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FINAL REPORT
CIRCUMFERENTIAL HEAT PIPE SYSTEMS
FOR LARGE STRUCTURES

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ABSTRACT

Application of heat pipe systems to manned spacecraft requires considerable development study, particularly in the areas of fabrication and testing of large scale heat pipe systems. This report describes a program for the design and fabrication of two full scale 50-foot circumferential heat pipes (one water and one ammonia) for test with the Subsystems Test Bed (STB) in the NASA/MSC Chamber A Thermal Vacuum Test Facility. Conventional heat pipe technology was applied to the problems of large scale systems, and manufacturing techniques (e.g., modularization) were developed to permit their fabrication. Thermal vacuum testing was directed toward an investigation of steady-state and transient response characteristics, and to explore potential problems associated with diffusion freezeout and start-up.

The major areas of accomplishment on this program were:

- The use of modularization concepts to fabricate large scale heat pipes was demonstrated.
- Two 50-foot circumferential systems, one with water as the working fluid and one with ammonia, were fabricated and operated as heat pipes in an ambient (laboratory) environment. Closure difficulties required removal of fluid from the ammonia heat pipe prior to thermal vacuum testing.
- Steady-state operation of the water heat pipe system under simulated space conditions was demonstrated.
- Valuable data were obtained on the diffusion freezeout phenomena of large heat pipes in a space environment. Start-up from a frozen state was successfully affected using auxiliary heaters.

Planned study of the use of an inert gas for frozen start-up of the ammonia heat pipe was not achieved due to failure of the ammonia pipe. Data collected for the water heat pipe system indicated the concept is feasible.

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1. INTRODUCTION

Heat pipe technology has progressed to a point of practical application to the thermal problems of unmanned spacecraft. This is amply demonstrated by a number of flight experiments (GEOS-B, ATS-E, OSO-PAK) and by upcoming plans for numerous other flight systems. Extension of this technology to manned spacecraft, on the other hand, will require considerable development study, particularly in terms of the physical size of the heat pipes involved. Potential programs which might benefit from heat pipes typically require units which are substantially longer than those for unmanned systems. While theoretical or phenomenological advances may not be necessary to accomplish this extension, it is obvious that demands on existing technology are more severe. Of particular concern are 1) the pumping requirements for the thermal loads and physical lengths involved; 2) the potential freezing problem during prolonged eclipse and the attendant problem of restart; and 3) the fabrication technology necessary to manufacture heat pipes of great lengths.

The primary objective of the program reported here was to design and fabricate two full scale 50-foot circumferential heat pipe systems for test with the Subsystem Test Bed (STB) in the NASA/MSC Chamber A Thermal Vacuum Test Facility. Specifically, the effort was directed to the following goals:

- Extend conventional heat pipe technology to the problems of large scale system applications,
- Develop manufacturing techniques (e.g., modularization) to permit fabrication of large systems,
- Investigate steady-state and transient dynamic response of large heat pipes under simulated spacecraft conditions, and
- Explore potential freezeout problems and the introduction of a noncondensable gas to effect start-up of very large heat pipes from a frozen state.

The units, shown in Figure 1-1 in Thermal Verification Testing, were designed to operate with water as the working fluid in one and ammonia as the working fluid in the other.



Figure 1-1. Photograph of the Heat Pipe System Under Test at TRW/Redondo Beach

The time schedule for this program was extremely short; three months were available for design, fabrication, preliminary testing and delivery of the units to NASA/MSC. Consequently it was necessary to short-cut many design/manufacturing activities without verification at the subscale level, a situation which undoubtedly contributed to numerous fabrication and testing difficulties. In particular, after experiencing several problems with the operation and finally closure of the ammonia heat pipe system, it was decided to remove the ammonia charge prior to shipment to NASA/MSC and replace with methanol. For the edification of the reader, the test data and rationale leading up to this as well as all major program decisions are documented in detail in this report.

2. HEAT PIPE SYSTEM DESIGN

The heat pipe systems designed and fabricated under this program must be considered developmental units in that they represented one of the first major attacks on the problems associated with large scale heat pipes. The design activities, therefore, were primarily directed toward the performance of the heat pipes themselves with proportionally less concern for auxiliary functions. There was, for example, no effort undertaken to build a "flight weight" system or to provide extensive quality assurance/quality control beyond that necessary in ordinary research and development programs. This program was also severely constrained by the schedule; only seven weeks were allocated for final design and fabrication efforts, and five weeks for processing, filling, and performance verification testing. Little time was available for resolving unanticipated developmental problems. It was necessary, therefore, to manufacture the systems as much as possible from existing technology and materials in order to satisfy these schedule commitments.

2.1 REQUIREMENTS

An overall diameter of 183 inches was selected for the heat pipe systems to coincide with the outer diameter of the STB. This resulted in a heat pipe 575 inches (48 feet) long. The diameter of the individual pipes was chosen to be approximately 1/2 inch as a reasonable tradeoff between weight, heat flux and fabrication restrictions. A support stand was also required to physically and thermally separate the heat pipe systems from each other and from the STB.

The design heat load for the heat pipe section was "one sun" (130 watts/ft²) perpendicular to the axis of hoop; a fin was attached to each pipe for absorption and rejection of thermal energy. In any real application, the system described would be one of a number of pipes spaced along the longitudinal axis of the spacecraft and joined by a thermally conducting fin. A fin was, therefore, necessary on each heat pipe system for realistic testing. However, since the primary concern of this program involved heat pipes, no attempt was made to establish an "optimum" weight fin geometry.

Since manned missions are of primary concern, an operating temperature of approximately 70°F was selected. For an isothermal cylinder in space sunlit on one side only, and neglecting earthshine and albedo, a simple energy balance gives

$$\sigma T^4 = \frac{S}{\pi} \frac{\alpha_s}{\epsilon} \quad (1)$$

where

T = absolute temperature, °R

S = solar constant, 442 Btu/hr-ft²

σ = Stephan Boltzmann constant, $.1713 \times 10^{-8}$ Btu/hr-ft² - °R⁴

α_s = solar absorptivity

ϵ = total hemispherical emissivity

This expression assumes that the 'hoop' is perfectly insulated on the internal surfaces.

Calculation shows that if

$$\alpha_s/\epsilon = 1, \sigma T^4 = \frac{S}{\pi} = \frac{442}{\pi} = 140.8 \text{ Btu/hr-ft}^2, T = 76^\circ\text{F}$$

Also, if

$$\alpha_s/\epsilon = 0.9, \sigma T^4 = 126.9, T = 62^\circ\text{F}$$

and if

$$\alpha_s/\epsilon = 1.1, \sigma T^4 = 154.8, T = 89^\circ\text{F}$$

Therefore, for α_s/ϵ of 1.0 ± 0.1 , the heat pipe system operating temperature would be bounded between 62 and 89°F, with a nominal value of 76°F.

The power requirement of the heat pipe systems can be determined from the following equation

$$Q_R = \epsilon A \sigma T^4 \quad (2)$$

where

Q_R = total radiated energy, Btu/hr

A = total heat rejection area, ft²

For a heat rejection area of 2300 in^2 (based on a four-inch wide fin which is 575 inches long) and $\epsilon = 0.26$

$$Q_R = 0.26 \times 2300 \times 0.287 = 172 \text{ watts}$$

Half of this energy is rejected directly from the sunlit side and the remaining 86 watts must be transported by the heat pipe to the shadow side. This requirement, discussed in greater detail in ensuing sections of this report, cannot be accomplished by utilizing conventional wicking systems. In order to transfer this power with a level heat pipe system, an arterial wick structure was necessary, coupled with modularization techniques.

2.2 DESIGN CONSIDERATIONS

2.2.1 General

The function of any heat pipe is to transport thermal energy from a source to a sink with a minimum of intervening thermal resistance. The design requirements for heat pipes on large structures can be illustrated by two examples:

- The heat pipe is made integral with the vehicle skin and is continuous. Such a pipe will equalize the circumferential temperature by transporting the energy absorbed on the sun side of the vehicle to the shaded side; e. g., effectively "spinning" the vehicle thermally. This application requires any section of the pipe to act both as a collector and dispenser of thermal energy depending on the vehicle orientation.
- Limited regions at each end of the pipe act as a source or sink with or without an intervening adiabatic section. This type of pipe would be used to connect a cold plate for electronic equipment to a radiator on the vehicle skin (the cold plate and radiator could also be heat pipes in a cascaded system).

In both of these examples an important aspect is the requirement for a heat pipe of unrestricted length. Once long heat pipes (over 40 feet) have become state-of-the-art, other refinements can be considered; e. g., cascading, constant temperature control, thermal switching, etc. The concern of this program was the first example above.

2.2.1.1 Modular Technique

The practicality of utilizing small heat pipes for functional spacecraft applications has already been established. However, the manufacturing techniques and processes used for these smaller devices did not readily lend themselves to large scale heat pipe structures of the type for use on manned space stations. Consequently, the problems associated with manufacturing, assembling and processing large scale heat pipe structures had not been fully explored. If large scale heat pipes are to be used with efficiency in the future, a modular approach to their manufacture must be developed. Once the problems associated with modular construction have been solved, there will be no limit to the size or geometry that can be manufactured. These "building blocks" can be used to customize a particular heat pipe application of any magnitude. Of course, as discussed later, there are practical limits to the hydrodynamic pumping distances which can be considered.

2.2.1.2 Arterial Design

If large heat pipes are to become a practical reality, the designs must depart from "conventional" type capillary structures. For distances much over a few feet, the viscous drag factor becomes larger than the available capillary pressure head. It is true that the viscous drag in the liquid can be reduced by increasing the pipe diameter and using thicker wicks. However, from a practical point of view, the thermal resistance between the source or sink through the liquid-wick interface may become excessive causing large temperature drops. Also, the system weight increases substantially. If the liquid viscous drag can be greatly decreased without adding to the temperature differential and weight, a practical heat pipe of high performance can be realized. One method for accomplishing this is known as the artery technique. Figure 2-1 shows four ways in which internal arteries can be introduced without greatly increasing the source-sink temperature differentials. The arrows denote the areas for absorbing and extracting the thermal energy.

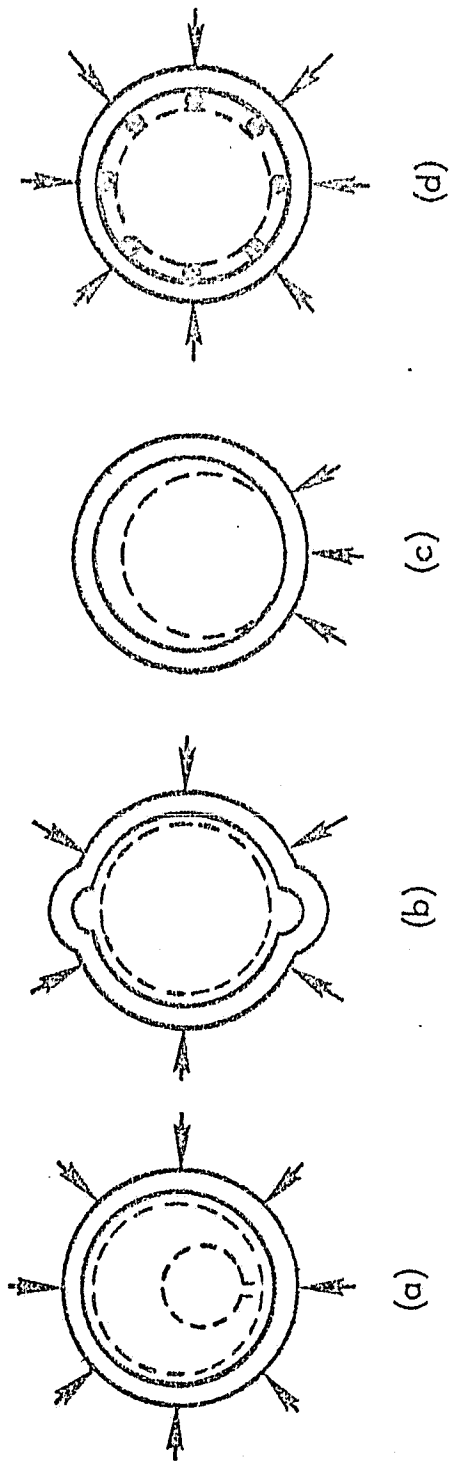


Figure 2-1. Schematic Drawing of Potential Arterial Geometries

In the first three methods, the arteries are located so as to minimize the thermal resistance through the fluid-wick combination. Type A, with a pedestal-mounted internal artery is an attractive approach in that it does not restrict the heat input zone. However, the state-of-the-art of heat pipe manufacturing at the onset of this program did not permit the joining of this type of artery in a modular manner because of its irregular cross section. Developing the necessary manufacturing skills during the course of the program was not possible because of the severe schedule and financial constraints. In type B, the wick is of cylindrical geometry and the arteries are formed by using an irregularly shaped tube. This, too, has disadvantages in that the heat pipe must be a pressure vessel. Thus, to maintain this geometry under pressure, the wall thickness must be increased, resulting in a weight penalty. The best pressure vessel is a simple cylinder, which suggests types C or D configuration. These are annular artery approaches which not only have an inherent artery by virtue of their geometry, but are excellent pressure vessels and, because of their purely cylindrical geometry, readily lend themselves to the joining of module interfaces. The configuration shown as type D has a number of manufacturing advantages, namely that fabrication and joining is significantly simplified over the other configurations and that careful control over the arterial thickness (δ) is possible through the use of spacer wires welded to the inside of the tube. The major disadvantage to type D is the relatively thick liquid layer through which heat input and heat rejection must take place. This drawback is particularly serious in the case of a high power density heat input where large ΔT 's across the liquid interface in the evaporator can cause boiling internal to the liquid artery with resultant failure of the pipe. Configuration D, however, is attractive for low power density applications such as for this program, provided the pipe/vapor ΔT in the evaporator can be maintained below that which would induce nucleate boiling. This approach was selected for the heat pipe designs on this program, the decision being heavily influenced by the time available for developing manufacturing techniques.

In all arterial heat pipes, the fine pores of the wick (e. g., fine mesh screen) provide the capillary pressure head required to pump the working fluid while the artery provides a low resistance longitudinal liquid flow path. Consequently, it is necessary that the artery be sealed over its entire envelope to prevent any openings or holes that are larger than the pores of the wick material. In other words, the available pumping head is determined by the largest opening in the wick. Careful control of manufacturing processes is required to insure that no openings larger than the wick pore size are allowed in the artery. This is of particular concern when joining the multiple heat pipe modules.

Care must also be exercised to assure that the spacing between the artery wick and the pipe wall is carefully controlled. If this spacing is not controlled, 1) the thermal resistance increases over portions of the tube and, under high enough loads, boiling can occur within the artery and 2) the pipe may not prime (as discussed later). Boiling in the artery not only leads to unstable heat pipe operation, but can ultimately cause a cessation of operation if the liquid return path is interrupted by the associated vapor lock.

Another aspect of arterial heat pipe design is the need for testing under one-g conditions. It is true that the artery, once filled with liquid, will remain full under expulsive forces (e. g., gravity, vibration, etc.) up to the maximum capillary head producible by the wick. However, should it be partially or fully depleted of liquid, it is necessary to design for automatic refill; i. e., the artery should be self-priming. This is a trivial matter under zero-g conditions, but it requires considerable attention in a one-g field.

2.2.1.3 Steady State Operation

From a steady state operational point of view, there are two principal effects of large scale heat pipes. First, the increased flow length causes significant pressure drops in both the liquid and vapor flow, which act to decrease the maximum heat transfer capacity of the pipe. Second, the increased pressure loss in the vapor transport phase manifests itself as an increase in the overall heat pipe temperature drop. The arterial wick structures previously discussed decrease the liquid flow losses

considerably. Such wicking systems have been successfully proven in the past. The fabrication of such wicks in very long heat pipes of modular type construction did, however, represent a major extension of previous technology.

The second factor, the tendency to increase the vapor pressure loss, can be overcome by either increasing the vapor flow area (i.e., pipe diameter) or by choosing a working fluid such that the slope of the vapor pressure curve (dp/dt) is very large at the desired operating temperature. From a system weight point of view, the latter is the preferable approach and was selected for this program.

2.2.1.4 Transient Operation and Start Up from a Frozen State

Transient operation of heat pipes poses entirely different problems than those related to steady state. One of the more serious of these is the ability of the heat pipe to restart from a frozen state. The danger in such a startup is that the evaporator section can become depleted of working fluid before the fluid in the condenser is completely melted. Fluid cannot return to the evaporator, leading to a dryout failure mode. This problem is further aggravated as the heat pipe length is increased because of the need to melt larger quantities of frozen working fluid. One approach to solving this problem, which has met with some success, is to introduce a quantity of noncondensable gas into the pipe to establish minimum pressure level. The presence of this gas causes a diffusion barrier to oppose the vapor coming from the evaporator during startup, causing it to condense in close proximity to the heat source. As the evaporator temperature rises, the vapor pressure of the fluid increases and compresses the noncondensable gas, resulting in a continually increasing "effective" heat pipe length. At operating temperature (and pressure), this gas should be compressed into a very small portion of the pipe, causing a minor loss of performance. Although this technique has been demonstrated on small heat pipes (15 feet and less), there was a major concern as to whether it would work successfully on the heat pipe systems developed on this program. First of all, to start very large pipes such as these, the amount of noncondensable gas becomes appreciable. Theoretically, the gas inventory must be such that, under frozen conditions, its partial pressure is greater than the vapor pressure of the working fluid at its melting point. This

requirement suggests the choice of a working fluid such as ammonia which has a very high ratio of vapor pressure between operational conditions (≈ 128 psia at 70°F) and its melting point (≈ 0.9 psia at -108°F) in order that the gas be compressed to an acceptably small volume when the pipe is under load.

Aside from some loss of condenser area, a large gas blocked zone can lead to a heat pipe failure mode which might be called "diffusion freezeout." Vapor continually diffuses into the noncondensable gas blocked region and condenses. Normally, the condensate is simply pumped back to the active zone by the wick and steady-state operation continues. If, however, part of the gas blocked zone falls below the freezing point of the fluid, the vapor diffusing into this region freezes and is lost to the system. Eventually, this process can deplete the evaporator of working fluid and the heat pipe fails. This phenomenon will be discussed in more detail in Section 4.4.2.2. By minimizing the ultimate gas blocked length, it is possible to avoid subfreezing temperatures and thus obviate the diffusion freeze-out problem.

Ammonia, as will be shown in Section 2.2.3, is a marginal fluid (from a hydrodynamic standpoint) for the type of artery system chosen. Secondly, ammonia, which has a relatively low thermal conductivity as well as a very low allowable superheat temperature is particularly susceptible to incipient boiling.

In a conventional heat pipe wick consisting entirely of a mesh or sintered type capillary structure, boiling can be tolerated to some extent. As the bubbles find their way to the surface of the wick and escape, they are replaced by new liquid through the capillary action of the wick. This is also true for the pedestal type artery mentioned previously. However, in the concentric type artery employed on this program, vapor generated by boiling fills the artery in preference to escaping through the wetted wick, resulting in a total vapor blockage of the heated zone.

Theoretically, in a perfectly pure system containing noncondensable gases or contaminants, a fluid can boil only by means of heterogeneous nucleation resulting from molecular agglomeration. Such nucleation, for most fluids, requires substantial superheat temperatures (on the order of

hundreds of degrees for some fluids). However, in the presence of non-condensable gases, the gas present in small cavities on the heated surface acts as a seed to promote what is known as "nucleation from a pre-existing gaseous phase." The presence of this gaseous 'seed' acts to sharply reduce the amount of superheat that a fluid can sustain without nucleating. The following expression approximates the critical superheat for incipient boiling if noncondensable gases are present in a heat pipe

$$^* \Delta T_{\text{crit}} = \frac{2 T_{\text{sat}} \sigma}{J \lambda \rho_v} \left[\frac{1}{r_n} - \frac{\cos \theta}{r_c} \right]$$

where

ΔT_{crit} = superheat temperature to induce boiling by gaseous nucleation, °F

T_{sat} = vapor saturation temperature, °F

σ = liquid surface tension at T_{sat} , lb/ft

J = universal conversion constant = 778 ft-lb/Btu

λ = latent heat of vaporization at T_{sat} , Btu/lb

ρ_v = vapor density at T_{sat} , lb/ft³

r_n = nucleus bubble radius, ft

θ = fluid wetting angle

r_c = capillary radius, ft

This equation states that, in the presence of noncondensable gas, a bubble nucleus of radius r_n , present in an artery constrained by a capillary radius r_c , will grow when the liquid surrounding it is superheated by the amount ΔT_{crit} . This is the onset of boiling, and will occur first at the region of highest superheat; in the present application, the inside surface of the wall.

For the concentric type artery used on this program, ΔT_{crit} was calculated to be 0.04 to 0.4 °F for ammonia over a typical value range of $r_n = 1 \times 10^{-3}$ to 10^{-4} inches, respectively, at $T_{\text{sat}} = 70^\circ\text{F}$. This range

* B. D. Marcus, "On the Operation of Heat Pipes," TRW Independent Research Report No. 9895-6001-TU-000, May 1965.

of superheat temperature is very low and was the basis for concern as to whether the noncondensable gas method of starting from a frozen state was applicable with this particular type artery.

2.2.1.5 Fluid/Material Selection

For heat pipe operation at temperatures near ambient, there are three fluids to be considered as good heat pipe fluids. They are: water, ammonia, and methanol. From a hydrodynamic liquid transport standpoint, water is best, followed by ammonia, with methanol a distant third. Although water has a higher liquid transport factor and thermal conductivity than ammonia, it does have some drawbacks. First of all, at 70°F, water has a vapor pressure of only 0.363 psia. Secondly, at this temperature level, the slope of the vapor pressure curve (dp/dt) is relatively small, meaning appreciable temperature losses are associated with very small pressure drops. Furthermore, at this low pressure, even a small quantity of noncondensable gas can occupy and effectively shut off a significant length of the condenser. For this reason, the noncondensable gas technique for restarting frozen water heat pipes was considered inapplicable.

On the other hand, ammonia has a very high vapor pressure in this temperature range (128 psia at 70°F). At this pressure, the quantities of noncondensable gas required for startup occupy a relatively small volume at the operating temperature of the pipe. However, as was pointed out in Section 2.1.1.4, the presence of this gas might cause nucleation with the type of artery being used. Another serious drawback with ammonia is its hazardous nature. Methanol was then considered, since its properties are a compromise of those of ammonia and water with respect to the problems just mentioned; however, its liquid transport factor is substantially below water or ammonia.

It was, therefore, decided to fabricate two heat pipe systems, one using water and the other using ammonia, to study the merits of each fluid with respect to the other. Since the program was not oriented toward fluid studies, it was further decided to keep as many things as possible similar between the two pipes. Of the materials known to be compatible with both ammonia and water, the 300 series stainless steels are considered best. Until recently, the stainless steels have presented a compatibility problem with water, due to the generation of hydrogen. This problem can be mitigated, however, by the use of selective passivation techniques.

2.2.2 Water Heat Pipe System

Figure 2-2 shows a cross section of the heat pipe-fin assembly. The basic heat pipe concept is that shown as type D in Figure 2-1. There are six spacer wires located between the wick and the heat pipe wall to form a low viscous drag fluid artery. The initial design called for 1-1/4 turns of 200 mesh screen to provide the capillary structure. The screen was held against the spacer wires by an internal spring.

As was mentioned previously, this type of artery must be self-priming. This means that careful control over the gap width δ must be maintained. Too small of a gap width results in a liquid pressure drop too high to permit adequate return of the fluid at the power levels required. Too large of a gap means the artery would not prime with liquid and the pipe would not function at any power level. Another crucial aspect of this design is that the maximum capillary head will be limited by the largest opening that exists between the vapor space and the liquid artery.

For a perfectly constructed system this opening corresponds to the holes in a 200 mesh screen. If any larger openings exist in the artery, the entire pipe may fail when that section becomes an evaporator. Vacuum chamber tests call for every point along the heat pipe to become an evaporator at one time or another.

With respect to the first concern, self-priming, it is clear that an optimization had to be made with respect to the arterial gap width δ . In order for the artery to self-prime, the following expression must be satisfied for a horizontal pipe

$$\delta \leq \frac{\gamma \cos \theta}{\rho_l R g / g_o} \quad (4)$$

where

γ = liquid surface tension, lb/ft

θ = wetting angle

ρ_l = liquid density, lb/ft³

R = radius of heat pipe wall, ft

g = acting gravity field, ft/sec²

g_o = earth's gravity = 32.2 ft/sec²

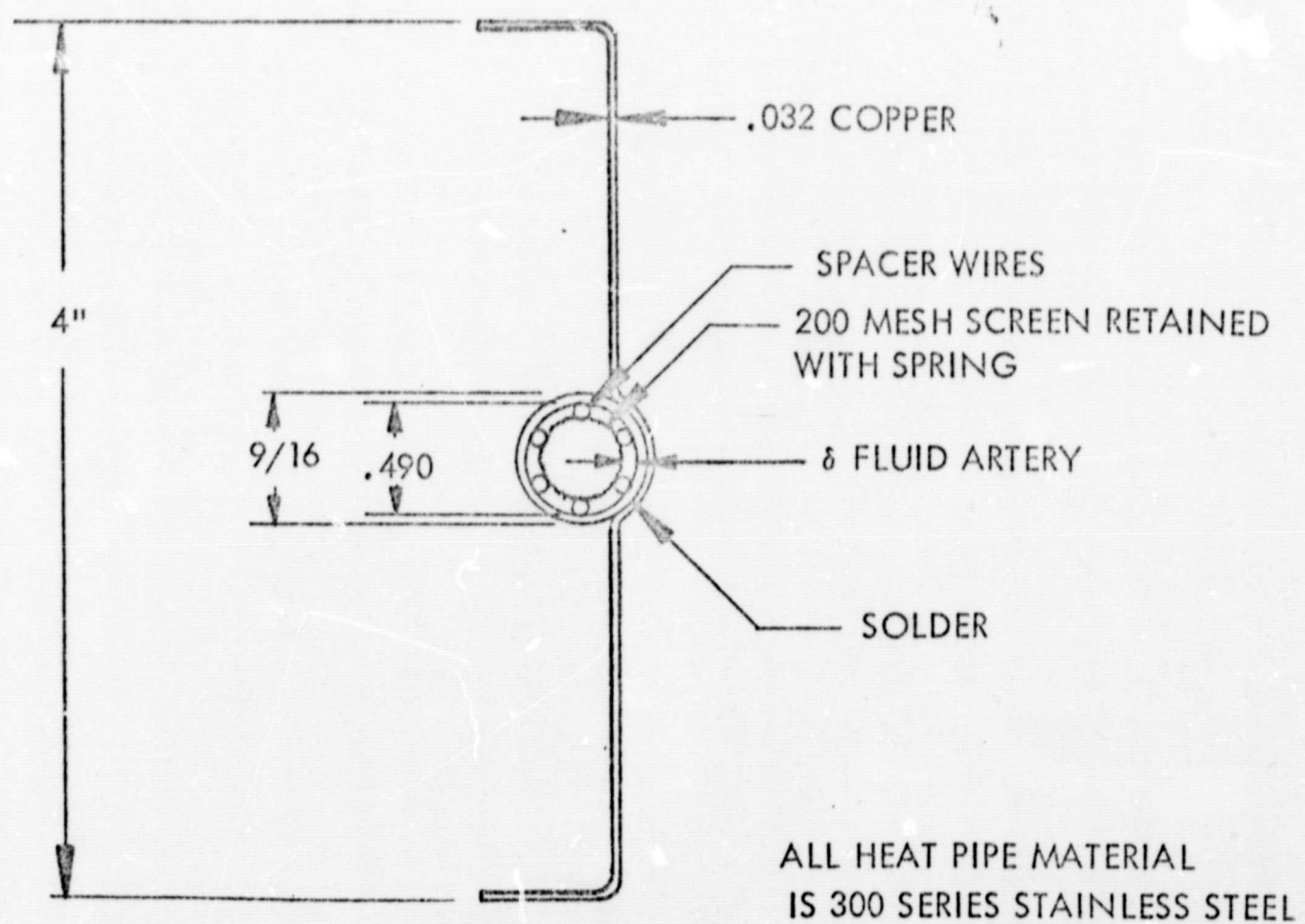


Figure 2-2. Cross-Sectional View of Fin/Tube/Wick Configuration

For earth-bound tests $g/g_0 = 1$ and the equation reduces to

$$\delta \leq \frac{\gamma \cos \theta}{\rho_f R} \quad (5)$$

For water at 76°F, and assuming a conservative wetting angle of 45 degrees

$$\delta \leq \frac{4.93 \times 10^{-3} \times 0.707 \times 144}{62.4 \times 0.25} = 0.032 \text{ inch}$$

For a safety factor of approximately 2

$$\delta = 0.015(R_{\max} = 0.5) \text{ inch}$$

With a gap thickness of 0.015 inch the water heat pipe would have to be kept flat within 0.25 inch ($R_{\max} - R_{\text{tube}}$) in order to be self-priming. The hydrodynamic characteristics of the system were then checked for this gap thickness to insure that an adequate factor of safety was available with respect to the liquid and vapor pressure drops versus the available capillary head. Figure 2-3 presents the performance curves showing the vapor pressure drop, liquid pressure drop, and available capillary head, versus the angle ϕ proceeding away from the sun normal point n for water at 76°F. Note that the summation of all the losses is well within the available capillary head and a theoretical factor of safety of 4.04 exists for this design at 76°F. At a calculated temperature of 104°F (the pipe operates at this temperature during laboratory testing), the pipe has an even higher factor of safety (see Figure 2-4).

2.2.3 Ammonia Heat Pipe System

For ammonia at 76°F and an assumed wetting angle of zero degrees

$$\delta \leq \frac{1.445 \times 10^{-3} \times 1.0 \times 144}{38.0 \times 0.25} = 0.022 \text{ inch}$$

For a safety factor of approximately 2, $\delta = 0.012$ inch was selected which gives a value of $R_{\max}$ of

$$R_{\max} = \frac{1.445 \times 10^{-3} \times 1.0 \times 144}{38.0 \times 0.012} = 0.45 \text{ inch}$$

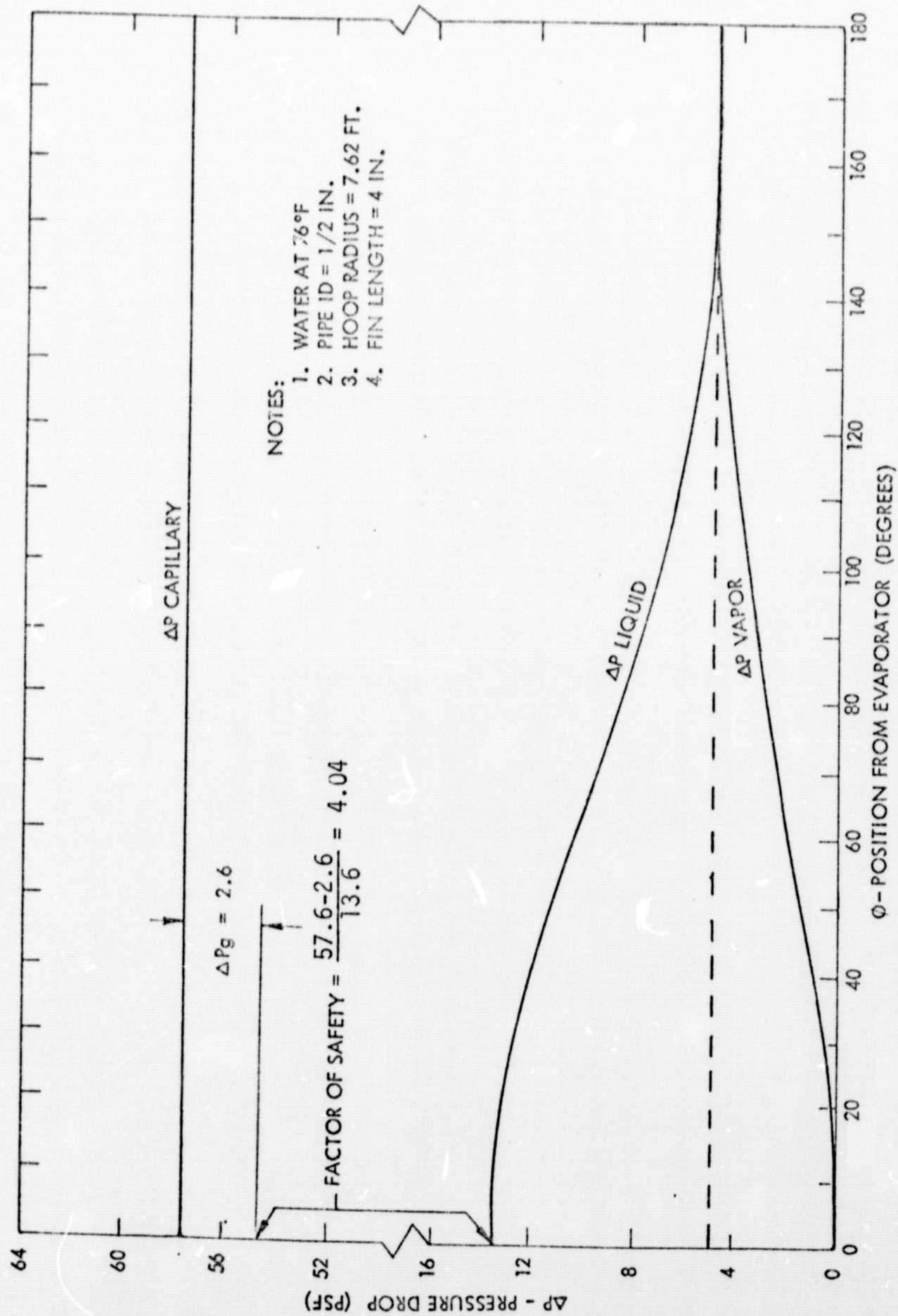


Figure 2-3. Hydrodynamic Pressure Losses - Water Heat Pipe System at 76°F

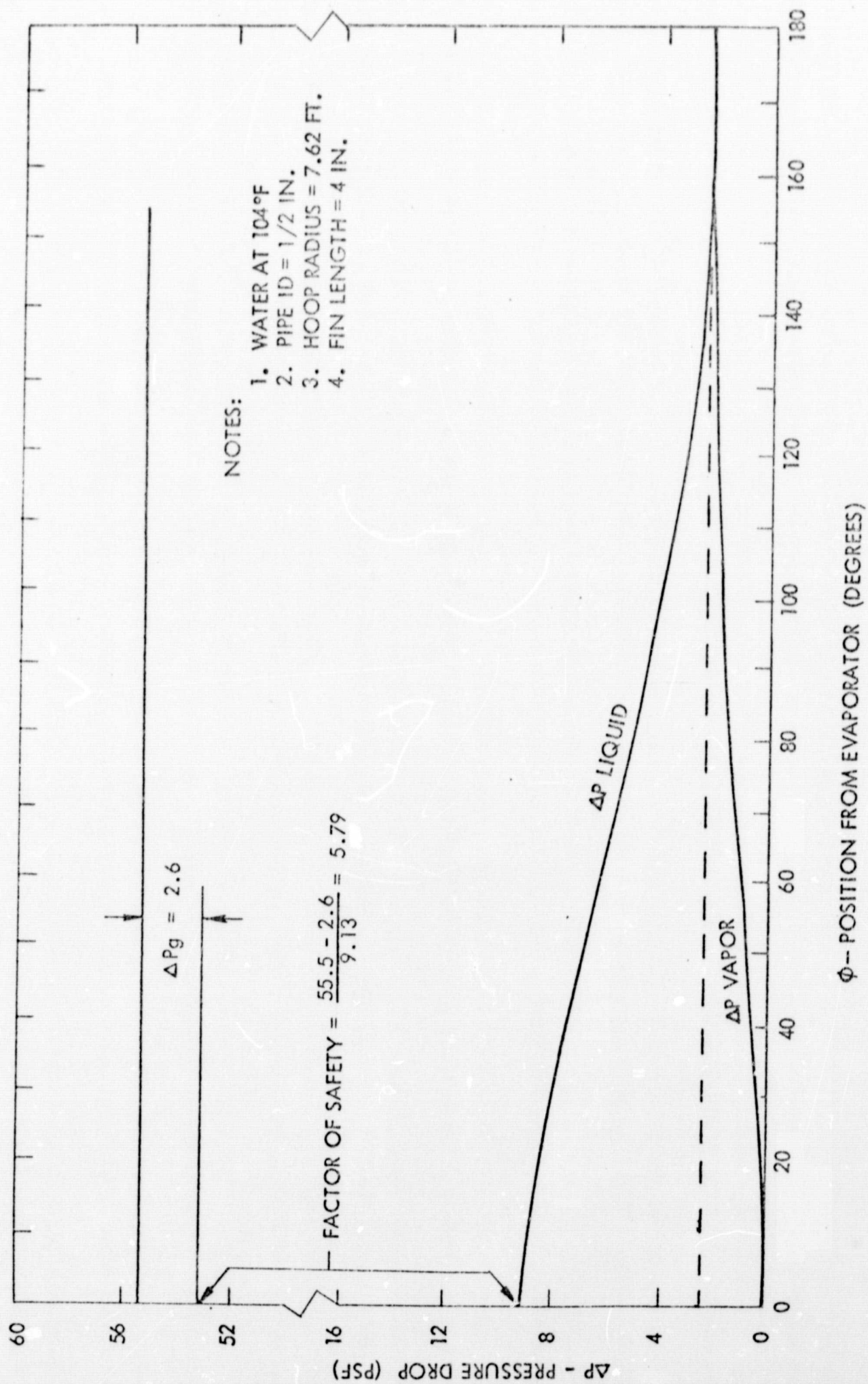


Figure 2-4. Hydrodynamic Pressure Losses - Water Heat Pipe System at 104°F

With this gap thickness the ammonia pipe must be kept flat to ± 0.20 inch ($0.45 - 0.25$); the associate factor of safety for self-priming is $0.45/0.25 = 1.8$.

A summary of the pressure drops for a gap thickness of 0.012 inch is shown in Figure 2-5. The hydrodynamic factor of safety is only 1.5 at 76°F. At a calculated temperature of 104°F, Figure 2-6 shown an even lower hydrodynamic factor of safety 1.15. In view of the possible manufacturing variations, it must be concluded that the design of the ammonia pipe was marginal. Fabrication had to be carefully controlled if this pipe was to prime and to operate at full load. Although the ammonia pipe design was indeed marginal, it must be recognized that there was not much opportunity to improve it with the arterial wick used. Increasing the pipe diameter would require a smaller gap width for self-priming, and this would lower the hydrodynamic capacity. Decreasing the pipe diameter to gain capacity with larger gaps would increase the ΔT into the evaporator and aggravate an already worrisome nucleation problem.

As noted in Section 3, it was necessary to fabricate the ammonia heat pipe system first while the passivation schedules for the water heat pipe system were being developed. It should be stated that there was substantially more confidence in the manufacture of the water system because of techniques learned in the manufacture of the ammonia system.

2.2.4 Design Summary

In summary, the water pipe design had a substantial theoretical hydrodynamic factor of safety for operation at steady state to absorb potential variations in the manufactured system. However, the results of the design study indicated that the water pipe would, in all probability, not restart from a frozen state.

The ammonia pipe design, which could potentially start from the frozen state using inert gas, had the following serious drawbacks:

- Marginal Hydrodynamic Design — Manufacturing variables that might occur could render the pipe inoperable at full load; or in fact cause the artery not to prime in a one-g environment.
- Nucleation — Introduction of the noncondensable gas to aid startup might cause nucleation in the artery with resultant failure at very low power levels.

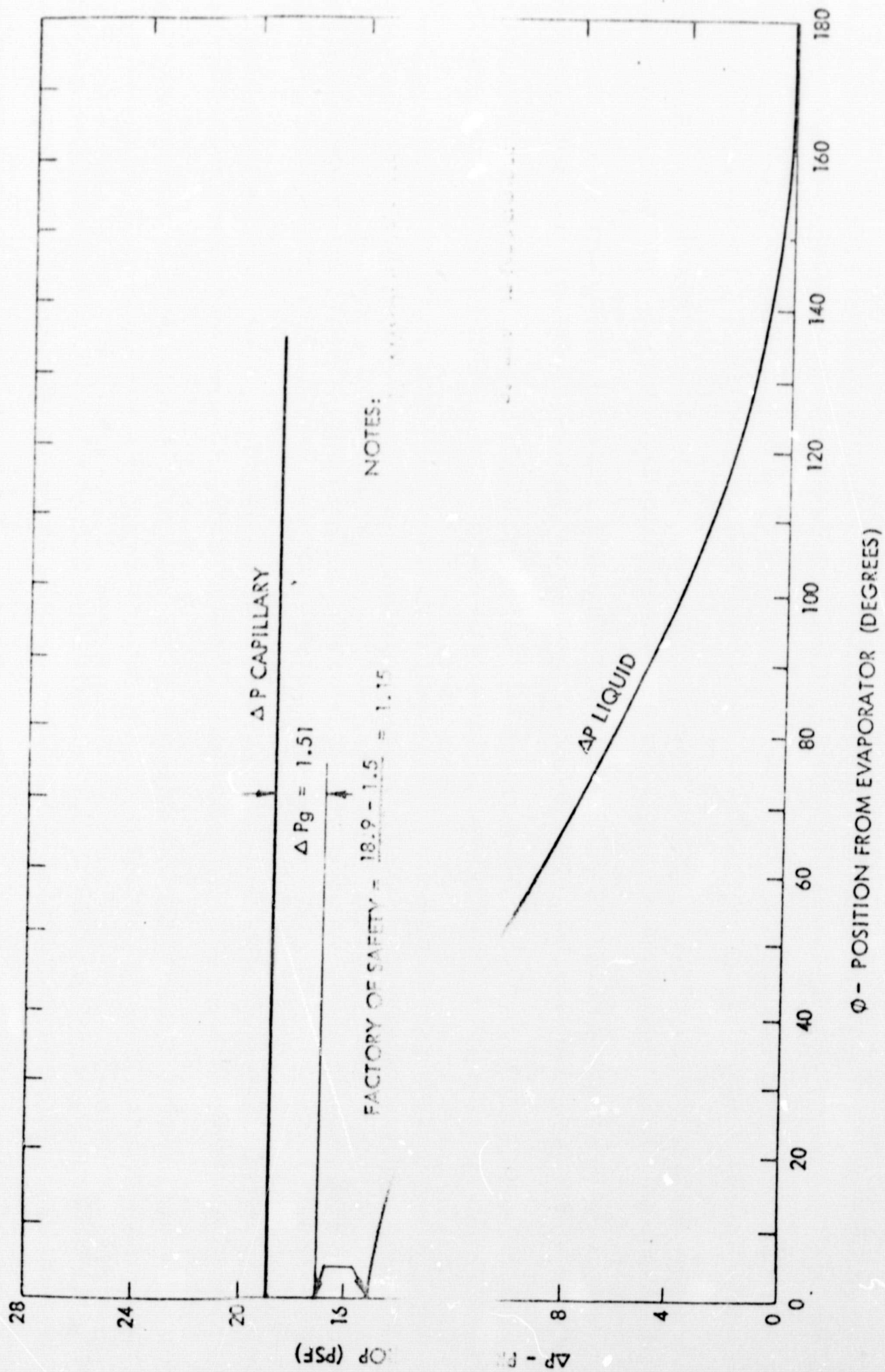


Figure 2-6. Hydrodynamic Pressure Losses - Ammonia Heat Pipe System at 104°F

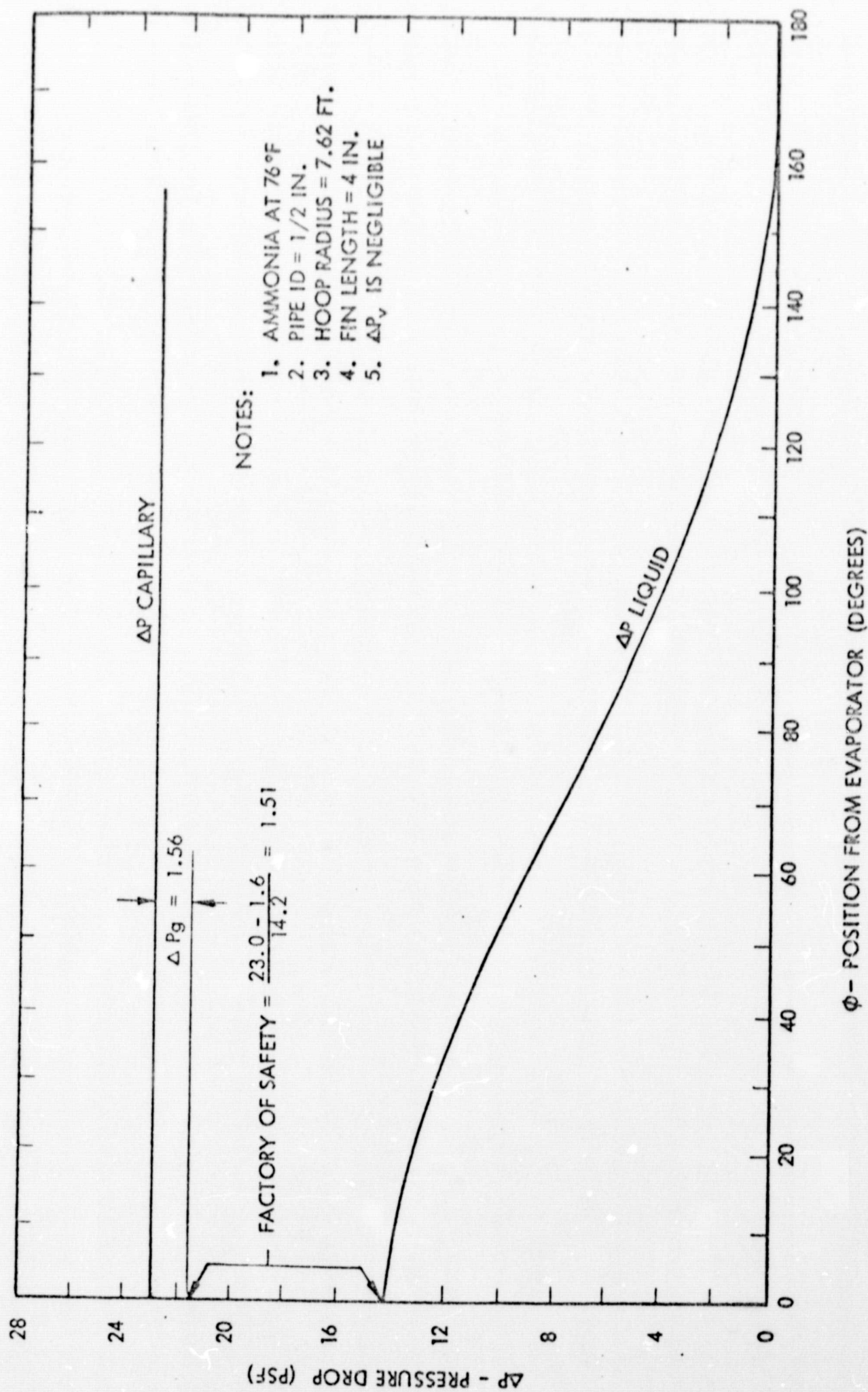


Figure 2-5. Hydrodynamic Pressure Losses - Ammonia Heat Pipe System at 76°F

2.3 DEVELOPMENTAL SUBSCALE TESTING

Developmental subscale testing was severely constrained by schedule commitments. However, some subscale testing was deemed essential to verify the design concepts presented in the last section and to evaluate manufacturing techniques for forming the artery and for fabrication of the modular joints. The following paragraphs delineate these tests and present the negative as well as positive results of this activity.

2.3.1 Wick Integrity Tests

A basic requirement for the heat pipe system design selected is that the artery be completely sealed with respect to the vapor core. It was necessary, therefore, to determine if the manufacturing methods and techniques employed satisfactorily met this requirement.

The initial design specified 1-1/4 turns of 200 mesh stainless steel screen held firmly in place against the wall by an internal spring. Six longitudinal spacer wires were spot welded to the mesh prior to its insertion and served to space the screen away from the wall. A 5-foot module of this type was fabricated and tests performed to determine the integrity of the wicking system. One end of the artery was sealed with an 'O' ring gasket, and the module then filled with ethanol from the sealed end. Ethanol was used instead of water for this test because the module had not been passivated to promote good wetting with water (ethanol has essentially a zero-degree wetting angle with unpassivated stainless steel. Care was taken to insure that the test module was kept flat well within the requirements for self-priming. The gap thickness for the test module was the 0.015 inch chosen for the water system.

This test established that the 0.015-inch gap would self-prime (liquid was observed at the unsealed end). However, the artery would not hold more than 1/2-inch elevation of the sealed end. Theoretically, with ethanol, a 200 mesh screen is capable of a height of 6-1/2 inches before the artery empties. It was clear in the test that either the welds (wires to screen) created holes in the artery system or that the spring did not effectively seal the overlap of the screen wire.

The spring, mesh, and wires were then removed from the module and a new mesh inserted. This mesh consisted of two continuous wraps

(actually 2-1/4) and the spacer wires were welded directly to the tube. It was felt that this approach would solve both of the aforementioned problems. The artery was then sealed in the same manner as before and the test repeated. This configuration could be raised to a height of four inches before the liquid column in the artery broke. This height, when converted to an equivalent water value, was well within the hydrodynamic requirement for the water system. The fact that the full height of 6-1/4 inches could not be achieved was attributed in part to the rubber 'O' ring, which it was felt was not completely sealing the artery.

The module was then placed on a forming platform and bent on a 15-foot diameter. The artery was refilled with ethanol and the lift test repeated. This time the liquid column in the artery broke at 4-1/4 inches, approximately the same as before within testing tolerances.

2.3.2 Water Heat Pipe Module

After successful completion of the wick integrity test, the 5-foot test module was converted into a water heat pipe. First, both ends of the artery were sealed by trimming back the wire spacers for about 1/2 inch from the end and spot welding the 200 mesh screen against the inside wall of the pipe. After the entire module had been chemically passivated, a wick integrity test was performed using triple distilled water. Results of this test indicated that the sealed artery system could hold a total gravity head of 11-3/4 inches. The maximum theoretically obtainable with 200 mesh screen and water is 15-1/2 inches, assuming perfect wetting and sealing. More typically, a wetting angle of 45° with water would yield a maximum height of 11 inches. The measured 11-3/4 inches showed that the artery seal was successful. The pipe was then provided with end caps and fill tube, and processed into a water heat pipe.

