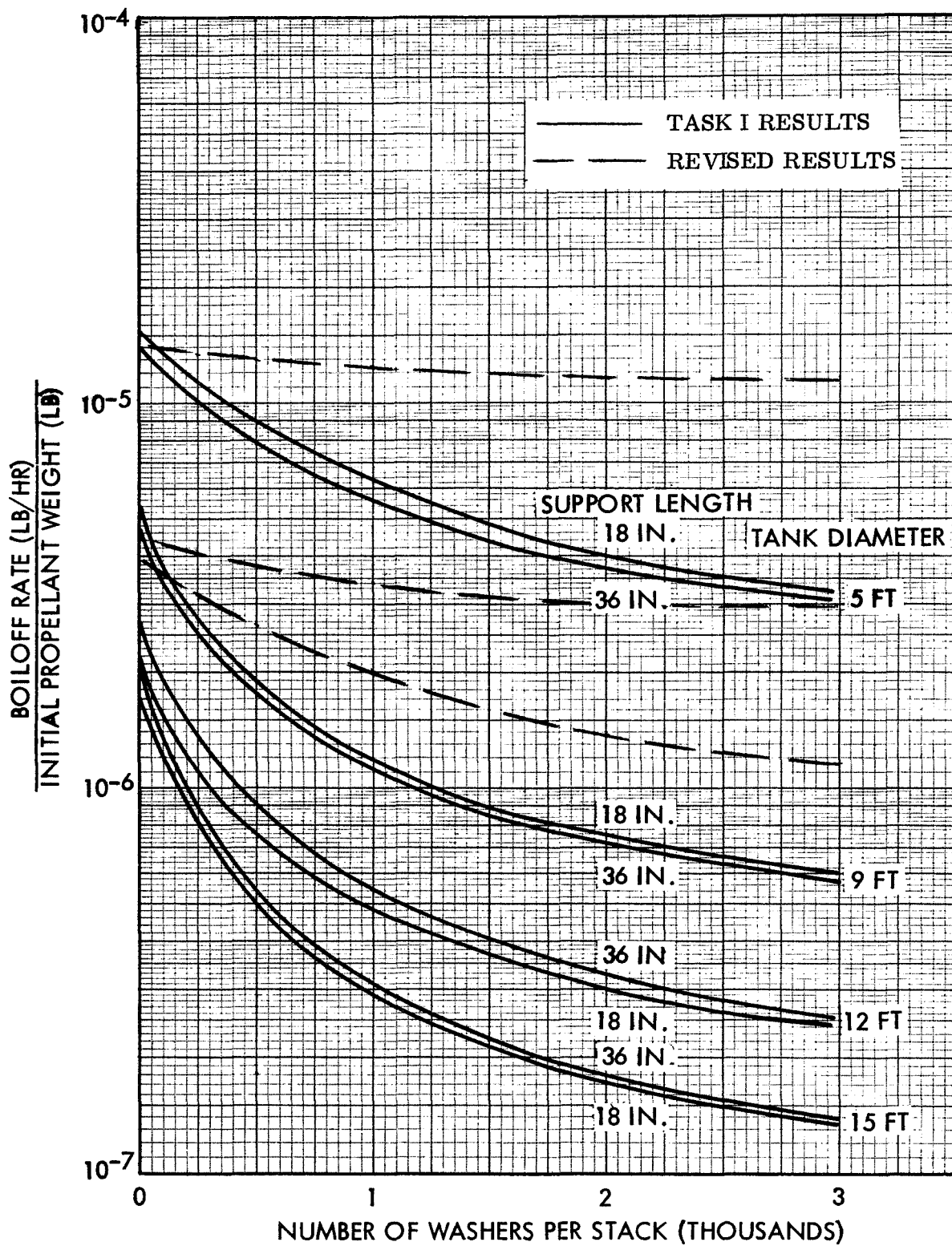


Fig. 66 Hydrogen Boiloff Rates With Stacked Washer Concept

Table 18

REVISED THERMAL PERFORMANCE FOR BALL AND CLAMP SUPPORT

Fluid	Tank Diam. (ft)	Support Length (in.)	Boiloff Rate
			(lb/hr x 10 ³)
H ₂	15	36	3.0
		18	2.5
	12	36	2.2
		18	1.5
	9	36	1.4
		18	1.6
	5	36	0.96
		18	1.6
CH ₄	15	36	3.2
		18	3.3
	12	36	2.6
		18	2.6
	9	36	2.0
		18	2.1
	5	36	1.2
		18	1.2

support was too high. They have been reduced according to the following schedule:

