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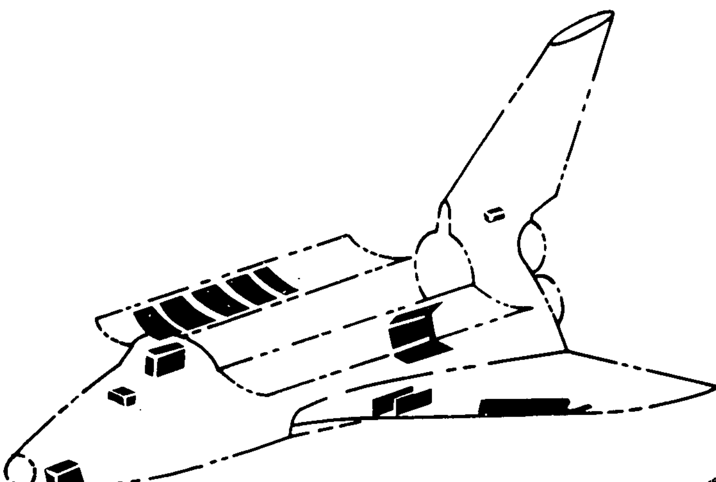
SPACE SHUTTLE ORBITER HEAT PIPE APPLICATIONS

VOLUME 1 SYNOPSIS



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SPACE SHUTTLE ORBITER HEAT PIPE APPLICATIONS

CR- 128497

VOLUME 1
SYNOPSIS

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FOREWORD

This report was prepared by Grumman Aerospace Corporation for the Manned Spacecraft Center of the National Aeronautics and Space Administration. The work was performed under Contract NAS 9-12034 and was administered by the Thermal Technology Branch of the Structures and Mechanics Division, with Mr. R. Bullock as Project Technical Monitor.

The work described herein was performed from July 6, 1971 to April 30, 1972. This report is the final report for Contract NAS 9-12034 and consists of two volumes:

Volume I: Synopsis of Final Report - a brief summary of the study and results

Volume II: Final Report - a detailed presentation of the heat pipe applications formulation, evaluation, supporting analyses and designs

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SECTION 1

ABSTRACT

An investigation was made to formulate and evaluate heat pipe applications for the space shuttle orbiter. Of the twenty-seven specific applications which were identified, a joint NASA/Grumman evaluation resulted in the selection of five of the most promising ones for prototype development. The formulation process is described, along with the applications which evolved. The bulk of the discussion deals with the "top" five applications, namely:

- heat pipe augmented cold rail
- avionics heat pipe circuit
- heat pipe/phase change material modular sink
- air-to-heat-pipe heat exchanger
- heat pipe radiator for compartment temperature control

The philosophy, physical design details, and performance data are presented for each concept along with a comparison to the baseline design where applicable. A sixth application, heat pipe space radiator for waste heat rejection, was also recommended for prototype development-but its development would be more efficiently handled under a separate contract.

This document is a brief synopsis of the final report, Volume II, which describes the results of the study in detail.

SECTION 2
INTRODUCTION

The heat pipe, as a component, is an extremely efficient thermal control device that can transfer heat with very little temperature drop. This heat transfer is accomplished by the evaporation, vapor transport, condensation and return by capillary action of a working fluid within a sealed container. In addition to superior thermal performance, heat pipes have no moving parts, require no electrical power and can be made self-regulating. These characteristics make heat pipes attractive for aerospace applications since they can benefit overall vehicle performance by providing thermal control systems that are lighter, are simpler and more reliable, require less power, operate at much lower noise levels, minimize fluid leak probabilities and have improved maintenance features.

In recent years there has been a veritable explosion of information about various heat pipes, their design and thermal performance. As a result, the feasibility of various types of heat pipe devices has been established. For example, flight hardware or working models exist for simple heat pipes, isothermalizers, cold and hot reservoir variable conductance pipes, diode pipes and feedback control heat pipes. They encompass a performance range from cryogenic to entry temperatures with corresponding thermal capacities from a few to a few thousand watts.