After processing and final closure, the pipe was tested for thermal and hydrodynamic performance. Under a 'scaled' thermal load, the heat pipe had a total temperature differential of 5°F (evaporator surface to condenser surface), while transferring a total of 32 watts at $\approx 176^\circ\text{F}$. This power input was distributed over the entire circumference of the first foot of the heat pipe, and corresponded to the same maximum power density the pipe would receive in the final system configuration. To scale

the hydrodynamic load the pipe was elevated at the evaporator end. Under a 32-watt load at 176°F the pipe could be elevated to a height of 10 inches before the artery failed. This condition represents approximately 100 percent of theoretical capacity.

These limited subscale tests substantiated the validity of the water heat pipe wick design both from the heat transfer (acceptable ΔT , no boiling) and the hydrodynamic point of view.

2.3.3 Ammonia Heat Pipe Module

Original plans called for a 10-foot section with modular artery joint to be processed into an ammonia heat pipe. The 10-foot section, however, was fabricated using the initial wicking system of one wrap of mesh with spacer wires spotted to it and did not pass the wick integrity tests. Because of schedule, it was then decided to convert the 5-foot water heat pipe into an ammonia pipe. The 5-foot module was drained of its water charge and vacuum baked for 48 hours under high vacuum. The pipe was filled with the appropriate amount of ammonia and sealed off. Subsequent thermal testing showed that, when filled with ammonia, the pipe operated successfully in the heat pipe mode (1/2 inch elevation) with power inputs exceeding 50 watts. These were rather severe test conditions since the arterial gap width was larger than specifications (0.015" versus 0.012") and the input heat flux was higher than maximum design conditions (e.g., 32-watt scaled load). Both of these factors increase the temperature drop into the evaporator and aggravate the potential nucleation problem. In spite of this, the pipe operated successfully with an input ΔT of 7°F. When the input power was increased to 70 watts, nucleation in the artery caused failure. Thus, it was shown that the ammonia heat pipe, in the absence of noncondensable gases, could sustain superheats of up to 7°F, which exceeded requirements. Of course, as discussed in Section 2.2.1.4, introduction of noncondensable gases was expected to considerably alter this result.

The pipe was then bent on a 15-foot diameter and the same tests repeated with no significant change in the results. At this point the designs of both pipes were frozen and full manufacturing efforts were exerted toward the fabrication of the two deliverable systems. The design freeze

decision was made because of schedule commitments even though no successful tests had been conducted for the modular joint configuration.

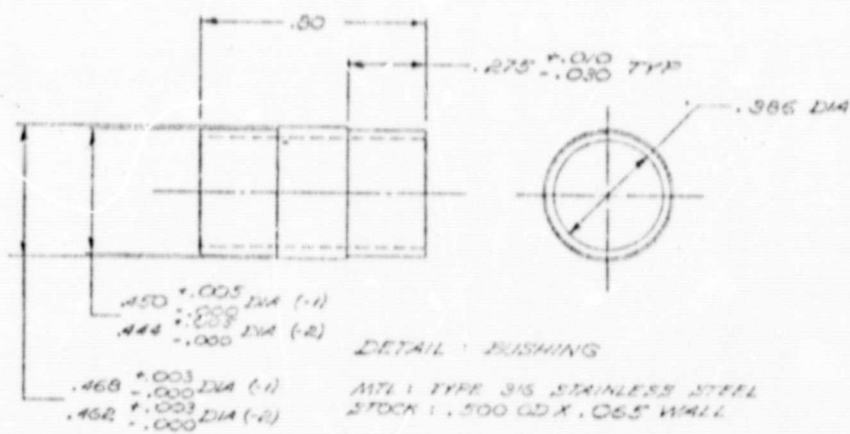
It is interesting to note that, well into the fabrication phase of this program, an additional potential failure mechanism was identified for gas loaded arterial heat pipes in general. It was hypothesized that the non-condensable gas might dissolve into the liquid at the cold, gas-blocked zone and be carried by the fluid to the evaporator. The gas should partially come out of solution in the evaporator due to the higher fluid temperature (gas solubility decreases with increasing temperature). Since it could not be expelled because of the wetted wick, the gas might cause blockage and failure. The only apparent mechanism for escape was by diffusion through the wetted wick as a consequence of the partial pressure difference which would exist across it. The question of potential failure was thus one of the relative rates of accumulation by virtue of solubility changes and removal by diffusion.

To explore this question, the same 5-foot module was reprocessed with water and a noncondensable gas charge (nitrogen) and placed on life test. Since it operated without failure for over six months before the test was terminated, it can be concluded that the removal rate by diffusion was sufficient to avoid gas accumulation in the artery.

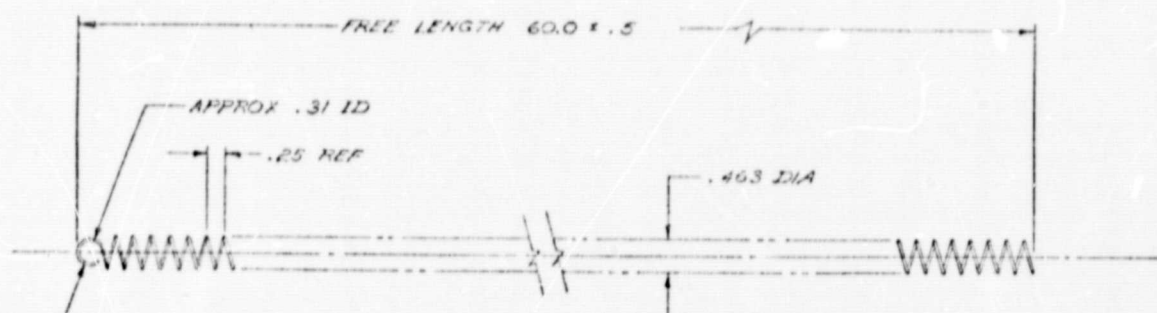
2.4 FINAL DESIGN

2.4.1 General

Each of the two 183-inch diameter systems are made up to ten equal length sections; i. e., modules. The manufacturing working drawing for the units is presented as Figure 2-7. A standard "module" is approximately 5-feet long (57-1/2 inch precisely) and subsequent reference to 5-foot or 10-foot sections will mean one or two standard "modules", respectively. The fabrication techniques associated with the 57-1/2 inch long modules are applicable to any length up to about 7 feet so the discussion in this section can be extended, without complication, to heat pipe modules in this range. As noted previously, designs are presented for water and for ammonia as the working fluid. General design information is presented in this paragraph; specific data on the individual pipes are presented in Paragraphs 2.4.2 and 2.4.3.

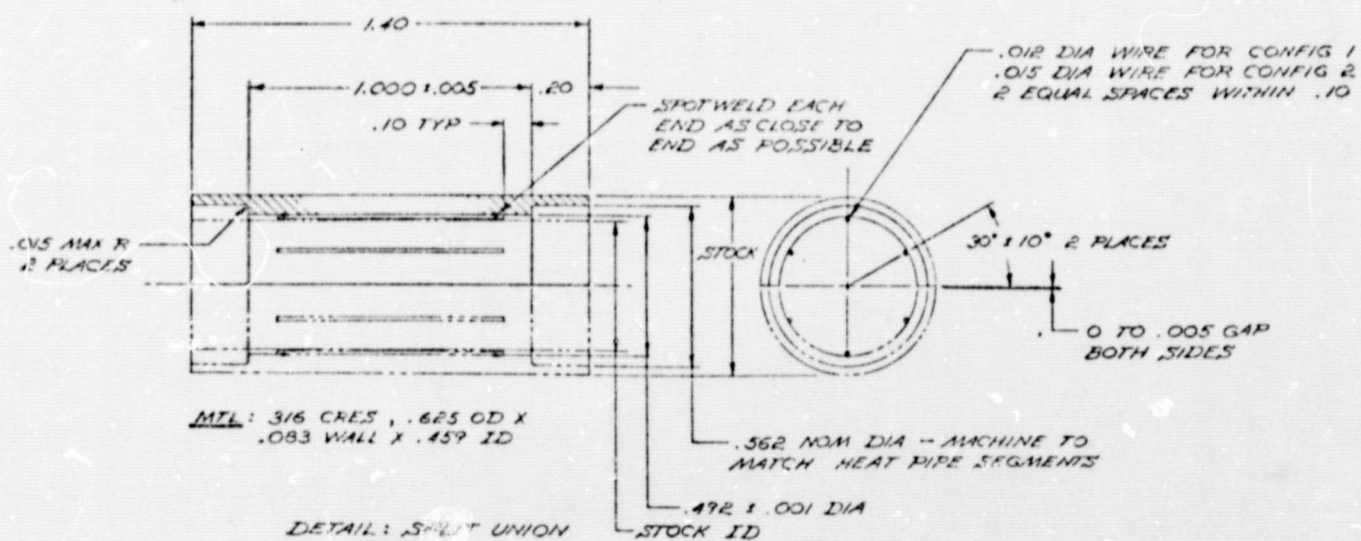


SCALE $\frac{4}{1}$



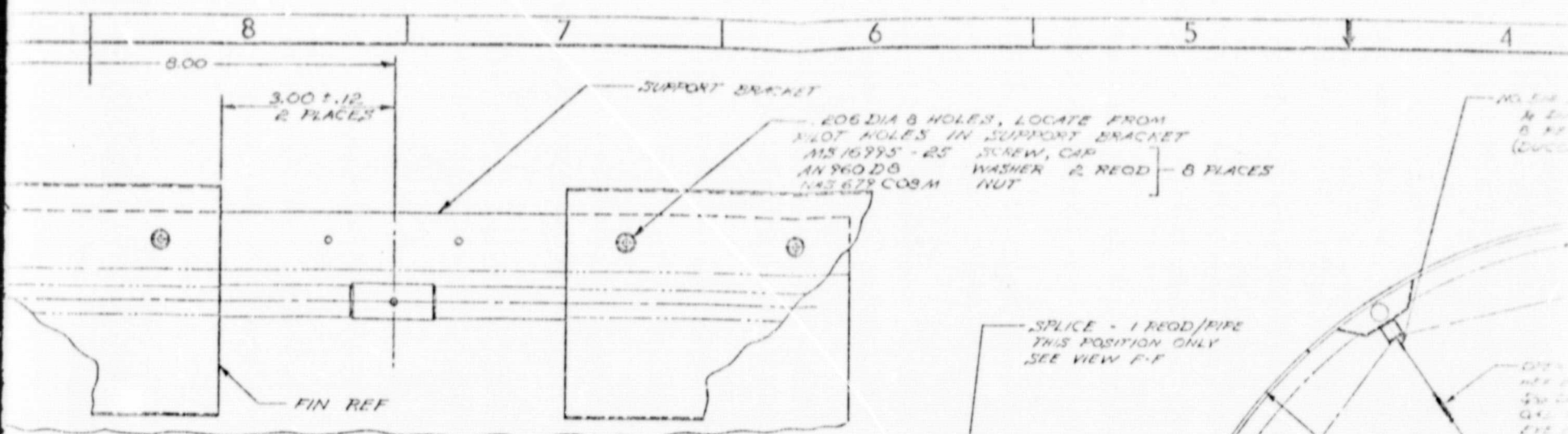
SPRING DATA

MTL	.0475 DIA 316 CRES WIRE
MEAN DIA	.415 ± .007
TOT'L NO. COILS	240 ± 10
INITIAL TENSION	0 LB
SPRING RATE	.39 ± .02 LB/IN
PITCH	.25
MAX EXTENDED LENGTH	87.00

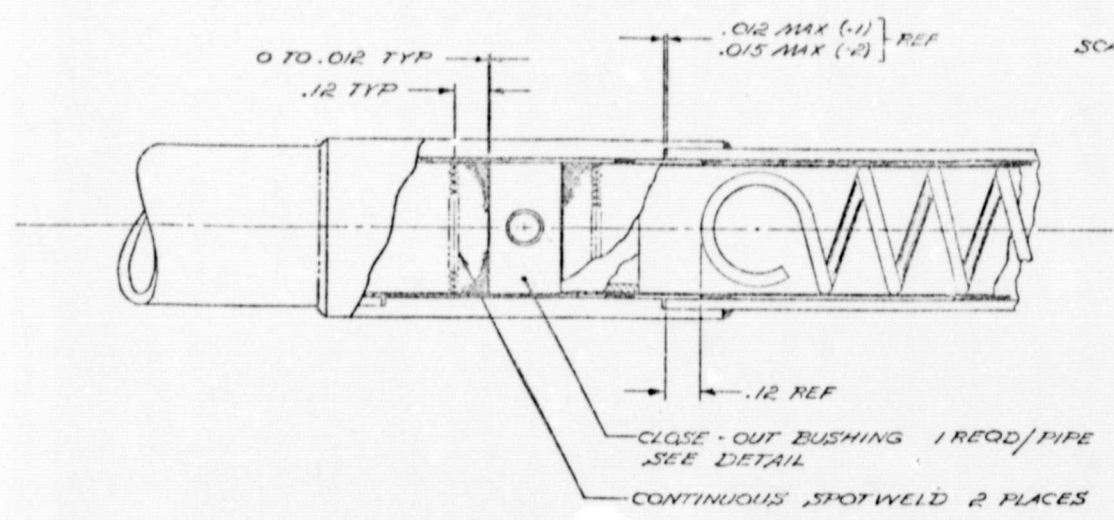


SCALE $\frac{4}{1}$

FOLDOUT FRAME

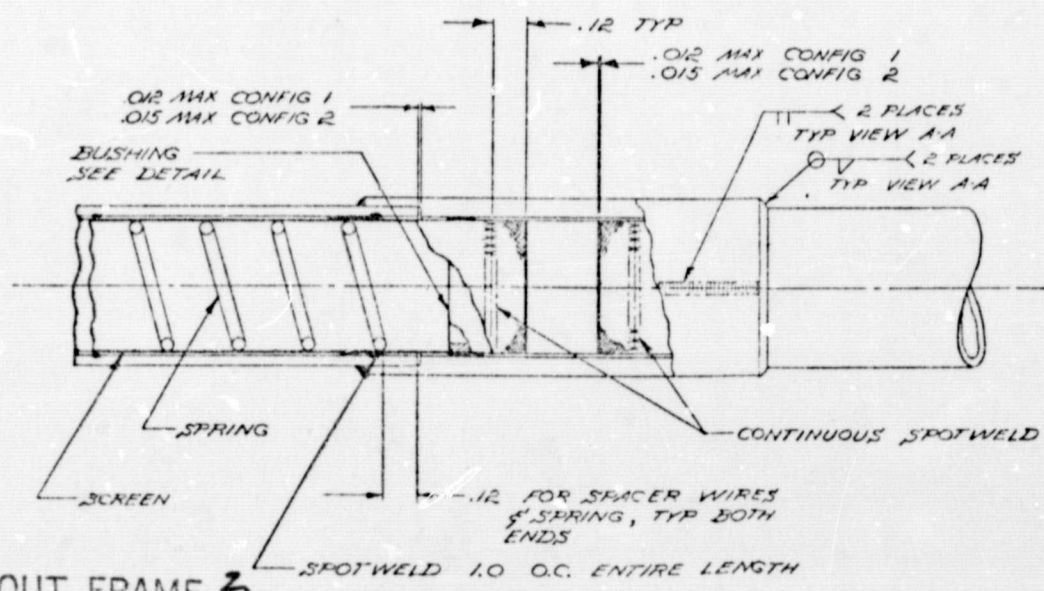


VIEW B-B
SCALE $\frac{1}{1}$



VIEW A-A
SCALE $\frac{4}{1}$

SHORT REED/PIPE
NO. 102 ANNEALED
SEE TYP QQ-C-576



FOLDOUT FRAME 3

DETAIL C SCALE $\frac{4}{1}$
TYP 9 PLACES/PIPE

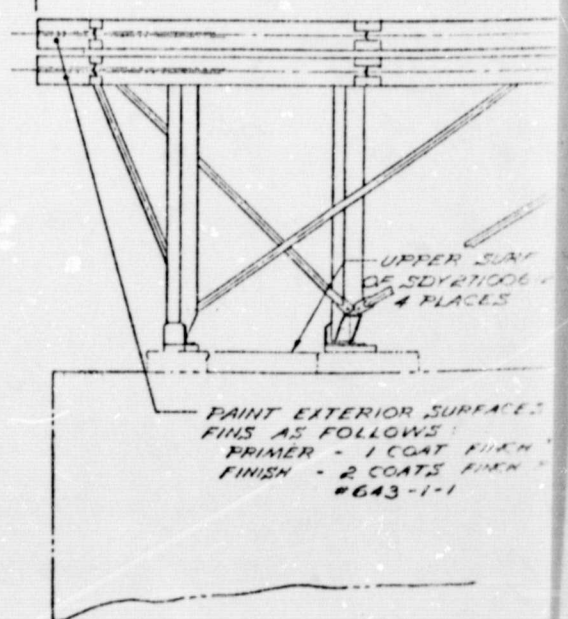
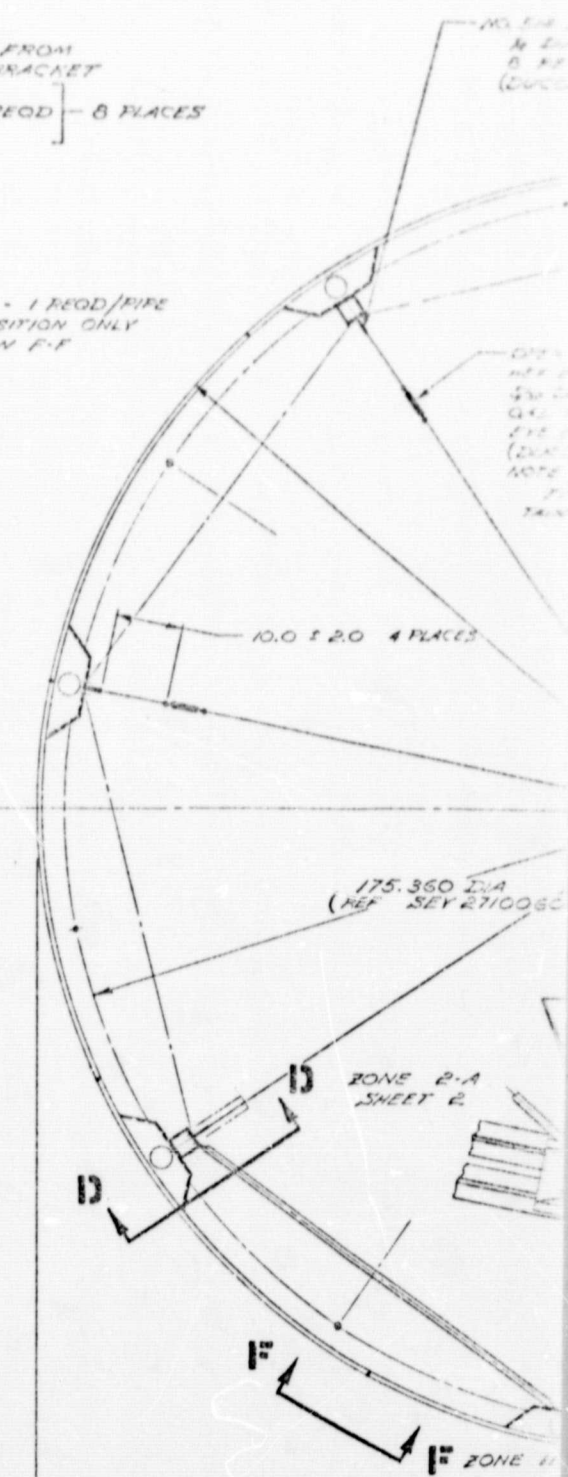


Figure 2-7.

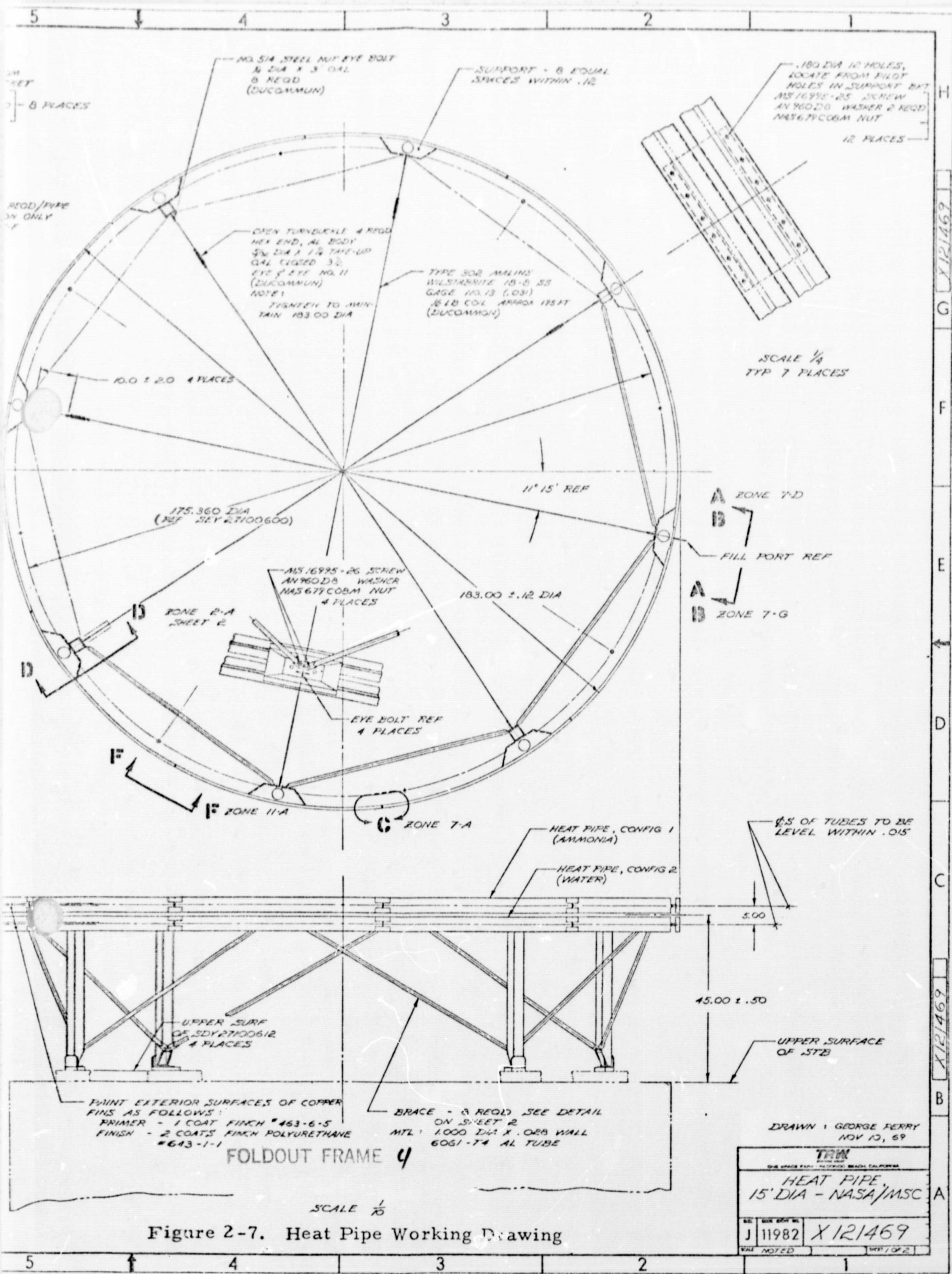


Figure 2-7. Heat Pipe Working Drawing

The concentric annular artery concept was achieved in an individual module by placing appropriately sized spacer wires between the walls of the tube and the screen surface providing the capillary driving force. Although a single layer of 200 mesh screen was theoretically adequate, two wraps were used to aid in sealing the artery and to eliminate potential problems due to 60 percent variation in wire spacing possible in commercial screenwire. The screen was forced outward against the spacer wires with a helical spring. Selection of six spacer wires was somewhat arbitrary; this number appeared to provide good control of the artery dimension without introducing excessive liquid pressure drop. The wires are all approximately parallel with the tube axis to minimize the liquid travel length. The wires were spot welded periodically to the inside of the tube to maintain their position during the bending operations.

Stainless steel (Type 316) was selected for all material in contact with the working fluids. This approach was taken even though ammonia is compatible with aluminum in order to have the fabrication of both pipes as nearly identical as possible. To this end surface passivation procedures were utilized for both systems although only the water pipe required this treatment. The intent of this action was twofold. First, welding schedules for joining the two pipes would be identical thereby reducing cost and adding to the confidence level of fabrication. Second, integrity checks of the wick joints during fabrication could be made using water rather than acetone or methanol. The passivation procedures utilized on the two heat pipe systems were not developed on this contract and are considered proprietary by TRW Systems.

The heat pipe systems described here would, in a real application, consist of a series of circumferential pipes on a cylindrical spacecraft. Spacing of these pipes and the thickness of the interconnecting fin would be the subject of an extensive weight "optimization" study where the tube and fin geometries would be "juggled" until the total weight per unit of heat rejection is a minimum. The primary objective of this study concerned specifically the operation of the heat pipe and a true weight optimization study considering the structural requirements and characteristics of the actual spacecraft was outside the scope of this program. For purposes of heat absorption and rejection, a short copper fin was included with

each of the heat pipe units. The dimensions (4 inch long by 0.032 inch thick) were selected to give a reasonable solar heat load on the system without experiencing a significant temperature drop in the longitudinal direction (this geometry has an effectiveness of essentially unity).

The fin also provided some stiffness to each heat pipe unit and was used as a mounting surface for the eight fiberglass supports which tied the ammonia and water pipes together. This aspect is described in more detail in Section 2.4.4. Thermal heaters used for testing of the heat pipes were also bonded to the copper fin (Section 2.4.7).

The joint between 5 foot heat pipe modules consisted of a 0.80 inch long solid bushing which had been stepped down on either end to receive the screen wick without a change in the artery dimension. Sealing the artery was accomplished by spot welding each screen to the sleeve; these spots must overlap to provide a continuous seal. Attempts were made to use a screen inner sleeve for joining two modules. This approach was desirable because vapor can escape at the joints and there is no danger of a vapor lock with subsequent failure of the artery. However, a screen sleeve was abandoned when it was clear that careful control of the artery could not be maintained and that there was a strong possibility of weld blow holes in the joint. Both conditions can bring about total failure of the artery. Insulation was provided around each joint to reduce the danger of vapor lock.

The final seal was made by welding a split sleeve collar to the two pipes. This sleeve was stepped down to conform to the ID of the tubes and had spacer wires welded to the ID to insure proper spacing of the inner sleeve-wick assembly. Great care must be used in welding to avoid burning the screens.

The fill tube is also shown in Figure 2-7 and consisted of a pipe between the outside and the vapor core. One welded seal was required to prevent leakage between the vapor core and the artery and another to prevent leakage between the artery and the external environment.

2.4.2 Water Heat Pipe System

The water heat pipe design specifies a 0.015 inch arterial gap thickness and two wraps of 200 mesh screen. The hydrodynamic pressure drops were shown in Figures 2-3 and 2-4. Figure 2-8 presents a plot of the vapor mass flow rate as a function of the heat pipe position ϕ (Sun normal is at $\phi = 0$). As can be seen, the highest mass flow rate (0.145 pound/hour) occurs at $\phi = 70^\circ$. This flow rate is substantially below the sonic and entrainment limits for water heat pipe systems.

For the fin configuration shown in Figure 2-2, the contact between the fin and the pipe is over only 1/3 the surface area of the pipe. Neglecting circumferential and axial conduction of the heat pipe wall and the axial conduction of the copper fin represents the "worst case" for computation of the overall temperature drop; Figure 2-9 shows the predicted temperature profile for operating with a "one sun" load at 76°F. The fluid inventory was calculated to be approximately 300 cc's. Hydrodynamic factor of safety for the water pipe is 4.0.

2.4.3 Ammonia Heat Pipe System

The ammonia pipe has a 0.012 inch arterial gap. The hydrodynamic pressure loss curves were shown in Figures 2-5 and 2-6. The calculated vapor mass flow rate for the "one sun" load is shown in Figure 2-10 as a function of pipe position. As for the water pipe, this flow rate is so small that no concern for the entrainment or sonic limit need be expressed. The "worst case" temperature profile predictions for "one sun" at 76°F is shown in Figure 2-11. As can be seen in Figure 2-11 the predicted temperature drop of the vapor core is negligible. The hydrodynamic factor of safety for this ammonia pipe is 1.5.

Calculations were also performed to 1) establish the charge of non-condensable gas to be used for start up from a frozen state, and 2) to estimate the time required for the startup with the solar input perpendicular to axis of the loop. These calculations were made using a gas front analysis program developed by TRW Systems to study gas filled variable conductance heat pipe systems. Neon was selected as the inert gas charge, having a molecular weight very near that of ammonia (20 versus 17). It was concluded from the analysis that the gas pressure must be at least

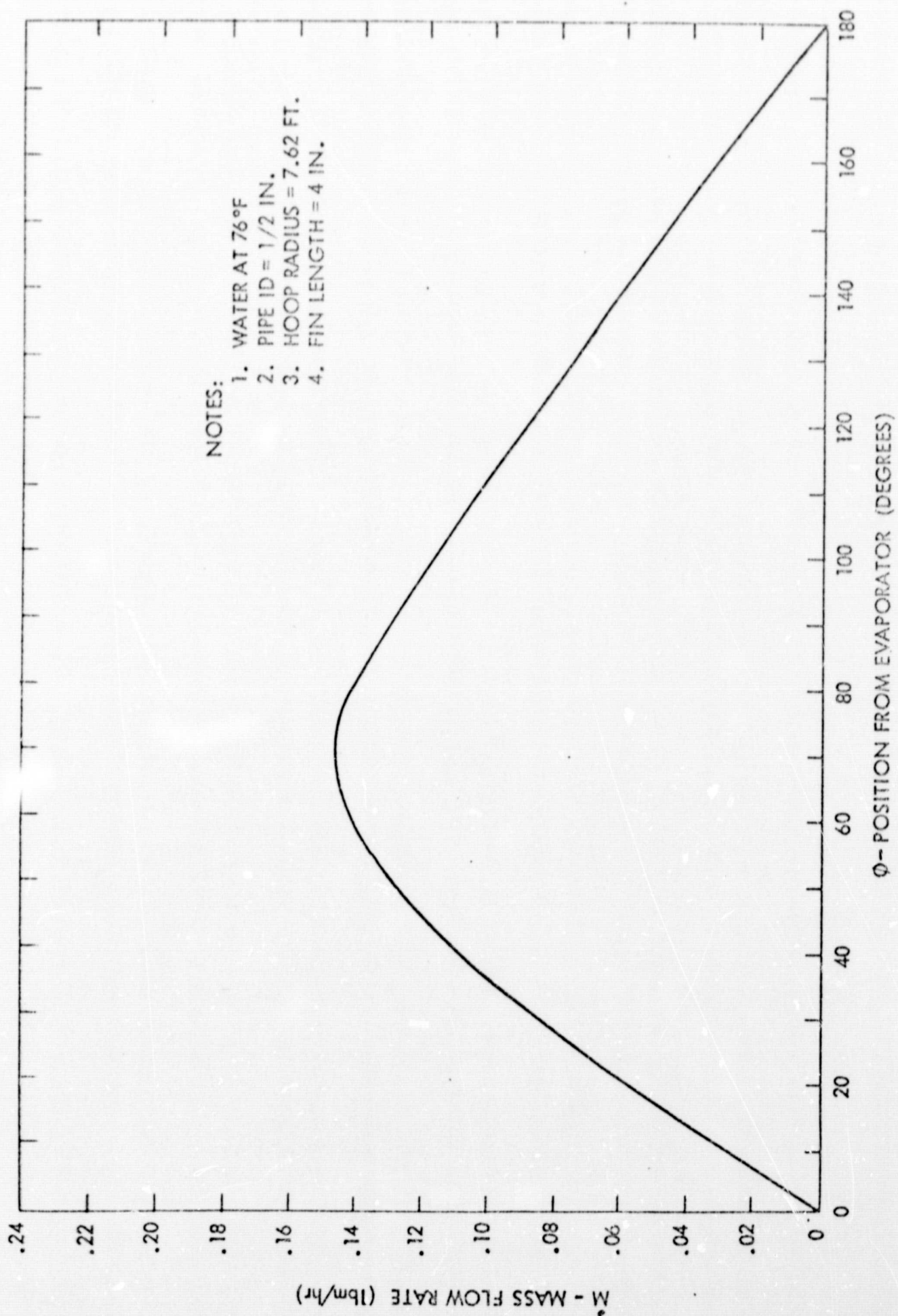


Figure 2-8. Mass Flow Rate Distribution - Water Heat Pipe System at 76°F

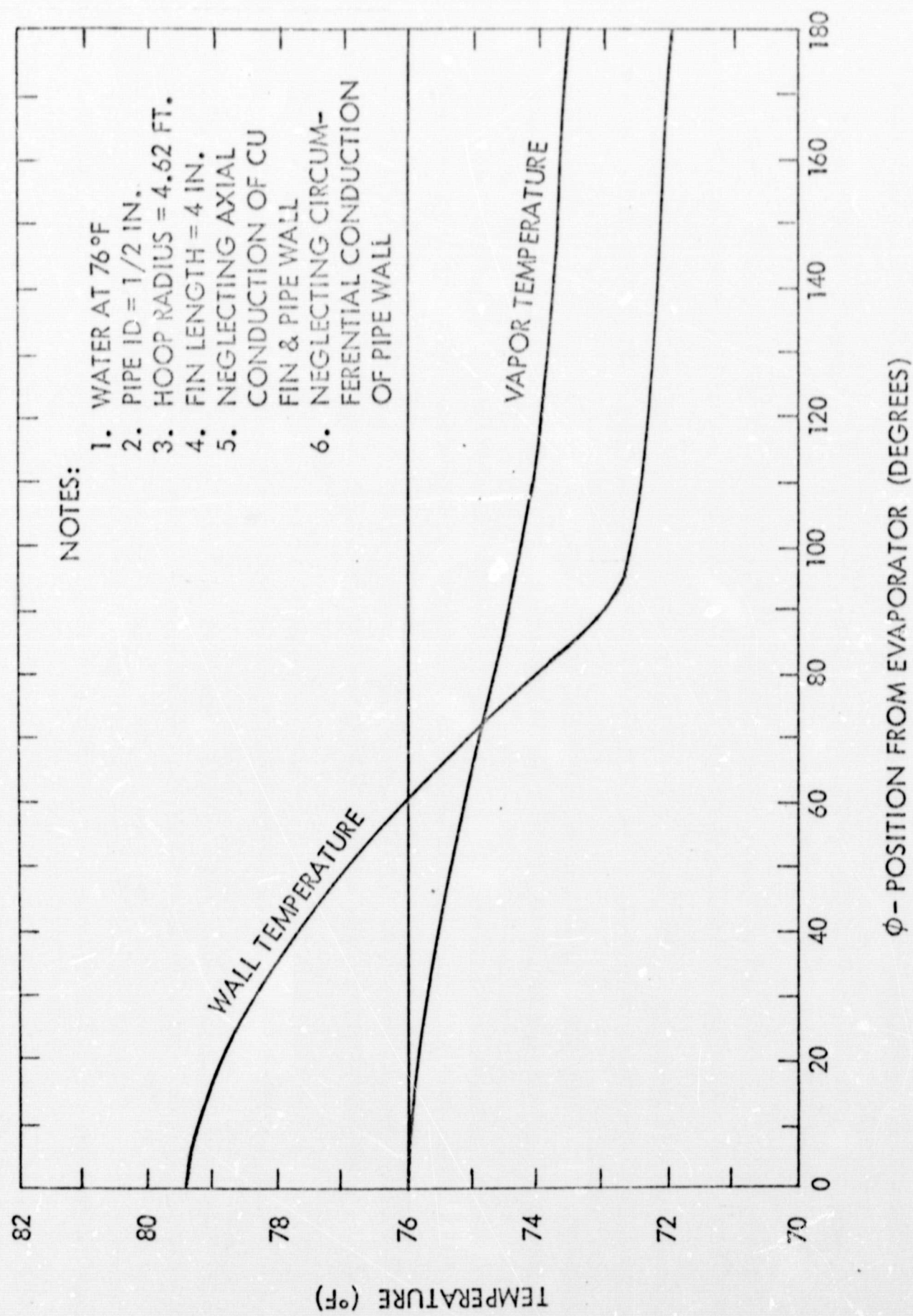


Figure 2-9. Temperature Profile - Water Heat Pipe System Under Thermal Vacuum Test Conditions

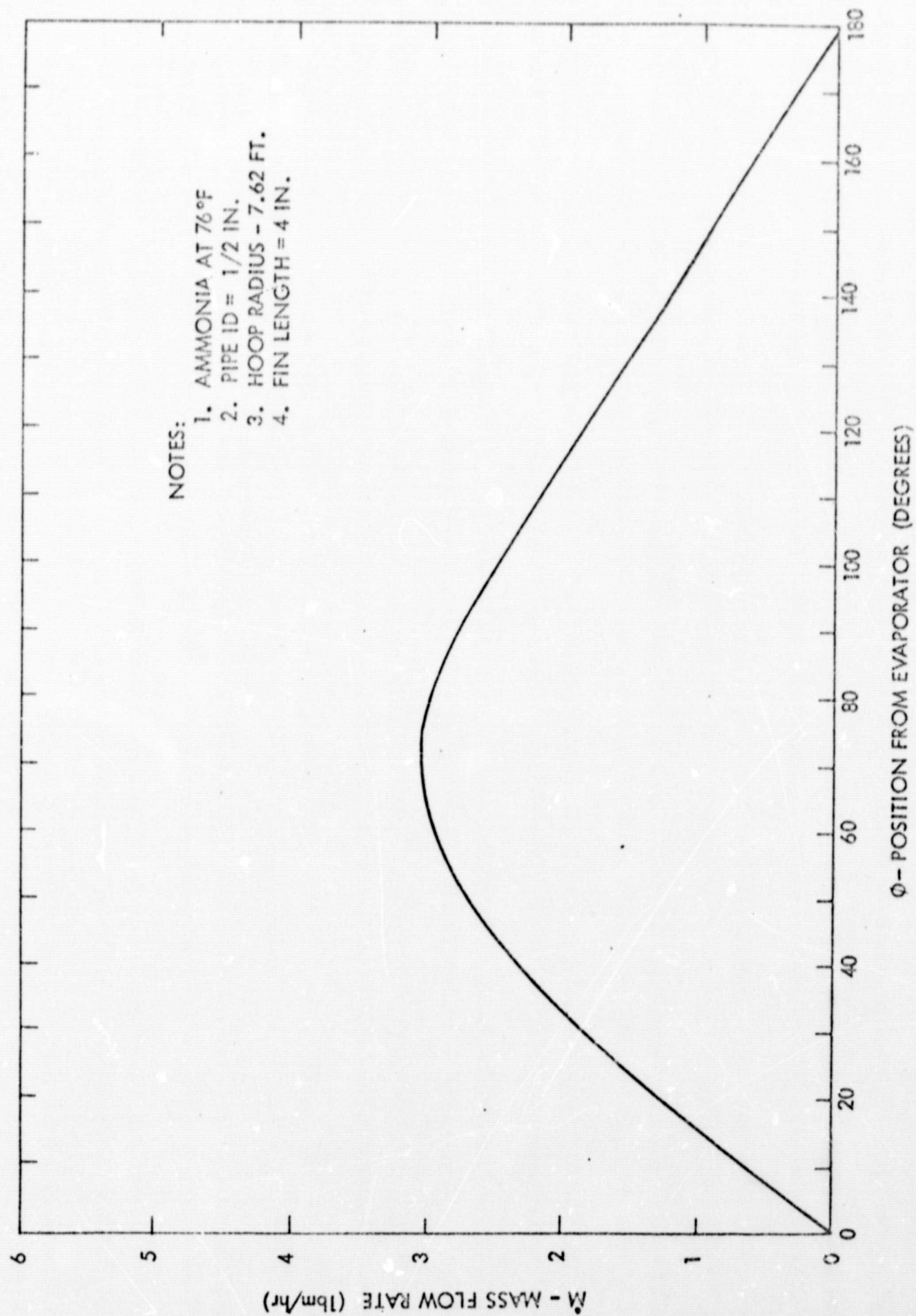


Figure 2-10. Mass Flow Rate Distribution - Ammonia Heat Pipe System at 76°F

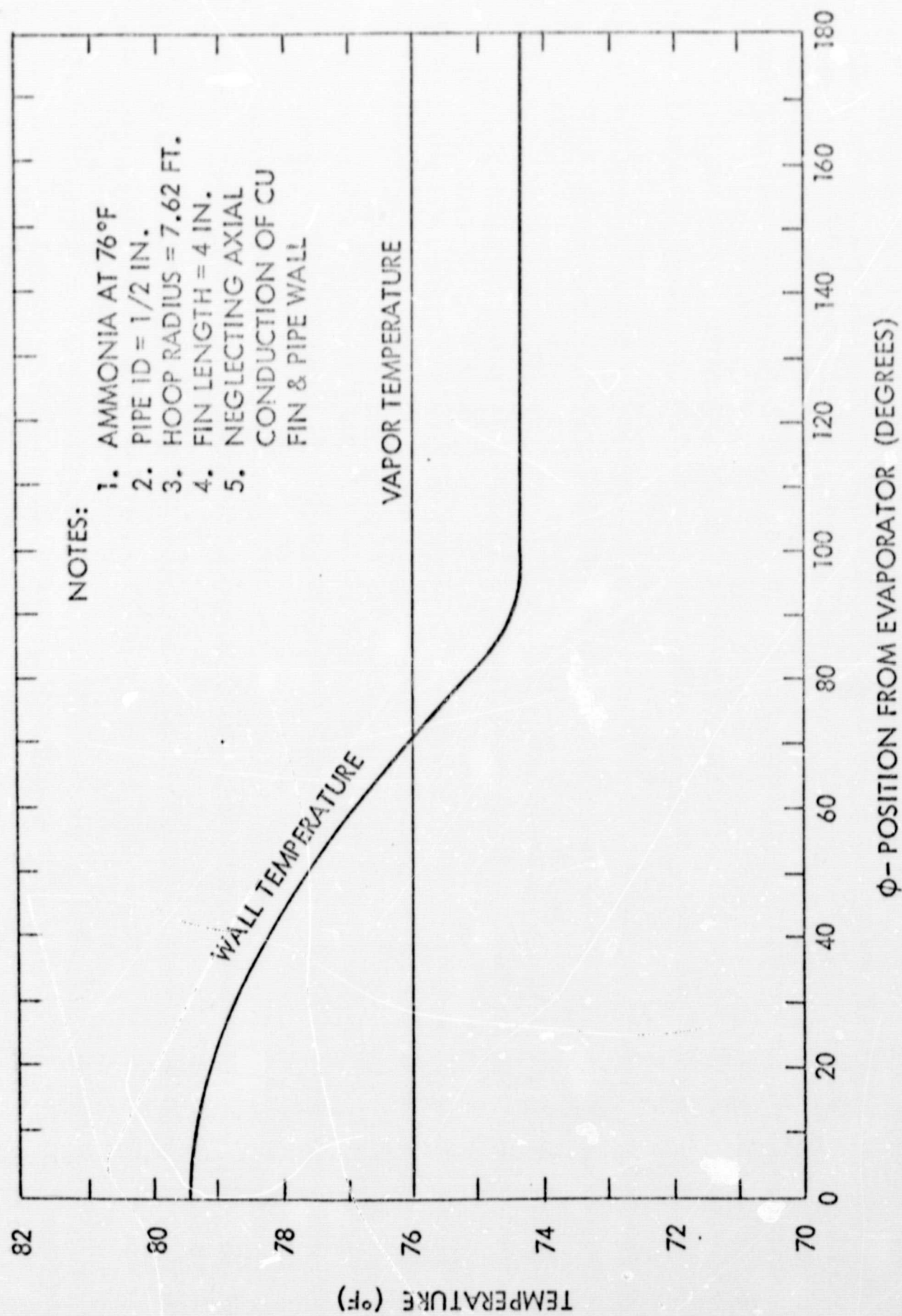


Figure 2-11. Temperature Profile - Ammonia Heat Pipe System Under Thermal Vacuum Test Conditions

1.0 psia at -300°F , a value slightly greater than vapor pressure of ammonia at its melting point (-108°F). The gas front during normal operation in the vacuum chamber would then occupy approximately 21 inches of the hoop (<4 percent). The front during ambient operation would be approximately twice as long because of the increased temperature level of the gas blocked region. Allowing for a small safety factor, a 4 psia gas pressure (at ambient temperature) was selected; the gas front for this pressure was predicted at 37 inches for chamber operation and 69 inches during ambient testing.

Predicted startup from -300°F is shown in Figure 2-12. It was assumed for this calculation that the pipe was uniform at -300°F when solar heating was initiated. Approximately 25 minutes are required for the first hoop segment to reach -108°F (ammonia melting temperature) and 275 minutes to completely compress the noncondensable gas and melt all of the solid ammonia. The relatively low equilibrium temperature resulted from the assumption in these calculations, that the edges of the fins would not be insulated. Figure 2-12 does demonstrate the relatively long time constant of the systems and suggests that a spinning heat pipe system (2 RPH) may restart without the aid of the noncondensable gas. The latter condition is shown as a dashed curve on Figure 2-12.

2.4.4 Support Stand

A support stand was required to attach the heat pipe-systems to the STB. The overall arrangement of the heat pipe/stand assembly on the STB is shown in Figure 2-7; the detailed parts of the stand are also shown in Figure 2-13. Conceptually this stand consisted of four aluminum posts (3 inches x 3 inches x 0.125 inches wall) which were attached, on one end, to four of the eight fiberglass brackets holding the two heat pipe units together. The other end contained a nylon insulator block which, in turn, connected to the STB. Support to the intermediate fiberglass brackets was provided by aluminum tubular trusses. Four stainless steel guywires, arranged in the form of spokes, provided rigidity to the assembly. The guywires also helped prevent droop in the pipe systems at the intermediate points not supported by the posts.