400°R - CH₄ - 14 percent
 520°R - CH₄ - 11 percent
 400°R - LH₂ - 19 percent
 520°R - LH₂ - 17 percent

The revised heat rates are shown in Tables 19 and 20 for the monocoque and honeycomb cones, respectively.

Table 19

REVISED HEAT RATES – FIBERGLASS CONE CONCEPT

Tank Diameter (ft)	Heat Rates (Btu/Hr)			
	LH ₂ Tanks		CH ₄ Tanks	
	L = 18 in.	L = 36 in.	L = 18 in.	L = 36 in.
5	1.8	1.1	1.5	0.9
9	3.5	2.1	3.4	1.7
12	4.7	2.9	6.6	3.5
15	7.0	3.7	11.9	6.4

Table 20

REVISED HEAT RATES – FIBERGLASS HONEYCOMB CONTINUOUS SUPPORT
(L = 36 in.)

Tank Diameter (ft)	Heat Rate (Btu/hr)	
	LH ₂	CH ₄
5	0.9	0.4
9	1.6	1.1
12	2.1	2.0
15	2.7	4.2

These heat rates are for an outer boundary temperature of 400°R. They must be increased 55 percent for hydrogen and 65 percent for methane, when the outer temperature is 520°R.

SYSTEMS COMPARISON AND RANKING

When selecting a support system for a particular propellant tank and a specific mission, the vehicle designer must consider the total weight penalty imposed by the system and

the cost and reliability of the system. However, in a conceptual design and screening study, such as the one described in this document, the realistic considerations are:

- Total system weight
- Thermal performance
- Capability of usefulness for a wide spectrum of tanks and missions

These can be quantitatively evaluated and coupled with a qualitative assessment of the mechanical complexities for the various systems, to select those worthy of further consideration.

A summary of the effective systems weights is presented in Table 21 for all the methane tanks and in Table 22 for all the hydrogen tanks. Two support lengths are included for each. The weights include both hardware (system plus tank stiffening weight) and boiloff: the latter being that which results if the sides of the support are perfectly insulated and the environment temperature is 400°R. The boiloff is the only quantity that was revised from the Task I results. For convenience, the concepts are identified by the same numbers previously used in Ref. 1 and listed in Table 1.

- | | |
|-----------------------------|--------------|
| ● Fiberglass Strut | — Concept 9 |
| ● Stacked Washer Strut | — Concept 8 |
| ● Ball and Clamp Strut | — Concept 4 |
| ● Cooled Titanium Strut | — Concept 11 |
| ● Fiberglass Cone | — Concept 7 |
| ● Fiberglass Honeycomb Cone | — Concept 6 |

These revisions reflect only a reevaluation of the thermal performance. Therefore, it is not surprising that significant changes result only for the longest mission duration of 10,000 hr. Even then, the incremental weight change is relatively small compared to the total system weight, with the exception of the stacked washer concept.

Results of this program clearly indicate that for the ellipsoidal propellant tanks between 5 and 15 ft, point-type supports result in the lowest weight penalty. The cooled titanium strut is theoretically the best of those concepts investigated. However, it requires