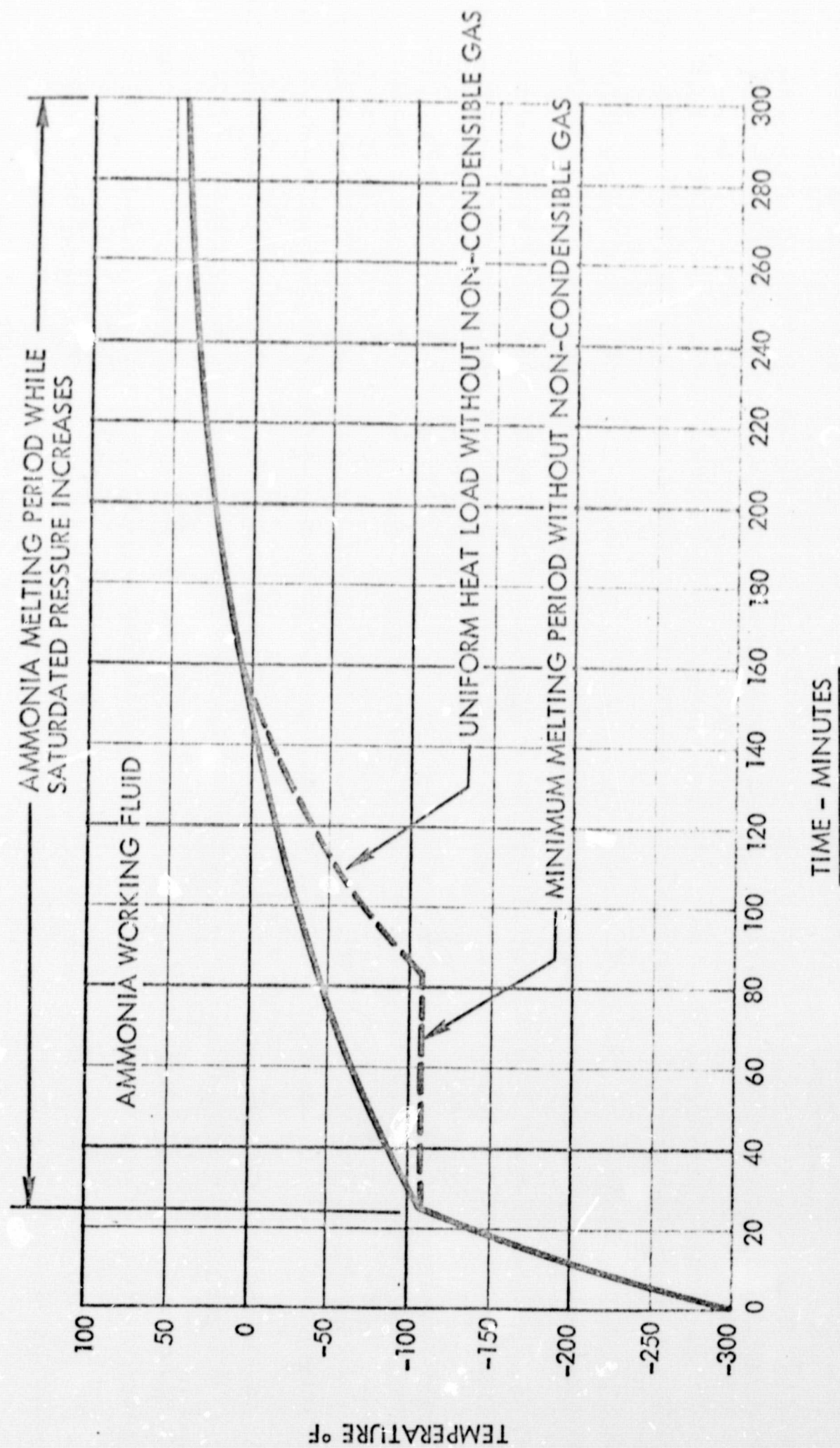
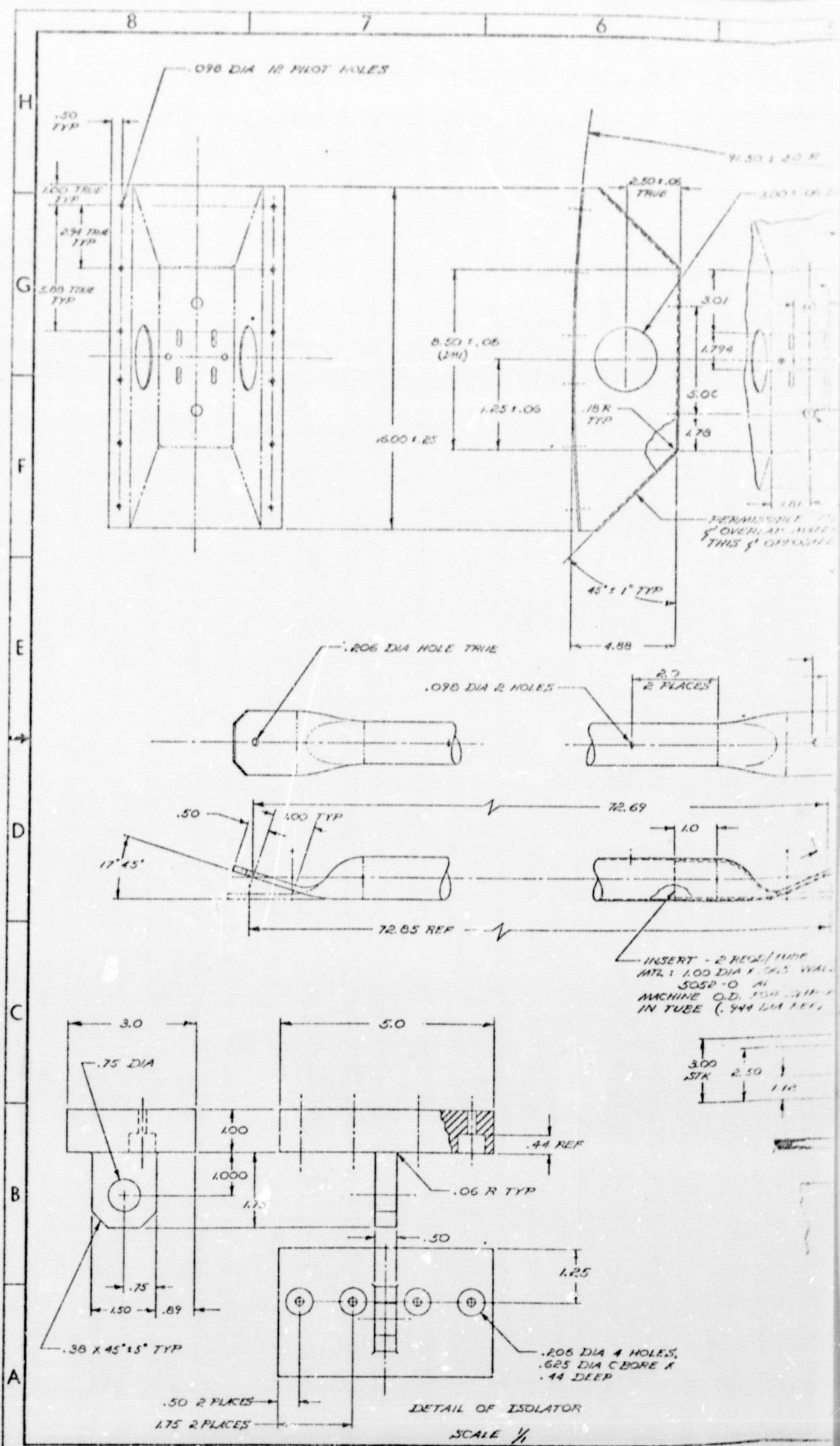
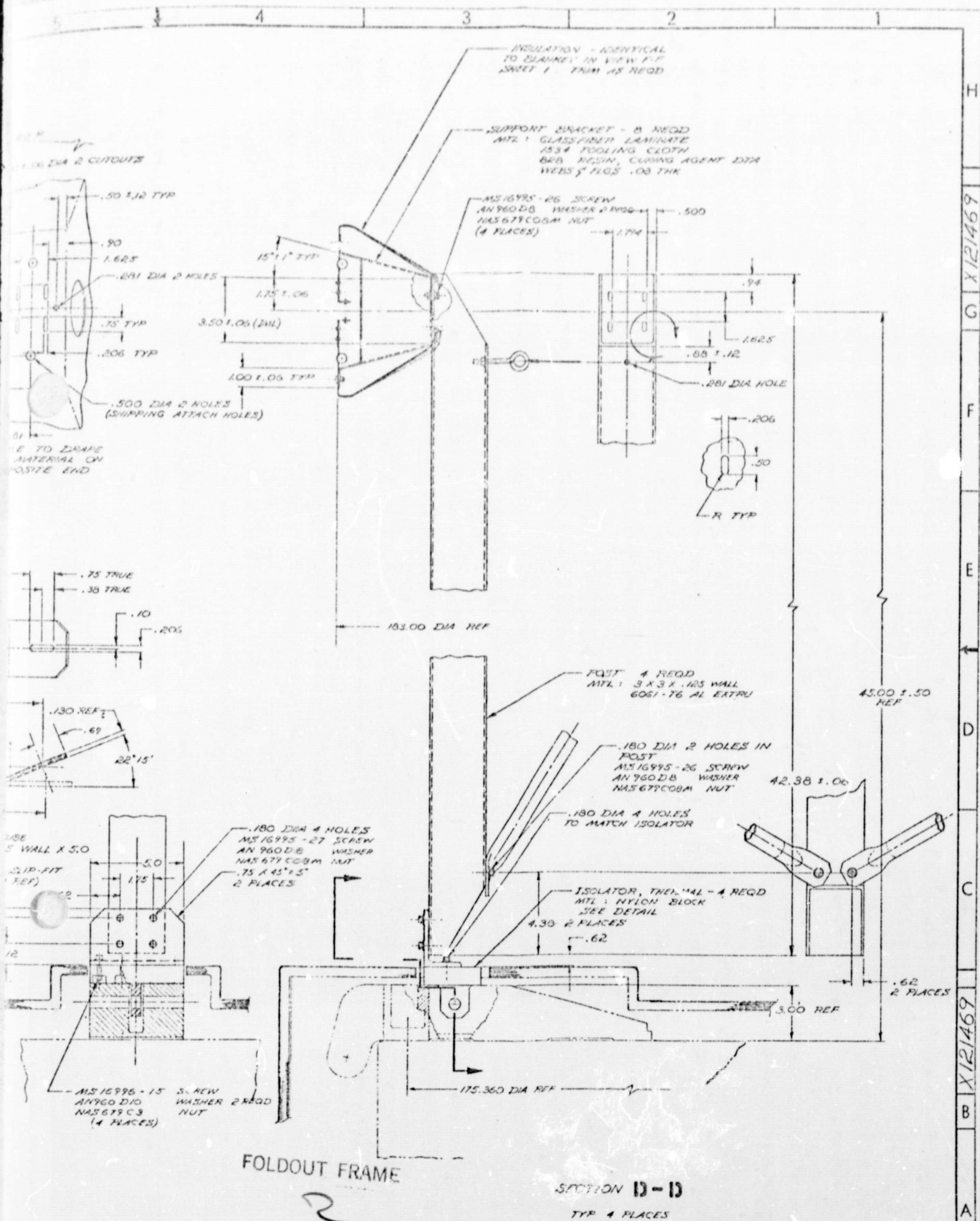


Figure 2-12. Ammonia Heat Pipe System - Transient Temperature During Startup

FOLDOUT FRAME





Evaluation of the STB heat leak due to the support stand depends, of course, on the temperature of the STB mounting points as well as interface resistances introduced at the relatively soft mount. Taking the extremely conservative assumption that the interface resistances are negligible (the only insulation is introduced by the nylon pad), that the STB mounting temperature is 50°F and that the uninsulated support rod "sees" only the LN₂ cold walls, the maximum heat leak from the STB was computed to be approximately 13 Btu/hour for each of the four support posts. It then concluded that the total STB loss for this condition will total less than 17 watts.

The two pipe systems were physically separated with eight fiberglass brackets; intermediate fiberglass blocks (1/2 inch x 3/4 inch x 1 inch) were also used to properly space the two pipes and to give additional support to the fins. The overall thermal resistance between the two pipes was estimated to be less than 3 hr-ft-°F/Btu. The net energy transfer with the two pipes 50°F apart in temperature would be less than 5 watts as compared to 172 watts absorbed solar load.

2.4.5 Insulation

Insulation on the back (i. e., inner) side of the pipe fin assembly consists of 10 layers of aluminized mylar which has been crinkled to provide separation between layers. The two outer layers are 3 mil thick to minimize the danger of damage; all inner layers are 1/4 mil. The 3 mil outer layer is reversed so the mylar side is facing outward to prevent excessive insulation temperature in case of sun impingement. The apparent conductance for this assembly is estimated to be less than 0.010 Btu/hr-ft-°R. Mylar tape was used to secure the insulation to the copper fin.

Each heat pipe system is insulated individually to enable separate testing at some later date. Separate insulation R_a+ sections were provided for each of the eight fiberglass bracket assemblies.

2.4.6 Radiative Coating—Fin

The fin required a coating have an α_s/ϵ of 1.0 to maintain 76°F with the sun incident normal to the side. A commercial aluminum paint (Finch Polyurethane Coating No. 643-1-1) was selected. According to

the manufacturers literature, the α_s and ϵ can vary from 0.18 to 0.26 with the α_s/ϵ ratio remaining between 0.9 and 1.1

The results of preliminary tests by the Thermophysics Laboratory of TRW Systems are summarized below:

α_s	ϵ^*	α_s/ϵ	Sample Description
0.26	0.32	0.81	1 mil, with primer
0.21	0.29	0.72	2 mil, with primer
0.24	0.32	0.75	1 mil, without primer

One mil of paint over a primer was selected for application and gives an equilibrium temperature of approximately 50°F. Average properties of samples coated at the same time the fin was painted were $\alpha_s/\epsilon = 0.217/0.237$. This gives an equilibrium temperature of approximately 65°F.

Finch Polyurethane Aluminum Paint is very susceptible to damage during handling. Care must be exercised to prevent contact with the surface after application; all moving, lifting, etc., should be accomplished using other, non-painted surfaces.

2.4.7 Test Heaters

Electric resistance heaters were provided on each 5 foot module of each heat pipe system. These heaters were bonded to the fin bottom half in the case of the ammonia pipe and top half in the case of the water pipe, and were approximately 50 inches long by 1 1/2 inch wide. Each heater consisted of approximately 2000 feet of Manganin heater wire wound on top of 3 mil teflon tape which had been bonded to the fin; a second layer of teflon tape was applied over the wire for total electrical insulation. Aluminum tape applied over the top of the assembly aided in holding the heaters in place. This design has been utilized with success on numerous TRW spacecraft thermal model tests. Each heater was capable of 100 watts at 28 volts (AC or DC) and requires control of the input voltage to achieve a specified power level.

* Hemispherical emittance determined by correction of near normal reflectance data.

Each heater was cut to have a resistance of 7.0 Ω at room temperature. The effect of temperature on heater resistance is shown in Table 2-1. Over the expected range the total variation in heater resistance is less than 0.52 Ω . Thirty feet of leads (20 gauge) are provided for each heater to permit mating with the NASA/MSC Chamber A Facility. These leads on the average increase the total resistance (leads + heater) to 7.6 Ω . The input voltage reading (ΔV) to the lead must be corrected per the following equation to determine the power (P_H) actually dissipated at the heat pipe module

$$P_H = \frac{7.0}{(7.6)^2} \Delta V^2$$

The above equation does not account for losses in the wiring system upstream of the heater leads.

Table 2-1. Resistance Change vs Temperature
Manganin Wire Stock

<u>Temperature °C</u>	<u>% Change in Resistance</u>
20	0
30	-.0036
40	-.0150
50	-.0357
60	-.0621
70	-.0963
80	-.1270
90	-.1890

The 40 heater leads were routed to a Cannon connector compatible with the NASA/MSC Chamber A Facilities. This connector, specified by NASA, was purchased from North American Rockwell and had the NAR designation ME414-0584-0001. Only 42 of the 60 pin locations were utilized. Details as to pin locations for each heater are given in Section 4.0 of this report. All leads were bundled together after leaving the heat pipe systems and were wrapped with multilayer insulation to minimize the heat leak. This bundle is located 90 degrees around from the fill tube area (clockwise as viewed from above).

2.4.8 Instrumentation

A total of 85 thermocouples were applied to the heat pipe/support stand assembly. Of these 30 were applied directly on each of the two heat pipe systems and 25 located at various places on the fin, insulation, fiberglass bracketry and support stand; the precise location of each thermocouple is given in Section 4 of this report.

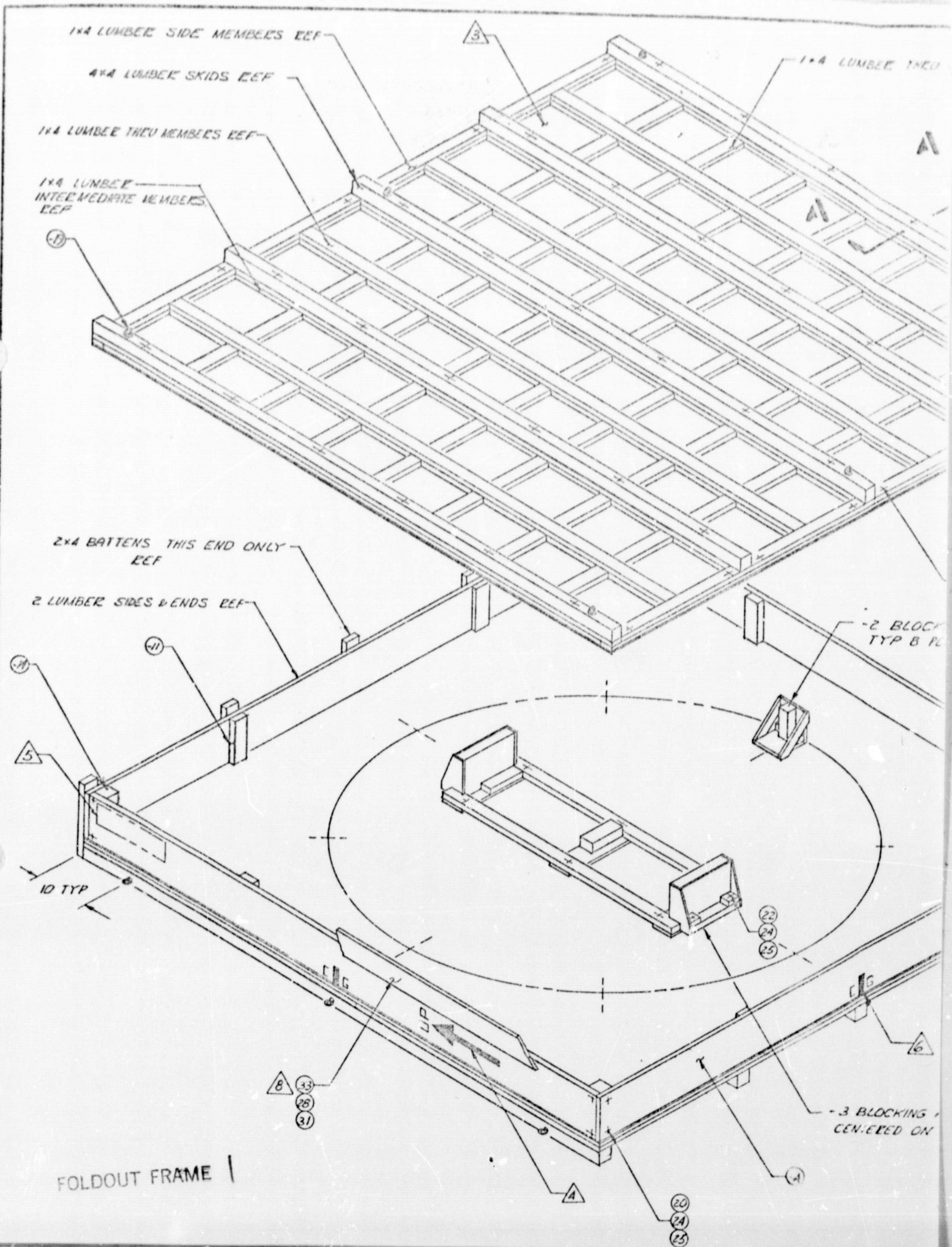
Each thermocouple was a 28 gauge, copper-constantan pair with fiberglass insulation. Junctions were formed by spot welding or by soft soldering. The individual couples were unshielded. However, the bundle of wires (located 270 degrees clockwise from the fill tube area as seen from above) was provided with an electrical shield and was wrapped with multilayer insulation to minimize heat losses. The electrical shield is grounded to the cable connector. Each thermocouple was provided with 50 feet of extra lead for mating with MSC chamber. Originally 30 feet was provided; the addition of an extra 20 feet was accomplished by splicing and soldering each thermocouple pair.

The connectors used to mate the thermocouples with the NASA/MSC Chamber A Facility were Cannon receptacles having the designation DPDMA41700-215/FSC5935-LN89772. These units were specified by NASA/MSC and had 20 gauge copper and constantan removable pins. Three connectors were used for the 85 thermocouples; the pin locations of each is specified in Section 4 of this report.

Thermocouple attachment in the case of the heat pipes or the copper fin was by direct spot welding or by soldering. Conducting epoxy with a covering of aluminum tape was used for all others. Where possible thermocouple leads were routed at least 12 inches along the attach surface to minimize measurement errors due to the thermocouple conduction losses.

2.4.9 Packaging and Shipping

Packaging and shipping, although not specifically tied to the development of the heat pipe systems, represented a substantial amount of effort because of the large size of the system. A drawing of the container is shown in Figure 2-14; the unit measures 16 feet x 16 feet x 1 foot and weighs approximately 2000 pounds.



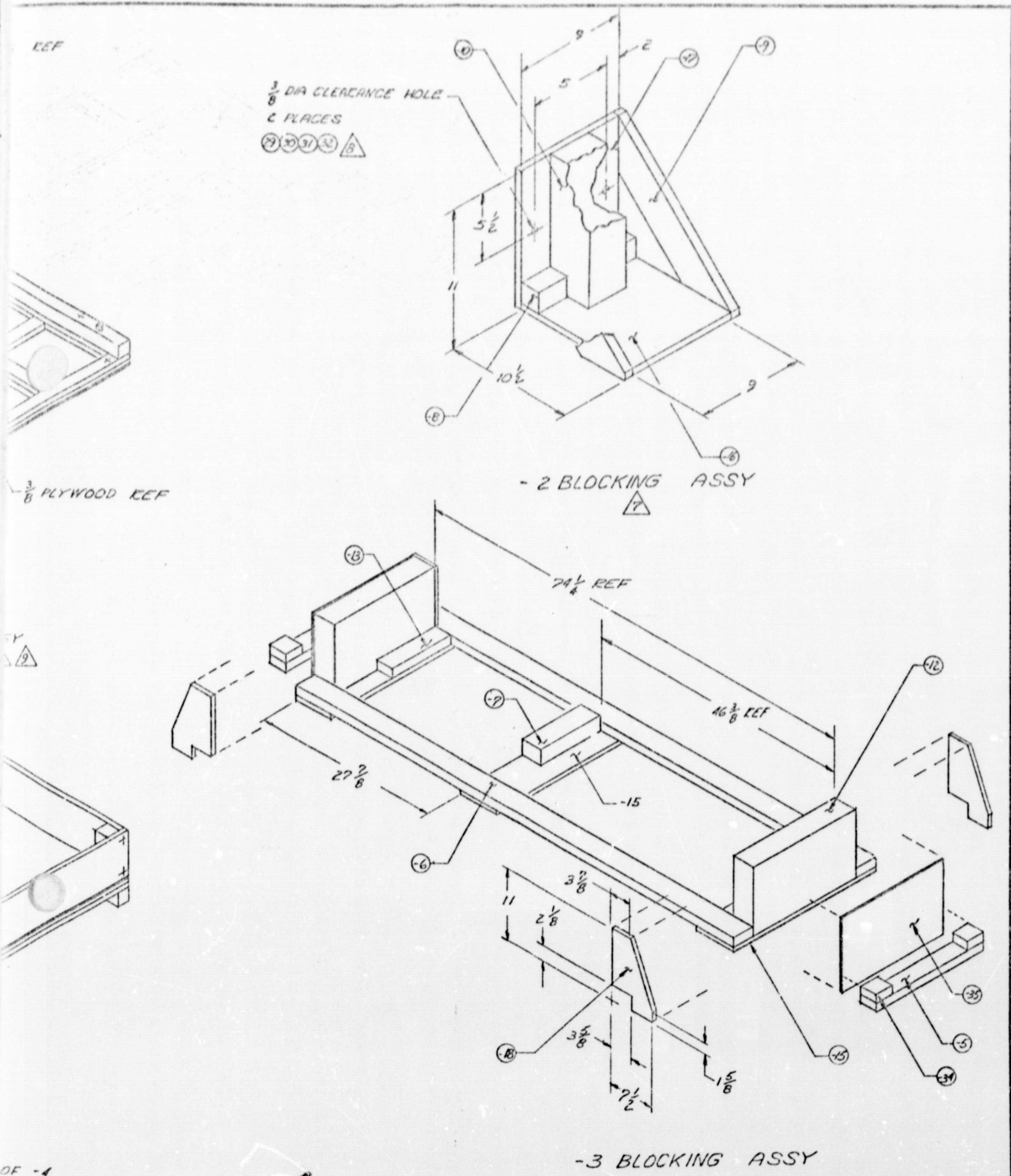


Figure 2-14. Shipping Container

SIZE	CODE IDENT NO.	SK2713
D	11982	
SCALE	NONE	
		SHEET 2 OF 2

3. HEAT PIPE SYSTEM FABRICATION

This section serves to highlight the fabrication effort and identifies some of the more significant development problem areas that had to be resolved. A number of fabrication and processing procedures had been developed by TRW Systems prior to initiation of this contract and are considered proprietary by the company. Proprietary concepts are not described in this report.

As nearly as was feasible, the fabrication of the ammonia and the water pipes was identical. The following discussion, therefore, applies to both. Necessary differences between the two are clearly identified.

3.1 DEVELOPMENT

Fabrication of heat pipe systems is typically carried out by skilled laboratory technicians working in a closely controlled environment. Building large scale heat pipes, on the other hand, necessitates use of more conventional manufacturing facilities and personnel. The first of the major fabrication difficulties associated with the 15-foot diameter heat pipe systems described in this report centered around education of shop personnel about the cleanliness required in heat pipe fabrication. Use of gloves, copious amounts of acetone or methanol, and great care to prevent contamination of the pipes were difficult to instill initially. After some experience, they became routine operating procedures.

Besides these normal difficulties of cleaning and handling, major development effort was required in two other general areas. The first involved retention of the six spacer wires during the bending process, and the second problem included the overall process of joining two heat pipe modules together. Each of these is described in the following paragraphs.

The initial design concept called out a single layer of 200 mesh screen with the spacer wires spot welded directly to the screen. The total assembly was then rolled up and inserted into the tube. Two problems resulted from this mode of construction. First, sealing of the artery was difficult with a single screen layer. Two layers were then specified after unsuccessful attempts to make a joint. Also, maximum wire spacing varies up to 60 percent in standard 200 mesh screen. Two

layers assures better control of artery pore size. The second problem came from the danger of a blow hole in the screen due to the weld. The capacity of the artery system is limited by the largest hole in the artery. Any hole larger than the 200 mesh screen would cause a reduction in that capacity.

It was, therefore, decided that the spacer wires would instead be spot welded to the ID of the tube. The six wires each had to be tacked every inch along the tube to prevent movement during bending. A special welding fixture was developed to make these spot welds. With this approach, extreme care had to be exercised when inserting the rolled screen in the tube to prevent binding and tearing.

The problem of joining two modular sections fundamentally reduced to sealing the arterial cavity. The first design called for the use of an inner screen sleeve to which screens from both modules were welded. This approach presented three difficulties. First, critical artery dimensions were difficult to maintain. Second, there existed a problem of weld blow holes which would degrade the performance of the artery. Finally, this approach required insertion of a bullet to support the joint during welding. Thus, the support spring could not be inserted in one of the two five-foot modules until after the joint had been made; the danger of damage to the joint was substantial.

The final configuration utilized a solid internal sleeve to which the module screens were welded. This sleeve needed no additional support during the welding operation, thereby eliminating the need for the welding bullet. No major problems in the joint were encountered after selection of the solid internal sleeve. It should, however, be recognized that at the solid internal sleeve there existed no way for vapor to evaporate from the artery surface, which posed the potential danger of nucleation and vapor lock of the artery. To reduce this danger, all junctions on the final heat pipe systems were insulated to prevent heat input. Figure 3-1 shows two views of a developmental screen/sleeve joint.

During the initial fabrication development effort, 5-foot modules were sealed and processed as heat pipes for detailed thermal tests; these tests are described in Section 2. The results of these tests, in fact, had a significant impact on the selection of the final design configuration.

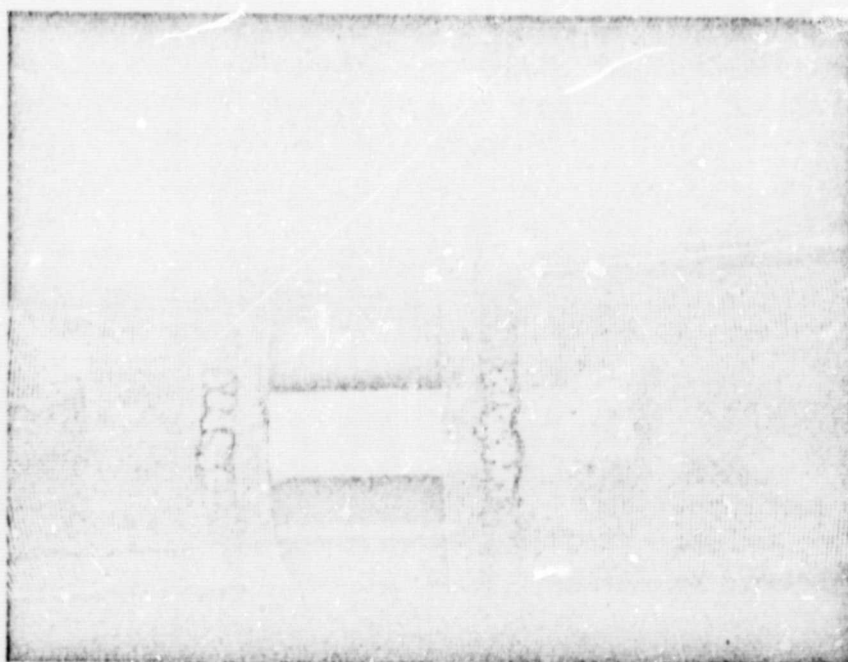


Figure 3-1. Screen Solid-Sleeve Junction Between Two 5' Modules

Additionally, to add to the confidence level of the joint fabrication process, wick integrity testing was performed on selected 10-foot sections. These tests, also described in Section 2, were hydrodynamic in nature in that they were designed to measure the maximum wicking head of the artery/screen/joint configuration. The artery was first sealed with a special end cap and plug, and the fluid introduced (water was used for both pipes). With the pipe in the level condition, the artery was allowed to prime. The unit was then lifted in 1/2-inch intervals, each time waiting until excess liquid drained past the spring coils, until the capillary head "broke" and the artery emptied. Typical heights for 10-foot modules tested were 8 to 10 inches. A nonsealed artery would not prime and could support no head. After successfully testing a series of joints, the confidence in the fabrication technique was sufficiently high that the time-consuming wick integrity tests were discontinued.

3.2 FABRICATION PROCEDURE

The procedure utilized in the manufacture of the complete heat pipe systems, of course, commenced with fabrication of the individual parts (cut tubes, screens, machine solid bushings, split couplings, etc.). The following sequence was then utilized to complete one 15-foot diameter heat pipe system:

- Clean all parts and anneal screens for easier handling.
- Spot weld spacer wires to tube ID.
- Passivate critical pipe components.
- Roll screens and insert in tubes. Force outward against spacer wires with progressively larger balls or bullets.
- Insert springs and trim screens and springs to length.
- Make joint between two five-foot modules. This operation consisted of welding the screens to the interconnecting coupling, inspection of the joint, installation of the split coupler and final welding.
- Wick integrity check the assembly. On a few joints for each of the pipes, a wick integrity check was performed to insure that the manufacturing procedure was producing acceptable joints.

- Make three 10-foot sections and a 15-foot section, which are then joined into a 45-foot straight length.
- Bend straight length around "Ferris Wheel" fixture. This fixture is shown in Figure 3-2, with water heat pipe system attached.
- Install the last five-foot module. Note that this closure contains the fill tube assembly which requires an additional weld on the split coupler.
- Evacuate the assembly and check for leaks with a mass spectrometer leak detector.
- Solder the 10 copper fin sections in place and rivet the interconnecting pieces.
- Pressurize the completed unit to check structural integrity. A pressure of 720 psi was used to check the ammonia pipe and 250 psi for the water pipe.

The first unit (ammonia) was then removed from the "Ferris Wheel" fixture, placed in a horizontal position for processing and filling, and the fabrication of the water pipe initiated.

3.3 PROCESSING

Processing of the fabricated heat pipe systems followed the procedure listed below. For the most part, this procedure was the same for the water and the ammonia pipes. Differences, when they do occur, are specified in the following write-up:

- Pump Down and Bake Out — The completed unit was subjected to a roughing vacuum (approximately 35 microns) and heated to 300°F for a minimum of four days to drive out volatile contaminants.
- Pump Down and Fill — Prior to fill with the appropriate working fluid, the system was subjected to a hard vacuum (10^{-4} torr or better) for approximately 24 hours and then back-filled with ammonia or water.

Vacuum valves had been inserted on both fill tubes prior to final pump down to provide a means to alter the liquid inventory and to "burp" out noncondensable gases, if required.

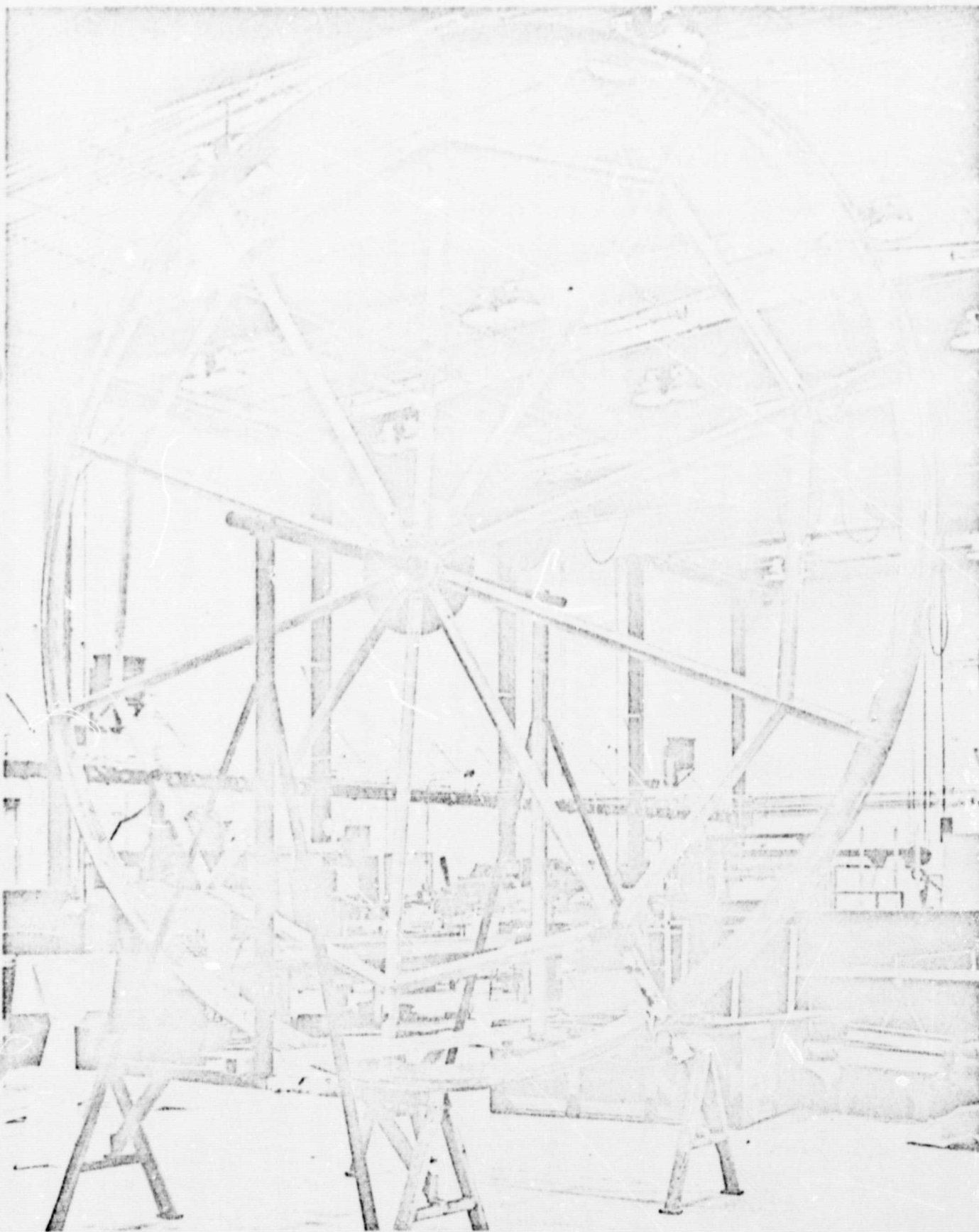


Figure 3-2. Water Heat Pipe System Formed into 15-ft Diameter on Special "Ferris Wheel" Assembly

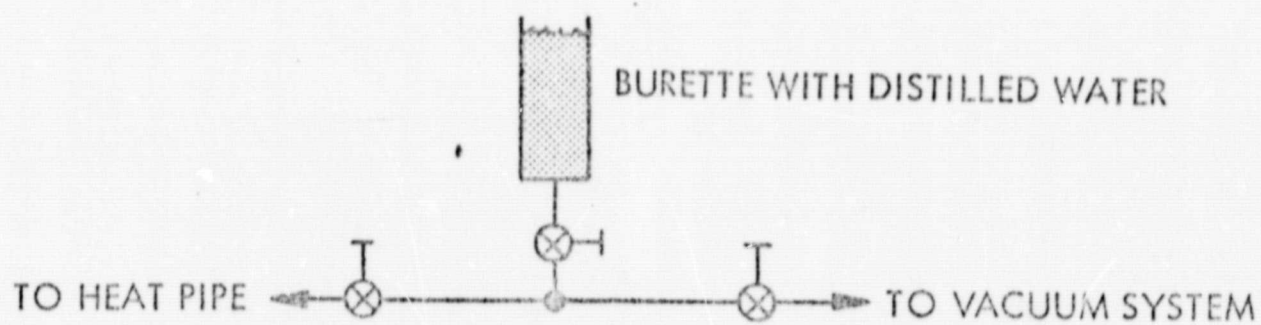
Filling the water heat pipe was relatively simple; the setup is shown schematically in Figure 3-3a. A measured amount of triple distilled water was filled directly into the evacuated pipe through a fill line which also had been evacuated. Both the pipe and the fill line were carefully checked for leaks prior to the fill operation. There was evidence that initial water fills introduced trapped gasses into the pipe. Subsequent fills were made with degassed water in a closed system (Figure 3-3b).

The fill of the ammonia pipe was substantially more complicated; a schematic of the setup is shown in Figure 3-3c. With the line to the water column and the regulator valve to the ammonia closed, and all other valves open, the heat pipe, lines and reservoir were evacuated and checked for leaks. The valves to the heat pipe and to the vacuum system were then closed, the reservoir chilled with an acetone and dry ice mixture, and the regulator opened to allow ammonia flow into the reservoir where it liquified. The regulator was closed after the reservoir was filled and ammonia distilled into the heat pipe by heating the reservoir and opening the heat pipe valve. The ammonia quantity entering the heat pipe each time was established by the volume of the reservoir; both 50 cc and 300 cc reservoirs were used in the fill operation.

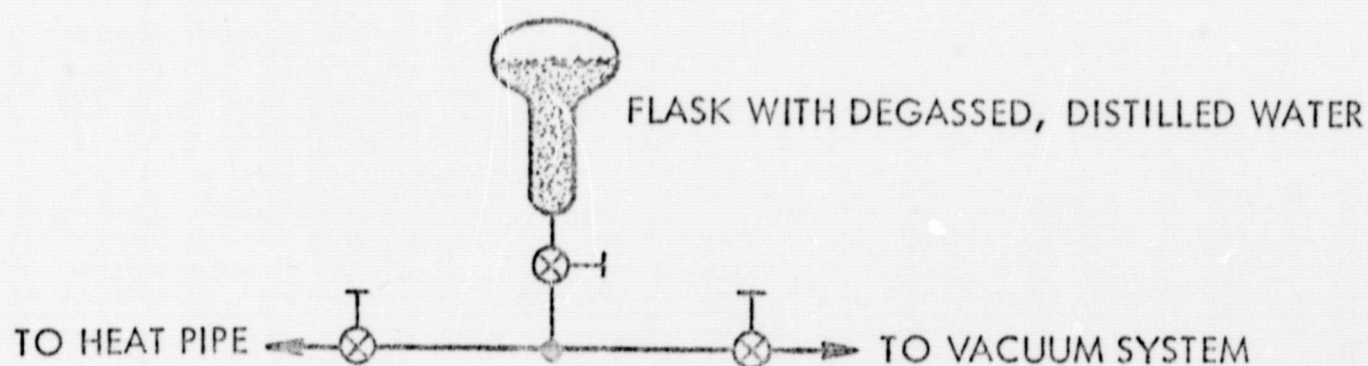
The initial fill was accomplished by freezing rather than liquifying ammonia in the reservoir. This approach was abandoned because of uncertainties in the manner in which the ammonia would freeze. This led to considerable uncertainty as to the exact quantity of ammonia in the system.

Introduction of the neon charge in the ammonia pipe after preliminary testing was accomplished by first measuring the pressure of the ammonia in the pipe and adding an overpressure of 4 psi of neon. This value represented approximately 2×10^{-5} pound moles of neon. The ammonia pressure at 75°F is 128 psi which made the addition of a 4 psi overpressure difficult to control. A preferable approach would have been to fill with gas before adding the ammonia. This procedure was not followed because of the fear (later justified) that the ammonia pipe would not operate with the gas load.

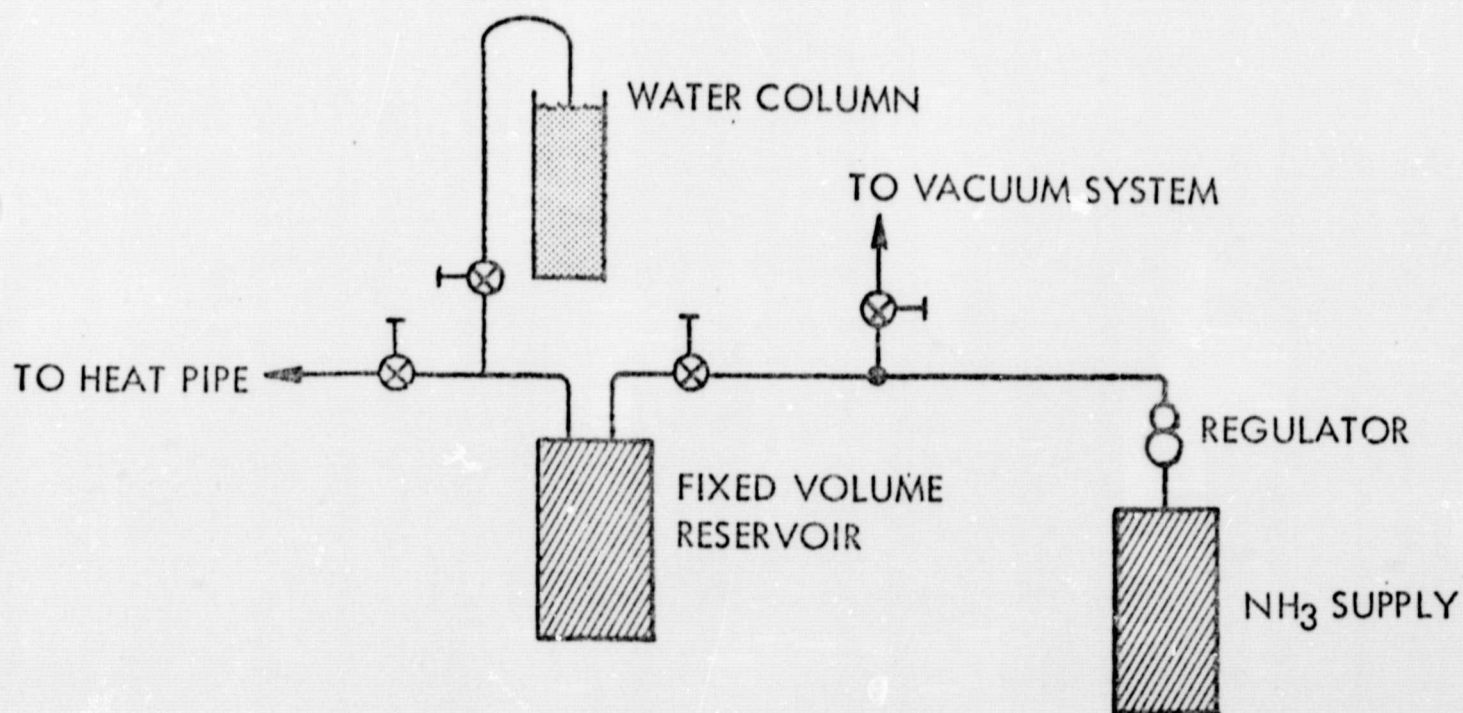
Each pipe was instrumented and the heaters installed prior to development testing (Section 4.2). See Sections 2.4 and 4.1 for a description of the instrumentation and the electric heaters.



A. INITIAL WATER FILL SYSTEM



B. FINAL WATER FILL SYSTEM



C. AMMONIA FILL SYSTEM

Figure 3-3. Heat Pipe System Fill Techniques

A final closure of the heat pipe systems was planned after completion of the development testing, but prior to the final assembly procedure described below. The closure technique utilized is considered proprietary to TRW Systems and will not be described here. Because of a sequence of difficulties, these closures were subsequently cut off the pipes and replaced with vacuum valves and a mechanical pinch off backup. The activities leading up to these circumstances are described chronologically in Section 4.

3.4 FINAL ASSEMBLY

Final assembly of the two heat pipe systems into a single test unit did not take place until development testing had been completed. This assembly included the following activities:

- Coupling of the two heat pipe systems with eight fiberglass supports. The supports are shown in Figure 3-4. Fiberglass spacers (3/4-inch wide) were also used between the supports to maintain constant fin spacing.
- Painting of the unit — An aluminum paint (Finch Polyurethane Coating No. 643-1-1) was used (one coat) with the appropriate primer (Finch No. 463-6-5 primer).
- Insulation of the back of the individual fins and of the individual joints. Figure 3-4 shows the back insulation and Figure 3-5 shows the insulation of the joints.
- Shielding and insulation of the power and thermocouple leads and installation of the connectors. Figure 3-6 shows a technician checking one of the three thermocouple connectors.

Figure 1-1 shows a view of the completed heat pipe unit undergoing thermal verification testing. The fiberglass supports could not be insulated until final installation with the support stand on the STB. Final installation of the thermocouples on the support stand was also accomplished at NASA/MSD.

3.5 PACKAGING AND SHIPPING

The heat pipe system being packed in the shipping container is shown in Figure 3-7; the unit mounted vertically on the rail flatcar is shown in Figure 3-8. The black cover is a tarpaulin to protect the unit from inclement weather.

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Figure 3-4. View of Heat Pipe System Fiberglass Supports and Multilayer Insulation

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Figure 3-5. Heat Pipe System Joints Between Modules

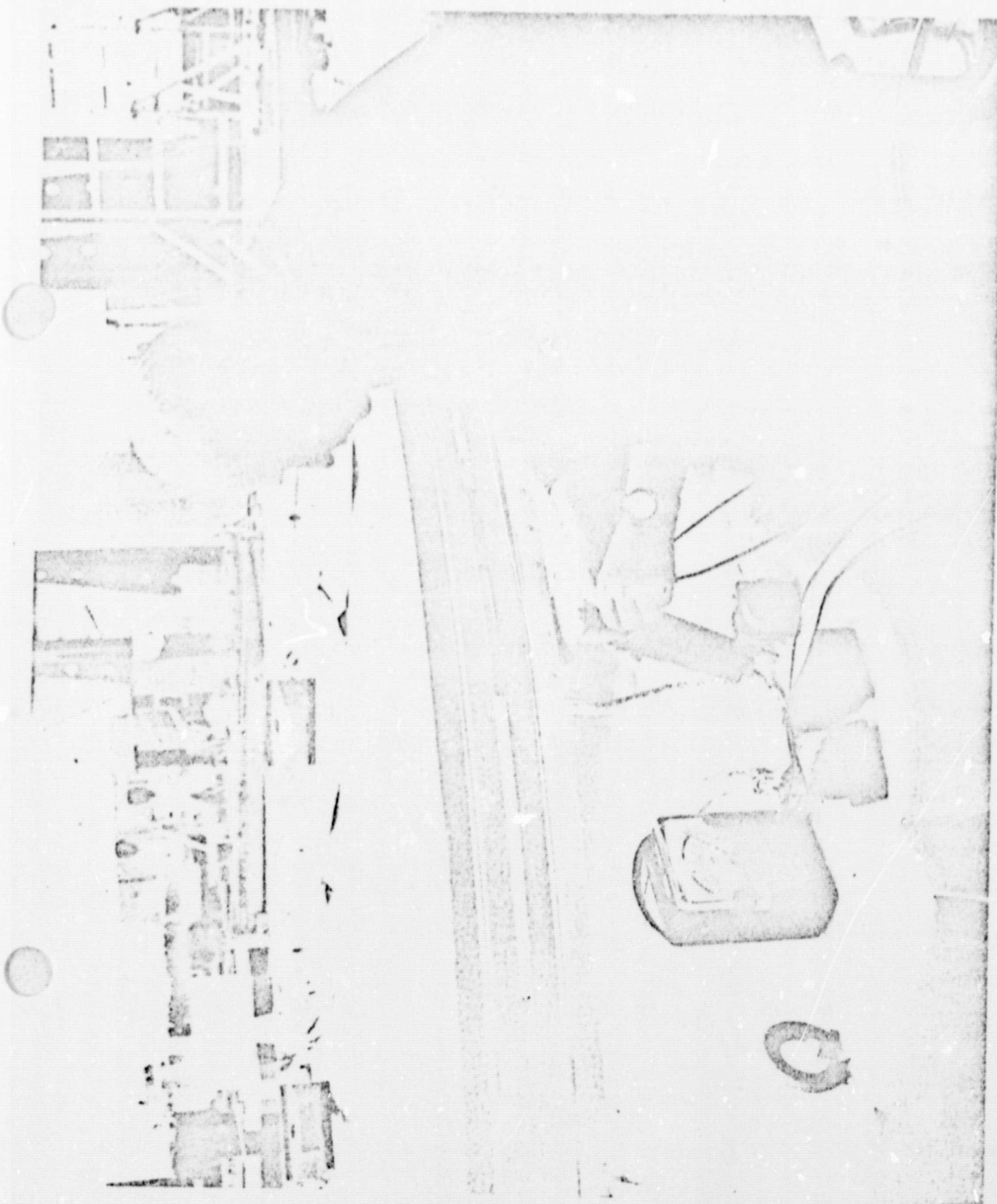


Figure 3-6. Thermocouple and Power Connectors for Thermal Vacuum Testing

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Figure 3-7. Heat Pipe System Installed in Shipping Container

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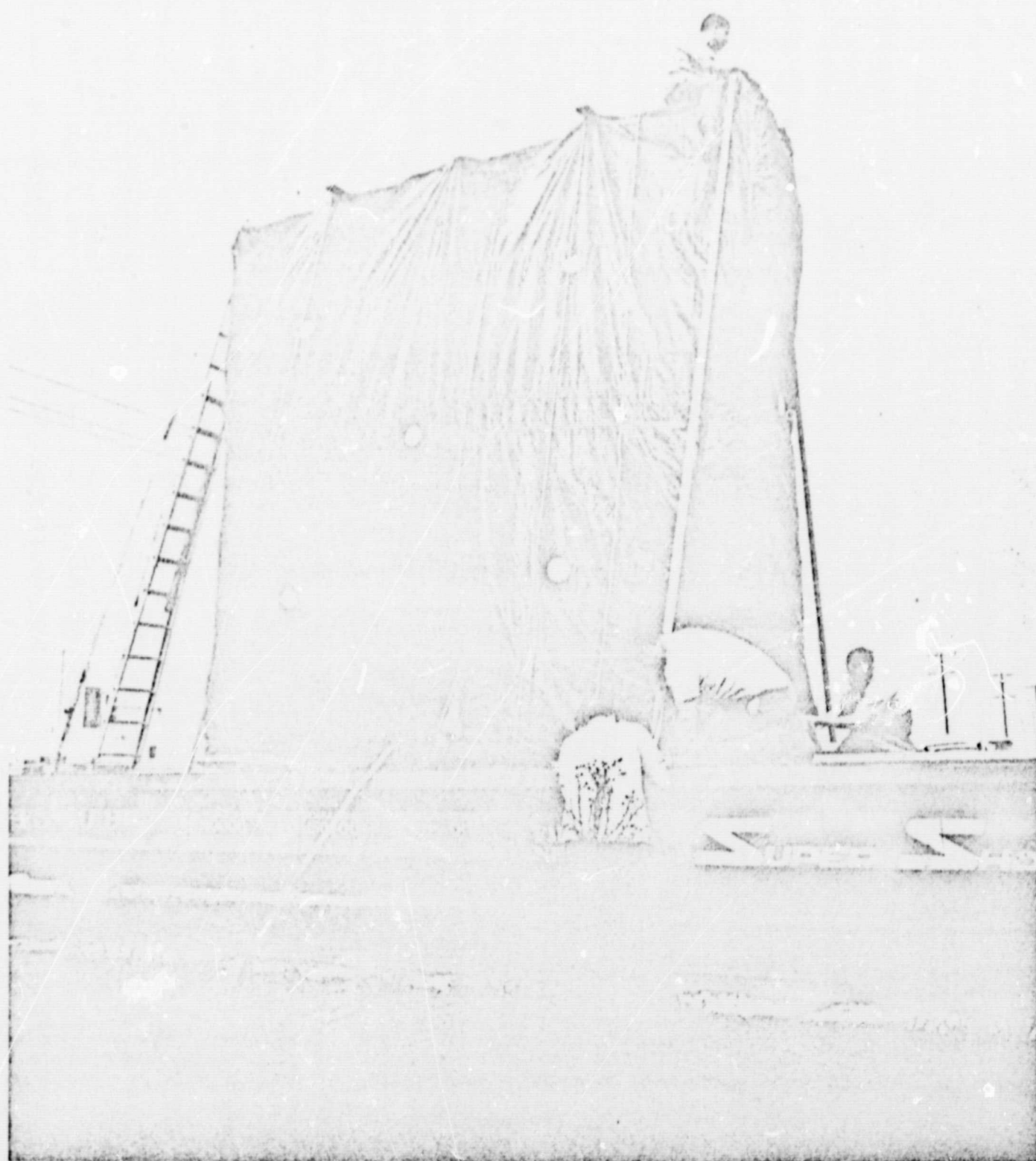


Figure 3-8. Heat Pipe Shipping Container Mounted on Rail Car

3.6 SUMMARY

In summary, it should be noted that the two units fabricated in this program represented the first major attempt at the manufacture of large scale heat pipe systems. Further, the time constraints prevented the substantial fabrication development effort which would normally accompany a program of this type. However, much has been learned during this program about reliable and economic fabrication of large-scale heat pipe systems which will be invaluable to subsequent programs. Much, in fact, was learned during the fabrication of the first heat pipe (ammonia) which substantially increased confidence in the manufacture of second pipe (water). The information presented in this section is, therefore, intended to provide sound foundation for subsequent development efforts on large scale heat pipe systems.

4. HEAT PIPE SYSTEM TESTING

Two types of thermal testing were formally required on the heat pipe system as part of this program. The first, called Thermal Verification Testing, was to demonstrate acceptable operation of each unit under nominal heat loads in an ambient environment. This test was performed at TRW/Redondo Beach prior to shipment and repeated at NASA/MSFC to identify any damage resulting from handling and shipping. The second test required was a Thermal Vacuum Test to be performed in the NASA/MSFC Chamber A facilities. Planning and conduct of the Verification Testing was the responsibility of TRW Systems (although NASA personnel actually performed the measurements at MSFC). The Thermal Vacuum Testing was the responsibility of NASA although TRW personnel were on hand to aid in the planning and conduct of the effort. Data evaluation was the responsibility of TRW Systems in both instances.

Before initiation of the verification testing, however, a substantial amount of development testing was necessary to;

- Determine if the two heat pipe systems were operational and under what conditions.
- Establish the appropriate liquid inventories.
- Reduce cold fronts in the water pipe presumably caused by generation of non-condensable gas during self-passivation.
- Determine the effects of the Neon gas charge on the performance of the ammonia pipe.

These tests and their results are described in this section. As will be seen, a number of changes to the basic pipes were necessitated by these results. The changes are summarized in a chronology of the pipe configurations leading up to Thermal Verification and Thermal Vacuum Testing.

As is clearly shown in the following sections, heat pipe systems of the type described in this report are in a development state and much is to be learned about the operational characteristics of these units. To this end, the thermal vacuum testing cannot be considered a demonstration of heat pipe systems, but an indepth study of the operational

characteristics of the units in a simulated space environment. Since analytical predictions for these type heat pipe systems are not fully established (particularly transient behavior), data from these tests will be invaluable in the design and application of future large scale heat pipe systems for manned spacecraft.

Before describing the test programs and their results, a word is necessary about the need to level the heat pipe systems. As noted in Section 2, the artery in the ammonia pipe can, theoretically, prime only .9" in a one "g" environment. Since the pipe I. D. was 1/2 inch and some recognition must be given to manufacturing tolerances, a leveling requirement of 1/8" of flat was established to assure artery priming (the water pipe requirement was 1/2 inch). Once the artery has primed, the pipes can, theoretically, pump several inches against gravity under limited heat loads. However, after a break develops in the artery, the pipes must be again leveled before the heat pipe action can be restarted. These requirements were difficult to achieve for heat pipes systems as large as 15' in diameter and leveling posed one of the more serious testing problems.

4.1 INSTRUMENTATION

For purpose of easy identification, the ammonia pipe was given the designation A and the water pipe the designation W. Instrumentation pertaining to the vacuum chamber support stand has been labelled S. Individual 5' modules of each heat pipe are also labelled; e. g., A1, A2, W8. . . , to permit easy reference to any part of either pipe system. Figure 4-1 shows this designation; conceptually, the modules are numbered sequentially around the pipe (clockwise as seen from above) with the fill tube being between Sections 5 and 6. Heater identification is identical with module designations; e. g., heater A3 is on Module A3 (third module of the ammonia pipe). Connector pin locations for each heater are provided in Table 4-1. More specific information regarding heater design and application is presented in Section 2.4.7.

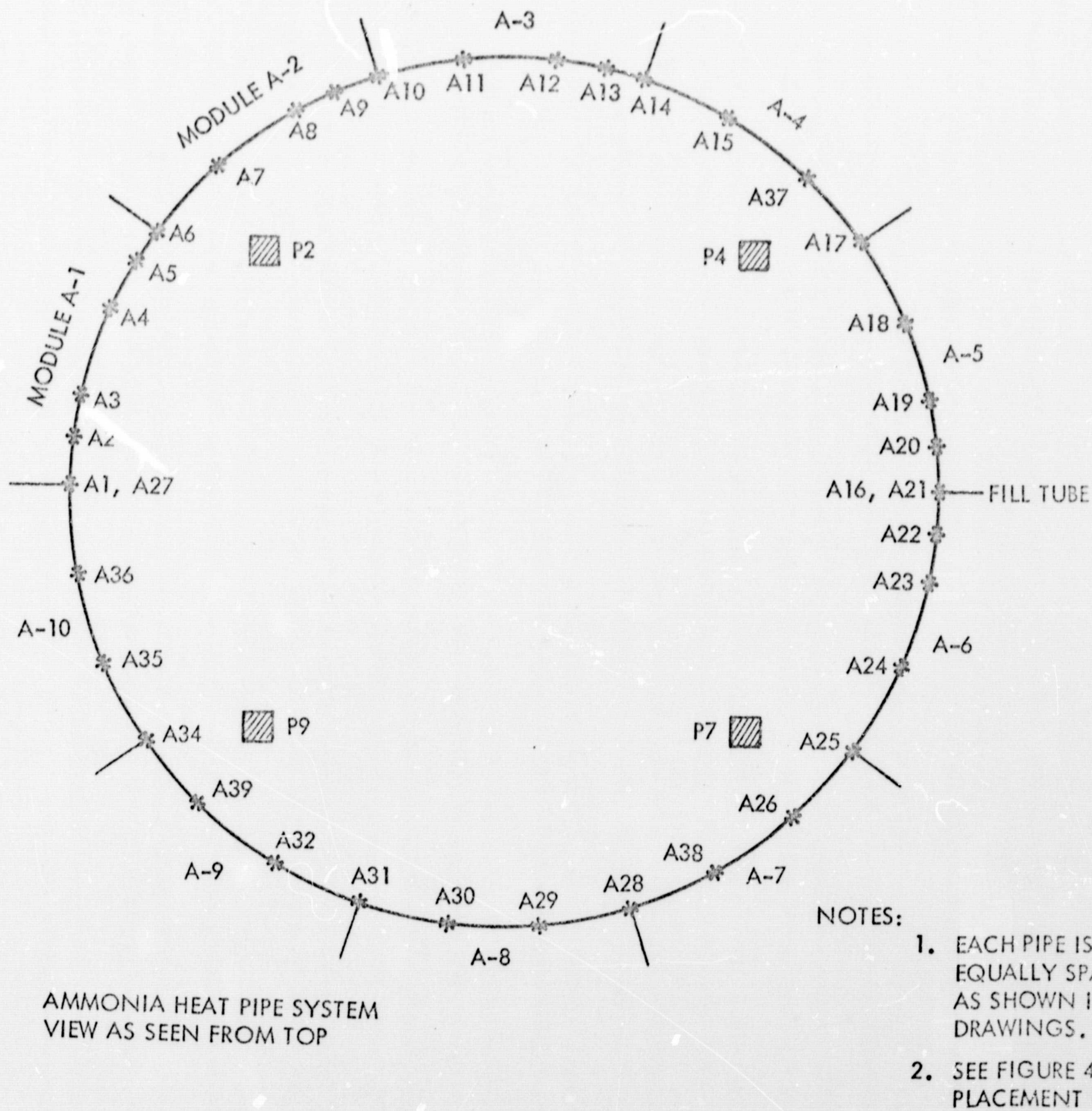
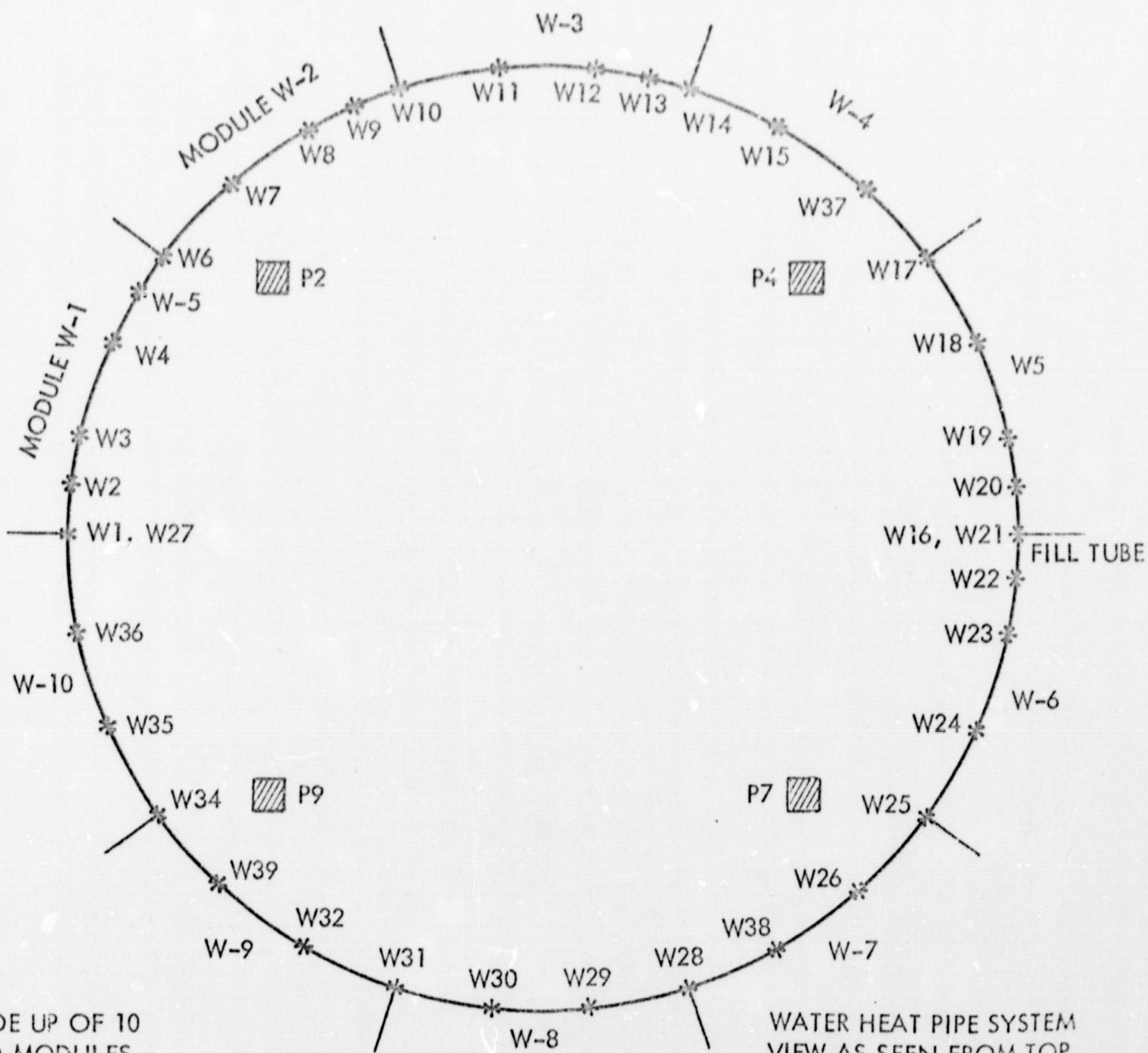


FIGURE 4-1: THERMOCOUPLE LOCATIONS -



* - THERMOCOUPLE LOCATIONS

▨ - VERTICAL SUPPORT POSTS

PIPE IS MADE UP OF 10
EQUALLY SPACED MODULES
SHOWN IN THE ABOVE
FIGURES.

FIGURE 4-2 FOR DETAILED
LAYOUT OF THERMOCOUPLES

FIGURES - NASA/MSC 15' HEAT PIPE SYSTEMS

Table 4-1. Pin Locations - Heat Pipe System Module Heaters and Grounds

Heater/Ground Identification	Pin Locations
<u>Heaters</u>	
A1	1 & 2
A2	3 & 4
A3	5 & 6
A4	7 & 8
A5	9 & 10
A6	11 & 12
A7	13 & 14
A8	15 & 16
A9	17 & 18
A10	19 & 20
W1	21 & 22
W2	23 & 24
W3	25 & 26
W4	27 & 28
W5	29 & 30
W6	31 & 32
W7	33 & 34
W8	35 & 36
W9	37 & 38
W10	39 & 40
<u>Grounds</u>	
Ammonia Pipe	41
Water Pipe	42

Thermocouples attached to the heat pipe units are also shown in Figure 4-1. Figure 4-2 shows the relative position of the pipe, fin and insulation thermocouples for a typical cross-section. As noted in Section 2.4.8, the pipe and fin thermocouples are spot welded or soldered to the surfaces to be measured. Figures 4-3 through 4-7 show the locations for the remaining thermocouples. Table 4-2 provides a summary of the locations with reference to the appropriate figure. Conducting epoxy was used to bond the thermocouples in place with aluminized tape over the top to reduce the measurement error.

Not all thermocouples were utilized in the Thermal Vacuum Testing. Table 4-3 presents the TRW designation, the NASA/MSC designation, the connector number and the pin location of each thermocouple utilized in the MSC test chamber. Pins 1 and 4 on each plug were used to ground the shielding.

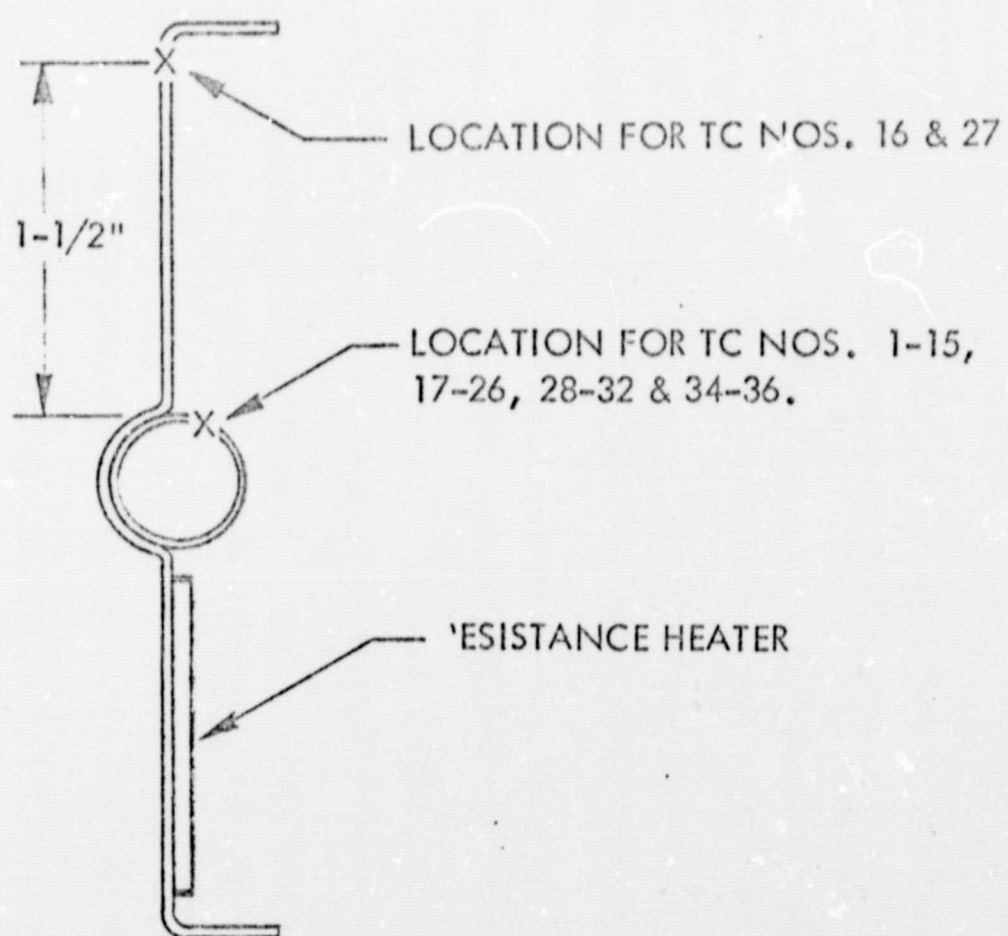
It is estimated that the copper-constantan thermocouples applied cannot be read with accuracy better than $\pm 1^\circ\text{F}$ at the Cannon connector. This means that all readings have an uncertainty of at least $\pm 1^\circ\text{F}$, and that the uncertainty of the reading/recording equipment must be added to this value. All stray EMF's generated between the connector and the reading equipment must also be added to the total for a final determination of the instrumentation error.

4.2 DEVELOPMENT TESTING

This section presents a detailed chronology of all full scale developmental testing activity associated with both heat pipe systems leading up to the thermal verification testing activities.

These data along with the environmental conditions associated with the tests show the failures as well as the successes, and are intended to pass on to the reader as much of the experience gained on this developmental program as possible. It is expected that these experiences (positive and negative) will be extremely valuable in planning and conducting the next large scale heat pipe program.

Section 3 describes the fabrication and processing of the two heat pipe systems up to and including the initial fill. The discussion in this section initiates at that point in the program.



SEE FIGURE 4-1 FOR PLAN VIEW OF TC'S.

Figure 4-2. Thermocouple Application for Typical Cross Section - NASA/MSFC 15-ft Heat Pipe System

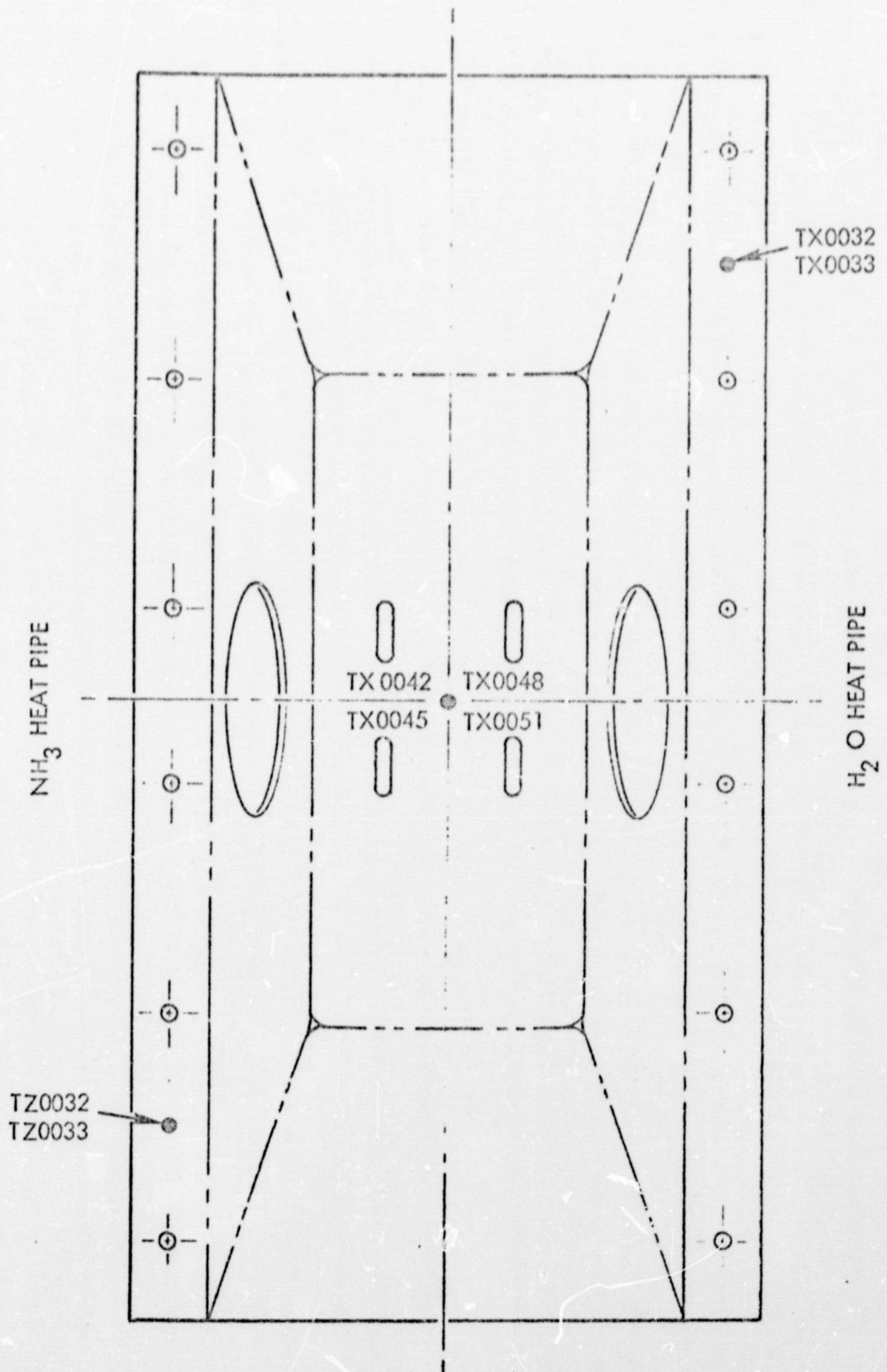


Figure 4-3. Thermocouple Locations - Fiberglass Support

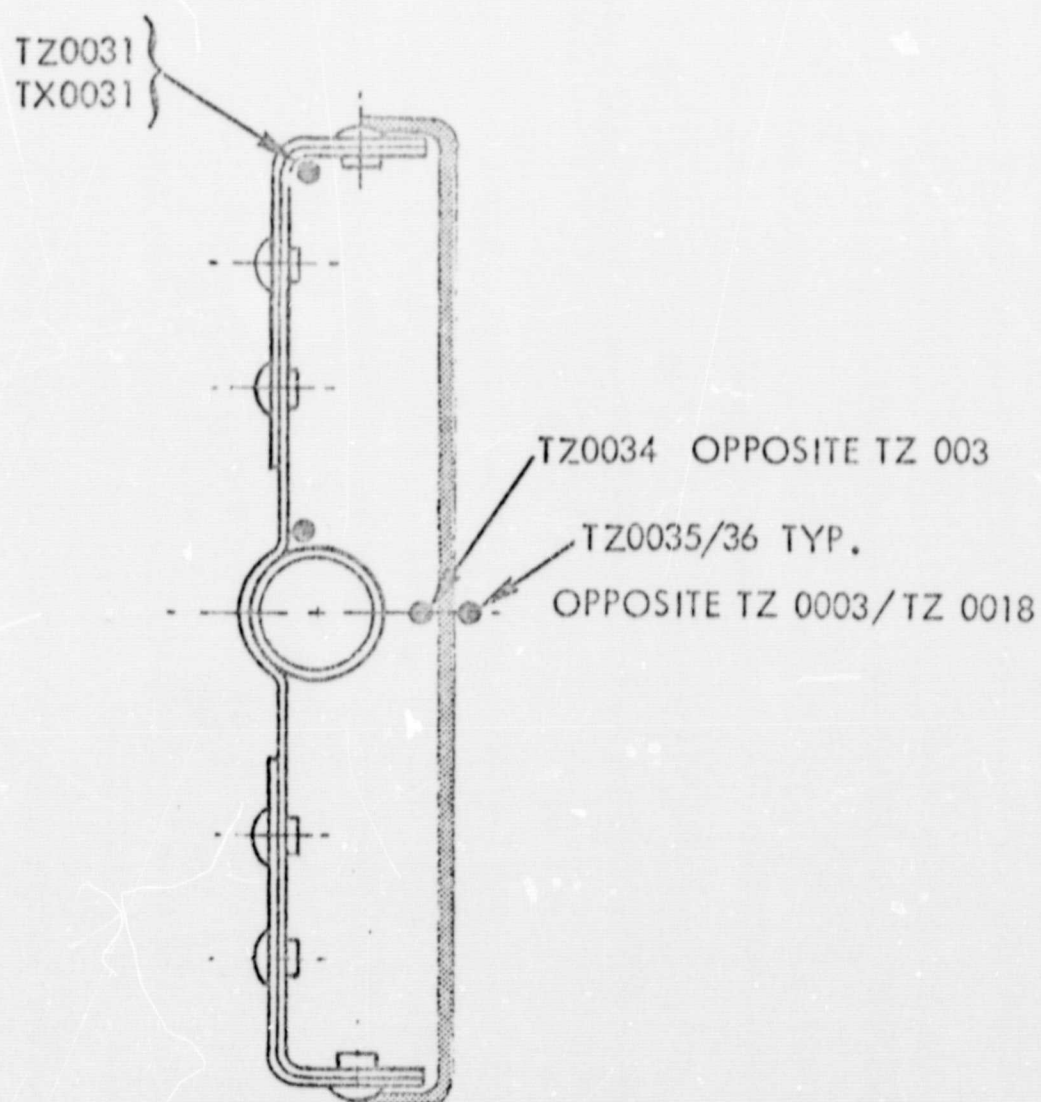


Figure 4-4. Thermocouple Locations - Multilayer Insulation

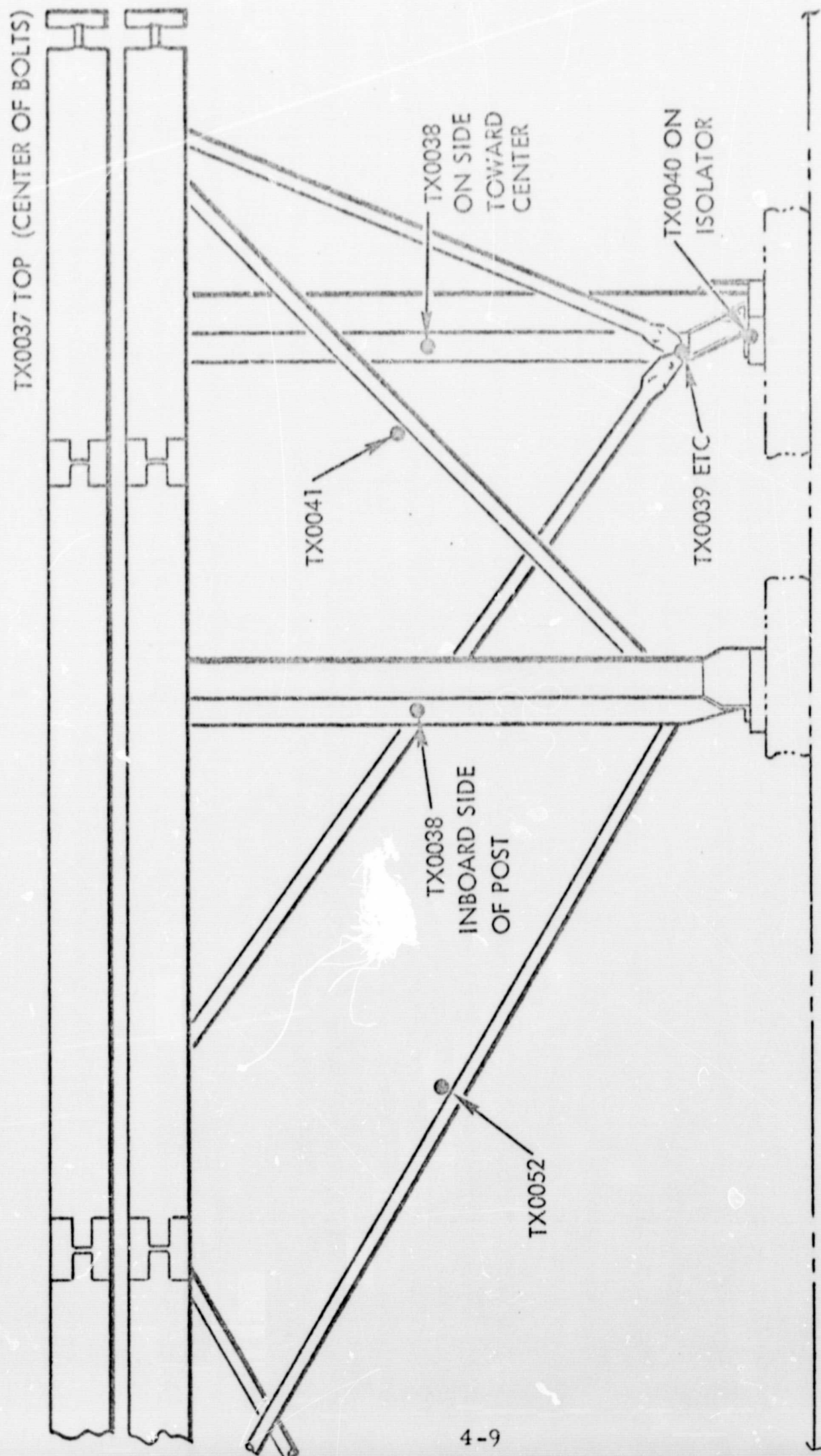


Figure 4-5. Support Stand Thermocouple Locations (Typical)

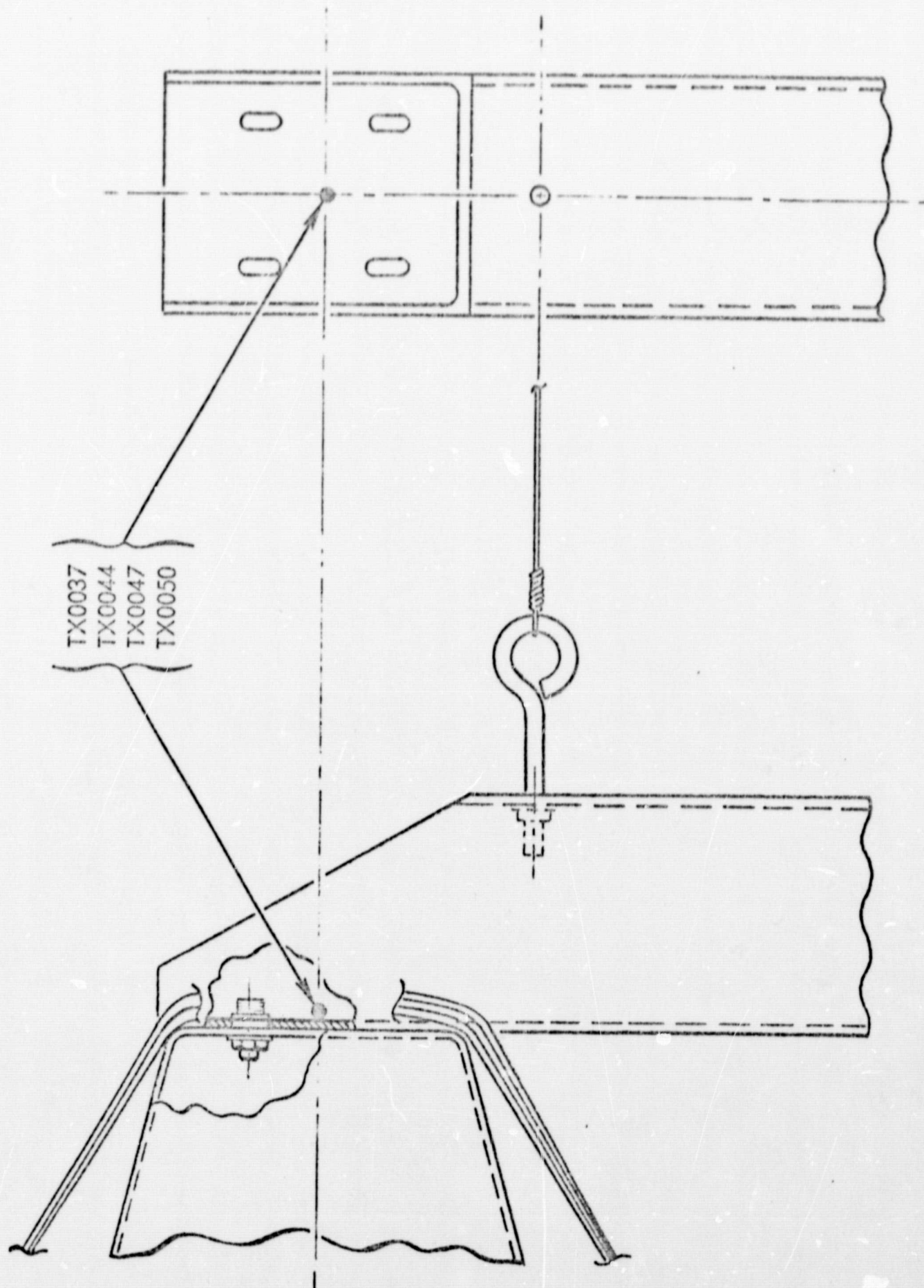


Figure 4-6. Support Stand Thermocouple Locations (Fiberglass Support)

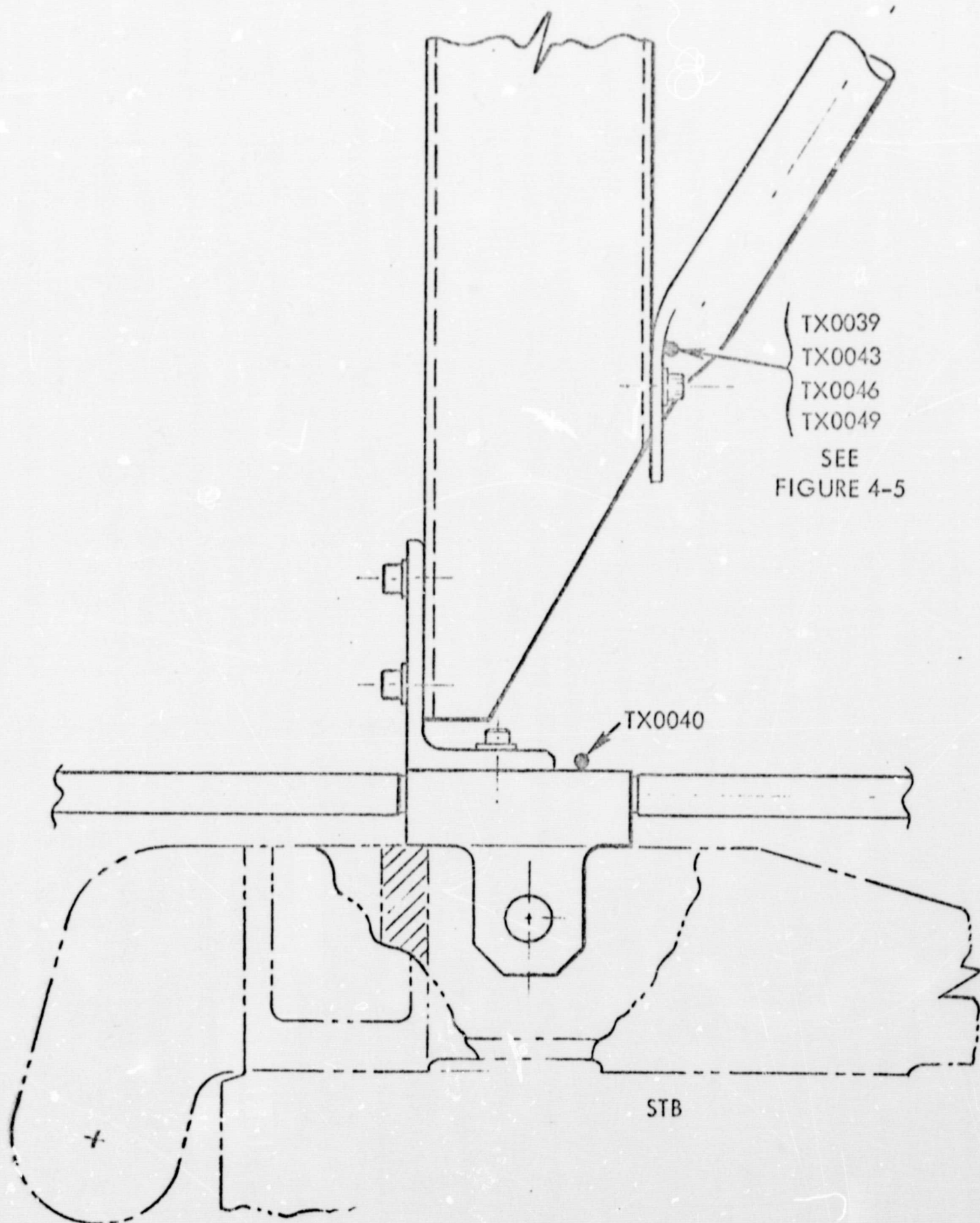


Figure 4-7. Support Stand Thermocouple Locations and STB Interface

Table 4-2. Physical Location-Auxiliary Thermocouples (S-Series)

TRW No.	MSC No.	Fig. No.	Location
S1	TX0033	4-3	Fiberglass Bracket between sections 1 and 10
S2	TZ0033	4-3	"
S3	TX0042	4-3	"
S4	TZ0031	4-4	Fin extremity near TC A4
S5	TX0031	4-4	Fin extremity near TC W4
S6	TZ0034	4-4	Inner blanket opposite TC A4
S7	TZ0035	4-4	Outer blanket opposite TC A4
S8	TX0041	4-5	Diagonal Post P2 to A1 (Midpoint)
S9	TX0032	4-3	Fiberglass bracket at post P2
S10	TZ0032	4-3	Fiberglass bracket at post P2
S11	TX0037	4-6	Post P2, Top
S12	TX0038	4-5	Post P2 (See Fig. 4-4)
S13	TX0039	4-7	Post P2 (See Fig. 4-4), Bottom
S14	TX0040	4-7	Post P2 (See Fig. 4-4), Isolator
S15	TX0051	4-3	Fiberglass brackets-Section 3
S16	TX0049	4-7	Post P4, Bottom
S17	TX0050	4-6	Post P4, Top
S18	TX0048	4-3	Fiberglass bracket-fill tube
S19	TZ0036	4-4	Outer Thermal blanket opposite A24
S20	TX0046	4-7	Post P7, Bottom
S21	TX0047	4-6	Post P7, Top
S22	TX0045	4-3	Fiberglass bracket-Section 8
S23'	TX0043	4-7	Post P9, Bottom
S24	TX0044	4-6	Post P9, Top
S25	TX0052	4-5	Diagonal Support-Post P2 to A3

Table 4-3. Summary of Heat Pipe Thermocouple/Cannon Plug Pin Locations

MSC No.	TRW Redondo No.	Pin Nos. CU CON		MSC No.	TRW Redondo No.	Pin Nos. CU CON	
TX0001	W1	11	12	TX0048	S18	40	39
TX0002	W3	2	3	TX0049	S16	29	30
TX0003	W4	14	15	TX0050	S17	43	42
TX0004	W6	5	6	TX0051	S15	44	69
TX0005	W7	17	18	TX0052	S25	46	45
TX0006	W8	8	9				
TX0007	W10	20	21	TZ0001	A1	11	12
TX0008	W11	34	33	TZ0002	A3	2	3
TX0009	W12	23	24	TZ0003	A4	14	15
TX0010	W14	37	36	TZ0004	A6	5	6
TX0011	W15	26	27	TZ0005	A7	17	18
TX0012	W37	40	39	TZ0006	A8	8	9
TX0013	W17	29	30	TZ0007	A10	20	21
TX0014	W18	43	42	TZ0008	A11	34	33
TX0015	W19	44	69	TZ0009	A12	23	24
TX0016	W21	46	45	TZ0010	A14	37	36
TX0017	W23	59	60	TZ0011	A15	26	27
TX0018	W24	49	48	TZ0012	A37	40	39
TX0019	W25	62	63	TZ0013	A17	29	30
TX0020	W26	52	51	TZ0014	A18	43	42
TX0021	W38	65	66	TZ0015	A19	44	69
TX0022	W28	55	54	TZ0016	A21	46	45
TX0023	W29	80	81	TZ0017	A23	59	60
TX0024	W30	71	72	TZ0018	A24	49	48
TX0025	W31	85	84	TZ0019	A25	62	63
TX0026	W32	74	75	TZ0020	A26	52	51
TX0027	W39	88	87	TZ0021	A38	65	66
TX0028	W34	77	78	TZ0022	A28	55	54
TX0029	W35	91	90	TZ0023	A29	80	81
TX0030	W36	103	102	TZ0024	A30	71	72
TX0031	S5	94	93	TZ0025	A31	85	84
TX0032	S9	106	105	TZ0026	A32	74	75
TX0033	S1	97	96	TZ0027	A39	88	87
				TZ0028	A34	77	78
TX0037	S11	11	12	TZ0029	A35	91	90
TX0038	S12	2	3	TZ0030	A36	103	102
TX0039	S13	14	15	TZ0031	S4	94	93
TX0040	S14	5	6	TZ0032	S10	106	105
TX0041	S8	17	18	TZ0033	S2	97	96
TX0042	S3	8	9	TZ0034	S6	109	108
TX0043	S24	20	21	TZ0035	S7	100	99
TX0044	S23	34	33	TZ0036	S19	112	111
TX0045	S22	23	24				
TX0046	S20	37	36				
TX0047	S21	26	27				

Note: Pin numbers 1 and 4 are ground pins for the cable bundle shield.

4.2.1 Water Heat Pipe System

The initial fill of the water heat pipe unit was 350 cc's and represented a 12% excess over a calculated requirement (312 cc) which was based on the idealized geometry of the system and on subscale developmental testing. Subsequent testing under "one sun" conditions* and with the pipe level gave a runaway condition; i. e., Section 1 was dry and was 20-30°F hotter than the other sections. The water inventory was then increased by 200 cc and the pipe retested in a level condition. Table 4-4 presents the results of these tests.** As can be seen, the pipe successfully handled the "one sun" input condition but had a substantial cold front (W19 → W26) which is approximately 96" in length. The true length cannot be determined accurately since the ends of the front are undoubtedly between thermocouple locations.

It was speculated, but not experimentally confirmed, that the cold front was caused by non-condensable gas generated as the tubes became self-passivated (the screens had been passivated prior to final fabrication). Based on discussions with the NASA/MSFC Technical Monitor, it was decided to keep the water pipe operational until the reaction had gone to completion and to "burp" the pipe once prior to final closure.

Prior to "burping" to minimize the cold front, however, a number of tests were conducted to determine the operating characteristics of the water pipe. These tests consisted initially of power density tests where maximum heat flux (54 watts) was applied individually to each section to see if boiling occurred in the artery and if there were manufacturing flaws in any section. Table 4-5 presents the results of these tests. A single thermocouple reading per section was taken during these tests to reduce the time required. A complete set of thermocouple readings (35) were taken with 54 watts applied to Section W3. Note that the cold front, although more difficult to distinguish because of the low temperature, was

* One sun conditions in this discussion is taken to mean 54 watts on Section 1, 43.7 watts on Sections 2 and 10, and 16.7 watts on Sections 3 and 9 (Total = 174.8 watts). These values were selected to approximate the magnitude and cosine distribution of the solar input.

** The heat pipes were not insulated during preliminary testing. Thus the thermocouples were exposed to ambient drafts, etc., causing some loss in accuracy.

Table 4-4. Temperature Profile - Water Pipe

Date: January 9, 1970

Conditions: One Sun Input, Max Heating on Section W1
Pipe Level

TC No.	Temp. of	TC No.	Temp. of
W1	93	W19	80
W2	94	W20	76
W3	95	W21	74
W4	94	W22	74
W5	94	W23	75
W6	94	W24	74
W7	94	W25	75
W8	95	W26	78
W9	95	W27	96
W10	94	W28	94
W11	95	W29	93
W12	94	W30	94
W13	95	W31	95
W14	95	W32	74
W15	94	W33	--
W16	67	W34	95
W17	94	W35	95
W18	95	W36	95

$P(W1) = 54$ watts
 $P(W2) = 43.7$ watts
 $P(W10) = 43.7$ watts
 $P(W3) = 16.7$ watts
 $P(W9) = 16.7$ watts

Table 4-5. Temperature Profile Water Pipe Power Density Test

Section TC No	1	2	3	4	5	6	7	8	9	10	Date
Heated Section	W3	W7	W11	W15	W18	W23	W26	W29	W32	W35	
W1	85	81	81	81	78	76	81	81	81	81	1/19/70
W2	81	81	81	81	81	81	77	79	81	81	
W3	80	80	82	80	80	80	80	76	77	80	
W4	81	81	81	81	81	81	81	82	77	81	
W5	80	80	80	80	81	81	81	80	77	77	
W6	75	80	80	80	80	81	80	80	80	80	
W7	80	75	80	80	80	81	81	81	81	81	
W8	81	81	76	81	81	82	82	82	82	82	1/20/70
W9	82	82	82	77	82	82	82	82	82	82	
W10	81	81	81	81	77	81	81	81	82	82	
TC	W1	W2	W3	W4	W5	W6	W7	W8	W9	W10	W11
Temp. °F	78	79	80	79	79	79	80	79	78	79	82
TC	W12	W13	W14	W15	W16	W17	W18	W19	W20	W21	W22
Temp °F	80	79	79	80	80	79	80	79	79	79	79
TC	W23	W24	W25	W26	W27	W28	W29	W30	W31	W32	W33
Temp °F	80	79	79	80	79	75	76	74	74	77	-
TC	W34	W35	W36	Section W3 Heated							
Temp °F	79	80	79								

Conditions: Pipe level, 54 watts applied to the heated section

still approximately the same (100") after ten days of nearly continuous operation. Boiling in the artery in this test series would have resulted in pronounced temperature rise of the heated section; in no test did this situation occur.

Table 4-6 shows the results of the "one sun" condition applied to the fill tube area (Section W6 = 54.0 watts, Sections W5 and W7 = 43.7 watts, Sections W4 and W8 = 16.7 watts). These data show the operation of the system as a heat pipe at evaporator elevations (Section 6) of 1/4", 1/2" and 3/4". The unit had clearly suffered an evaporator dryout at 1" elevation. Actually, it may be argued that the pipe had started to run away at 3/4"; the temperature on Section A6 had increased 4°F in about 75 minutes.

Theoretically, the water pipe should have operated at elevations up to 8.3 inches under this load. Failure at 3/4" elevation suggests that the arterial wick only generated about 23 percent of the theoretical capillary head. Thus, it appears that the artery was not perfectly sealed, reducing the capillary pumping head achievable. The fact that the pipe did operate in an elevated heat pipe mode, however, indicates that the artery did prime and function, albeit not as well as hoped.

When tested at higher heat loads in a level position, the water heat pipe operated successfully at 1.25 "suns" but quickly suffered a hydrodynamic failure at 1.5 "suns". This result suggests an actual capillary head equal to about 25 to 30 percent of theoretical. This is in close agreement with the elevation tests described above, lending support to the thesis that the artery was not perfect, seriously reducing the pipe's heat transport capacity. It did, however, appear to be sufficient to handle the design load of 1 sun at anticipated operating temperatures.

A series of "burping" operations were then performed to reduce the cold front prior to final closure of the water pipe. It was noticed during this effort that the front length changed very little with each burp leading to the speculation that the front was caused by excess liquid or that the noncondensable gas, if any, was hung up in the artery and couldn't get out to the vapor space (or to the fill tube area). However, the pipe was operated level, in the heat pipe mode (+1/2") and the refluxing mode (-1/2") during these burping operations without substantial change in the front. This supported the preliminary conclusion that the front was indeed

Table 4-6. Temperature Profile - Water Pipe - Heat Pipe Mode

Section TC No	1	2	3	4	5	6	7	8	9	10	Date 1/22/70
Level Operation	W3	W7	W11	W15	W18	W23	W26	W29	W32	W35	
	92	93	92	93	94	92	94	94	92	92	
1/4" Elevation Heat Pipe Mode	82	91	92	92	93	93	93	93	92	93	
1/2" Elevation Heat Pipe Mode	78	89	89	90	90	91	90	89	90	89	
3/4" Elevation Heat Pipe Mode	77	91	91	90	90	95	91	91	90	90	
1" Elevation Heat Pipe Mode	76	88	89	89	89	106	90	89	89	89	After 75 minute

Complete Temperature Profile - Level Operation

TC No.	W1	W2	W3	W4	W5	W6	W7	W8	W9	W10	W11	W12	W13
Temp. °F	93	93	92	78	82	93	93	93	94	94	92	94	94
TC No	W14	W15	W16	W17	W18	W19	W20	W21	W22	W23	W24	W25	W26
Temp. °F	94	93	95	95	94	94	94	93	94	93	94	94	94
TC No.	W27	W28	W29	W30	W31	W32	W33	W34	W35	W36			
Temp. °F	92	94	94	94	94	92	-	94	92	94			
Heaters	55 watts W6 43.7 watts W5 & W7 16.7 watts W4 & W8												

gas, and pointed up what may be a serious problem with arterial heat pipes. In charging such heat pipes with inert gas, either for start-up from the frozen state, or for variable conductance control, there exists the potential for some of the gas to enter the artery. If this happens, the flowing vapor is isolated from the gas by the artery wall, and the gas is not pumped to the end of the condenser. Thus, a much larger portion of the condenser can be blocked by gas than anticipated. Also, as in this case, it becomes very difficult to purge the heat pipe of noncondensable gas by the common practice of "burping" it.

The cold front had been reduced to about 40" in length prior to final sealing; in the chamber, this length would be reduced to about 20" (if gas) because of the reduced temperature of the "shut off" section. During the sealing operation, however, a leak developed in the closure causing the water pipe to fill with air. It was, consequently, totally non-operative. Opening the pipe required that the fill tube be cut off and the positive closure replaced with a vacuum valve.

The system was again pumped down and baked out at about 175°F for approximately 2 hours before refilling with 470 cc's of degassed water. The water heat pipe with this charge would not handle the "one sun" condition level without runaway; no cold front was observed. Addition of 250 cc's more of degassed water caused the pipe to operate properly (level and in the heat pipe mode). However, a cold front of approximately 86" was observed in the level condition. Subsequently, the pipe was tipped to the refluxing position and, at an elevation of 4" the front disappeared completely. It could then be concluded that excess liquid inventory was the cause of the cold front.

The volume of the vapor space in the water pipe is about 0.159 in³/in of length and 86" will hold approximately 225 cc's of liquid. It then appears that the 470 cc's initially charged in the pipe was only slightly below the required inventory and that most of the 250 cc's added stayed in the cold area and, in fact, bridged the vapor space to form a plug. We were unable to disperse this plug by "chasing" the cold front with heaters. Also, after operating the unit in a reflux mode (4"), the front quickly reformed at the cold zone when the system was leveled.