Table 21

SUPPORT SYSTEM WEIGHTS FOR METHANE TANKS

Support Concept →			4	6	7	8	9	11
Tank Diam. (ft)	Support Length (in.)	Mission Time (hr)	Effective Weight (lb)					
5	36	10	17	38	31	9	8.8	9
		10^2	17	38	31	10	8.8	9
		10^3	18	40	35	11	9.3	9
		10^4	29	54	69	31	13.6	9
	18	10	16	—	19	7.5	7	7.5
		10^2	16	—	20	8	7	7.5
		10^3	17	—	25	10	7.5	7.5
		10^4	28	—	79	29	11.9	7.5
9	36	10	28	62	76	17	19	17
		10^2	28	62	77	17	19	17
		10^3	30	67	83	23	20.5	17
		10^4	48	106	142	60	31	17
	18	10	23	—	55	9.5	11	9.5
		10^2	23	—	56	10	11	9.5
		10^3	25	—	69	15	12.5	10
		10^4	44	—	191	53	26	11
12	36	10	64	159	152	53	52	52
		10^2	64	160	153	56	52	52
		10^3	67	167	166	82	54	52
		10^4	90	239	292	129	77	52
	18	10	52	—	107	40	38	40
		10^2	52	—	110	43	38	40
		10^3	56	—	133	69	41	41
		10^4	86	—	371	116	69	45

Table 21 (Cont.)

Support Concept →			4	6	7	8	9	11
Tank Diam. (ft)	Support Length (in.)	Mission Time (hr)	Effective Weight (lb)					
15	36	10	176	288	274	145	145	145
		10 ²	176	290	277	148	145	145
		10 ³	179	305	300	174	148	145
		10 ⁴	208	456	530	300	178	145
	18	10	152	—	190	135	123	135
		10 ²	152	—	195	138	123	135
		10 ³	155	—	238	164	128	140
		10 ⁴	185	—	666	318	175	162

Table 22

SUPPORT SYSTEM WEIGHTS FOR LIQUID HYDROGEN TANKS

Support Concept →			4	6	7	8	9	11
Tank Diam. (ft)	Support Length (in.)	Mission Time (hr)	Effective Weight (lb)					
5	36	10	17	34	25	8	7.5	8
		10^2	17	35	26	8	7.5	8
		10^3	18	39	31	13	8	8
		10^4	27	80	81	56	11.4	8
	18	10	16	—	15	7	7	7
		10^2	16	—	16	7	7	7
		10^3	18	—	26	12	8	7
		10^4	32	—	107	55	17	7
9	36	10	17	65	52	9	8.8	9
		10^2	17	66	53	10	8.9	9
		10^3	18	73	64	18	9.7	9
		10^4	31	147	160	76	18	9
	18	10	16	—	31	7	7	7
		10^2	16	—	33	8	7	7
		10^3	18	—	49	16	8	7
		10^4	32	—	211	74	17	7
12	36	10	20	93	76	11	11	11
		10^2	20	94	78	13	11	11
		10^3	22	104	91	25	12.7	11
		10^4	42	201	224	106	28	11
	18	10	18	—	49	7	7	7
		10^2	18	—	51	9	7	7
		10^3	20	—	73	21	8.7	7
		10^4	33	—	289	102	25	7

Table 22 (Cont.)

Support Concept →			4	6	7	8	9	11
Tank Diam. (ft)	Support Length (in.)	Mission Time (hr)	Effective Weight (lb)					
15	36	10	26	122	100	16	16	16
		10^2	26	123	102	18	16	16
		10^3	29	136	119	34	18	16
		10^4	56	260	290	134	36	16
	18	10	22	—	69	10	9	10
		10^2	22	—	73	12	9.2	10
		10^3	25	—	105	28	11.2	10
		10^4	47	—	428	128	31	10

continuous venting of the propellant tank, which is not practical for long missions where such a concept might be required. The theoretical performance advantage of the cooled strut over that of the fiberglass strut is marginal, and the latter is much more practical and completely passive; consequently, it should be inherently more reliable. The indicated thermal performance of the stacked washer concept is inferior to that of the fiberglass strut. However, it is a passive support system and its performance might be improved, with a different washer material, providing that methods can be developed for obtaining repeatable performance from the thin, processed washers. The indicated thermal performance of the bail and clamp thermal disconnect concept is generally inferior to that of the fiberglass support. Also, because of its mechanical complexity it is inherently less reliable, in addition to being heavier.

The performance of the two fiberglass cones is comparable over most of the range of tank sizes and mission duration although the honeycomb is most advantageous for the longest support length and mission duration in excess of 10^3 hr.

On the basis of these results, it is concluded that the fiberglass strut support concept offers the most promise for cryogenic tank supports within the sizes considered in this program.

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