Based on this information, the excess liquid was diminished by centering the cold zone on the fill area and "burping" with a vacuum pump through a LN_2 cold trap. A total of 200 cc's were taken out leaving approximately 520 cc's and a cold front of around 14". It is concluded that there are still about 35 cc's excess water in the pipe. Since some excess is necessary to assure a saturated artery; this was judged acceptable. The vacuum valve was then closed and the line mechanically pinched off before the valve; i.e., the valve was between the pinch off and the pipe. This was the final water pipe configuration shipped to NASA/MSFC; test data showing its operation for the "one sun" condition are presented in the next section.

It should be noted that all development test data collected in the laboratory environment for the water pipe are valid provided the water pipe had sufficient fluid inventory to be operational as a heat pipe. Substantially, the only differences from one fill condition to another were the lengths of the cold front. Consequently, the data presented earlier of maximum flux in the level condition (power density tests) and the loads at varying elevations are also considered applicable to the final configuration of the water heat pipe system.

4.2.2 Ammonia Heat Pipe Systems

The initial fill of the ammonia heat pipe unit was with 300 cc's and represented a 20% excess over the calculated requirement.

As noted previously, the initial fill procedure utilized a fixed volume reservoir in which NH_3 was frozen prior to distilling in the heat pipe itself. This approach may not have given a true indication of the liquid volume since the density of the frozen ammonia is strongly dependent on whether the NH_3 freezes as "ice" or "snow". Subsequent fills of NH_3 were made by liquifying in the reservoir rather than freezing.

The system was not operational with the initial NH_3 fill and it was decided to add another 300 cc's. After considerable difficulty in getting the heat pipe level to within $1/8"$ ($\pm 1/16"$), the unit was able to run in the "one sun" condition in the level position; Table 4-7 shows equilibrium temperature data taken for this condition. No data were recorded with the ammonia pipe operating in a true heat pipe mode with the heated end elevated.

Table 4-7. Temperature Profile - NH₃ Pipe

Date: January 9, 1970

Condition: One Sun Input, Maximum Heating on Section A1
Pipe Level, No. Inert Gas

TC No.	Temp. of	TC No.	Temp. of
A1	88	A19	87
A2	89	A20	85
A3	90	A21	87
A4	95	A22	87
A5	94	A23	87
A6	88	A24	87
A7	88	A25	85
A8	91	A26	88
A9	90	A27	93
A10	88	A28	89
A11	87	A29	88
A12	88	A30	86
A13	88	A31	88
A14	88	A32	89
A15	87	A33	--
A16	87	A34	88
A17	87	A35	93
A18	88	A36	92

P(A1) = 54 watts

P(A2) = 43.7 watts

P(A10) = 43.7 watts

P(A3) = 16.7 watts

P(A9) = 16.7 watts

A series of power density tests similar to those reported for the water pipe were conducted on the system with the pure ammonia charge. As before, 54.0 watts power was input sequentially to each of the sections; temperature data for these tests are presented in Table 4-8. Inadvertently no data were taken with Section A3 heated. The data in Table 4-8 show a tendency to be hotter at the heated zone but the system did not runaway and did not nucleate. In all instances, the pipe was level.

An objective of this program was to investigate the application of a non-condensable gas into the ammonia heat pipe system to aid in start-up from a frozen state. Conceptually, as explained in Section 2, the gas expands throughout the pipe as the NH_3 cools and freezes. Upon reheating, the gas front contracts slowly restricting the heat pipe action to the heated area until all the NH_3 melts and restart is accomplished. This concept was utilized in the ATS-E Solar Cell heat pipes, apparently with success.

Unfortunately, the addition of a non-condensable gas in the NH_3 pipe can have a deleterious effect in a concentric artery system. Calculations show (see Section 2) that the nucleate boiling limit of NH_3 in the presence of gas may be extremely low. In the case of a concentric artery system, nucleation is disastrous; vapor generation in the artery will eventually cause the artery to empty in the evaporation section and the pipe to fail completely. After discussion with the NASA/MSFC Technical Monitor, it was decided to charge the system, and to experimentally determine if these fears were real. With a molecular weight of 20 (NH_3 is 17), neon was selected for the inert gas fill. This fill was accomplished by measuring the pressure of the pure NH_3 charge and adding a overpressure of 4.0 psi. The rationale and anticipated results associated with this neon charge are presented in Section 2.

Table 4-9 presents the temperature distribution within six hours and 24 hours after introduction of neon gas. The pipe was operated level with the following heat inputs:

54.0 watts	-	Section A1
43.7 watts	-	Sections A2 and A10
16.7 watts	-	Sections A3 and A9

Table 4-8: Temperature Profile—NH₃ W/O Neon—Power Density Tests

Section— TC No. Heated Section	1	2	3	4	5	6	7	8	9	10	Date	
A1	80	78	77	78	78	78	78	78	79	79	1/14/70	
A2	80	80	83	80	80	80	80	79	80	80	1/16/70	
A3												
A4	83	83	83	87	84	83	84	83	84	84		
A5	84	84	84	84	87	84	84	84	84	84		
A6	81	81	81	81	81	82	81	81	81	81		
A7	78	78	79	79	79	78	84	80	79	79		
A8	78	78	78	78	78	78	78	85	79	79		
A9	78	78	78	78	78	78	78	78	82	79		
A10	79	79	79	79	79	79	79	76	79	81		
Section 9 Heated	TC °F Temp.	A1 77	A2 76	A3 78	A4 76	A5 76	A6 76	A7 78	A8 77	A9 77	A10 77	1/19/70
	TC °F Temp.	A11 78	A12 77	A13 77	A14 77	A15 78	A16 78	A17 77	A18 78	A19 76	A20 76	
	TC °F Temp.	A21 76	A22 76	A23 78	A24 76	A25 77	A26 78	A27 77	A28 77	A29 78	A30 77	
	TC °F Temp.	A31 77	A32 82	-	A34 77	A35 79	A36 77					

Conditions: Pipe Level, 54.0 watts applied to heated section.

Table 4-9. Temperature Distribution - NH_3 Pipe with Neon Fill

Date: 1/20 & 21/70

Conditions: One Sun Input, Max. Heating on Section A1
Pipe Level

TC No.	Section	Temperature $^{\circ}\text{F}$	
		After 6 hours	After 24 hours
A3	1	91	94
A7	2	90	92
A11	3	89	90
A15	4	89	91
A18	5	89	91
A23	6	89	91
A26	7	78	81
A29	8	76	77
A32	9	90	91
A35	10	92	93
A16	Fin	88	89

A1 = 54.0 watts

A2 = 43.7 watts

A10 = 43.7 watts

A3 = 16.7 watts

A9 = 16.7 watts

The ammonia heat pipe did not runaway nor was there any indication of nucleation. However, left operational at "one sun" over a full weekend, the pipe failed and could not be restarted in the level position. Apparently sufficient neon had accumulated in the evaporator to cause nucleation and pipe failure. The measured cold zone was approximately 10' long which was almost twice the anticipated length under ambient test conditions. This was likely due to a portion of the gas residing behind the artery where it could not be pumped by flowing vapor, as discussed in the last section. In addition, it is very difficult to accurately control the quantity of gas introduced when adding it to a pipe already filled with a two-phase fluid. This typically results in the addition of too much gas.

It was concluded at this point that the ammonia heat pipe with a neon charge would not operate successfully at the required heat load, and the decision was made to remove the gas prior to final closure. Three sections of the pipe were immersed in LN_2 to freeze the NH_3 and the fill tube connected to a vacuum pump. The smell of ammonia at the outlet of the pump, however, indicated that not all the ammonia had been frozen. Removal of the neon was eventually accomplished by completely emptying the pipe (neon + ammonia) and refilling with a pure ammonia charge.

The final ammonia fill consisted of 450 cc's (138 cc theoretical excess) and was done using the fill procedure outlined in Section 3. The fixed volume reservoir was chilled with dry ice and methanol to condense the NH_3 from the container. Ammonia was not frozen in the reservoir prior to fill into the evacuated pipe. Once the ammonia fill had been completed, three sections of the heat pipe were immersed in LN_2 to reduce the pressure and the final closure was made. This closure did not include a valve as with the water pipe because of the hazard of ammonia.

Due to schedule pressure, the ammonia heat pipe system was then not operated for several days while final painting and insulation were completed. When turned on, it was found that the unit would not operate as a heat pipe even at 30% of the "one sun" condition and, further, that the failure appeared to be attributable to nucleation in the artery. Steady growth of a cold front with time served to confirm that a substantial amount of gas was being generated in the pipe.

The gas front it was felt could only have been caused by the presence of water in the pipe. Ammonium hydroxide was created which, in turn, attacked the passivated stainless steel screen and pipe. Ultimately, if left unchecked, the reaction could corrode through the wall and dump ammonia to the environment with a considerable safety hazard. Consequently, the pipe was opened, the ammonia emptied, a valve put on the fill tube and the system evacuated.

Accidental introduction of water in the pipe must have come from the water column used to capture NH_3 during the emptying process (see Figure 3-3). During a normal dumping operation, NH_3 is "bubbled" into the water column forming NH_3OH to minimize atmospheric contamination. In all cases the ammonia was first condensed from the pipe to the reservoir before opening the valve to the water; the heat pipe valve was always closed prior to the dump. As the pressure in the reservoir decreases, however, it is possible for water to back up into the reservoir where it can remain until the next ammonia fill into the pipe. It is postulated that water was introduced into the ammonia pipe by this process.

At this point, time did not permit another attempt to fill the pipe with ammonia. The fill tube had already been used (it can only be sealed once). Even if an adapter were welded over it to allow another seal, the pipes had been painted and insulated which prohibited the freezing operation necessary with the closure technique used. Finally, the alternative of filling with ammonia and using a pinch-off or valve seal was deemed unacceptable from a safety point of view.

It was decided after discussions with NASA/MSFC to refill the empty ammonia pipe with methanol. This was accomplished and the pipe closed in the same way as the water pipe (valve plus pinchoff). No thermal data were collected on the methanol system due to the shipping schedule. Calculations indicate that self-priming of methanol in a 0.012" artery is marginal. In any case, the thermal load which can be handled at 70°F is minimal (~ 0.3 suns). Further, the performance drops off sharply with temperature. Consequently the methanol filled, ammonia heat pipe was not expected to provide meaningful data in the thermal vacuum testing but can perhaps be utilized in laboratory testing at partial sun loads.

4.3 THERMAL VERIFICATION TESTING

The primary objective of the Thermal Verification Testing was to provide a simple means to verify acceptable performance of the heat pipe systems for nominal operating conditions. By acceptable operation is meant:

- The maximum power density associated with one sun incident on one 5' module does not induce boiling in the artery.
- The unit is capable of operating level with one sun incident on one side of the 15' diameter cylinder with an overall ΔT not exceeding 10°F over the active length.*

In all cases the tests were conducted in air and utilized heaters and thermocouple instrumentation installed for the thermal vacuum test in the NASA/MSC chamber A test facility (see Section 4.1).

The thermal verification testing was not intended to provide extensive data regarding the operating characteristics of the heat pipe system under a wide range of environmental conditions. These data must be provided by development testing of the type described in Section 4.2 or the thermal vacuum testing described in Section 4.4.

4.3.1 Test Plan and Procedure

A detailed test plan and procedure for the thermal verification testing is presented in Appendix A; the overall philosophy and general procedure are repeated in this section. Instrumentation and heater placement are presented in Section 4.1 of this report.

Thermal verification testing calls for two different types of tests to be performed. In the first, or power density test, 54 watts thermal energy was supplied to a single section with all other heaters off. This value represents the maximum flux absorbed by one module with the sun

* Active length is defined to be the working portion of the heat pipe and does not include that portion blanketed with excess liquid inventory or by non-condensable gas.

incident perpendicular to the fin. If the units do not experience nucleation or boiling in the artery under this condition, we are assured that it will not happen for the normal operating condition with one sun on half the circumference of the cylinder.

For initial testing at TRW/Redondo Beach, each module was subjected to the power density test to determine the manufacturing variance from one module to the next. Subsequent verification testing at NASA/MSC was conducted by examining only selected modules.

In the second series of tests, electric heaters were powered to give one solar absorbed load over half the circumference of the pipe. A total of five heaters were used and adjusted to approximate the cosine distribution of solar input. The following table shows the distribution based on a fin surface coating having α_s/ϵ of 1.0 and $\alpha_s = 0.25$:

<u>Location</u>	<u>Power, watts</u>
Center Heater (e. g., Module 1)	54.0
Adjacent to Center (e. g., Modules 2 and 10)	43.7(2)
Outer Heaters (e. g., Modules 3 and 9)	<u>16.7(2)</u>
Total	174.8

Sections 1, 2 and 10, 3 and 9 were selected since they were opposite the fill tube area and were to receive the solar input for the major amount of the Thermal Vacuum Testing at NASA/MSC.

The heat pipe units were operated level for both portions of the Verification Testing described here. It is not intended that this test will serve to demonstrate operation of the units as heat pipes; this demonstration was provided as part of the development testing described in Section 4.2.

A detailed plan and procedure for the conduct of the Verification Testing is provided as Appendix A of this report. The interested reader is referred to this area for more specifics on these tests.

4.3.2 Analytical Predictions

Analytical predictions of the pipe performance in air are, of course, heavily dependent on the laboratory environment and on the heat

transfer characteristics of multilayer insulation. The latter was intended for vacuum operation and is fairly ineffective in ambient conditions except to block convection currents. An additional uncertainty is the position and extent of the excess liquid or non-condensable gas cold front. The subsequent analysis, therefore, neglects the cold front; the length of this front was experimentally determined.

The input heat flux assumed in the predictions for the Verification Testing is shown in Figure 4-8. The profile is symmetrical and only half is shown. It was estimated that both pipes would operate near 100°F ; fluid properties were therefore evaluated at this condition. Figure 4-9 presents the calculated temperature drop for the water pipe for the "one sun" verification condition. Note that the temperature drops across the artery (evaporator and condenser ends) account for 5.0°F of the predicted total of 5.6°F . The vapor temperature (pressure) drop is only 0.6°F . The total predicted ΔT is, thus, very sensitive to the assumptions of heat transfer into the system. For the calculations presented in Figure 4-9 it was conservatively assumed that heat input through the artery occurred only over $1/3$ of the circumference of the pipe.

Predicted temperature drop of the ammonia pipe for the "one sun" verification conditions is shown in Figure 4-10. The maximum predicted ΔT is 5.0 and is almost totally due to heat transfer across the artery; ΔT due to vapor pressure drop is negligible for these conditions. Consequently, the predicted ΔT for the ammonia pipe is subject to the same uncertainties as the predictions for the water pipe.

4.3.3 Test Results

4.3.3.1 Water Heat Pipe System

Results of the power density tests on the water heat pipe system are reported and discussed in Section 4.2; these results are repeated in Table 4-10 for convenience. As expected, it is clear that nucleation was not a problem with this unit.

Results for the "one sun" heat input condition are shown in Table 4-11. As noted, the maximum input (54.0 watts) occurred on Section W1. Sections W2 and W10 had 43.7 watts, and Sections W3 and W9 had

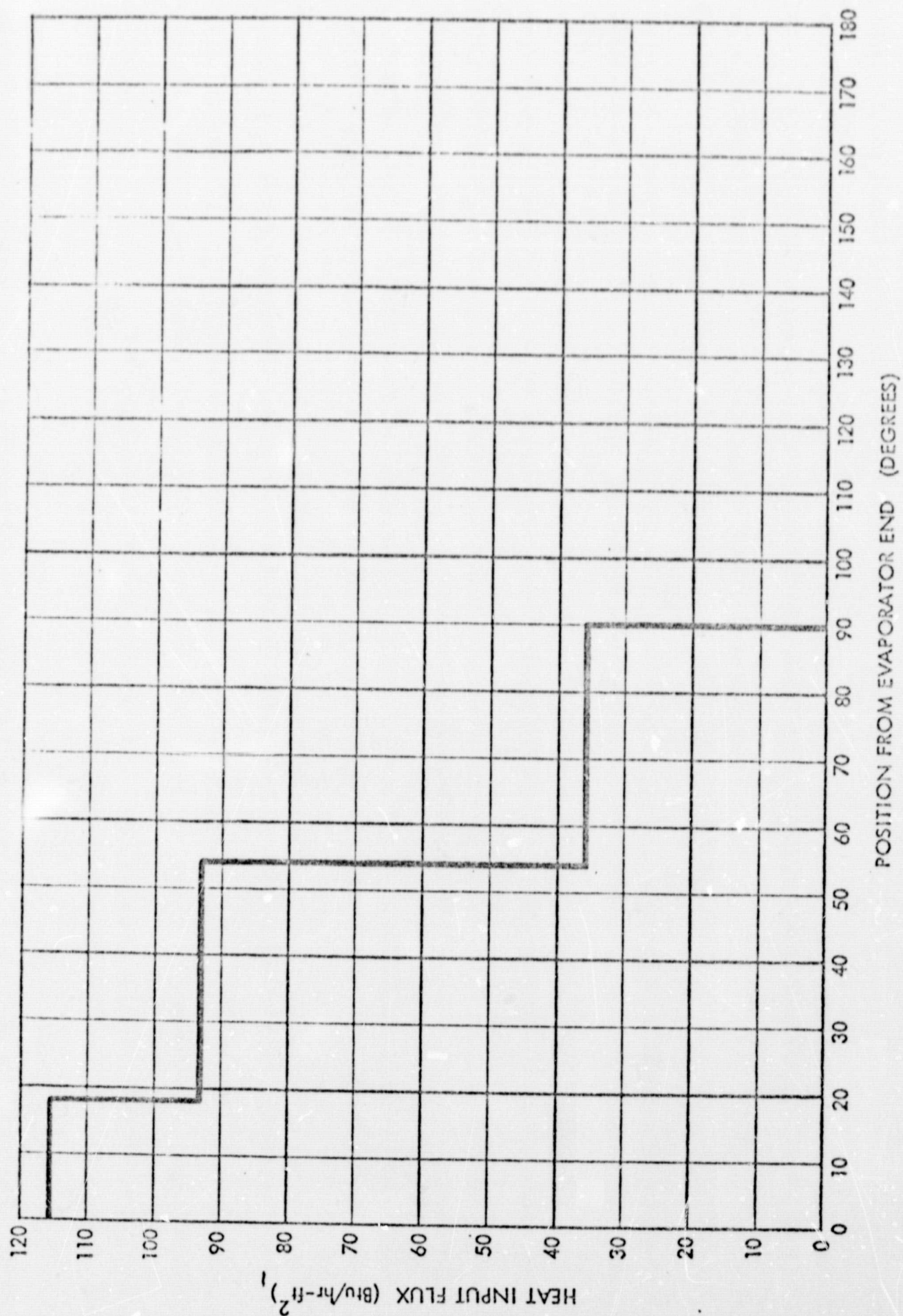


Figure 4-8. Electric Heater Input - Heat Pipe Thermal Verification Testing

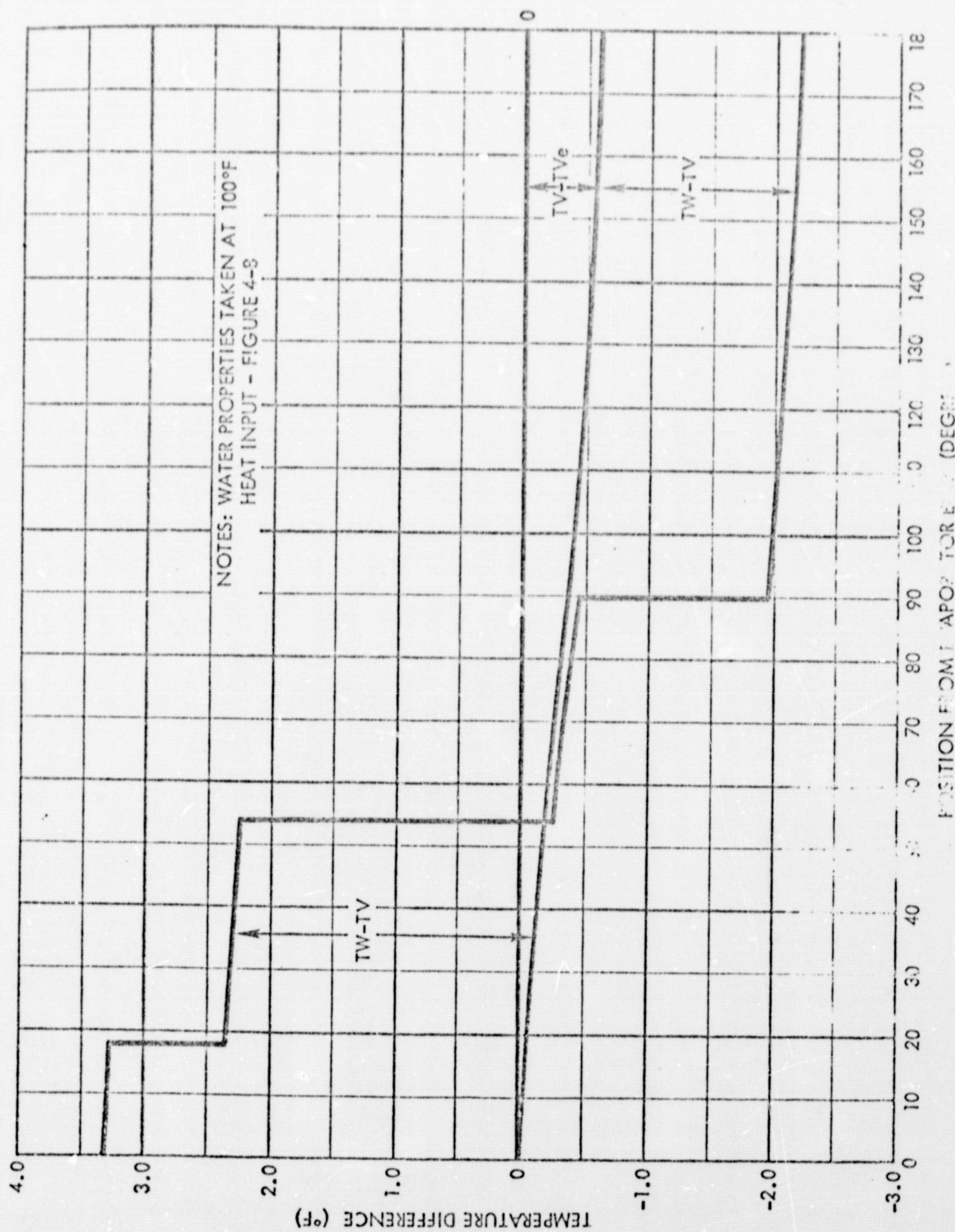


Figure 4-9. Calculated Temperature Distribution Water Heating Pipe Thermal Verification Testing

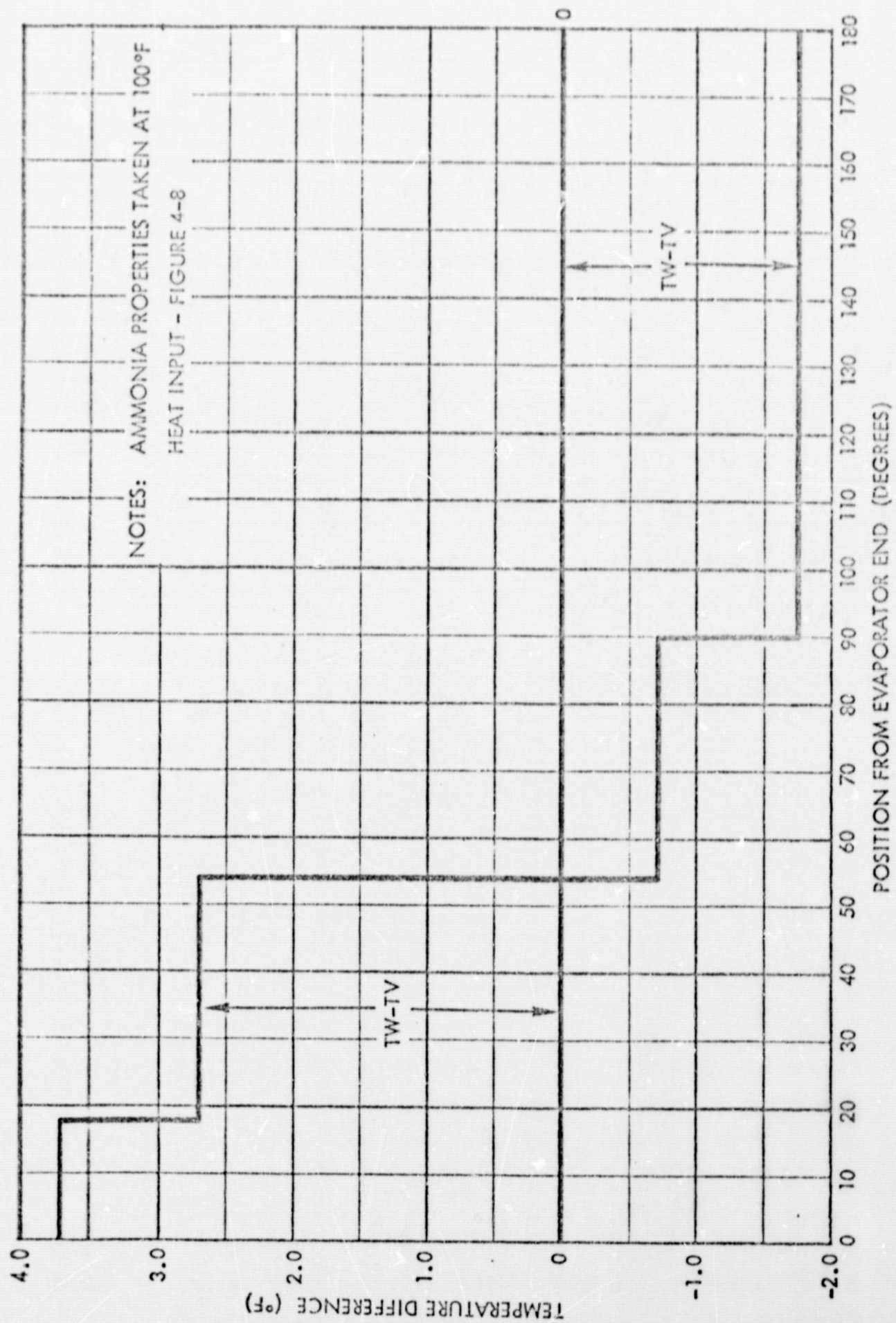


Figure 4-10. Calculated Temperature Distribution - Ammonia Heat Pipe Thermal Verification Testing

Table 4-10. Temperature Profile Water Pipe - Power Density Test (Thermal)
Verification

Section	1	2	3	4	5	6	7	8	9	10	Date
TC No. Heated Section	W3	W7	W11	W15	W18	W23	W26	W29	W32	W35	
W1	85	81	81	81	78	76	81	81	81	81	1/19/70
W2	81	81	81	81	81	81	77	79	81	81	
W3	80	80	82	80	80	80	80	76	77	80	
W4	81	81	81	81	81	81	81	82	77	81	
W5	80	80	80	80	81	81	81	80	77	77	
W6	75	80	80	80	80	81	80	80	80	80	
W7	80	75	80	80	80	81	81	81	81	81	
W8	81	81	76	81	81	82	82	82	82	82	1/20/70
W9	82	82	82	77	82	82	82	82	82	82	
W10	81	81	81	81	77	81	81	81	82	82	
TC °F Temp.	W1 78	W2 79	W3 80	W4 79	W5 79	W6 79	W7 80	W8 79	W9 78	W10 79	W11 82
TC °F Temp.	W12 80	W13 79	W14 79	W15 80	W16 80	W17 79	W18 80	W19 79	W20 79	W21 79	W22 79
TC °F Temp.	W23 80	W24 79	W25 79	W26 80	W27 79	W28 75	W29 76	W30 74	W31 74	W32 77	W33 --
TC °F Temp.	W34 79	W35 80	W36 79								

Section W3
Heated

Conditions: Pipe Level, 54.0 watts applied to the heated section

16.7 watts. The maximum ΔT shown is $2F^\circ$ with all but three readings being $92^\circ F$ (the instrumentation system had an estimated uncertainty of at least $\pm 1F^\circ$ for these temperature readings). It is clear therefore, that the overall ΔT is substantially below the predicted value of $5.6^\circ F$, a not unexpected result since the calculations in Section 4.3.2 were extremely conservative in nature.

It should also be noted that the water heat pipe system had a 12-14" cold front caused by excess liquid inventory. This front does not appear in the data of Table 4-11 because it was located between thermocouples. The front was identified by carefully feeling along the pipe and noting which areas had a temperature significantly below the rest of the pipe.

4.3.3.2 Ammonia (Methanol) Heat Pipe System

No verification data were collected after filling the ammonia heat pipe system with methanol. All data obtained for the ammonia filled pipe were presented in Section 4.2.

4.4 THERMAL VACUUM TESTING

The heat pipe thermal vacuum tests had three primary objectives: 1) to investigate the steady-state operation and transient dynamic response of large heat pipes under simulated spacecraft conditions, 2) to explore the potential fluid freezeout problem due to vapor diffusion into a portion of the condenser blocked by noncondensable gas (see Section 2.2.1.5), and 3) to investigate the feasibility of using noncondensable gas to effect start-up of very large heat pipes from a frozen state (see Section 2.2.1.4).

The first two objectives were successfully achieved with the water heat pipe. The third objective could not be satisfactorily accomplished for reasons discussed later in this section.

The thermal vacuum test program embodied several tasks other than the T/V tests themselves. These included inspection of the heat pipes after delivery to NASA/MSC, verification testing at MSC in the laboratory, post test data reduction, analysis of results and comparison with analytical predictions. Documentation of some of these tasks plus the thermal vacuum test timeline is presented in the Appendices of this report. This section will deal only with a brief description of the experiment and selected test results which are pertinent to an understanding

Table 4-11. Thermal Verification Test Results Water Heat Pipe System

Heat Inputs: Section W1 54.0 watts (2.78 amps)
 Section W2 & W10 43.7 watts (2.50 amps)
 Section W3 & W9 16.7 watts (1.55 amps)

TRW TC No.	NASA TC No.	Pins	Temp. (°F)	TRW TC No.	NASA TC No.	Pins	Temp. (°F)
W1	TX0001	11&12	92	W21	TX0016	45&46	92
W3	TX0002	2&3	92	W23	TX0017	59&60	92
W4	TX0003	14&15	92	W24	TX0018	48&49	92
W6	TX0004	5&6	92	W25	TX0019	62&63	92
W7	TX0005	17&18	92	W26	TX0020	51&52	91
W8	TX0006	8&9	92	W38	TX0021	65&66	92
W10	TX0007	20&21	92	W28	TX0022	54&55	92
W11	TX0008	33&34	92	W29	TX0023	80&81	92
W12	TX0009	23&24	92	W30	TX0024	71&72	92
W14	TX0010	36&37	92	W31	TX0025	84&85	92
W15	TX0011	26&27	92	W32	TX0026	74&75	92
W37	TX0012	39&40	92	W39	TX0027	87&88	92
W17	TX0013	29&30	93	W34	TX0028	77&28	92
W18	TX0014	42&43	92	W35	TX0029	90&91	92
W19	TX0015	44&69	92	W36	TX0030	102&103	92

- Notes: 1) Temperature level depends on environmental conditions
 2) 12-14" cold front not shown on above data because centered between thermocouples
 3) Data collected 2/3/70

of the heat pipe's performance and its relationship to the primary test objectives previously identified.

4.4.1 Experimental Approach

The two full-scale experimental systems were tested with the Subsystem Test Bed (STB) in the MSC Space Environmental Simulation Laboratory Chamber "A" during the period March 10 - 23, 1970. Figure 4-11 depicts the heat pipe assembly in the "piggyback test" configuration atop the STB test vehicle.

Although both heat pipes were subjected to thermal vacuum testing, it had previously been established in the laboratory that the methanol filled "ammonia" heat pipe did not function adequately, and successful heat pipe operation was not anticipated. However, structurally the two heat pipes had to be mounted as a unit. Further, the nonoperating methanol-filled pipe would serve as a reference for evaluating the performance of the water heat pipe. Since the methanol heat pipe was not operational, discussion in this section is generally limited to the water heat pipe test results.

In all thermal vacuum testing, heat transfer from the heat pipes was primarily by radiation to the liquid nitrogen cooled chamber walls. Heat transfer to the heat pipes was provided either by electrical heaters (see Section 2.4.7), solar simulation or both.

Since the heat pipe test was "piggyback" and secondary to the STB test program, the heat pipe timeline was developed about the STB timeline. Solar simulation, cold soaks, roll and stationary modes were all dictated by the STB tests. However, it was possible during the numerous cold soak conditions to utilize the heat pipe heaters to generate various heat source distributions and power levels. Thus, the test program permitted the acquisition of data for a sufficient number of conditions to characterize the heat pipe's performance.

The heat pipe instrumentation is described in Section 2.4.8 and 4.1. However, some comment regarding the temperature measurements is necessary to enable proper interpretation of the data. The data acquisition system for the entire STB test was programmed to yield a resolution of approximately $\pm 3^{\circ}\text{F}$ for the thermocouples. This is not really adequate

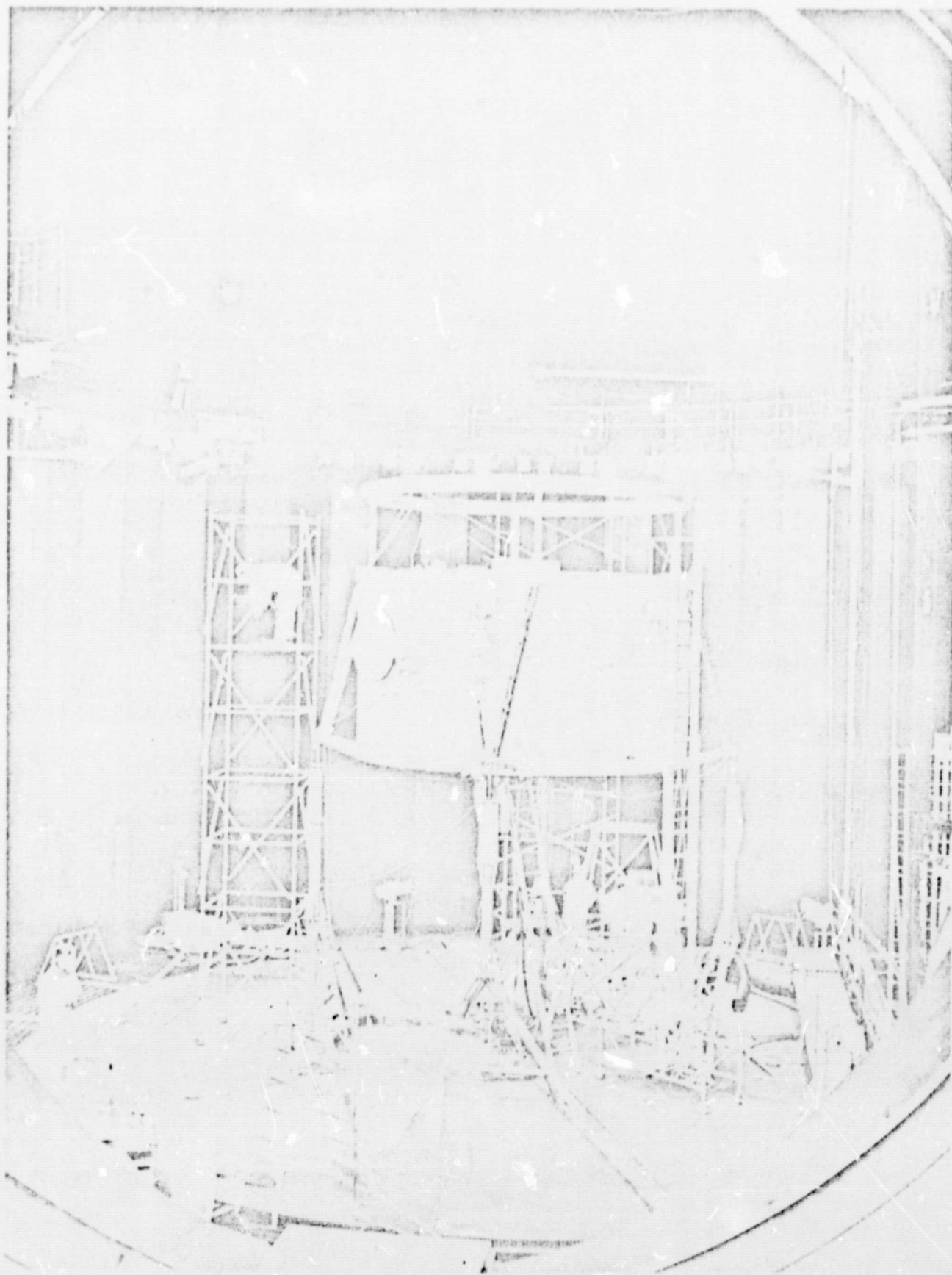


Figure 4-11. Heat Pipe Assembly Atop the STB Test Vehicle in the MSC Space Environmental Simulation Laboratory Chamber "A"

for heat pipe performance testing where one is looking for a much higher degree of isothermal operation. In an attempt to compensate for this situation, a data smoothing technique was employed. The data acquisition system recorded the thermocouple signals every second. The smoothing technique was to compute a line-averaged value for five consecutive readings of each thermocouple during the test periods of interest, and to plot this average as a single point on the graphs.

4.4.2 Test Results

The method of heat pipe test data presentation has been selected such that the temperature distribution about the heat pipe periphery are readily apparent at specific times. On all graphs the local temperature is plotted against the NASA thermocouple number. Since thermocouple Nos. 1 through 30 represent 30 thermocouples uniformly spaced around the heat pipe, the data graphs represent "developments" of the heat pipe. That is, position No. 1 and position No. 31 represent the same physical point on the heat pipe, and the abscissa is, essentially, a linear representation of circumferential position.

Each figure presents several temperature profiles of the heat pipe with time as the parameter. The code used in referring to time is the day number, hour (24-hour clock) and minute, (e.g., 70:23:02 means 23 hours and 2 minutes of the 70th day). This code is consistent with that used in the test timeline (Appendix) and allows correlation between the event and the overall timeline.

Superimposed on each graph are the heater loads in watts applied to each of the ten heat pipe modules during the particular test phase involved.

4.4.2.1 Cold Zone Due to Noncondensable Gas in the Pipe

The first test results obtained were during the initial pump down and chilling of the test chamber. A series of water heat pipe temperature distributions during the most significant portion of this test are presented in Figure 4-12.

Far reaching observations can be made from this graph. The first time slice (70:23:02) shows the heat pipe operating at near isothermal conditions except for the region between thermocouples 15 and 18. This cold zone, which is located diametrically opposite from the region of

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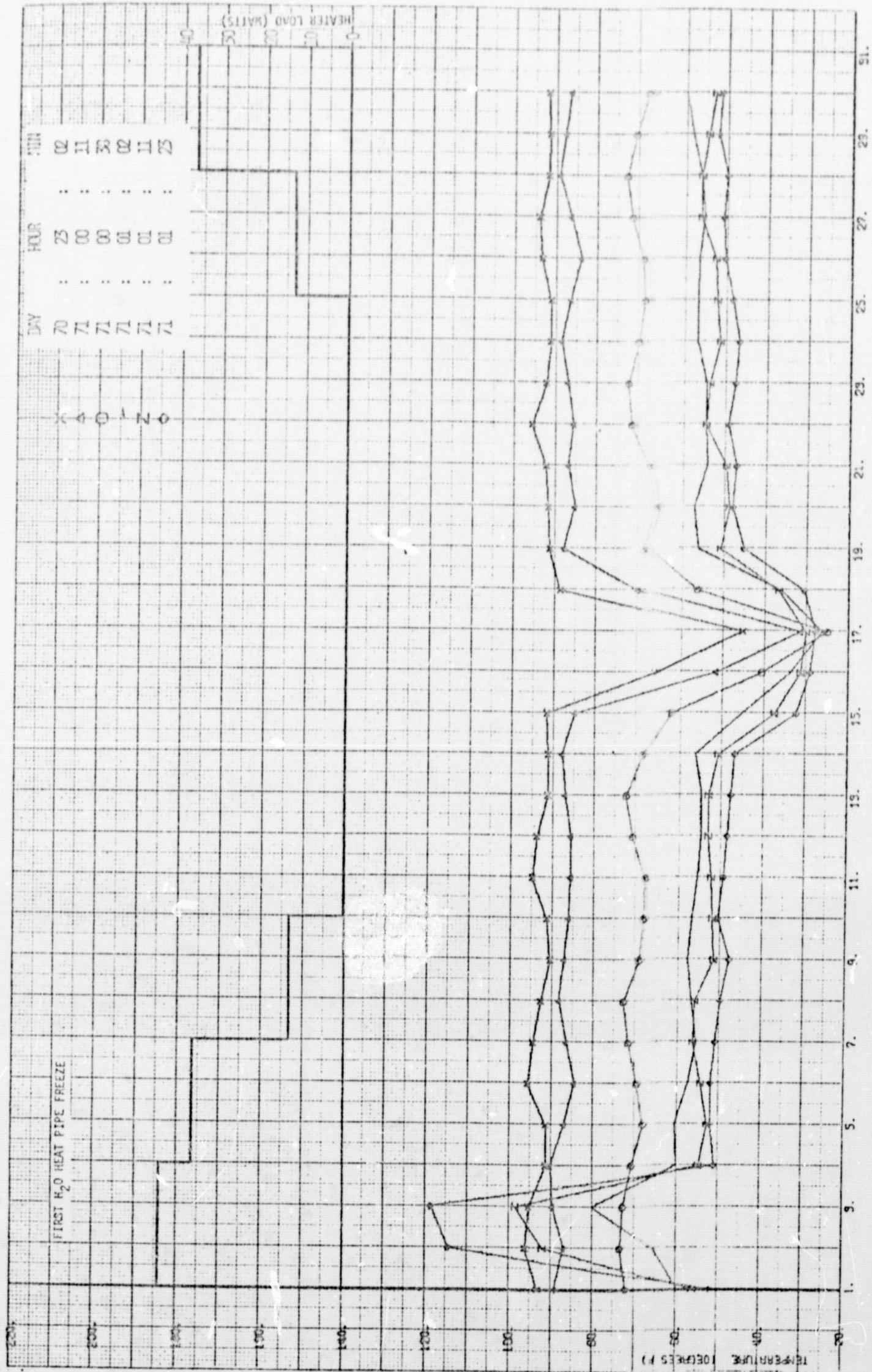


Figure 4-12. Pumpdown Phase, First H₂O Heat Pipe, Freeze

highest heat input, is characteristic of the presence of noncondensable gas in the heat pipe. The gas is pumped by flowing water vapor to the region opposite the heat input zone where it forms a gas slug. This gas slug acts as a diffusion barrier impeding the flow of water vapor into the region which it blankets. By reducing the partial pressure of water vapor in this region, the equilibrium saturation temperature, and hence the heat pipe wall temperature, falls to a lower value than the rest of the heat pipe.

The next two time slices on the graph (71:00:11 and 71:00:38) supply additional evidence for the presence of noncondensable gas. For a given quantity of gas in the heat pipe, the volume it occupies varies as an inverse function of the total pressure in the system. Thus, as the heat pipe operating temperature falls, the gas blankets an increasing portion of the heat pipe. This is clearly shown by the data.

Another situation which could yield a cold region on the heat pipe is a slug of excess fluid bridging the vapor core. As mentioned in Section 4.2.1, the heat pipe did contain a small quantity of excess fluid when shipped. However, if excess fluid caused the cold zone, it would not be expected to grow in size with decreasing heat pipe temperature. Thus, whether or not the heat pipe did contain some excess fluid, the data shows it clearly also contained some noncondensable gas.

Final verification of this gas load was obtained subsequent to this test program when the heat pipe was opened and its contents analyzed. This is discussed later in the report.

4.4.2.2 Diffusion Freeze-out

In addition to effecting a cold zone, the presence of noncondensable gas in a heat pipe has further implications when the heat sink temperature is below the freezing point of the working fluid. Depending on the magnitudes of the heat sink temperature, the quantity of gas in the pipe and the axial conductivity of the pipe and attached fins, it is possible to establish a condition where the wall temperature falls below the freezing point of the working fluid over a portion of the noncondensable gas slug region. Under these conditions, vapor continually diffuses through the gas to regions of lower vapor pressure, condenses and freezes on the wicks. Once frozen, it cannot be pumped back to the evaporator system and the

active portion of the heat pipe loses fluid inventory. If allowed to continue, this process will eventually cause the evaporator to dry out and the heat pipe will fail.

The actual time scale involved depends on the fluid freezeout rate, the fluid inventory in the heat pipe, and the nature of the wick. For homogeneous (conventional) wicks, partial desaturation can still allow satisfactory operation. However, for arterial wicks, the artery must be completely full. Any desaturation of an artery represents arterial failure and hence heat pipe failure. Thus, for example, the arterial wicks used in the heat pipes in this program could only afford a fluid loss corresponding to part of that held by the double layer of 200 mesh screen forming the annular artery. When this double layer of screen is depleted the artery fails. Unfortunately, this is a relatively small quantity of fluid, meaning that these heat pipes cannot tolerate a diffusion freezeout phenomenon for very long without failure.

This failure is clearly seen in Figure 4-12. The curves show that the coldest portion of the gas-blocked region reached the freezing point of water at 71:00:11. By 71:01:02, the evaporator temperature began to increase, indicating evaporator dryout and failure of the heat pipe.

Clearly diffusion freezeout phenomena in gas-loaded heat pipes is a serious problem which requires careful consideration in heat pipe design. Some techniques for avoiding this problem and their implications are discussed below:

- Eliminate noncondensable gas. Clearly, the problem exists only when noncondensable gases are present in the pipe. However, eliminating noncondensable gases is not always a satisfactory solution since they have been shown to be necessary to allow start-up of long heat pipes from a frozen state. If frozen start-ups are not anticipated (low freezing point and/or high effective sink temperature), the diffusion freezeout phenomenon will also be obviated.
- Provide for sufficient fluid inventory to allow the process to continue as long as necessary.

If the heat sources and sinks move around the heat pipes (e.g., slowly rotating spacecraft, changing sun position during orbit), it may be possible to tolerate some freezeout because this inventory will be melted and

recaptured as the active portion of the heat pipe moves over the frozen fluid. Thus, it simply becomes necessary to provide sufficient fluid inventory to accommodate the degree of freezeout anticipated.

Additional fluid inventory is a simple and straightforward solution with homogeneous wicks since, as mentioned earlier, they can operate in a partially desaturated state. In the case of arterial wicks, however, special provisions are required. It is possible, for example, to provide capillary excess fluid reservoirs, as developed by TRW for the Ames Heat Pipe Experiment, at various positions along the pipe. These reservoirs are small homogeneous wick structures with pores sized so that they retain excess fluid in the pipe but do not draw liquid from the primary wick or artery structure. Since these are homogeneous wicks which can operate in a desaturated mode, they will supply liquid to the primary wick when needed for freezeout but will retain this fluid when not frozen. Thus they permit filling the heat pipe with excess fluid without incurring adverse dynamic effects due to sloshing and without having the excess fluid pumped by flowing vapor to the cold region where it would simply freeze and be unavailable to serve its evaporator make-up function. Figure 4-13 shows schematically how one might employ such excess fluid reservoirs in circular heat pipes of the type studied in this program.

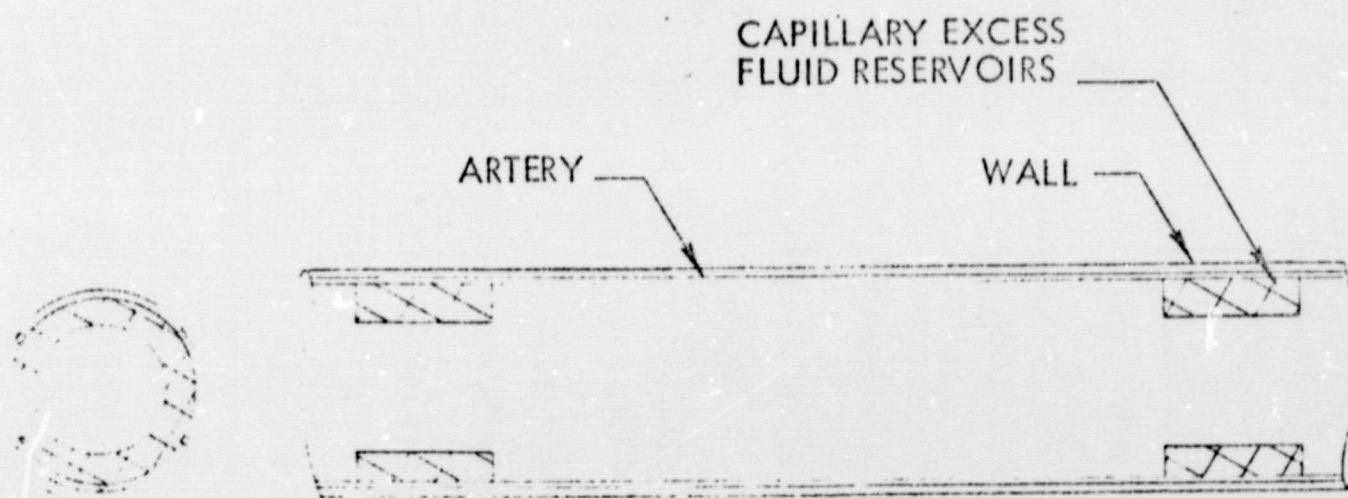


Figure 4-13. Schematic Diagram Shows Capillary Excess Fluid Reservoirs in Concentric Artery Heat Pipe

There was approximately 35 cc excess liquid in the water heat pipe system. However, this liquid was pumped by the vapor to the cold zone, froze and was lost to the system. The use of capillary reservoirs would, of course, prevent this condition. There must be a sufficient number of reservoirs so freezeout of one or two in the cold zone does not so delete the liquid inventory that the pipe fails.

- Assure sufficient axial conductivity in the pipe wall and fin such that the gas-blocked region does not fall below the fluid freezing point.

The noncondensable gas in the heat pipe impedes vapor flow into the gas-blocked region. However, heat is still conducted to this region, primarily by the pipe wall and fins. If this thermal conduction is high enough to prevent the pipe wall from falling below the freezing point of the working fluid, the freezeout problem cannot exist.

This approach is most promising where the fluid has a low freezing point and/or a high operating pressure. A low freezing point means the wall temperature can be permitted to fall close to the sink temperature minimizing the axial conduction requirements. A high operating pressure means the quantity of noncondensable gas required for frozen start-ups will be compressed to a small segment of the pipe, again minimizing the axial conduction requirements.

In the case of the water heat pipe, the fluid exhibits both a high freezing point and a low operating pressure. Thus, under design conditions, even the axial conductivity of the copper fin was insufficient to prevent a portion of the gas-blocked region from falling below the freezing point. Note, however, that this problem was recognized in the design phase of this program and the water heat pipe was not processed with a noncondensable gas inventory. The gas present in the pipe was a consequence of 1) leakage through the valve, 2) generation as a result of chemical reaction between the water and stainless steel, or both. Since analysis of the gas by MSC personnel subsequent to testing showed it to be nitrogen, it appears that leakage and not hydrolysis (which would yield hydrogen) was the gas source.

The design phase called for a gas-dosed ammonia heat pipe to study frozen start-up and the diffusion freezeout phenomenon. Ammonia has both a low freezing point and a high vapor pressure, making it a good candidate for this purpose. Unfortunately, as discussed in Section 4.2.2, the introduction of inert gas into the ammonia filled concentric annulus arterial heat pipe caused nucleation in the artery and pipe failure. This, in addition to other difficulties, resulted in not delivering a gas-dosed ammonia heat pipe. Thus, in one respect, the leakage of gas into the water heat pipe was fortuitous, for it provided a substitute with which to study the diffusion freezeout phenomena.

4.4.2.2.1 Side Sun Soak

A principal phase of the test program was to demonstrate the ability of large heat pipe systems to "isothermalize" a spacecraft under an incident solar load on one side. Thus, the test time-line included conditions wherein the heat pipes were held stationary and exposed to a simulated one sun side load without power input from any of the electrical heaters.

Theoretically, under these conditions the heat pipes should serve to transport absorbed heat from the sun side to all portions of the pipe for re-radiation, resulting in near isothermal operation.

Results of this test for the water heat pipe are shown in Figures 4-14a and 4-14b. It is apparent that the heat pipe did not perform as desired. However, it is clear that the reason for this failure is the same diffusion freezeout phenomenon discussed earlier. As seen in Figure 4-14a, the operating temperature of the heat pipe was so low under these heat input conditions that a portion of the gas-blocked region fell below the freezing point (32°F). Thus, as vapor diffused into the gas zone and froze, the evaporator became depleted of working fluid and exhibited a dryout condition.

The anticipated steady-state operating temperature of the heat pipe under solar simulation conditions was calculated to be 65°F assuming $\alpha_s / \epsilon = 0.217 / 0.237$ for the fin surface (Section 2.4.6). However, due to

H2O HEAT PIPE VACUUM TEST

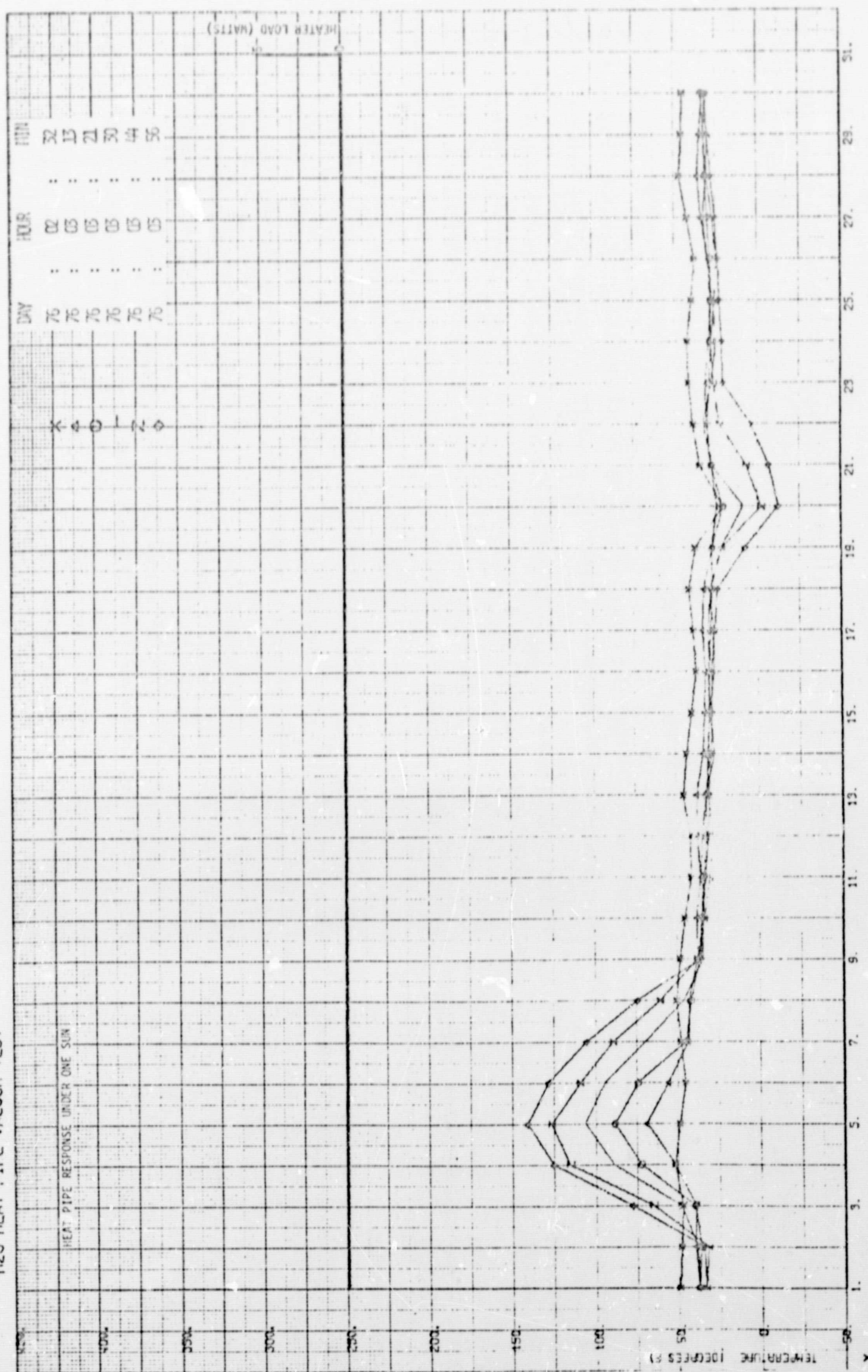


Figure 4-14a. Side-Sun Soak Phase

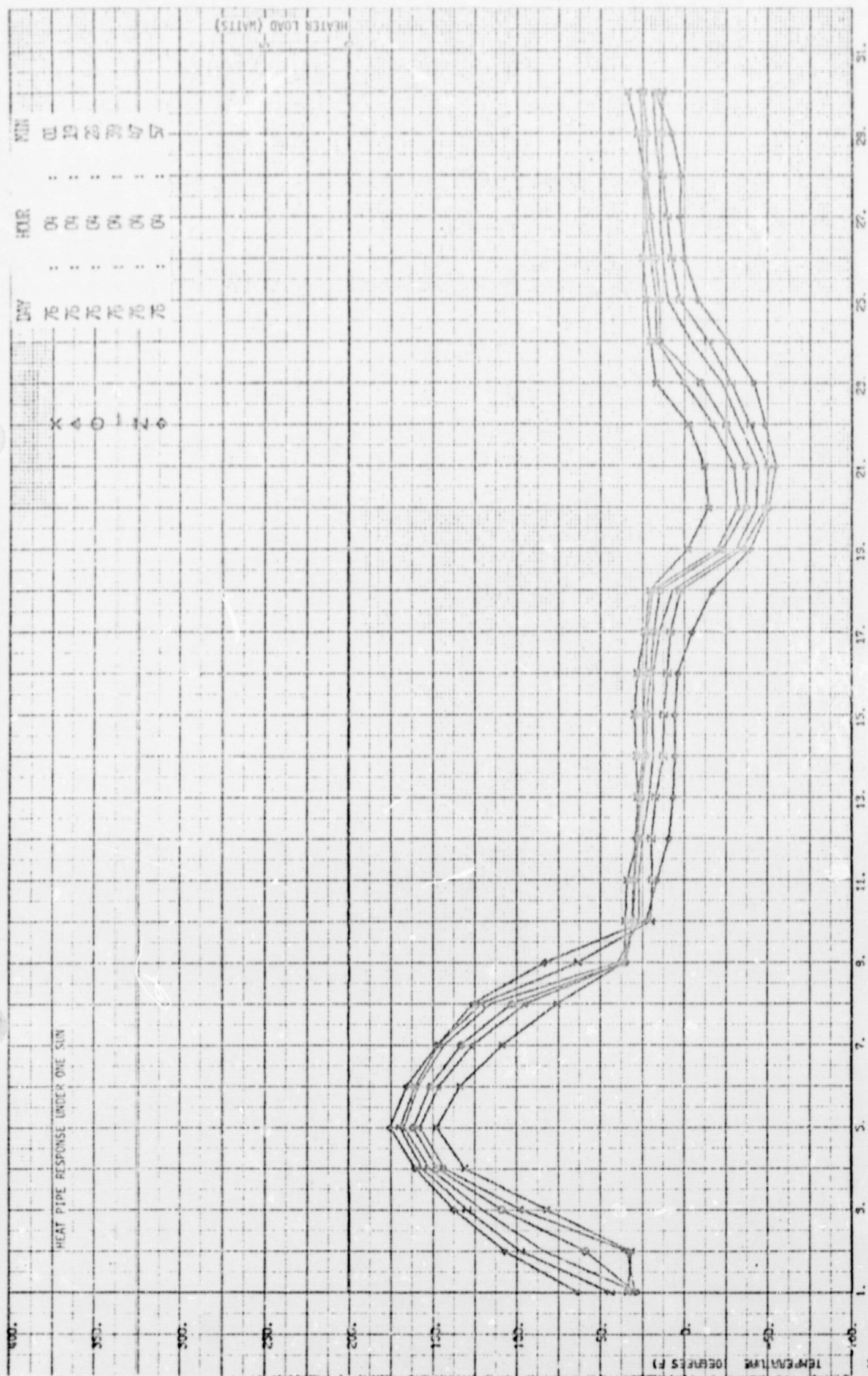


Figure 4-14b. Side-Sun Soak Phase

heat leakage and possible changes in surface properties (post test measurements by MSC yielded $\epsilon = 0.27$), the actual equilibrium temperature was considerably lower. From side-sun roll tests (discussed later), the mean steady-state temperature for the water pipe at 77:12:10 after long exposure to the side-sun was measured to be 29°F . Being below the 32°F freezing point made it impossible for the water heat pipe to operate properly in the side-sun condition.

4.4.2.3 Steady-State and Transient Operation

4.4.2.3.1 Operation at Higher Heat Input Loads

The previous discussion has shown that the combination of 1) non-condensable gas, 2) a high freezing point, and 3) larger than expected heat losses, did not permit successful steady-state operation of the water heat pipe under solar simulation conditions. However, to demonstrate successful heat pipe operation, one way of overcoming this problem was to increase the input heat load so as to raise the operating temperature of the pipe (particularly the gas-blocked section) an amount sufficient to avoid freezing. In other words, conditions were established such that axial conduction in the fin and tube wall was sufficient to keep the gas-blocked region above the freezing point.

An example of such a test is shown in Figure 4-15. This test was run during a stationary cold soak portion of the STB timeline, so all heat input was supplied by the electrical heaters. Three heaters, spaced in a "Y" configuration around the heat pipe, were energized at 90 watts each. This compares with a calculated total heat input of 174.8 watts under solar simulation conditions, which was insufficient to avoid freezeout.

Figure 4-15 shows that, under these conditions, the water heat pipe operated successfully, exhibiting only about 12°F total ΔT around its active circumference. The gas-blocked region was located approximately at the center of the longest condenser section, as expected. The length of the gas-blocked region cannot be established since only one thermocouple (T/C No. 2) was affected. However, it clearly was less than 3.2 feet, the distance between thermocouples 1 and 3.

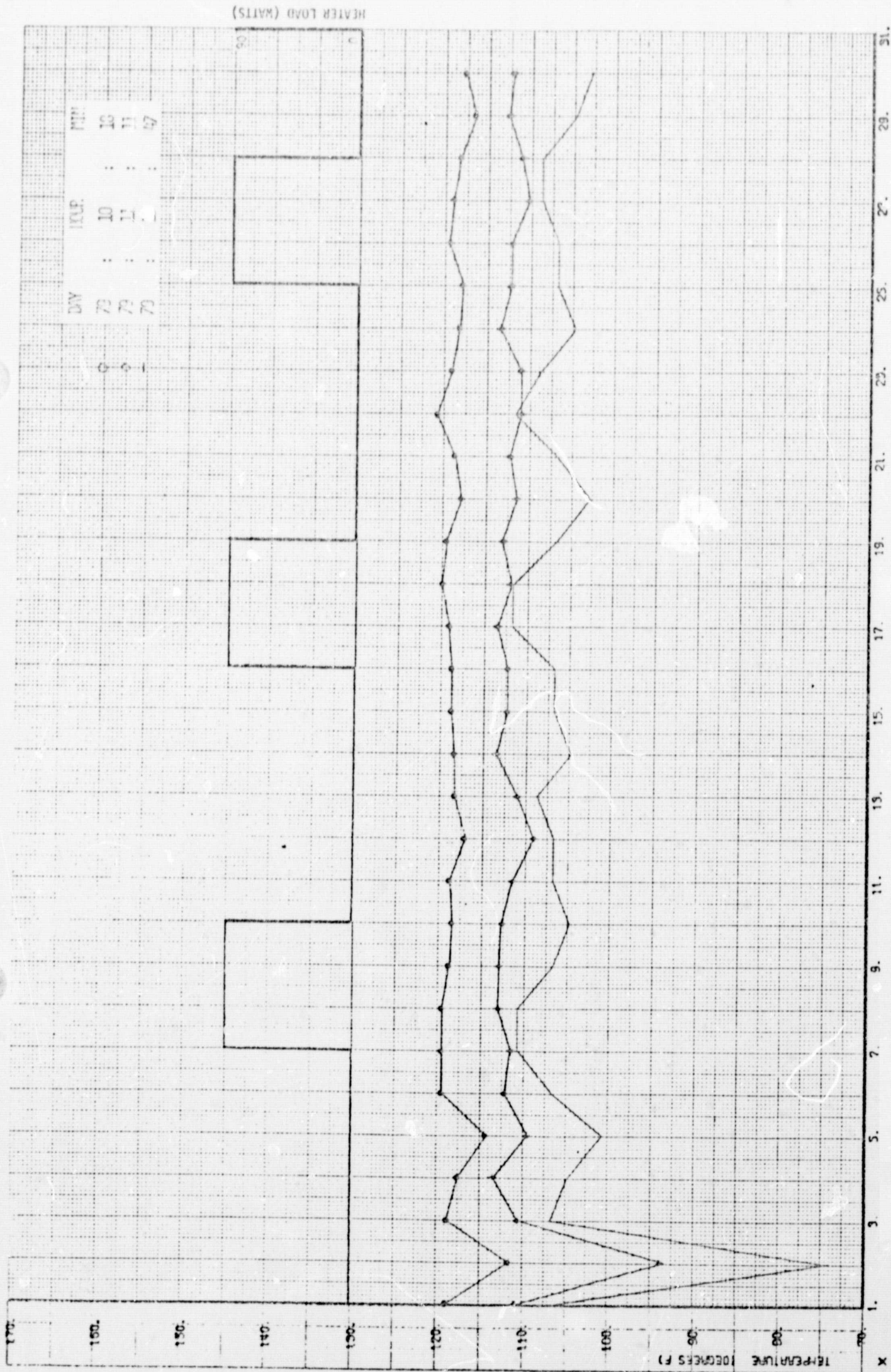


Figure 4-15. Cold Soak/1.8Y Load Phase

Of particular interest is the fact that, at the higher operating temperature and pressure associated with this increased heat load, the coldest point of the gas-blocked region remained above the freezing point. Thus, vapor diffusing into the gas-blocked region condensed but did not freeze, allowing continuous operation of the heat pipe.

Along these lines a particularly interesting test result was obtained with the same three heaters (Y configuration) energized at 62.5 watts each. The test began at 77:21:15. Figure 4-16 shows the heat pipe temperature profiles after 57 minutes (77:22:12), 1 hour and 47 minutes (77:23:2), and 16 hours and 47 minutes (78:14:2) of continuous operation.

The nominal heat pipe operating temperature was about 55°F. As one can see from Figure 4-16, this was not high enough to prevent part of the gas-blocked region from falling below the freezing point. At the end of this test phase, the minimum temperature in the gas-blocked region was about -10.5°F with a total of two thermocouples recording subfreezing temperatures. In spite of this, the heat pipe operated in this mode for over 16 hours (when the test was terminated) without suffering a diffusion freezeout failure.

The reason for this behavior has not been conclusively determined. However, it is possible to hypothesize a mechanism which is consistent with the results. This would be for the fluid to have frozen in such a way as to bridge and block the heat pipe on both sides of the noncondensable gas slug, as shown in Figure 4-17. The ice would then prevent diffusion of vapor into the gas zone where it would continuously condense and freeze. The ice plugs and liquid on their hot sides would serve as end caps on a shortened heat pipe, thus eliminating diffusion effects and permitting steady-state operation.

To reiterate, this hypothesis has not been proven to be the cause of the observed behavior. However, it is consistent with the results of Figure 4-16, as well as a later test (see Figure 4-26b), where a portion of the pipe operated in an "isothermal" heat pipe mode while other portions were clearly frozen or running hot due to local evaporator dryout.

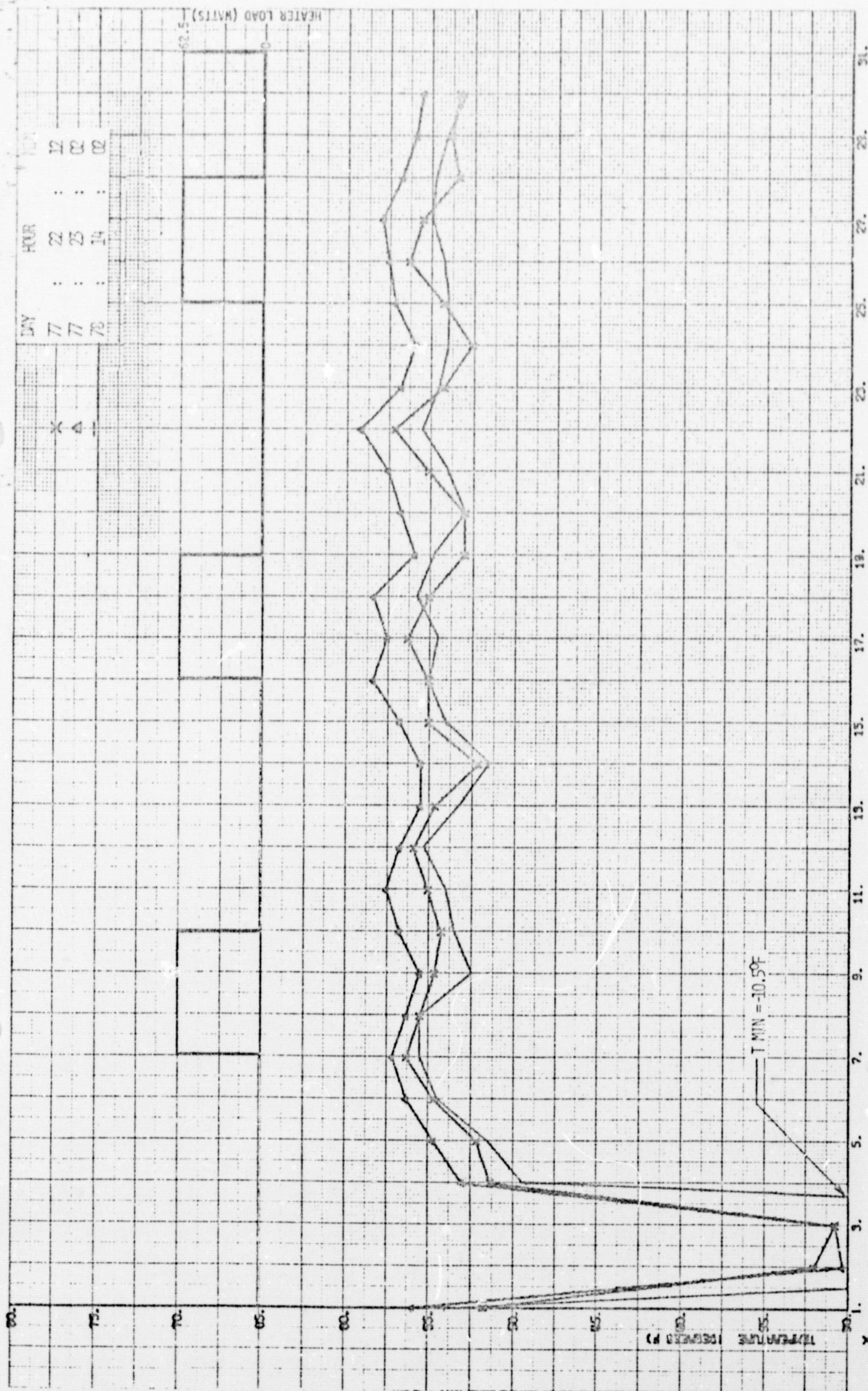


Figure 4-16. Cold Soak/1.3Y Load Phase

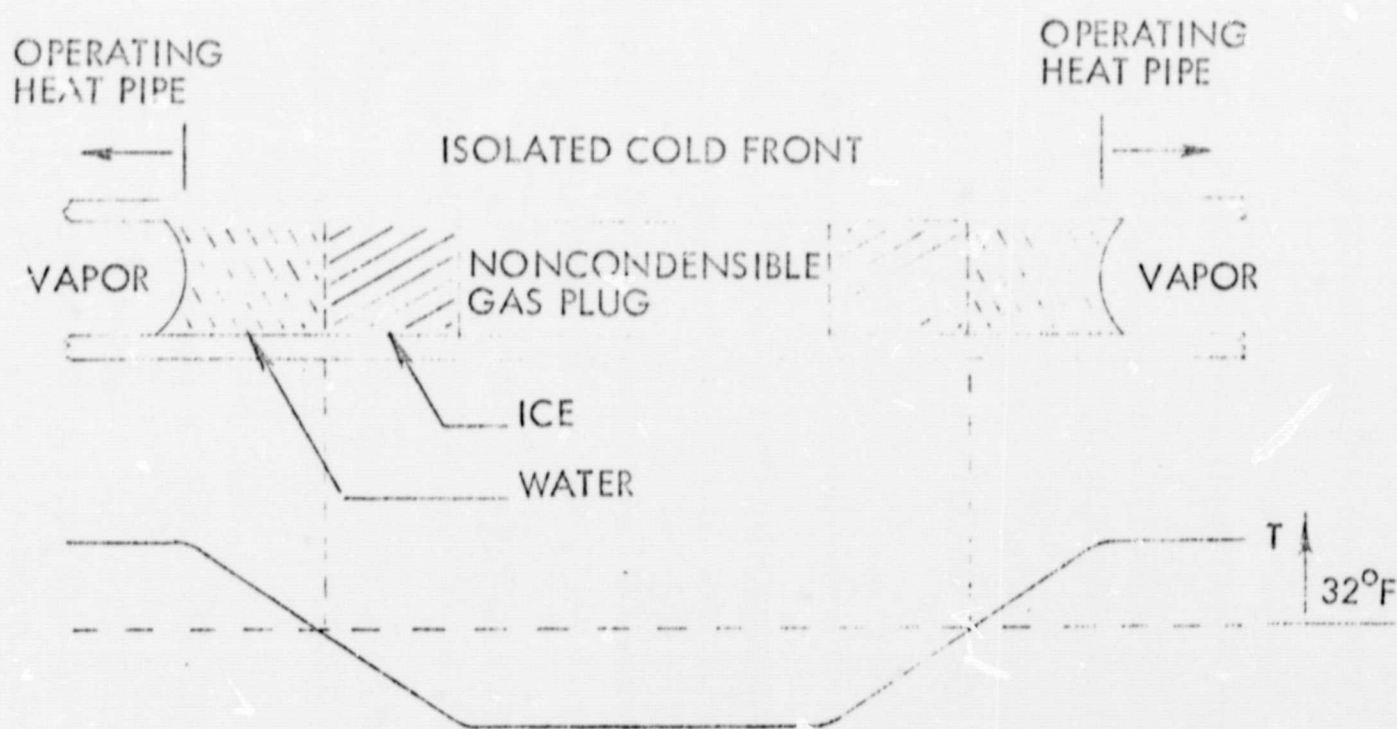


Figure 4-17. Hypothesized Model for Steady-State Operation with Subfreezing Gas-Blocked Section

4.4.2.3.2 Operation with Uniform Baseline Heat Input to Avoid Local Freezing

The previous section demonstrated that diffusion freezeout failure could be avoided, and steady-state operation achieved, by increasing the heat input load and operating temperature such that the gas-blocked cold zone remained above the freezing point. However, it was also of interest to examine heat pipe operation at heat input conditions equivalent to solar simulation and below. To accomplish this, all ten heaters on the water pipe were uniformly energized so as to keep the entire heat pipe above the freezing point. After some experimenting, it was found that a heat input of 15.5 watts per heater was sufficient to maintain the pipe at a "uniform" (see Figure 4-18) temperature of 43°F. This was established as the baseline heat load.

Note that under baseline heating conditions, the heat pipe ideally transfers no heat. Heat is uniformly radiated to the cold walls, matching the uniform heat input, and thus no axial heat transport is required. Under

H2O HEAT PIPE VACUUM TEST

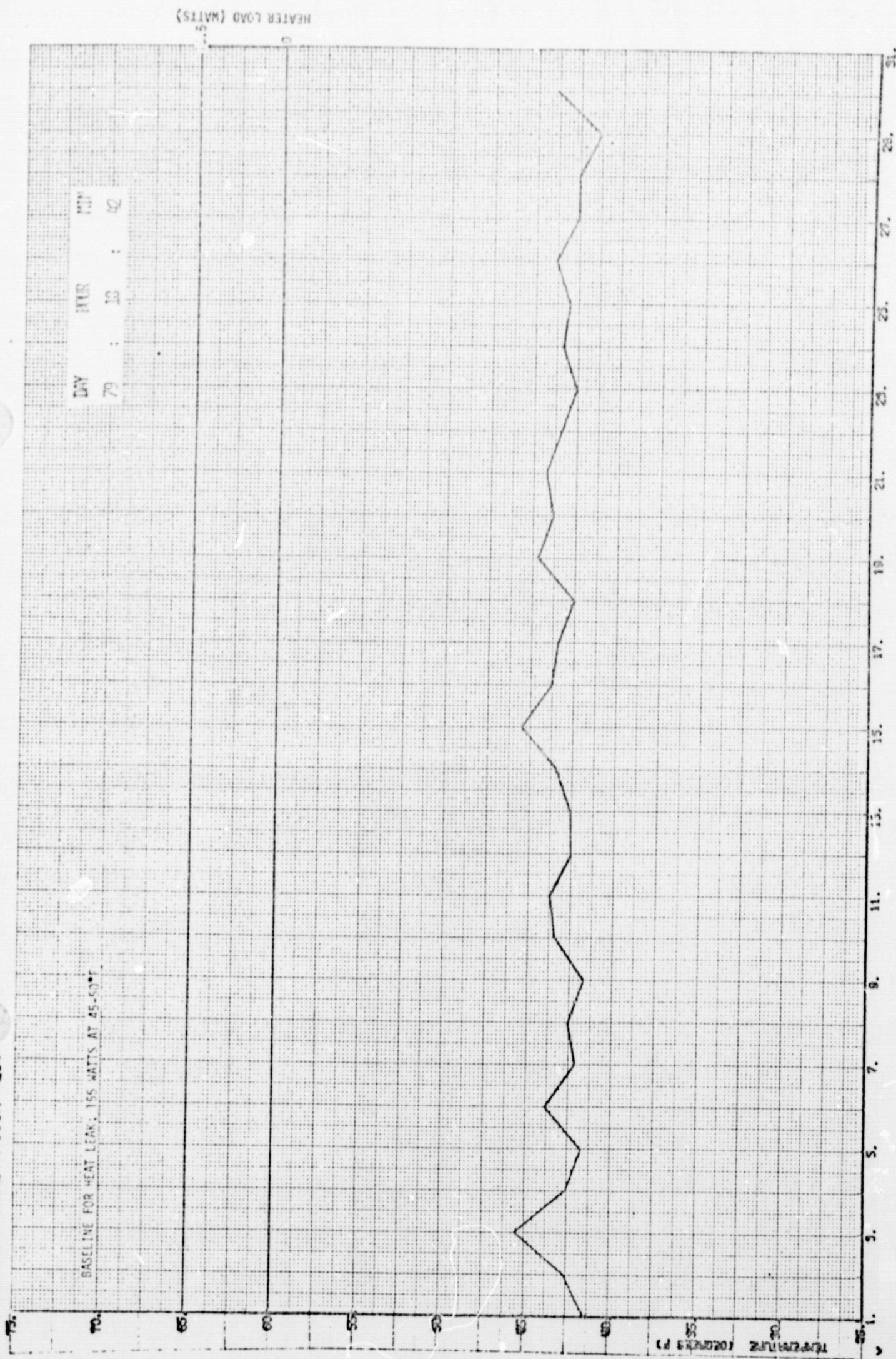


Figure 4-18. Cold Soak/Uniform Heating Phase

actual test conditions, localized heat pipe action probably occurs at numerous positions around the pipe due to nonuniformities in heat input, radiative properties, etc. However, these are second order effects in terms of total heat pipe capacity.

With the baseline heat load established, it was then possible to operate the heat pipe at varying net heat loads by increasing the power input to selected heaters. Thus the five heaters on one side of the pipe were energized to simulate a cosine heat input function characteristic of various fractions of a solar load.

The steady-state results for total net heat inputs (heat input in excess of baseline level) of 45, 87.5 and 131 watts are shown in Figure 4-19. It is seen that the pipe operated successfully at each of these conditions.

Of particular note was the transient behavior of the heat pipe in responding to input power changes during these tests. For example, Figures 4-20a and 4-20b show the transient response in going from the baseline condition (Figure 4-18) to the 45-watt net throughput condition. It should be noted that the test conditions were changed before equilibrium was achieved.

These curves clearly demonstrate one of the major positive features of heat pipes. That is, the heat pipe responds to nonuniform changes in heat load by uniformly altering its operating temperature level.

The steady state data at the end of the transient also show:

- 1) The resolution of the thermocouple measurements is only about 5°F, and all data presented should be viewed in that light. (There is no physical basis for the oscillatory behavior seen.)
- 2) At a mean operating temperature of about 64°F, the gas occupies less than 1.6 feet of the pipe (it cannot be observed at 79:20:32, indicating it was bounded by two adjacent thermocouples with this spacing).

Following the 131-watt case, an attempt was made to operate with a net throughput of 183 watts which would have exceeded design conditions of 174.8 watts (albeit at a higher temperature). However, after a short period at this power, the center evaporator section, with highest input

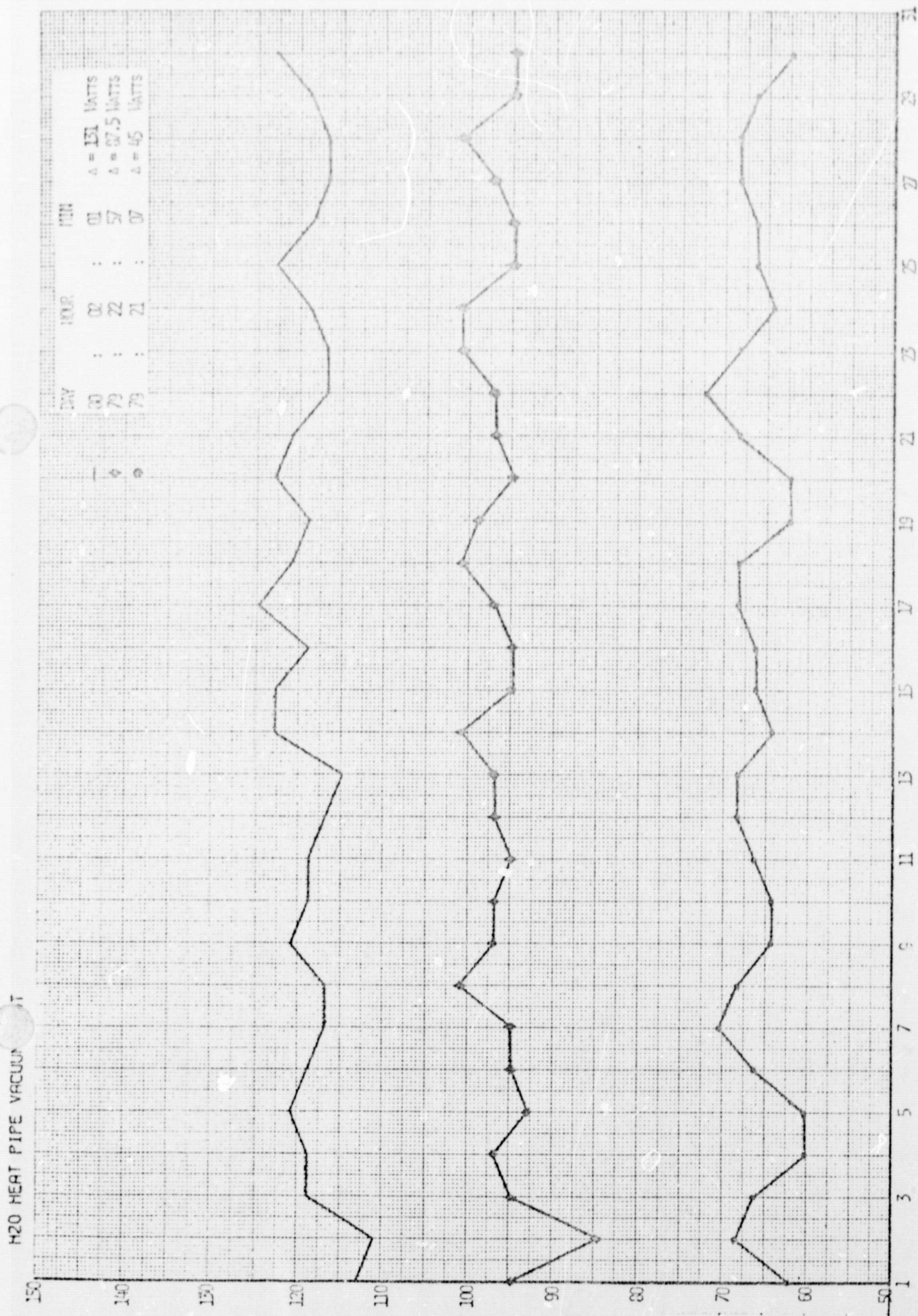


Figure 4-19. Steady-State Operation ΔS Loadings

H2O HEAT PIPE VACUUM TEST

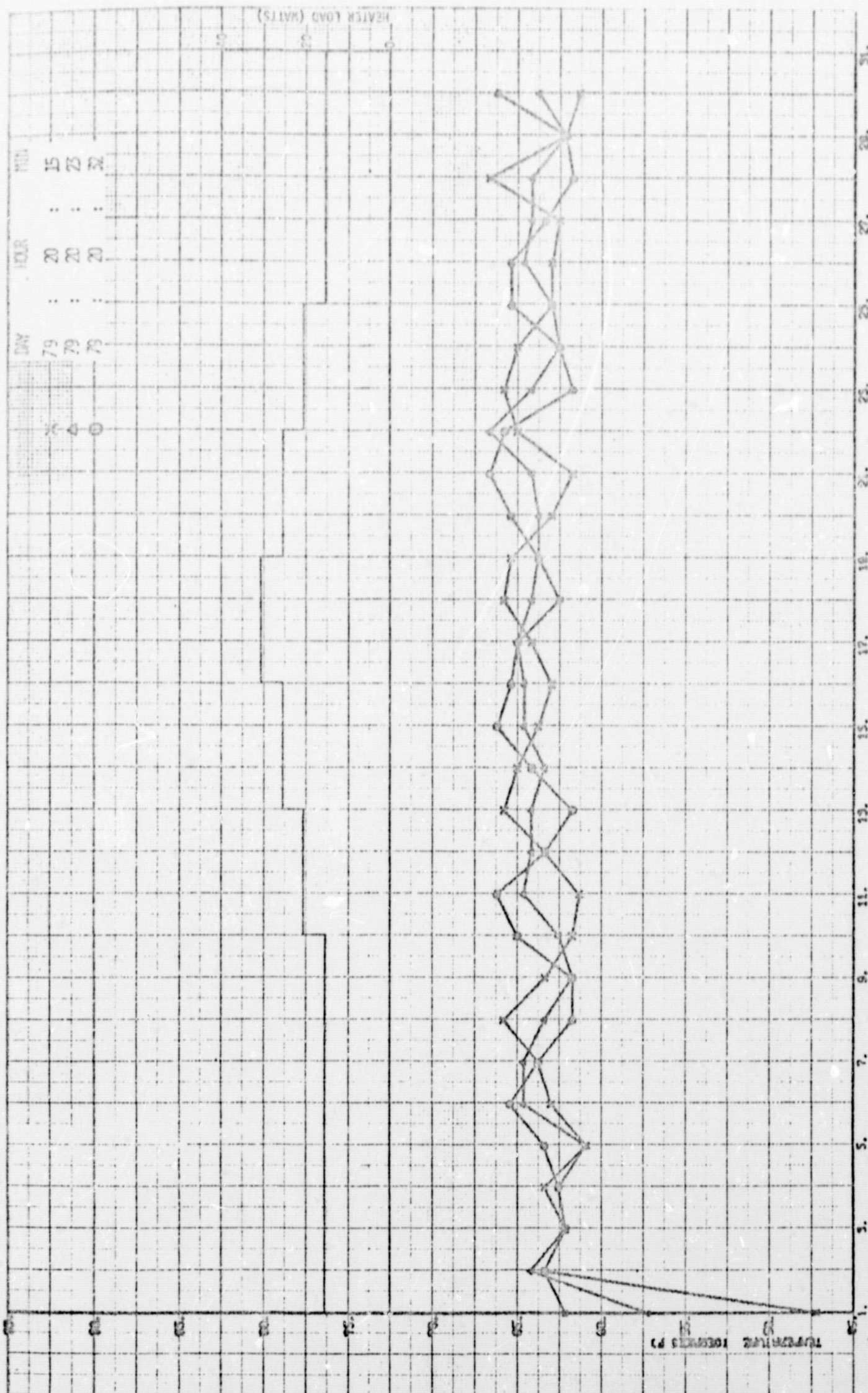


Figure 4-20a. Cold Soak Roll/.3 ΔS Phase

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power (75.5 watts total), exhibited a rapidly increasing temperature indicating pipe failure. This failure was apparently a hydrodynamic crisis rather than arterial nucleation since the pipe had previously operated successfully with a higher input power (90 watts) to the same module (see Figure 4-15).

After reestablishing heat pipe operation (see timeline for details), a more modest attempt was made to operate with a net throughput of 154.5 watts. The transient results of this test are shown in Figures 4-21a and 4-21b. During the initial portion of the test, which began at 80:04:00, the heat pipe operated successfully. However, the operating temperature was quite high—about 194°F—due to the high input power used to reestablish heat pipe operation after the failure at 183 watts. With passing time the heat pipe fell towards an equilibrium temperature corresponding to the 154.5-watt net plus 155-watt baseline load. As this occurred, the heat pipe continued to operate successfully until it reached an average temperature of 153°F (80:04:48), when it again failed near the center of the evaporator.

This transient yields further evidence that these failures are due to a capillary pumping limit (hydrodynamic crisis) rather than nucleation in the artery. The ability of a liquid to withstand superheat increases with decreasing pressure.* Thus, a nucleation failure is more likely at high temperature and pressure (where the pipe operated successfully) than at lower values (where it failed). On the other hand, within this temperature range the hydrodynamic heat transfer capacity of water heat pipes increases with increasing temperature. This is consistent with the observed behavior.

The highest successful net cosine load (131 watts) represents only about 13 percent of theoretical capacity. This compares with about 25 percent of theoretical obtained in laboratory testing prior to shipment from TRW. It was believed that the pipe had suffered some wick degradation during rail transport from Redondo Beach to Houston. However, laboratory testing by NASA subsequent to the thermal vacuum tests

* B. D. Marcus, On the Operation of Heat Pipes, TRW Report No. No. 9895-6001-TU-000, May 1965.

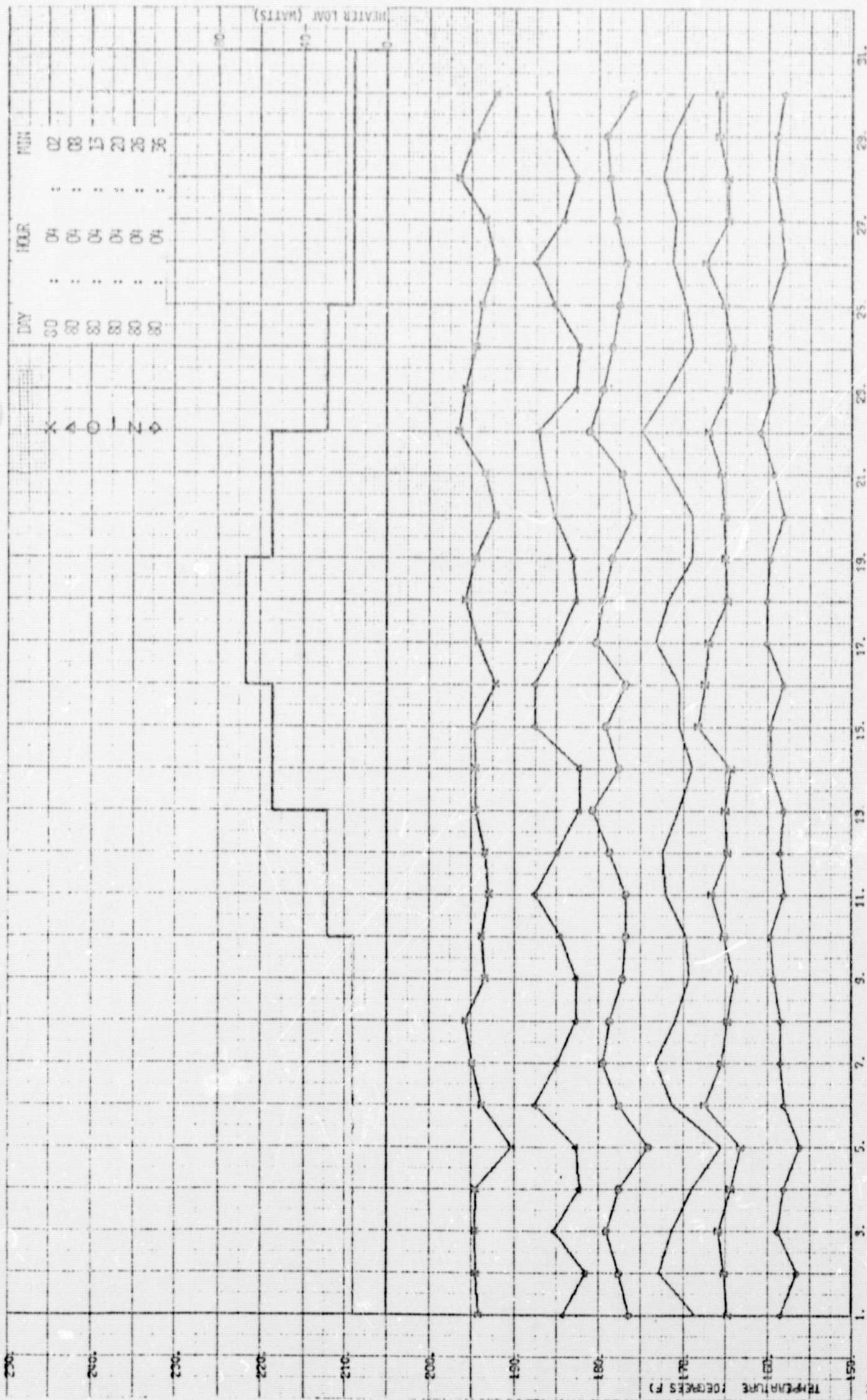


Figure 4-21a. Cold Soak Roll/1.1 ΔS Phase

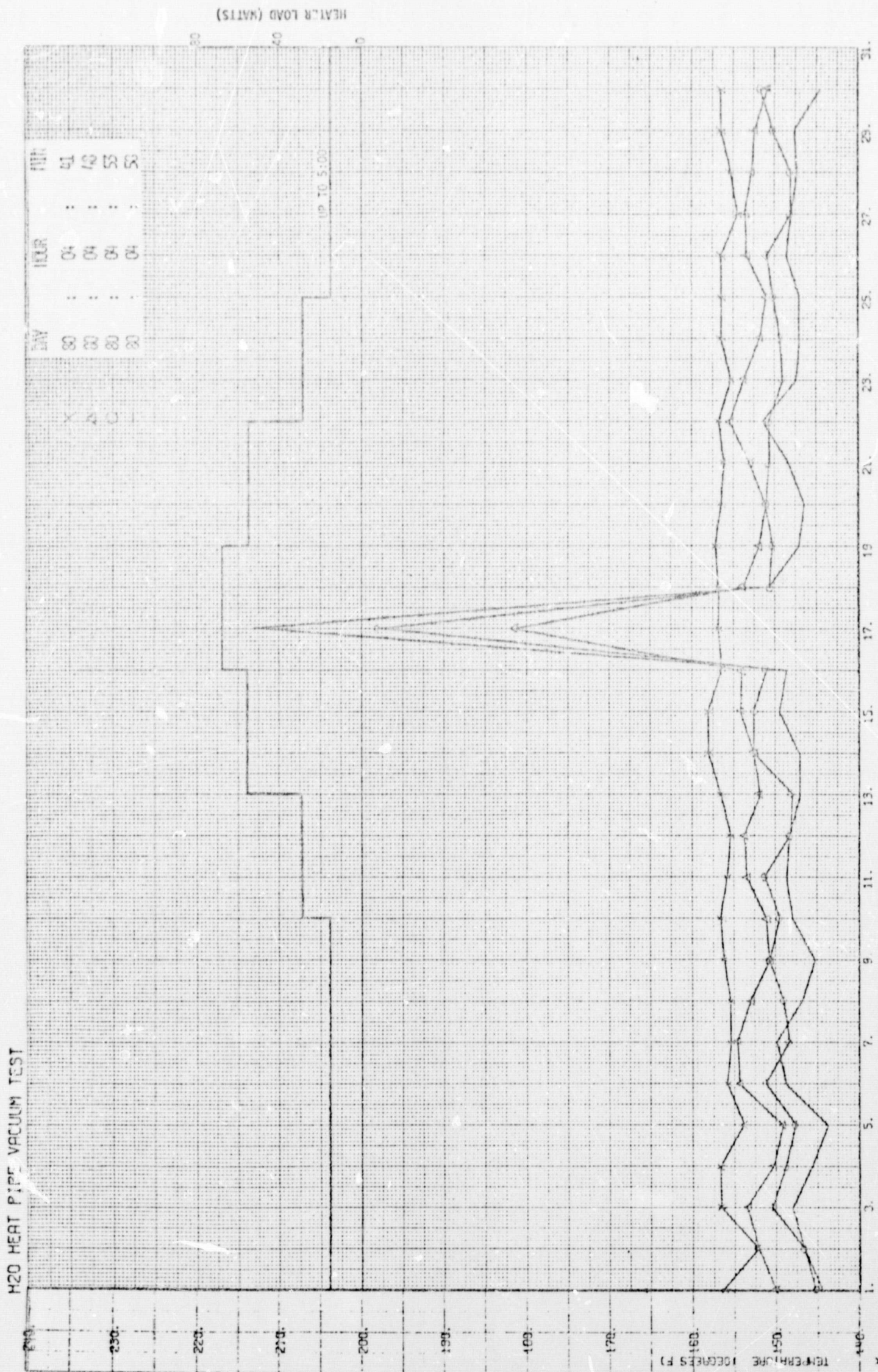


Figure 4-21b. Cold Soak Roll/1.1 Δ S Phase

confirmed results obtained in a laboratory environment at TRW. The differences between laboratory and thermal vacuum test results have not been explained.

4.4.2.3.3 Side-Sun Roll Tests

The side-sun soak tests (Section 4.4.2.2.1) were unsuccessful because a portion of the gas-blocked region fell below the freezing point of the water, causing a diffusion freezeout failure. However, as mentioned in Section 4.4.2.2, this situation can potentially be overcome by rotating the heat pipe so that the cold zone constantly changes position. Thus, any fluid that freezes is remelted shortly thereafter. The amount of fluid frozen at any given time depends on the rotational speed, among other variables. If the speed is high enough, the frozen fluid inventory can be rendered low enough to avoid evaporator dryout and permit successful heat pipe operation. This condition is not, of course, desirable from an application standpoint since one of the reasons for inclusion of heat pipes on a spacecraft was to eliminate the requirement for rolling. However, this test phase presented an opportunity to study the transient aspects of diffusion freezeout. The rotational speed of 2 revolutions per hour (RPH) with a return of 15 RPH was part of the STB timeline.

The results are shown in Figures 4-22a and 4-22b. The test began with the pipe at a high temperature due to previous application of electrical heat. With the passage of time the temperature fell towards an equilibrium value consistent with a simulated solar input. Since this temperature was below 32°F the total pipe system was frozen. However, the roll rate of 2 RPH was high enough to prevent a significant temperature gradient around the circumference of the pipe. The inoperative methanol pipe (Figure 4-23) demonstrated approximately the same temperature gradient although the average temperature level was slightly higher (29°F versus 32°F).

4.4.2.4 Frozen Start-up Attempts

As mentioned earlier, one of the objectives of this program was to study start-up of very long heat pipes from the frozen state. Of particular interest was the use of noncondensable gases to aid start-up by controlling the rate at which vapor flowed from the thawed to the frozen section of the pipe.

H2O HEAT PIPE VACUUM TEST

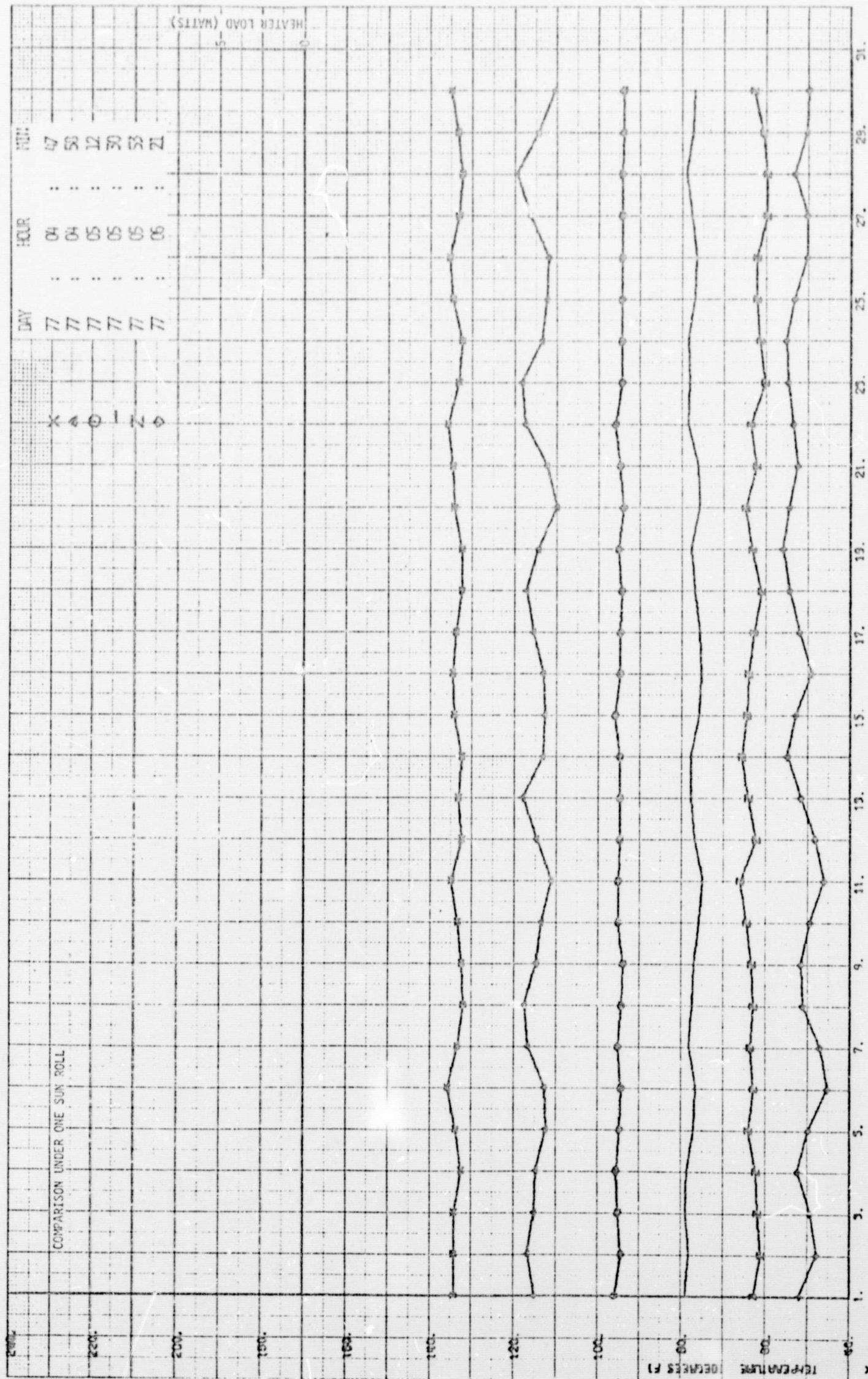


Figure 4-22a. Side-Sun Roll Phase

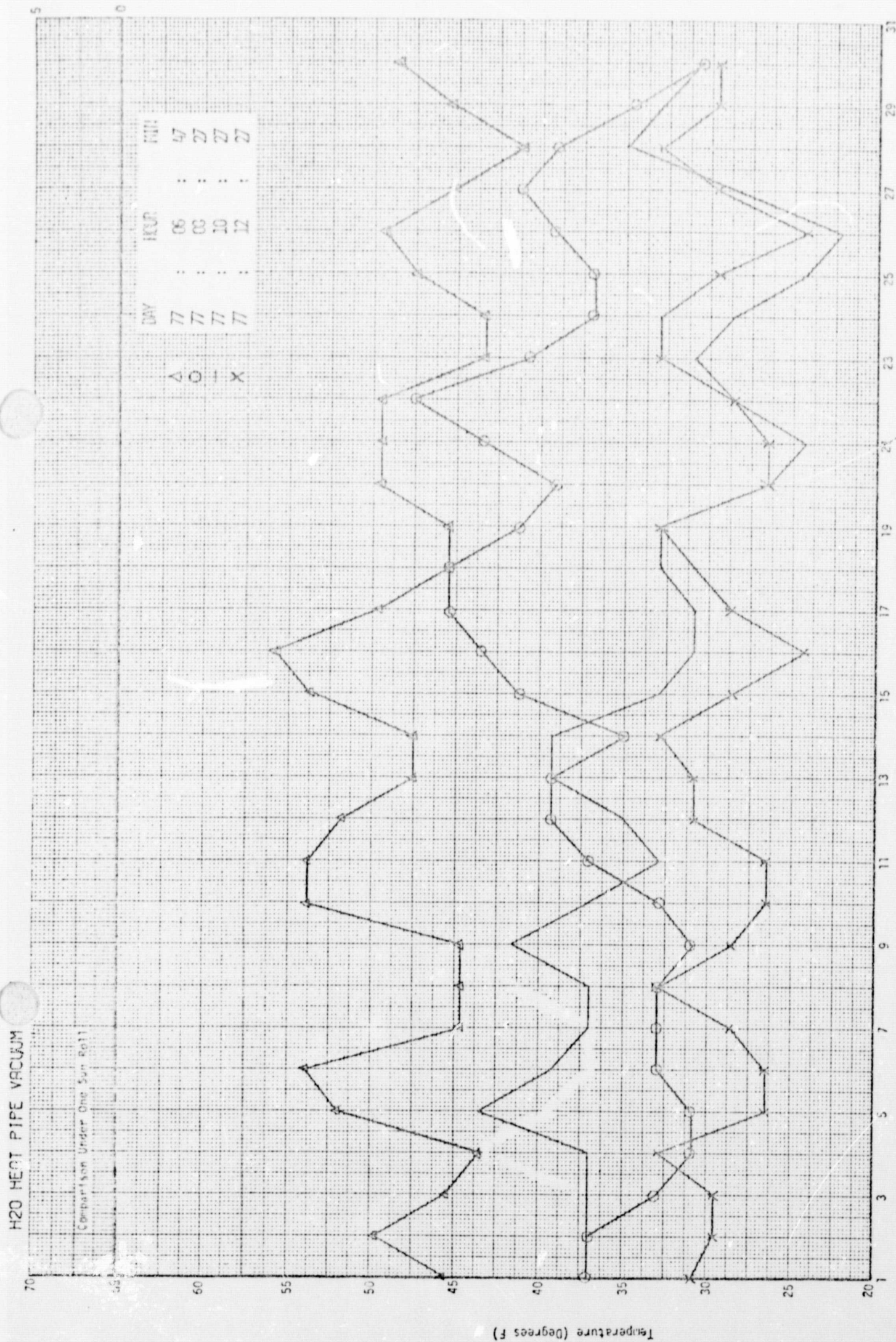


Figure 4-22b. Side-Sun Roll Phase

METHANOL HEAT PIPE

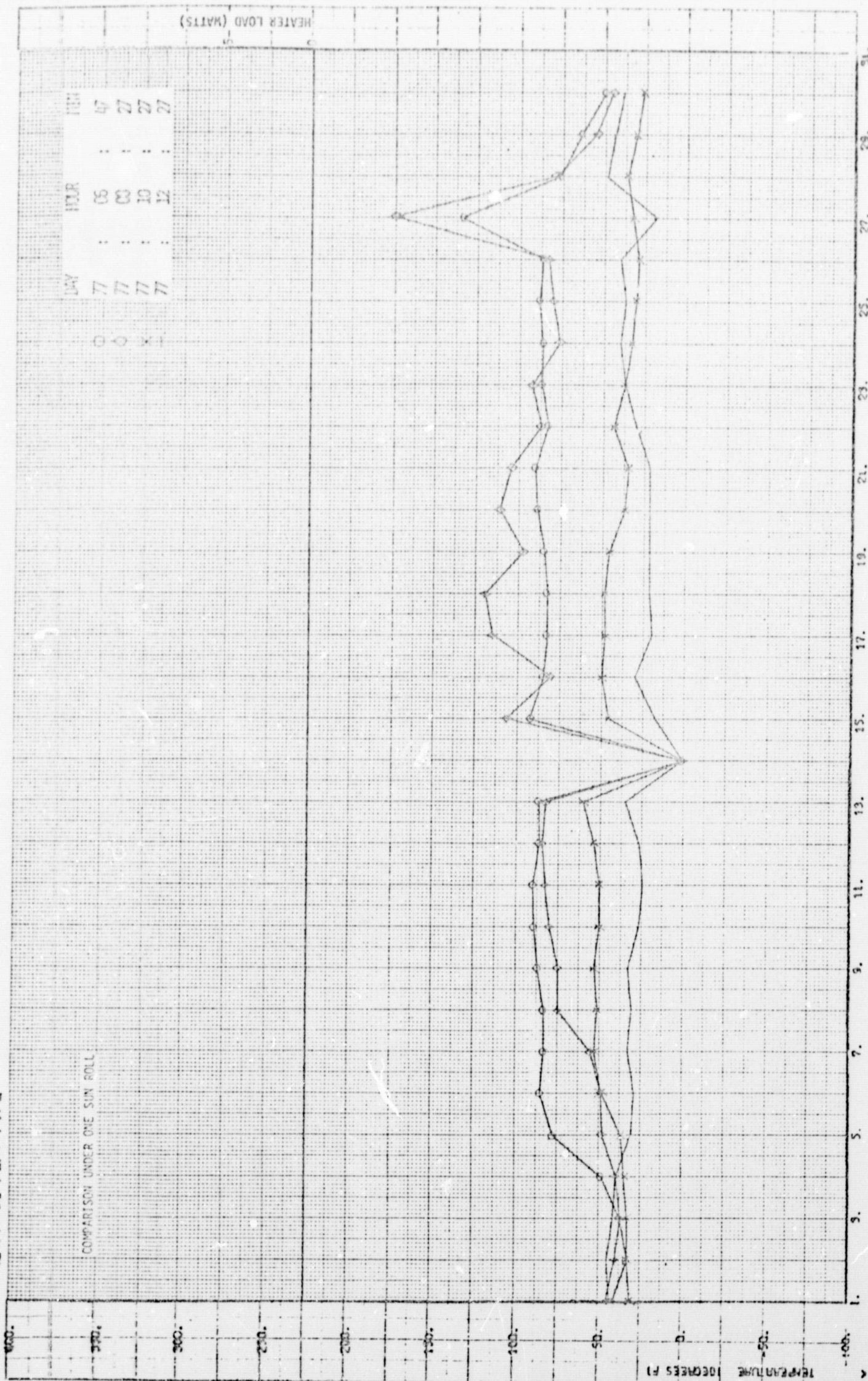


Figure 4-23. Side-Sun Roll Phase

The program plan called for performing these studies with a gas-dosed ammonia heat pipe. Ammonia's low freezing point and high operating pressure at design temperature promised a high probability of success for this start-up scheme (see Section 2.2.1.4).

Unfortunately, as discussed in Section 4.2.2, the gas-dosed ammonia heat pipe would not function properly due to nucleation within the artery. Thus, the planned gas-aided frozen start-up studies could not be performed with an ammonia heat pipe.

It was possible, however, to perform several start-up tests with the operational water heat pipe. Clearly, successful start-ups under a solar heat load distribution were not to be anticipated. Without non-condensable gases the high freezing point of water requires a great deal of heat transport to thaw the pipe, and the evaporator section would be depleted of sufficient working fluid to desaturate the artery before this could be accomplished. With sufficient noncondensable gas to effect a progressive start-up, the gas-blocked zone under steady-state operating conditions would be so large that a portion of it would be below the freezing point, leading to a diffusion freezeout failure.

Indeed, it was this predicted inability to start up long, frozen, water heat pipes which prompted the attempt to build the gas-dosed ammonia pipe. Thus, the primary objective of the design load start-up tests with the water heat pipe was not to demonstrate successful start-up but to allow observation of the transient behavior. In addition, several distributed load start-up tests were run to determine whether frozen start-up could be affected under less severe conditions.

Several start-up attempts were made under varying heat input conditions. One such attempt was with no electrical heater power and a radiant heat load (side-sun modules 19-23) with the heat pipe stationary. The results are shown in Figures 4-24a and 4-24b.* After allowing a

* T/C No. 16 exhibits what appears to be anomolous behavior on this and several other graphs. This is likely a result of its location at the insulated valve and fill tube. The large mass of the valve and the fact that it was insulated gives rise to substantial thermal lag at low temperatures when the heat pipe is not operating. Also, there was a 6-inch segment at the fill tube area with no interconnecting fin.

H2O HEAT PIPE VACUUM TEST

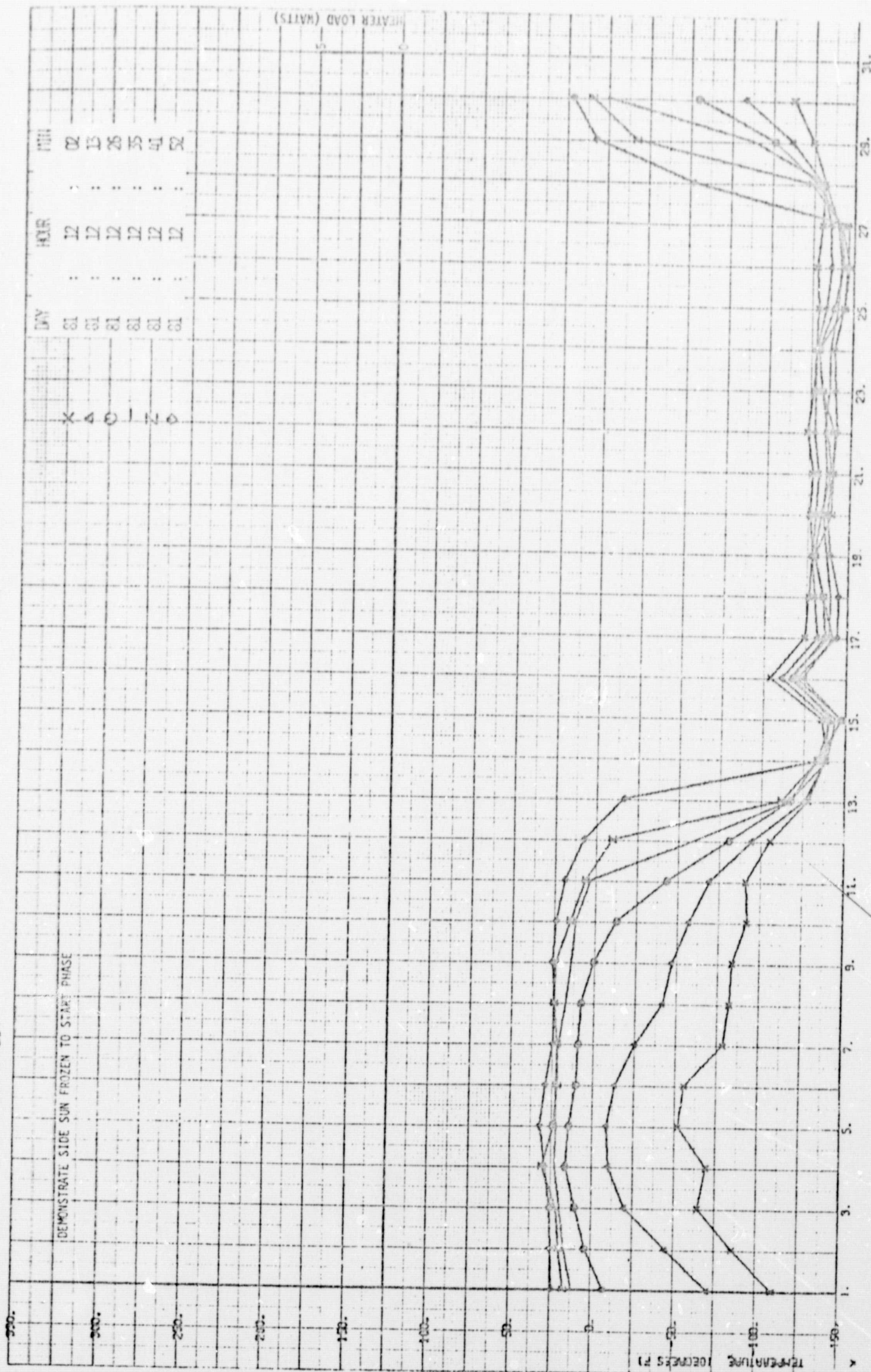


Figure 4-24a. Side-Sun Modules 19-23, Side Sun/No Roll Phase

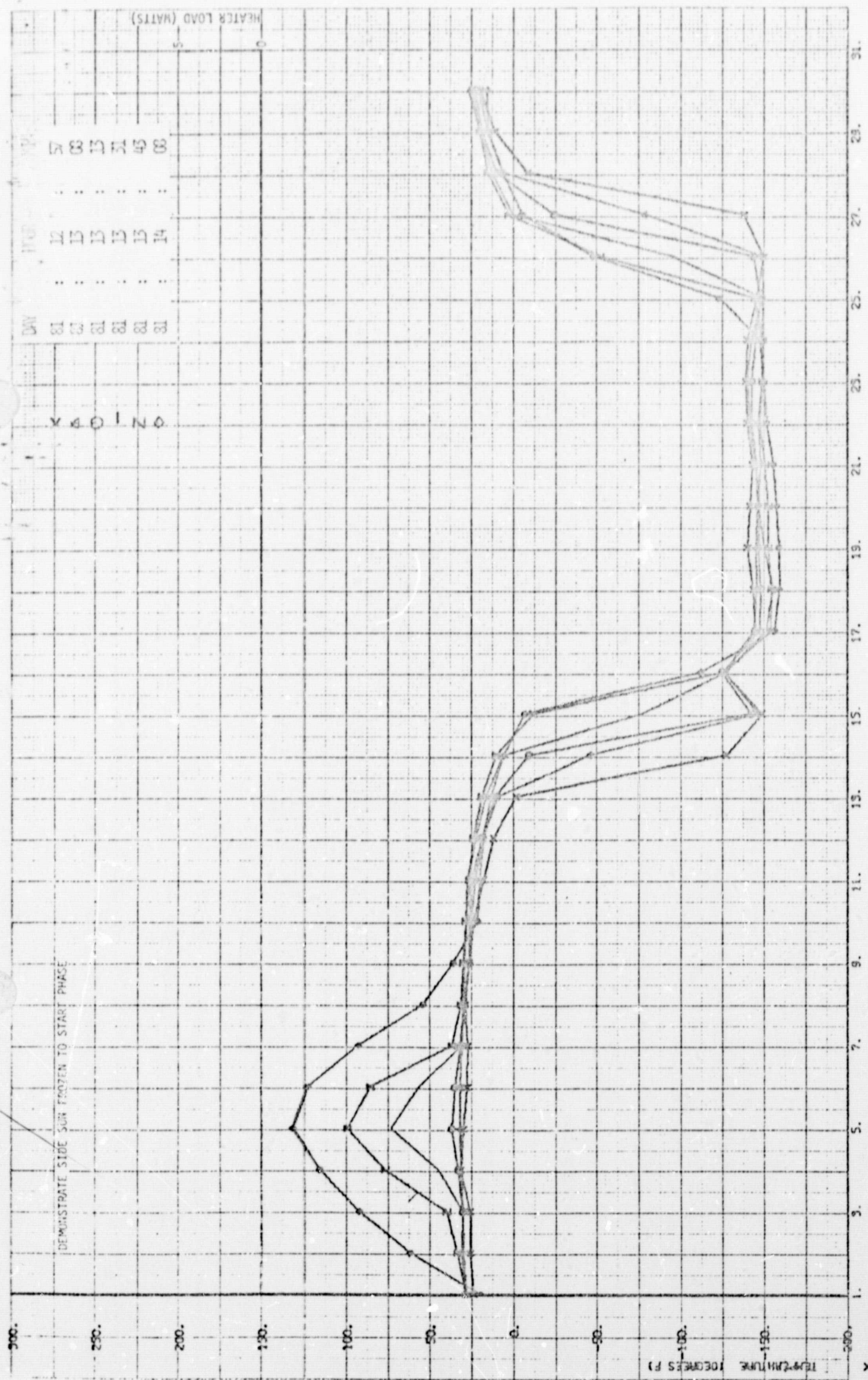


Figure 4-24b. Side-Sun Modules 19-23, Side Sun/No Roll Phase

cold soak period to uniformly freeze the heat pipe, the radiant load was applied at 81:11:58. The first time slice on Figure 4-24a is four minutes later (81:12:02) where one sees that the illuminated side of the heat pipe is increasing in temperature while the shadow side is not. Note that as time passes the temperature profile on the illuminated side is first peaked in the center and then progressively flattens out as the temperature increases and the hot zone grows. Since the heat input varies as a cosine function, it appears that the heat pipe was operating in the hot zone as temperatures above 0°F were attained. Furthermore, the progressive growth of the hot zone is characteristic of a frontal start-up caused by the noncondensable gas present in the pipe. Heat pipe action in the illuminated side would cause the formation of a gas front which moves out with increasing temperature and pressure. This behavior is evidenced by the data.

Note also that for all of the time slices on Figure 4-24a, the fluid is frozen—even on the hot side. Clearly, frozen working fluids do not mean that heat pipes cannot transfer heat. Vapor simply sublimates from the ice surface rather than evaporate from a liquid surface. Of course, it is not possible for the wick to pump a frozen fluid, and thus such operation is limited to a transient basis. It can continue only until the evaporator region depletes itself of working fluid whereupon its temperature rises, signifying pipe failure. This, too, is evidenced by the data of Figure 4-24b and during the side-sun roll tests (Figure 4-21).

One might surmise from the above discussion that, had the amount of noncondensable gas in the pipe been somewhat higher, forcing the illuminated side to operate at a higher temperature before the front could move out, the fluid would have thawed and a successful start-up have been achieved. This is not the case, however, since it was previously shown that, even with the quantity of gas already in the pipe, it fails by the diffusion freezeout mechanism at solar simulation loads because the gas-blocked cold zone remains below the freezing point.

Additional frozen start-up attempts were made during a cold soak roll period of the timeline using the electrical heaters. In one case, two diametrically opposed modules were heated with inputs of 90 watts each.

In this way the heat pipe was effectively divided into four symmetrical sections (as opposed to two under design conditions). This reduces the maximum length of nonheated sections, thus reducing the energy transport required to thaw that section. The results of this test are shown in Figure 4-25. The behavior is seen to be similar to that of the solar simulation start-up attempt. The evaporator zones first increased in temperature with a peaked distribution (81:01:12); as temperatures exceeded 0°F the hot zones flattened and spread showing the beginnings of a frontal (gas-aided) start-up (81:01:25); and finally, the evaporator temperatures ran away due to depletion of working fluid as a consequence of the absence of a liquid return mechanism (the entire pipe except for the heated zones was below the freezing point). Dividing the pipe into four symmetrical sections was not sufficient to affect start-up from the frozen state.

In another case, three modules in a "Y" configuration were used for start-up, each with 90 watts of input power. The results of this test are shown in Figures 4-26a and 4-26b.

As with the two heater start-up, the evaporator temperatures increased and, after exceeding 0°F, the hot zones flattened and grew. However, in this case, only two of the three evaporator regions exhibited "dryout". The third hot zone continued to operate as a heat pipe and continued to grow in length due to melting of the frozen regions on each side of it. Apparently, the bulk of the noncondensable gas was in the long condenser section (TC's 1-7 and 28-31), as expected, allowing the opposite hot zone to operate without freezing out. This process continued until the two short condenser sections (TC's 10-16 and 19-25) were completely melted, whereupon the two runaway evaporator sections were able to reprime with working fluid and "recover". The fact that the runaway evaporator sections could recover under load is of considerable significance. This type of behavior was also observed by McDonnell-Douglas Corporation in developing the pedestal artery heat pipes for the ATS-E spacecraft.

By the end of this test phase (80:21:01), the bulk of the pipe, including all three evaporator zones, was operating successfully as a heat pipe. Only a portion of the longest condenser (TC's 1-5) was frozen. However,

H2O HEAT PIPE VACUUM TEST

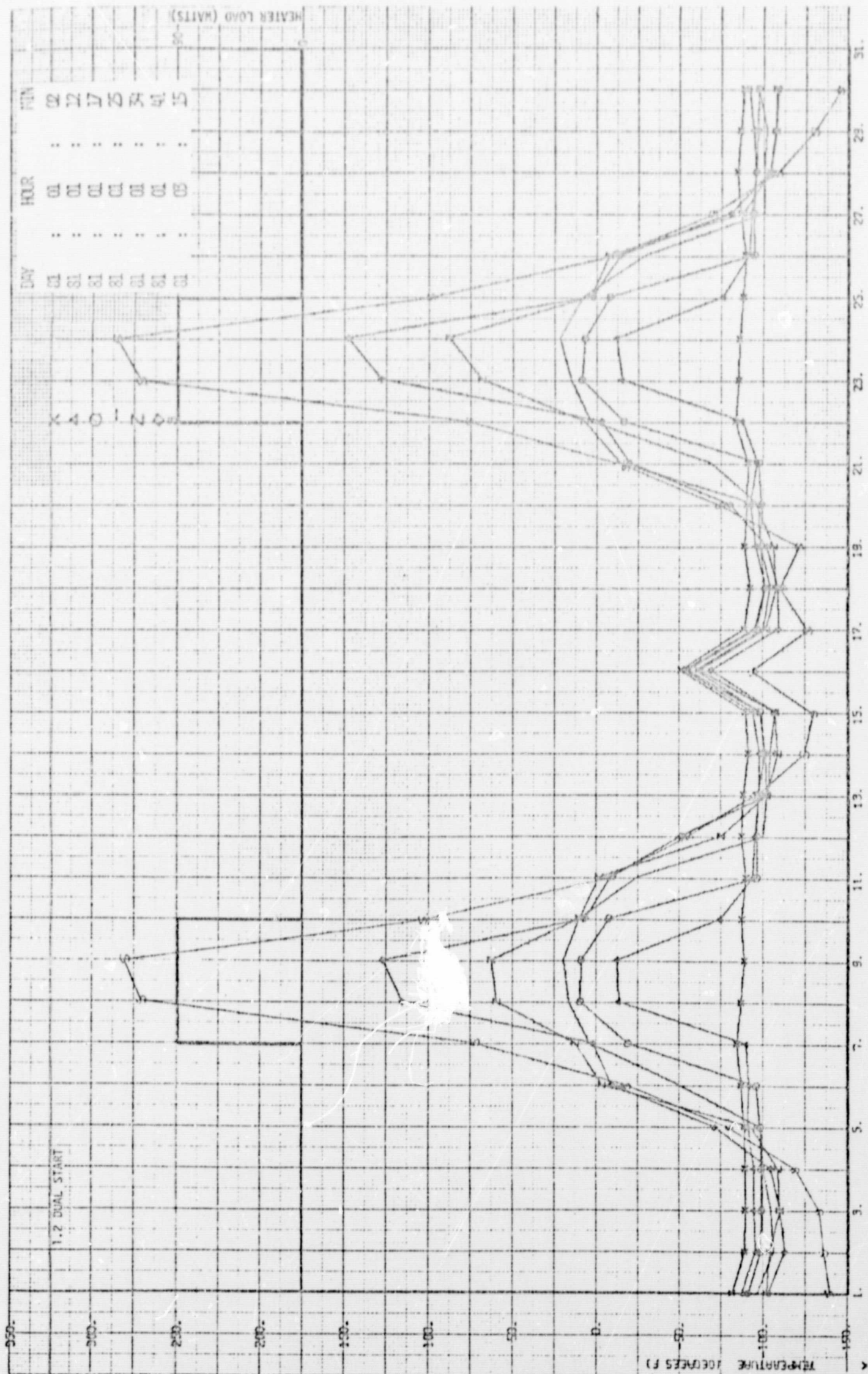


Figure 4-25. Frozen Start Phase

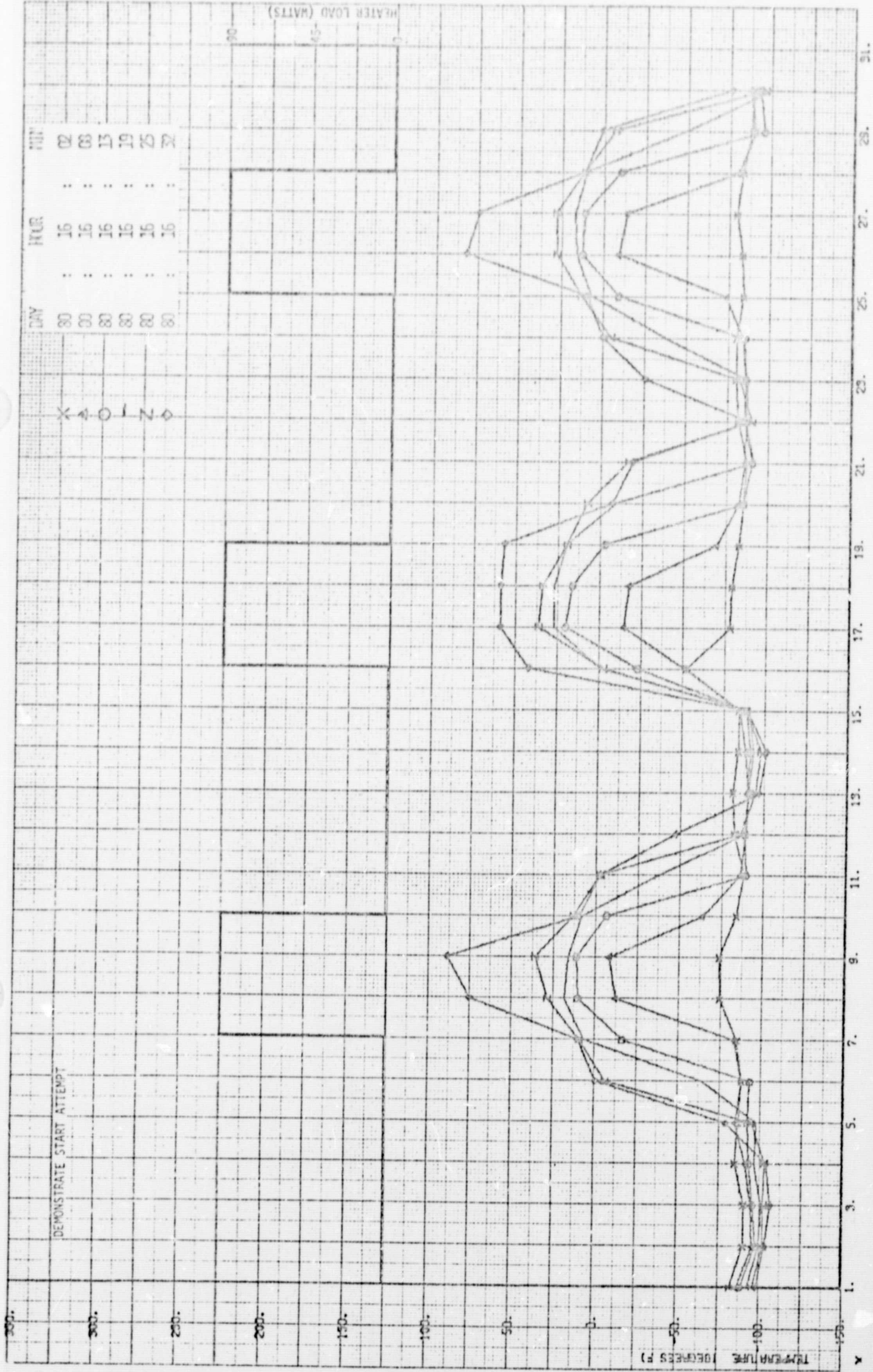


Figure 4-26a. Frozen Start Phase

H2O HEAT PIPE VACUUM TEST

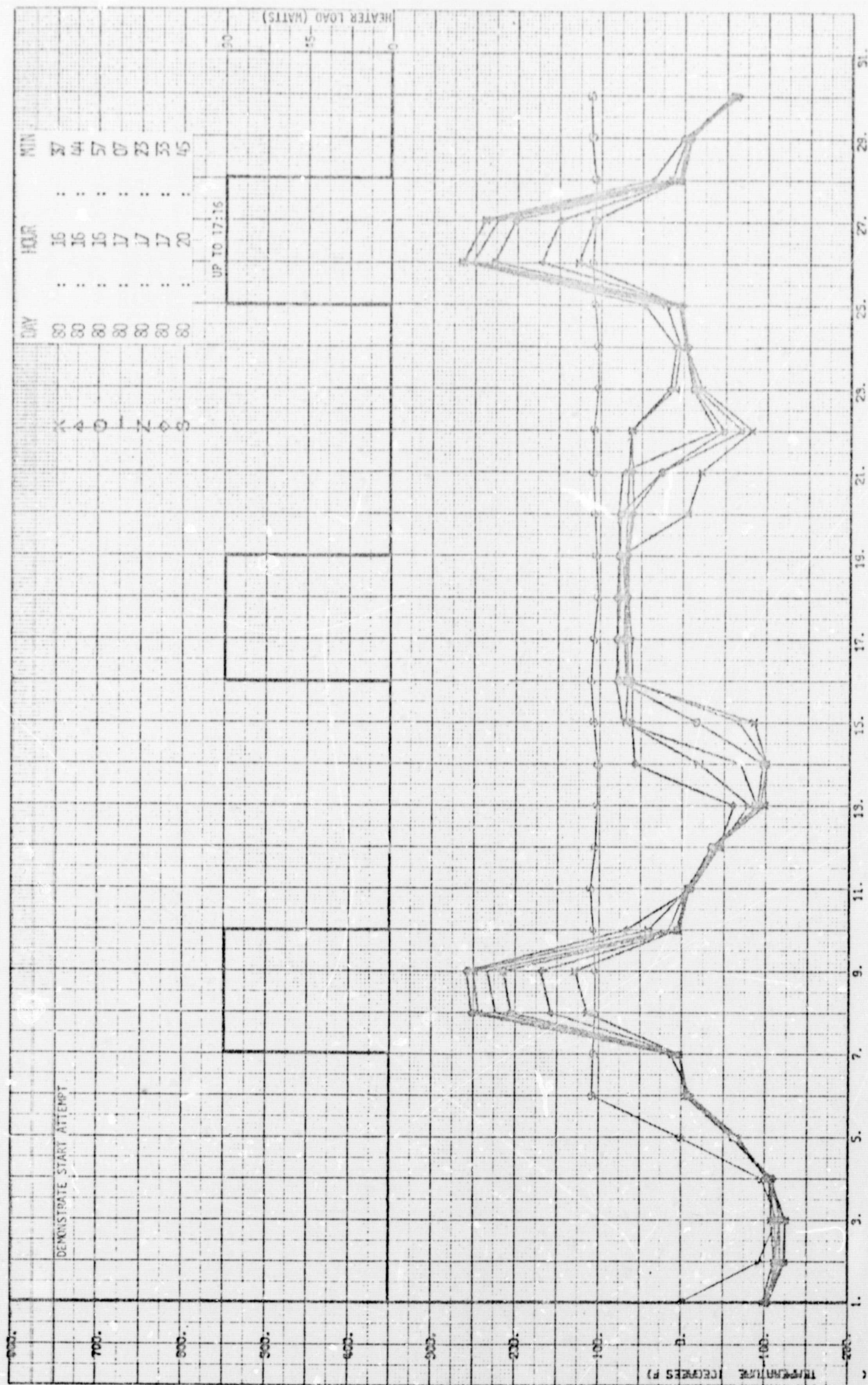


Figure 4-26b. Frozen Start Phase

as discussed in Section 4.4.2.3.1, by hypothesizing ice blockages as shown in Figure 4-17, steady-state operation with the gas-containing portion of the pipe frozen is entirely feasible (see Figure 4-16).

The last start-up attempt was with a radiant heat load (side-sun modules 19-23) while the STB and heat pipes were rotating at 2 RPH. Since it was previously shown (Section 4.4.2.3.3) that rotating the heat pipes altered the evaporator position rapidly enough to prevent significant temperature gradients, one might have expected a successful start under these conditions. This probably would have been the case but for the fact that the radiant load was insufficient to keep the water heat pipe above the freezing point. An attempt to supplement the radiant heat load with an electrical input of 67 watts to one of the heaters was still insufficient to bring the entire pipe above freezing and a freezeout failure followed. This is shown in Figures 4-27a and 4-27b.

4.4.3 Summary and Discussion of Thermal Vacuum Test Results

Thermal vacuum testing of the operational water heat pipe provided a significant advance in the technology of large heat pipe systems. Meaningful data were obtained with respect to each of the primary objectives of the program.

4.4.3.1 Steady-State and Transient Operation

Successful steady-state operation of the heat pipe under simulated cosine net heat loads up to 131 watts, and "Y" heat loads up to 270 watts, established the fact that very long, arterial wick heat pipes can be successfully fabricated using modular concepts. Although the 131-watt net cosine load represented only about 13 percent of theoretical capacity, this must be viewed in light of the extraordinarily short time period available to design, develop, fabricate, test and deliver these heat pipes (3 months). Thus, neither design nor manufacturing techniques could be optimized to assure perfect integrity of the arterial wick structure, which has extremely large influence on performance. There is little doubt that with further manufacturing development, much higher heat transport capacities can be achieved.

H2O HEAT PIPE VACUUM TEST

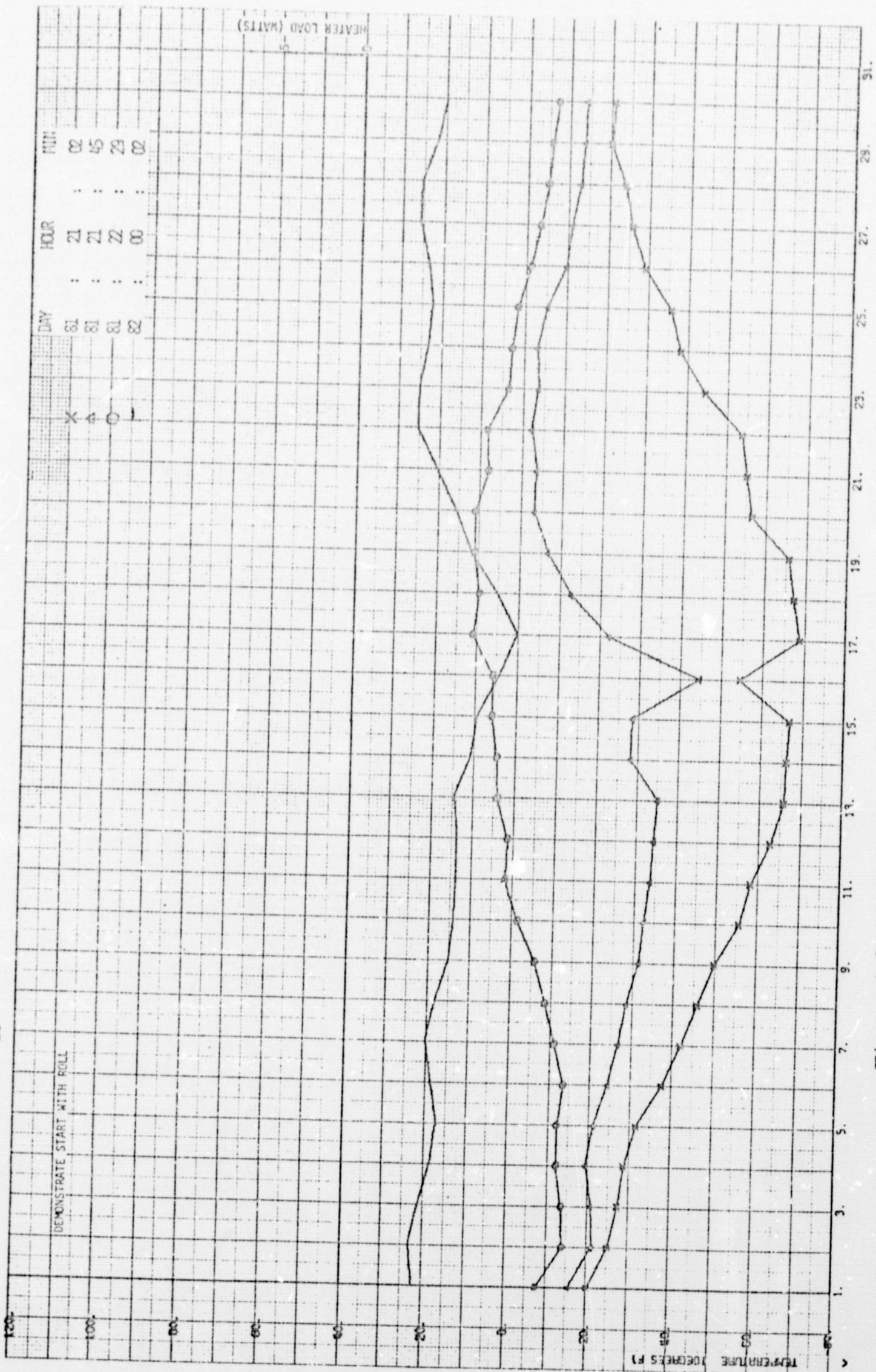


Figure 4-27a. Side-Sun Modules 19-23, Roll Phase

H2O HEAT PIPE VACUUM T

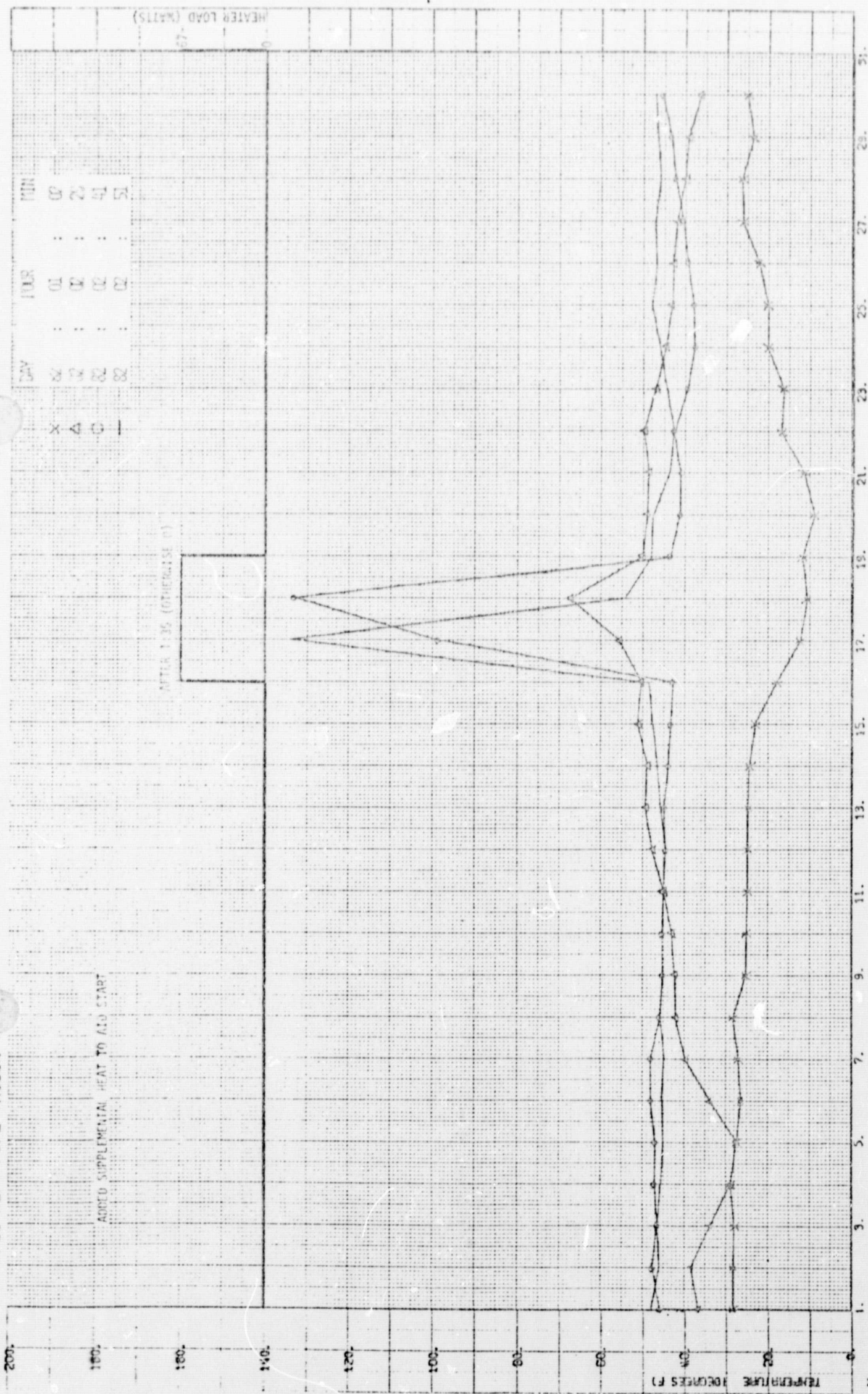


Figure 4-27b. Side-Sun Modules 19-23, Roll Phase

In addition to successful steady-state operation, the tests also demonstrated the uniform temperature transient response under changing loads, characteristic of heat pipes.

4.4.3.2 Diffusion Freezeout

Several of the test phases provided demonstrations of the diffusion freezeout failure mode. This is of major importance in large heat pipe systems such as might be used on Space Base. If such pipes are to be required to start-up from a frozen state, they will likely contain non-condensable gases to aid the start-up process. It was amply demonstrated in these tests that, if low sink temperatures cause a portion of the gas-blocked region to fall below the freezing point of the working fluid, the pipe can fail by continuous freezing of vapor which diffuses into the gas zone, resulting in evaporator dryout.

It was also shown that such failures can be avoided by:

- 1) Rotating the heat pipe such that (a) the cold zone continuously changes position, allowing remelt and recapture of the frozen working fluid, or better still, (b) a gas-blocked cold zone does not even form.
- 2) Establishing a combination of working fluid, cold zone length, pipe to sink temperature differential and axial conductivity of pipe and fin such that no portion of the gas-blocked region falls below the freezing point of the working fluid.

From an applications standpoint, the latter procedure is clearly preferable.

4.4.3.3 Start-up from the Frozen State

The program plan called for studying this aspect of large pipe technology with a gas-dosed ammonia heat pipe. Because such a heat pipe was not available (see Section 4.2.2), whatever was to be learned about start-up had to be accomplished with the water pipe. Fortunately, the water pipe contained some noncondensable gas which allowed the gathering of some meaningful test data.

Several frozen start-up attempts were made, under various combinations of stationary or rotational conditions with both radiant and electrical heat loads. A successful start-up was achieved under stationary conditions with a Y-configuration electrical load (3 heaters), thus

proving the feasibility of frozen start-ups with very long heat pipes. However, even in the unsuccessful start-up attempts, it was possible to establish that the appropriate start-up mechanisms were operating, suggesting that success would be achieved under proper conditions.

The stationary start-up transients under both electrical and radiant heat loads clearly showed the formation of noncondensable gas fronts which resulted in the desired progressive, rather than uniform, temperature transients. However, the quantity of gas in the pipe was such that the evaporator remained below freezing as the hot zones expanded, leading to eventual dryout in the absence of a fluid return mechanism.

A different approach to start-up, that of rotating the heat pipe with respect to a radiant heat source, also yielded promising results. Rotation prevented evaporator dryout, as expected. The start-up was not successful for the simple reason that the radiant heat input load was insufficient to bring the pipe above the freezing point. If the working fluid had a lower freezing point (e.g., ammonia, methanol, Freon), this start-up technique would have certainly succeeded.

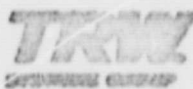
5. CONCLUSIONS

Specific conclusions associated with each aspect of the program are embodied in the text. Major conclusions derived from this study were:

- Modular type construction of long arterial heat pipes is feasible and was successfully demonstrated.
- Very long heat pipes can be fabricated with concentric annular arteries to handle heat loads of interest in large spacecraft. However, development of modularization techniques for internal-type nonsuperheated arteries promises superior performance.
- At spacecraft thermal control temperatures, ammonia can sustain at least 7°F superheat if the heat pipe is carefully processed. However, the presence of non-condensable gases drastically reduces this superheat tolerance. On the other hand, water can tolerate substantial superheat even in the presence of non-condensable gas.
- Use of inert gas for start-up of long frozen heat pipes is feasible. However, with ammonia as the working fluid, the heat pipe must be designed to avoid appreciable superheating of the artery so as to prevent arterial nucleation.
- Diffusion freezeout is a significant problem which must be considered in the design of heat pipes using noncondensable gas. The preferable design approach is to assure that the gas-blocked region does not fall below the freezing point of the working fluid, at least for long periods of time while the heat pipe is under load.
- If diffusion freezeout and arterial nucleation are avoided, stable operation of arterial pipes is possible in the presence of noncondensable gas. Gas transport to the evaporator as a result of temperature dependent variations in solubility in the liquid does not necessarily lead to arterial failure.
- If gas is present in an annular-type artery, larger than anticipated lengths of the condenser can be "shut off" and shielded from the pumping action of the flowing vapor. This condition would not be a problem with internal-type arteries where gas blockage does not interfere with condensing heat transfer.

APPENDIX A

THERMAL VERIFICATION TEST PLAN AND PROCEDURE



INTEROFFICE CORRESPONDENCE

TO: Distribution

CC:

DATE 70-8526.13-1
1970 January 7

SUBJECT: Test Plan & Procedure - Thermal Verification
Test - NASA/MSC 15' Heat Pipe Systems

FROM: G. W. Clausen
BLDG: B4 MAIL STA: 2028 EXT: 313

TRW Systems is building two 15' diameter hula hoop heat pipe systems for NASA/MSC. The primary objective of the thermal verification tests described in this report is to verify acceptable operation of these heat pipe systems. By acceptable operation is meant the ability of the pipe to minimize the temperature gradient around the circumference for the condition of one sun incident on one side of the cylinder.

In all cases the tests are to be conducted in air for simplicity and are to utilize the heaters and instrumentation installed for the Thermal Vacuum Test in the NASA/MSC Chamber A Test Facility.

TEST UNITS

The test units are both 183" diameter hula hoop heat pipes with heat rejection fins attached. One of the pipe systems uses water as the working fluid, the other uses ammonia. The units may be tested together with the 8 interconnecting fiberglass supports, or may be tested separately; the thermal interchange between the two during normal operation is negligible.

The fins are coated with an aluminum paint (Finch Polyurethane Coating No. 643-1-1) which is very susceptible to damage during handling. Consequently, there should be no contact with this surface after application; all moving, lifting, etc., is to be accomplished using other, non-painted surfaces. It is not necessary that the heat pipe systems be painted for verification testing.

Thermocouples attached to the test unit are shown in Figure 1; the ammonia pipe has the prefix A and the water pipe the prefix W. There are 35 copper-constantan TC's (28 gauge) attached to each heat pipe system. The TC's are spot welded directly to the heat pipe on the insulation side of the copper fin and are strung along the fin for at least 12 inches from the point of application to minimize conduction errors. Exceptions are TC's 16 and 27 which are positioned at the extremity of the fin to measure the longitudinal gradient of the respective fins; see Figure 2 for the application of the fin TC's. All thermocouples are bundled together at Module 3 and supplied with 30' of extra cable for mating to the appropriate connector in the NASA/MSC Chamber A Thermal-Vacuum Facility. All TC's are unshielded. However, the bundle will have RF shielding to prevent induced EMF's.

Thermocouples A33 and W33 will be positioned on two legs (P4 & P9) of the support stand. These TC's are not used during verification testing.

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Test Plan & Procedure - Thermal Verification Test - NASA/MSC 15' Heat Pipe Systems

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Heat input is provided with electric resistance heaters bonded to each 5' module. A typical heater is 50" long (the fin length is 57- $\frac{1}{2}$ " long) by 1- $\frac{1}{2}$ " wide and consists of approximately 200 feet of Manganin heater wire. The wire is wound on top of 3 mil Teflon tape with a second layer applied over the top of the wire for total electrical insulation. Aluminum tape applied over the top of the assembly aids in holding the heaters in place. The design has been utilized with success on numerous TRW Spacecraft thermal model tests. The heater designation is identical with the module designation; i.e., heater A3 is on Module A3 (third module of the ammonia pipe). Each heater is capable of inputting 100 watts at 28 VDC input (measured resistance = 7.0 Ω) and will require control of the input voltage to achieve the specified power input. The effect of temperature on heater resistance is shown in Table 1. Over the specified range the total variation in heater resistance is less than .05 Ω .

Heater leads (20 for each pipe) are bundled together at Module 8 and provided with 30' of extra length for mating with the NASA/MSC Chamber A Facility. The average resistance of each heater and lead wire is 7.6 Ω .

TEST INSTRUMENTATION

Test instrumentation required to conduct the Thermal Verification Test is listed below:

- Recording potentiometers to read and record the individual copper-constantan thermocouples. The potentiometers must be capable of providing a reference point (ice point) for reading the TC's.
- Two 28 volt DC or AC power supplies, one for each heat pipe system. Each unit should have a 15 amp capacity.
- Six DC variacs for adjusting power inputs to the individual heaters. Three variacs (or separate, adjustable power supplies) are required for each heat pipe system.
- Two voltmeters (0 to 35 volts) and two ammeters (0 to 10 amps) one for each pipe unit are required for power measurements.

The hookup for the power supplies and variacs is shown in Figure 3.

TEST PROCEDURE

Detailed procedures for conduct of the Thermal Verification Test are as follows:

- Level the heat pipe systems to be tested. All elements of the pipe must be leveled to within 1/8 inch to insure priming of the liquid artery.
- With all power supplies off and the heat pipe systems in thermal equilibrium read all 72 thermocouples. Note deviations from the mean.
- Adjust variacs A1, A3, W1 & W3 to zero and adjust variacs A2 and W2 to slowly provide 55 watts input heat to heaters A1 and W1. Measure all 70 thermocouples every 30 minutes until reading does not

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Test Plan & Procedure - Thermal Verification Test - NASA/MSC 15' Heat Pipe Systems.

1970 January 7

deviate from the last reading by more than 2°F (called equilibrium in subsequent discussions). Record also the ΔV across the variac and ΔI to the heater.

- Connect the A2 and W2 variacs to heaters A3 and W3, respectively, and adjust the input power to 55 watts as before. Record all thermocouples until equilibrium is established. Record also the ΔV and ΔI . The voltage measurements at the power supply must be corrected to account for line losses.
- Repeat the above tests for heat inputs to A5 and W5, A7 and W7 and A9 and W9. For each test, the preceeding heater should be set to zero; e.g., A1 and W1 power=zero for A5 and W5 test.
- Connect the power supplies as shown in Figure 3. Slowly, adjust the variacs to give power inputs per the following table.

Variac	Heaters	Power, Watts
A1, W1	A2, A10, W3, W10	43.7 Watts
A2, W2	A1, W1	54.0 Watts
A3, W3	A5, A9, W3, W9	16.7 Watts

- Record all TC's every 30 minutes until equilibrium is established. Measure also the ΔV 's across each variac and the ammeter readings (A1 and W1). Voltage measurements must be corrected to account for line losses.

PREDICTED TEST RESULTS

A non-operating heat pipe system of the test geometry with 55 watts heat input to a single module will have a predicted overall ΔT of approximately 100°F . These assumptions are sensitive to ambient air temperature and velocity, and can vary somewhat from one test to another. An operational pipe (water and ammonia) is calculated to have a ΔT of less than 15°F for the same test condition. This ΔT is to be measured over the active portion of the pipe; i.e., that portion not blanketed with excess liquid or non-condensable gas. In the case of the water, the active length is essentially the entire length (95%); the active portion of the ammonia pipe will be determined during the thermal verification testing at Redondo Beach.

For the one sun solar input run (last case) the total pipe ΔT should also not exceed 15°F over the active portion of the pipe. The active portion of the pipe is defined as before. A non-operating pipe under the same conditions is expected to have a maximum ΔT of approximately 100°F . Note that these calculations are based on assumed environmental conditions and convective film uncertainties. The temperature gradients, however, should remain valid over reasonable variations in the absolute temperature level.

OPERATING RESTRICTIONS

The water and ammonia heat pipe systems should not be subjected to temperature in excess of 300°F . The lower limit can be as low as LN_2 temperatures

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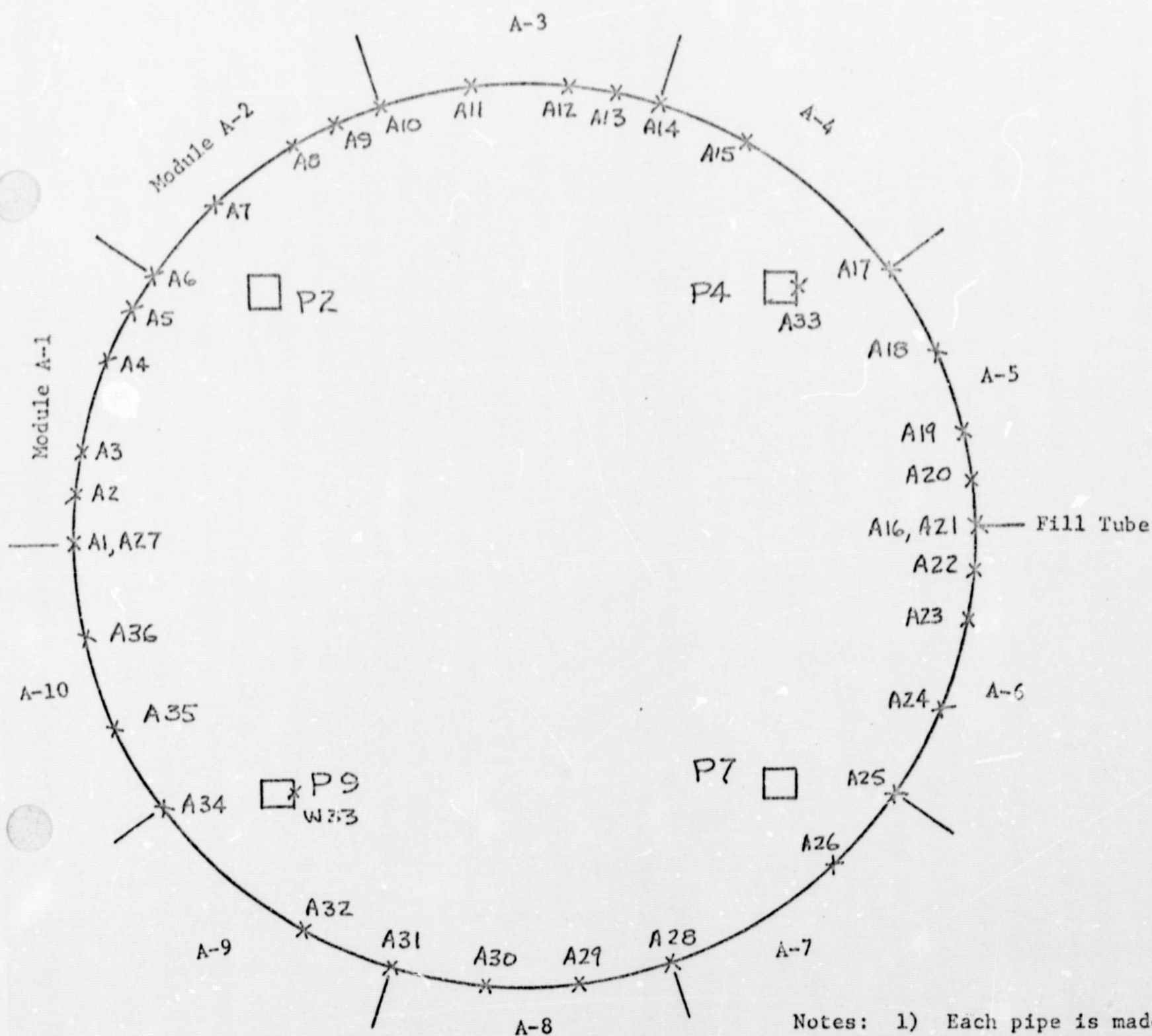
-330°F) but the
-105°F). Restart
Chamber A Thermal
from a frozen condition
that breaking the ice
allow priming to

will be frozen (water at 32°F and ammonia at
ices is uncertain and is one objective of the
Testing. The conditions and effects of a restart
cannot be reliably predicted at this time. Note
during testing requires shut down of the pipe to
can be resumed.

RESPONSIBILITIES

This Test
and variations
for the Redondo
Verification Test

is under the control of the appointed Test Conductor
cedure must be authorized by him. The Test Conductor
ification Test is O. W. Clausen and W. Piske for the
conducted at NASA/MSC.

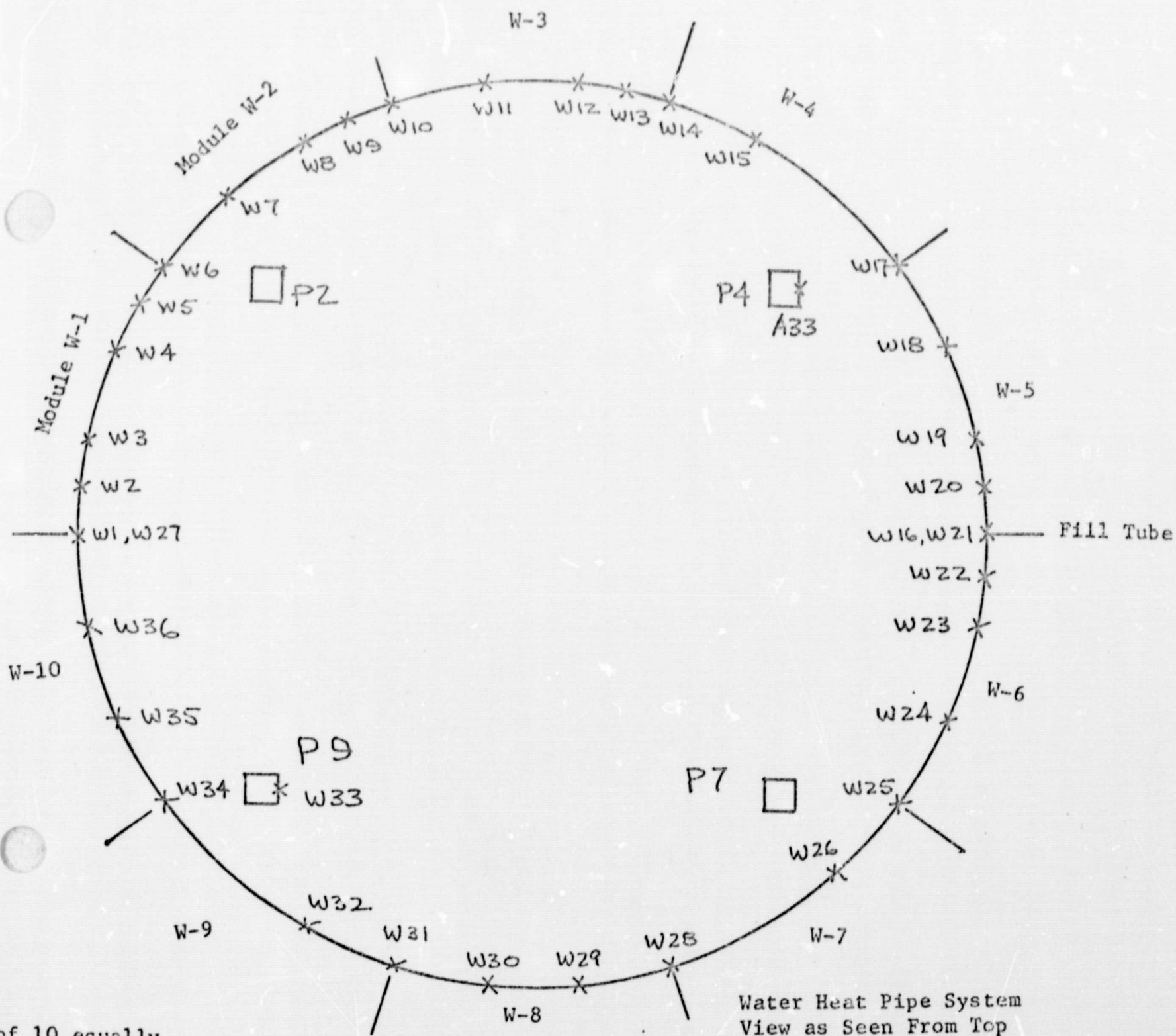


Amonia Heat Pipe System
View as Seen From Top

FOLDOUT FRAME

- Notes:
- 1) Each pipe is made up of spaced modules as shown drawings.
 - 2) See Figure 2 for details of thermocouples.
 - 3) TC's A 33 & W33 are located support stand legs (See

FIGURE #1: THERMOCOUPLE LOCATIONS - NASA/MSFC 15' HEAT

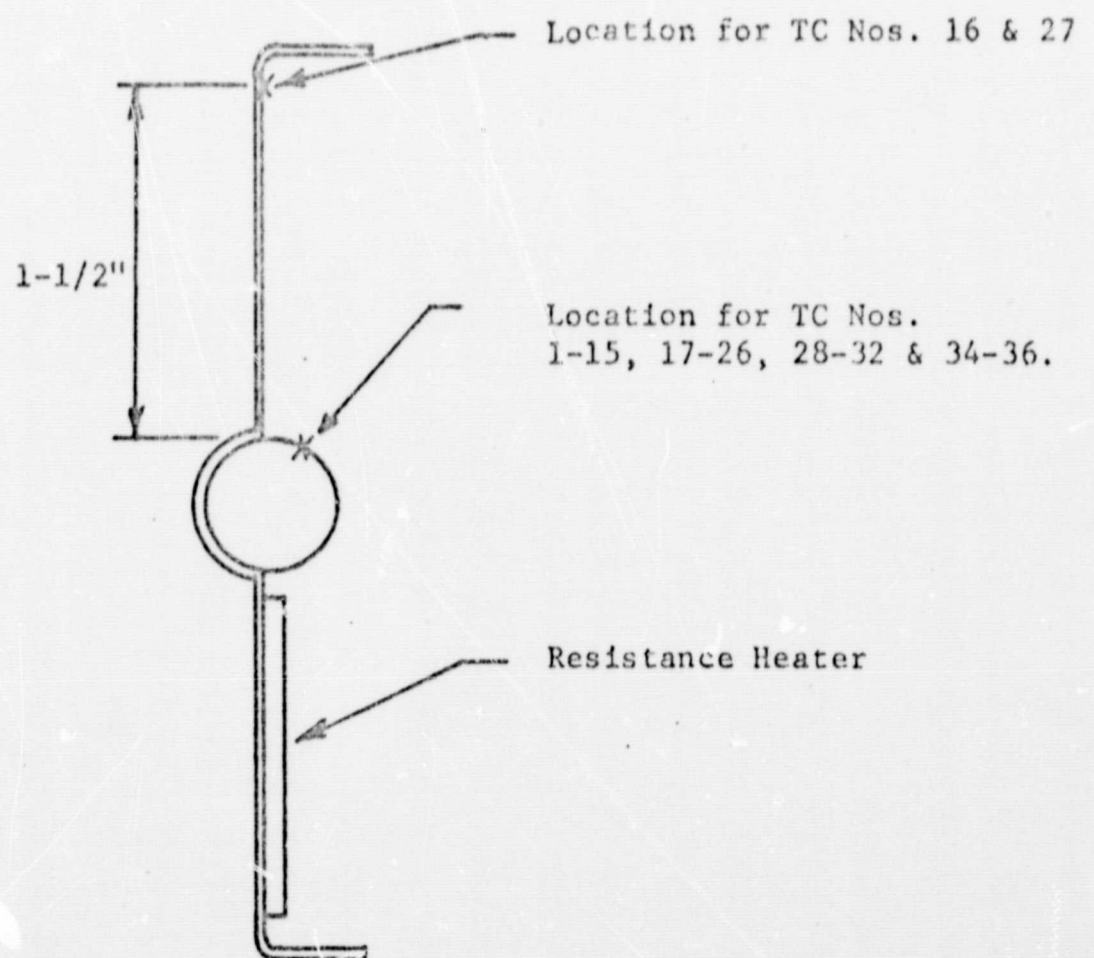


FOLDOUT FRAME

X -- Thermocouple Locations

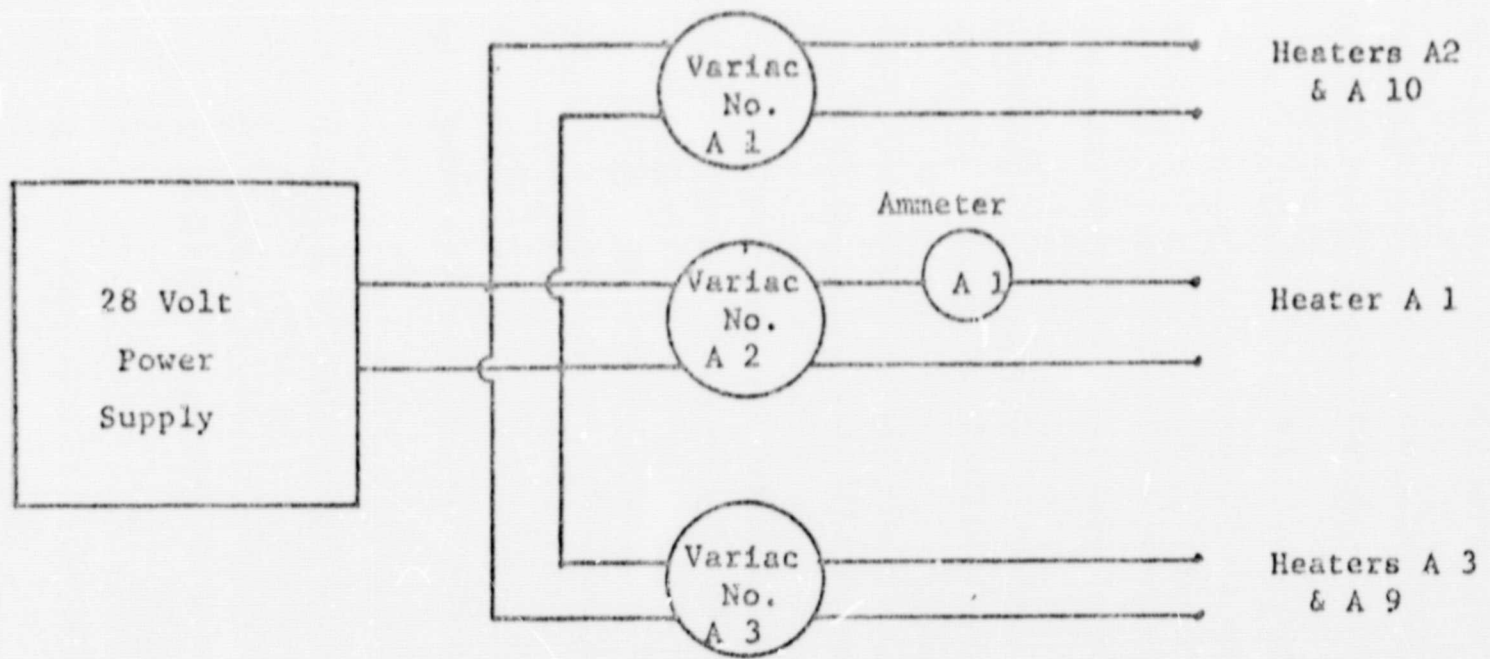
□ -- Vertical Support Posts

up of 10 equally
shown in the above
etailed placement
e located on the
(See Figure 1)

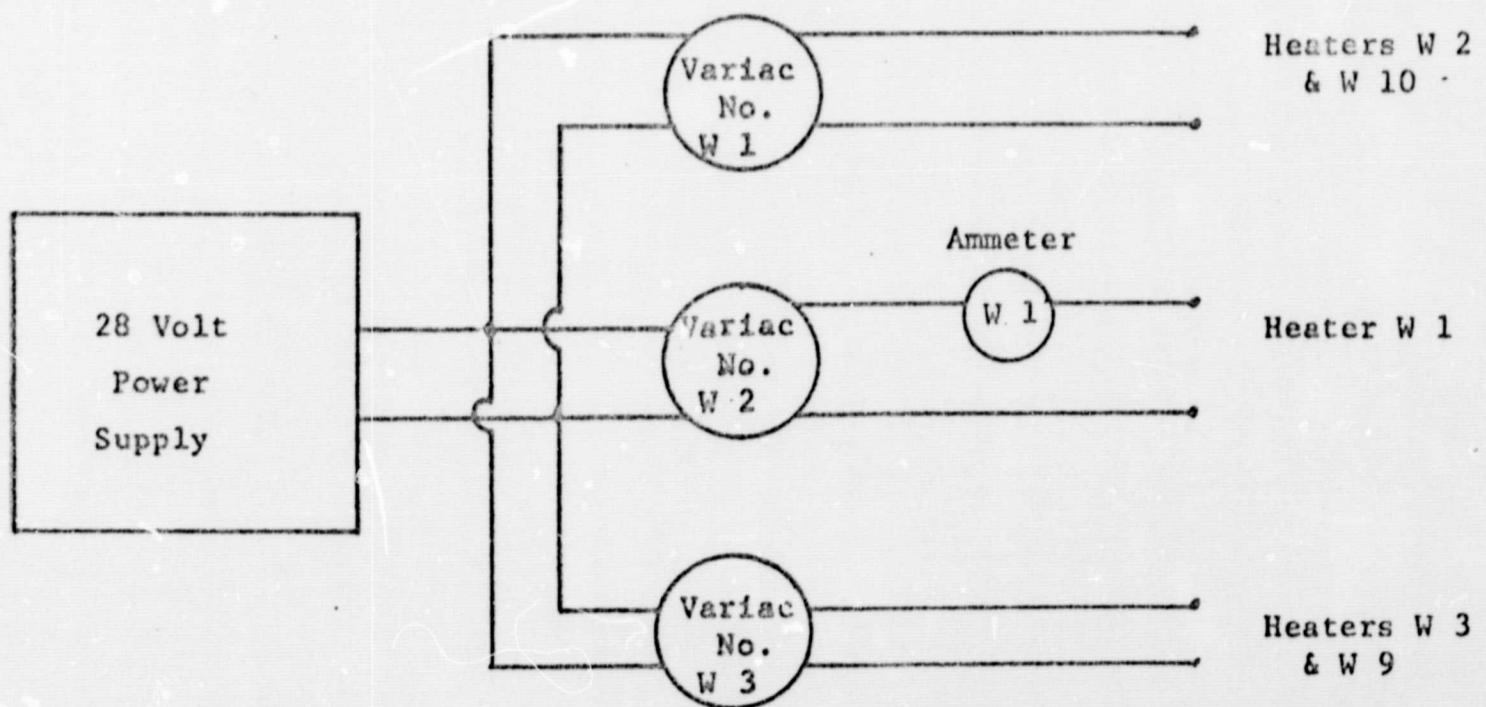


See FIGURE 1 for Plan View of TC's.

FIGURE #2: Thermocouple Application - NASA/MSX 15' Heat Pipe Systems



AMONIA PIPE



WATER PIPE

FIGURE #3: Power Supply for Heat Pipe System Test Heaters

TABLE 1: RESISTANCE CHANGE VS TEMPERATURE
MANGANIN WIRE STOCK

Temperature °C	% Change in Resistance
20	0
30	-.0036
40	-.0150
50	-.0357
60	-.0621
70	-.0963
80	-.1270
90	-.1890

APPENDIX B
THERMAL VACUUM TEST TIMELINE

APPENDIX B

THERMAL VACUUM TEST TIMELINE

The following pages depict the actual test timeline. It presents the test events and all heater settings with appropriate remarks to explain the rationale used during the test to implement the timeline. The timeline, as implemented, was basically trial and error changes to explore many alternative heat load profiles under which the water heat pipe would operate. The planned test timeline is contained in the Subsystem Test Bed Thermovacuum Test, Test Procedures, Test No. 25-A-70, Space Environment Test Division, MSC, NASA, dated March 6, 1970. Because the planned test timeline was not carried out during the test, it has no relationship to the recorded data and, therefore, will not be repeated here.

ACTUAL HEAT PIPE TIMELINE

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/10/70(69)*			
1120 hrs.	Pumpdown started	No power	
3/11/70(70)			
1346 hrs.	Pumpdown	A1 = 0 W1 = 2.5 2 = 0 2 = 2.3 3 = 0 3 = 1.4 4 = .8 4 = 0 5 = 1.3 5 = 0 6 = 1.4 6 = 0 7 = 1.3 7 = 0 8 = .8 8 = 0 9 = 0 9 = 1.4 A10 = 0 W10 = 2.3	"Hold" setting per Table 1, STB T/V Test Procedures Test No. 25-A-70.
1805	LN ₂ cooldown begins	Same power	
2330	Pumpdown + LN ₂	Same power	TX0017 reading 40°F; the first indication of H ₂ O pipe freezing.
3/12/70(71)			
0530 hrs.	Same	A1 = 0 W1 = 1.5 A2 = 0 W2 = 1.5 A3 = 0 3 = 1.5	Heaters set to "Thaw" setting.

* () is the day number of the year

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/12/70(71)	Pumpdown + LN ₂	4 = .5 4 = 1.5 5 = .5 5 = 1.5 6 = .5 6 = 1.5 7 = .5 7 = 1.5 8 = .5 8 = 1.5 9 = 0 9 = 1.5 A10 = 0 W10 = 1.5	Heaters set to "thaw" setting.
0530 (cont.)			
0645		A1,2,3,9,10 = 0 A4,8 = 0.8 A5,7 = 1.3 A6 = 1.4 W1,2,3,9,10 = 0 W4,5,6,7,8 = 1.8	
0900		"A" = same W1 to 10 = 1.8	
0946		"A" = same W1,2,3,9,10 = 0 W4,8 = 1.4 W5,7 = 2.3 W6 = 2.5	
228	Same	"A" = same W1,2,3,4,5,7,8,9,10 = same W6 = 2.3	W6 reduced because TX0018 started to boil out. (123°F) versus 88 & 89°F.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/12/70 (71) 1252 hrs.	Pumpdown + LN ₂	"A" = same W1 = 2.5 W2,10 = 2.3 W3,9 = 1.4 W4,5,6,7,8 = 0	Moved "Hold" setting to opposite side of H ₂ O pipe to recover boil out.
1846	Same	A1 to A10 = 0 W1 to W10 = 0	Test abort due to inability to reduce chamber pressure below 10 ⁻² torr.
3/16/70 (75) 0828 hrs.	Begin pumpdown Pumpdown	A1,2,3,9,10 = 0 A4,8 = 0.8 A5,7 = 1.3 A6 = 1.4 W1 = 2.5 W2,10 = 2.3 W3, 9 = 1.4 W4,5,6,7,8 = 0	"A" and "W" heaters set to "hold".
0906	Same	A1,2,4,5,7,9,10 = 0 A3,6,8 = 0.8 W1,2,4,5,7,8,10 = 0 W3,6,9 = 1.4	"A" heaters set to ".3Y" setting. "W" heaters set to ".3I" setting (see planned timeline). This setting changed because top & side sun pre-operations being performed.
1134	Same	A1,2,3,9,10 = 0 A4,8 = 0.8 A5,7 = 1.3 A6 = 1.4	"A" and "W" heaters returned to "hold" setting prior to entering cooldown at 1154 hours. Lunar plane rotation started at 1310 hours 2rph with return at 15 rph.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS.	REMARKS
3/16/70(75)	Pumpdown	W1 = 2.5 W2,10 = 2.3 W3,9 = 1.4 W4,5,6,7,8 = 0	
1134 (cont.)			
1440	Pumpdown + LN ₂ + roll	A1 to 10 = same W1 = 2.6 W2,10 = 2.4 W3,9 = 1.5 W4,5,6,7,8 = 0	"W" heaters set to 1.1S (1.1 equivalent sun) in an effort to maintain cold region above freezing.
1500	Pumpdown + LN ₂ + roll	A1 to 10 = same W1 = 2.7 W2,10 = 2.5 W3,9 = 1.5 W4,5,6,7,8 = 0	"W" heaters set to 1.2S (1.2 equivalent sun) to maintain cold region above freezing. Cold region - continues to decrease.
1515	Same	A1 to A10 = same W1 = 2.9 W2,10 = 2.6 W3,9 = 1.6 W4,5,6,7,8 = 0	"W" heaters set to 1.3S for same reason. Cold region continues to decrease as evaporator starts boil out.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/16/70 (cont.) 1535 hrs.	Pumpdown + LN ₂ + roll	A1 to A10 = same W1,2,3,9,10 = 0 W6 = 2.5 W7,5 = 2.3 W4,8 = 1.4	"W" heater set to "Rev Hold" to reduce boil out and eliminate freezing. This method was very efficient in recovering the H ₂ O pipe.
1715	Same	A1 to A10 = same W1 = 2.5 W2,10 = 2.3 W3,9 = 1.4 W4,5,6,7,8 = 0	"W" heaters set to "Hold" opposite side of pipe for recovery.
1758	Same	A1 to A10 = same W1 = 2.5 W2,10 = 2.3 W6 = 1.4 W3,9 = 1.4 W4,5,7,8 = 0	W6 heater activated to maintain cold region above freezing.
1840	Pumpdown + LN ₂ + roll	A1 to A10 = same W1,2,3,9,10 = 0 W6 = 2.5 W5,7 = 2.3 W4,8 = 1.4	"W" heaters set to "Rev Hold" for recovery.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/16/70 (cont.) 1925 hrs.	Same	A1 to A10 = same W1 = 2.1 W2,10 = 1.9 W3,9 = 1.4 W4,5,6,7,8 = 1.0	"W" heaters set to "Warm" to maintain cold side above freezing.
1935	Same	A1,2,3,9,10 = 0 A4,8 = 1.0 A5,7 = 1.7 A6 = 1.8 W1 to 10 = same	Increase power to "A" heaters to .5S to attempt to operate methanol at higher heater setting.
2025	Same	A1 to A10 = same W1 to W10 = 1.5	Set "W" heaters to "Thaw" in attempt to recover H ₂ O pipe.
2050	Same	A1 to A10 = same W1 to W10 = 1.7	Increase "W" heaters to speed up recovery. "Thaw +" setting.
2153	Same	A1 to A10 = same W1 = 2.8 W2,10 = 2.5 W3,9 = 1.6 W4,5,6,7,8 = 0	Set "W" heaters to TRW thermal verification heater settings.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/16/70 (cont.)			
2300 hrs.	Same	A1 to A10 = same W1 to W10 = 1.7	Set "W" heaters to "Thaw +" for recovery.
3/17/70(76)			
0015 hrs.	Pumpdown + LN ₂ + roll	A1 to A10 = same W1 = 2.8 W2,10 = 2.5 W3, 9 = 1.7 W4,5,6,7,8 = 1.0	Set "W" heaters to "Hot" setting to see if H ₂ O could be maintained above freezing.
0100	Pumpdown + LN ₂ + roll	A1 to A10 = same W1,2,3,9,10 = 1.0 W4,8 = 1.7 W5,7 = 2.5 W6 = 2.8	"W" heaters set to "Rev Hot" to recover low temp (TXC017 = 30°F) in cold region.
0140	Test conditions met P chamber = 5 x 10 ⁻⁵ torr coldwall stabilized side sun soak	A1 to A10 = 0 W1 to W10 = 0	All heaters set to zero for beginning of side sun burn with no roll.
0210	Same	A1 to A10 = same W1 = 1.2 W2,10 = 1.0 W3,9 = 0.6 W4,5,6,7,8 = 0	"W" heaters set to .2S to supplement side sun burn of 80%.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/17/70 (cont.)			
0225	Same	A1,2,3,9,10 = 0 A4,8 = .6 A5,7 = 1.0 A6 = 1.2 W1 to W10 = same	"A" heaters set to .2S to have same heat input as H ₂ O pipe.
0240	Same	A1 to A10 = 0 W1 to W10 = 0	All zero because side sun operating at approx. 1 equivalent sun.
0820	Same	A1 to A10 = 0 W1 to W10 = 1.7	"W" heaters set to "Thaw".
0830	Same	A1 to A10 = 0 W1 to W10 = 0	"W" heaters set to zero to continue side sun soak.
1224	Same	A6 = 2.5 A5,7 = 2.3 A4,8 = 1.4 A1,2,3,9,10 = 0 W7 = 2.5 W6,8 = 2.3 W5,9 = 1.4 W1,2,3,4,10 = 0	"A" heater settings now modified to correspond to "W" heaters for comparison in "Rev Hold". "W" heaters set to "Rev Hold" with max power rotated opposite sub solar.
1625	Same	A1 to A10 = same W1,5 = .93 W2,4 = .91 W3,6,7,10 = .94 W8,9 = .95	"W" heaters set to maintain H ₂ O pipe above freezing by placing a uniform heat load on each section. Amperages vary for each heater because of different resistance values for each. Resistances measured during acceptance test.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/17/70 (cont.) 1931	Same	A1,2,3,9,10 = 0 A4,8 = 1.4 A5,7 = 2.3 A6 = 2.5 W1,2,3,4,10 = 0 W5,9 = 1.4 W6,8 = 2.3 W7 = 2.5	"W" & "A" heaters set to "Rev Hold" to recover H ₂ O pipe. "W" heaters rotated one heater section to correspond to 180° sub solar.
2158	Same	A1,2,3,9,10 = 0 A4,5,6,7,8 = 1.3 W1,2,3,4,10 = 0 W5,6,7,8,9 = 1.3	
2305	Same	A1 to A10 = 0 W1,2,3,4,10 = 0 W5,6,7,8,9 = 1.1	Reduced power because of boil out. All "A" heaters zero. No data expected.
2320	Same	A1 to A10 = 0 W1,2,3,4,10 = 0 W5,9 = 1.4 W6,8 = 2.3 W7 = 2.8	To recover from boil out at subsolar;
2335	Same	A1 to A10 = 0 W1,2,3,4,10 = 0 W5,6,8,9 = 2.5 W7 = 2.8	To speed recovery.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/17/70 (cont.)			
2349 hrs.	Same	A1 to A10 = 0 W1,2,3,4,10 = 0 W5,6,7,8,9 = 1.1	To operate and maintain cold portion above freezing.
3/18/70(77) 0341 hrs.	Same	A1 to A10 = 0 W1,2,3,4,10 = 0 W5,6,8,9 = 2.5 W7 = 2.8	H ₂ O pipe was freezing. "W" heaters increased to recover prior to side sun with roll.
0445	Side sun roll	A1 to A10 = 0 W1 to W10 = 0	Begin side sun roll.
0806 to 0814	Side sun roll	A1,2,3,9,10 = 0 A4,8 = 1.3 A5,6,7 = 1.4 W1 to W10 = 0	No reason for "A" heaters activated for 8 min. All heaters at zero after this error.
1213	Same	A1,2,4,5,7,9,10 = 0 A3,6,8 = 1.4 W1,2,4,5,7,8 = 0 W3,6,9 = 1.4	"A" and "W" heaters set at ".31" load to add total heat input, because H ₂ O pipe uniformly at freezing temperature from roll mode.
1443	Same	A1 to A10 = 0 W1,2,3,9,10 = 1.1 W4,8 = 1.4 W5,7 = 2.3 W6 = 2.5	"A" heaters set at zero since no recovery setting can be made due to heaters A1,2,9,10 being inoperative. "W" heaters set to "Bias Hold" in an effort to find a setting for heat pipe operation.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/18/70 (cont.) 1610 hrs.	Cold Soak roll	A1,2,4,5,7,9,10 = 0 A3,6,8 = 2.5 W1,2,4,5,7,8,10 = 0 W3,6,9 = 2.5	"A" heaters set to .9Y load for comparison. "W" heaters set to .9Y load to test for heat pipe operation.
2007	Same	A1,2,4,5,7,9,10 = 0 A3,6,8 = 2.5 W1,2,3,9,10 = 1.1 W4,8 = 1.4 W5,7 = 2.3 W6 = 2.5	"W" heaters set to "Bias Hold" for precondition prior to going to 1.3Y load.
2043	Same	A1 to A10 = same W4,5,6,7,8 = 1.1 W3,9 = 1.4 W2,10 = 2.3 W1 = 2.5	"W" heaters set to "Rev. Bias Hold" for same reason.
2115	Cold Soak roll	A1 to A10 = 0 W3,6,9 = 3.0 W1,2,4,5,7,8,10 = 0	"A" heaters set to zero because no recover setting. "W" heaters set to 1.3Y load.
3/19/70(78) 1427	Same	A1 to A10 = 0 W3,6,9 = 3.2 W1,2,4,5,7,8,10 = 0	Increased "W" heaters to establish max power density in "Y" load configuration, 1.46Y

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/19/70 (cont.) 1847 hrs.	Same Top sun on (2316)	A1 to A10 = 0 W3,6,9 = 3.4 W1,2,4,5,7,8,10 = 0	"W" heaters set to 1.65Y. Top sun on (2316) - this increased operating temp. considerably.
3/20/70(79) 1010 hrs.	Top Sun off (1009) Cold soak roll	A1 to A10 = 0 W3,6,9 = 3.6 W1,2,4,5,7,8,10 = 0	"W" heaters set to 1.8Y.
1400	Same	A1 to A10 = 0 W1 to W10 = 0.8	Seeking a setting for "W" heaters to maintain above freezing.
1440	Same	A1 to A10 = 0 W1 to W10 = 0.9	Same
1505	Cold soak roll	A1 to A10 = 0 W1 to W10 = 1.1	Below freezing.
1520	Same	A1 to A10 = 0 W1 to W10 = 1.5	Below freezing, must get out latent heat of fusion.
1550	Same	A1 to A10 = 0 W1 to W10 = 1.7	Same
1600	Same	A1 to A10 = 0 W1 to W10 = 1.9	Same
1623	Same	A1 to A10 = 0 W1 to W10 = 1.7	Seeking a setting for "W" heaters to maintain above freezing.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/20/70 (cont.) 1651 hrs.	Same	A1 to A10 = 0 W1 to W10 = 1.5	Same After ~ 2 hours operation at 45-50°F this setting established as base line.
1840	Same	A1 to A10 = 0 W1,2,3,9,10 = 1.5 W4,8 = 1.7 W5,7 = 1.9 W6 = 2.1	"W" heaters set to 0.3ΔS.
2108	Same	A1 to A10 = 0 W1,2,3,9,10 = 1.5 W4,8 = 1.9 W5,7 = 2.3 W6 = 2.4	"W" heaters set to 0.6ΔS.
2309	Cold soak roll	A1,2,9,10 = 0 A4 = 2.9 A3,5 = 2.6 A6 = 2.0 A7,8 = 1.5 W1,2,3,9,10 = 1.5 W4,8 = 2.0 W5,7 = 2.6 W6 = 2.9	Set "A" heaters to simulate 0.9ΔS in an attempt to obtain comparison data for H ₂ O pipe. "W" heaters set to 0.9ΔS. Gradient under heaters A4,5,6,7,8 can be used as comparison to W6,7,8,9,10.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/21/70(80) 0202 hrs.	Same	A1,2,9,10 = 0 A4 = 3.3 A3,5 = 3.0 A6 = 2.1 A7,8 = 1.5 W1,2,3,9,10 = 1.5 W6 = 3.3 W5,7 = 3.0 W4,8 = 2.1	"A" & "W" heaters set to 1.3ΔS.
0300	Same	A1,2,9,10 = 0 A4 = 2.9 A3,5 = 2.6 A6 = 2.0 A7,8 = 1.5 W3,4,5,6,7,8,9 = 2.5 W2,10 = 3.0 W1 = 3.3	"W" heaters changed to recover from "boil out."
0400	Cold soak roll	A1,2,9,10 = 0 A4 = 3.1 A3,5 = 2.8 A6 = 2.0 A7,8 = 1.5	"A" and "W" heaters set to 1.1ΔS.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/21/70			
0400 (cont.)	Cold soak roll	W1,2,3,9,10 = 1.5 W6 = 3.1 W5,7 = 2.8 W4,8 = 2.0	"A" and "W" heaters set to 1.1ΔS.
0500	Same	A1,2,9,10 = 0 A4 = 2.9 A3,5 = 2.6 PC = 2.0 A7,8 = 1.5 W3,4,5,6,7,8,9 = 2.5 W2,10 = 3.0 W1 = 3.3	Set "A" & "W" heaters to recover prior to entering "Dual" setting.
0555	Same	A1,2,4,5,6,7,9,10 = 0 A3,8 = 3.3 W1,2,4,5,6,7,9,10 = 0 W3,8 = 3.3	Set "A" & "W" heaters to "1.06 Dual" setting. Heat input = 1.06 equivalent sun on two diametrically opposed heaters. This divided the pipe into 4 active heat pipes which reduces the wick pressure drop for each.
0905	Same	A1,2,4,5,6,7,9,10 = 0 A3,8 = 3.6 W1,2,4,5,6,7,9,10 = 0 W3,8 = 3.6	Increased "A" & "W" heaters to "1.2 Dual" setting.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/21/70 (cont.) 0956 hrs.	Cold soak roll	A1,2,9,10 = 0 A3,4,5,6,7,8 = 2.5 W3,4,5,6,7,8,9 = 2.5 W2, 10 = 3.0 W1 = 3.3	Set "A" & "W" to recover from boil out.
1029	Same	A3,8 = 3.6 A1,2,4,5,6,7,9,10 = 0 W3,8 = 3.6 W1,2,4,5,6,7,9,10 = 0	"A" & "W" set to "1.2 Dual".
1200	Same	A1,2,9,10 = 0 A3,4,5,6,7,8 = 2.5 W1 = 3.3 W2,10 = 3.0 W3,4,5,6,7,8,9 = 2.5	"A" & "W" heaters set to recover from boil out and prepare for cold soak.
1300	Same	A1 to 10 = 0 W1 to 10 = 0	All power zero for cold soak.
1602	Same	A3,6,8 = 3.6 A1,2,4,5,7,9,10 = 0 W3,6,9 = 3.6 W1,2,4,5,7,8,10 = 0	"1.8 Y" heater start from frozen state.
1716	Same	A3,6,8 = 3.4 A1,2,4,5,7,9,10 = 0 W3,6,9 = 3.4 W1,2,4,5,7,8,10 = 0	All heaters reduced because of approaching max temp on heater W3 and A3 (~250°F).

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/21/70 (cont.) 2103 hrs.	Cold soak roll	A1 to A10 = 0 W1 = 3.3 W2,10 = 3.0 W3,4,5,6,7,8,9 = 2.5	"W" heaters set to recover for next cold soak.
2201	Same	A1 to A10 = 0 W1 to W10 = 0	All heaters zero for cold soak.
3/22/70(81) 0100 hrs.	Same	A3,8 = 3.6 A1,2,4,5,6,7,9,10 = 0 W3,8 = 3.6 W1,2,4,5,6,7,9,10 = 0	"A" & "W" heaters set to "1.2 Dual" heater start from frozen state.
0346	Same	A3,8 = 3.4 A (others) = 0 W3,8 = 3.4 W (others) = 0	Power reduced because of max temp being approached under heaters A3, A8, W3, W8. Pipe is not starting.
0646	Same	A1 to A10 = 0 W1 = 3.3 W2,10 = 3.0 W3,4,5,6,7,8,9 = 2.5	"W" heaters set to recover prior to next cold soak.
0746	Same	A1 to A10 = 0 W1 to W10 = 0	All heaters zero for cold soak.

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/22/70 (cont.) 1158 hrs.	Side sun on no roll	A1 to A10 = 0 W1 to W10 = 0	Solar start with modules 19-23.
1645	Side sun off roll (cold soak)	Same	End of solar start attempt.
1657	Cold soak roll	A1 to A10 = 0 W1 = 3.3 W2,10 = 3.0 W3,4,5,6,7,8,9 = 2.5	"W" heaters manipulated for recover prior to cold soak.
1737	Same	A1 to A10 = 0 W1,2,3,4,5,9,10 = 2.5 W7 = 3.3 W6,8 = 3.0	Same
1805	Same	A1 to A10 = 0 W1,10 = 1.5 W2,3,4,5,9 = 2.5 W6,8 = 3.0 W7 = 3.3	Same
1817	Same	A1 to A10 = 0 W2 = 0 W3,4,5,9 = 2.5 W1,10 = 1.5 W6,8 = 3.0 W7 = 3.3	Same

TIME	TEST CONDITIONS	HEATER SETTINGS - AMPS	REMARKS
3/22/70 (cont.) 1828 hrs.	Cold soak roll	A1 to A10 = 0 W1,2,10 = 0 W3,4,5,9 = 2.5 W6,8 = 3.0 W7 = 3.3	"W" heaters manipulated for recover prior to cold soak.
1834	Same	A1 to A10 = 0 W1 to W10 = 0	Cold soak prior to solar roll start.
2046	Side sun modules 19-23 roll	A1 to A10 = 0 W1 to W10 = 0	Solar start with roll. 2rph with return at 15rph.
3/23/70(82) 0004 hrs.	All side sun modules "on" roll	Same	
0135	Same	A1 to A10 = 0 W6 = 3.7 W1-5,7-10 = 0	H ₂ O would not start. Applied ~ 75 watts to one heater. After 40 min. pipe started; followed by a boil out under heater W6.
0154	Side sun off No roll	Same	
0300	Repressure with cold wall	A1 to A10 = 0 W1,2,3,9,10 = 1.5 W4,8 = 1.9 W5,7 = 2.3 W6 = 2.4	"W" heaters set to keep pipe above freezing for repressure phase. Maintained this until ambient conditions 3/24/70. 0800 hrs.

APPENDIX C

DELIVERY AND PRELIMINARY INSPECTION AT NASA/MSC

APPENDIX C

DELIVERY AND PRELIMINARY INSPECTION AT NASA/MSC

Upon arrival at the Houston rail receiving agency, the heat pipe shipping container was examined to determine the extent of externally observable damage. No visible damage was apparent except that one of the three steel tie-down straps had broken at the top of the shipping container, probably due to a fatigue break at a bend in the strap. The heat pipe shipping container was then trucked on a flat bed in a horizontal position (side-load) to the receiving depot at MSC. After delivery to Building 32, the heat pipes were removed from the shipping container, and assembled in the high bay area adjacent to Chamber "A" of the Space Environment Test Division. The assembly was optically leveled to $\pm 1/8$ inch using shim stock under the nylon thermal isolator pads.

The verification test procedure, Appendix A, was not followed due to a shortage of power supplies. Instead, two heater sections were energized until the water heat pipe operated in the heat pipe mode within the acceptance level established by the MSC monitor.

When power was applied to the four methanol pipe heaters (A1, A2, A9, A10), there was a current leak from the heaters through the heat pipe structure to all T/C's. This leakage was large enough to cause the thermocouple recorder to severely deflect all T/C's to an off-scale condition. Electrical resistance measurements showed the values to be much less than the heater design value of 7.0 ohms. In order to meet Chamber "A" test scheduling requirements, it was necessary to completely eliminate the use of these heaters during vacuum testing.

A word here is appropriate regarding the history of the test heaters. Initially, stainless steel wire was used for all twenty heaters (ten on each system). However, there was a tendency for the units to peel up at the corners after prolonged use because of the spring of the steel wire. All twenty heaters were then removed and rewound with manganin wire. Also, a layer of heavy aluminum tape was used over the top layer of teflon tape to aid in keeping the wire from lifting off the surface. Visually there was no difference in the twenty units. Also, no further difficulties were encountered in the development testing effort.

During the operation to seal the ammonia (methanol) pipe, it was necessary to freeze the ammonia in the area opposite the fill tube (see Section 4.2.2). To accomplish this, modules A1, A2, A9 and A10 were submerged in LN₂; these four were, in fact, the inoperative heaters for the thermal vacuum chamber testing. No heater which had not been subjected to cycling in and out of LN₂ malfunctioned prior to or during the testing. It should be noted that all heaters including the four inoperative units were operated at TRW/Redondo Beach after the freezing operation. Also, the resistances of all twenty heaters were checked just prior to packaging and shipment.

It is speculated that the LN₂ immersion was the principal contributor to the failure of the four heaters although subsequent attempts to duplicate the thermal cycling has not produced a similar malfunction. Possibly, a secondary contributor was the shock and vibration encountered in the rail transport to NASA. However, the combination of events causing the malfunction of the four heaters could not be reconstructed from the limited evidence available.

The unit was then installed in the NASA/MSFC Chamber "A" Thermal Vacuum Facility using a special handling fixture provided by TRW Systems and leveled to within the required $\pm 1/8$ inch. Leveling was accomplished with a laser beam.

The real-time test support function performed by TRW was to man the heat pipe console in the Chamber "A" control room, monitor the continuous on-line visual display cathode ray tube (CRT) which displayed all heat pipe thermocouples, verify the heater setting with the power control operator, record the thermocouple data in real-time in the event it was required prior to the availability of a hard copy, act as a consultant to the Test Conductors regarding heat pipe operation and assure the heat pipe timeline was accomplished. In addition to the cathode ray tube readout, the Chamber "A" Data Acquisition System supplied on-line data printout, Thermal Uplink Data Display (THUDD), every half-hour with about a half-hour to one-hour lag from real-time. The Heat Pipe Monitor maintained a file of this data in addition to the other duties.

Test results are presented and discussed in Section 4.4 of the main body of this report.