Heat pipes have reached the point where their unique performance qualities can benefit space shuttle orbiter thermal control systems. With this in mind, a study was undertaken to formulate, evaluate, and design practicable heat pipe systems offering tangible benefits over baseline designs, with a realistic chance of being implemented. The primary objectives of this study were:

- identify potential heat pipe applications for the space shuttle orbiter
- evaluate the applications and recommend the most promising ones for further development
- perform detailed design and analysis on the recommended applications
- prepare design drawings with necessary material specifications to permit fabrication of prototype hardware for at least three of the recommended applications
- prepare test plans for performance verification of the three or more prototype applications

Secondary objectives were to:

- evaluate a general design concept employing "off-the-shelf" heat pipe components to be used in minimizing costs, in the event of an extensive commitment to heat pipe systems
- create study plans for the development of prototype heat pipe hardware for space station, space shuttle and common shuttle/station applications (including space radiators)

SECTION 3
SUMMARY

Each of the shuttle subsystems, i.e., structure, propulsion, avionics, power and environmental control and life support were reviewed in detail, with possible heat pipe applications areas indicated by the heat sources and sinks located throughout the shuttle vehicle. Twenty-seven initial applications were defined, from which eleven were chosen for further design and analysis. The procedure used to evaluate these eleven was based on a better than/worse than comparison with the baseline system for each of six criteria: temperature gradient, capacity margin, power requirements, control requirement, weight, and safety. Because of the lack of factual data, parameters such as cost, maintainability, reliability, durability, and development risk were only evaluated on a secondary basis.

The eleven prime contenders are briefly summarized below:

1. Isothermalization of the leading edge of the wing to lower peak temperature and to increase mission life
2. Wheel well radiators to maintain minimum temperatures sufficient for tire survival by supplying waste heat
3. A design similar to (2) for the air breathing engine compartments
4. A HP avionics circuit to collect and transfer the thermal load from electronics boxes to the heat transfer system
5. Modular heat sinks for cooling remotely located components without the need for long extensions of the pumped coolant system
6. An adaptation of (5) for the flight/voice recorders located in the tail
7. A modular heat pipe heat exchanger system for adapting air-cooled commercial and military avionics to the shuttle
8. An all HP radiator system for waste heat rejection
9. A modified version of (8) incorporating a pumped fluid loop header

10. A HP augmented cold rail capable of absorbing an order of magnitude greater local power density when compared to a simple fluid cold rail

11. A high temperature heat rejection system for the fuel cells

The preliminary design studies of these prime contenders included a description of the overall system, supporting drawings showing the heat pipe systems and shuttle interfaces, and heat pipe design details including capacity requirements, working fluids, wick design, pipe lengths and diameters.

Further evaluation resulted in six of the eleven concepts being selected for detailed design and analysis. These six are noted in Table 3-1, which summarizes the results of the evaluation process.

Original 27 Candidates	11 Prelim. Design Study	6 Detail Design
1. TPS leading edge	TPS leading edge	HP radiator for compartments
2. Landing gear	Landing gear	HP radiator, waste heat**
3. Avionics HP circuit	Avionics HP circuit	HP circuit, avionics*
4. Modular sinks	Modular sinks	HP augmented cold rail*
5. ATR equipment	ATR equipment	Modular sink (voice rec)*
6. Flight and voice recorders	Flight and voice recorders	Air cooled equip rack
7. HP radiator W/HP header	HP radiator with HP header	
8. HP radiator with integral HP/fluid header	HP radiator with integral HP/fluid header	
9. ECS cold rail	HP Augmented cold rail (high capacity)	
10. HP radiator for fuel cell	HP radiator for fuel cell	
11. Air breathing engine compartment	Air breathing engine compartment	
12. OMS LH_2 boiloff		
13. High intensity lights		
14. Battery		
15. Tracking radar		
16. Fluid evaporator		
17. Fuselage TPS, interference heating		
18. TPS panel		
19. Control surface pivots		
20. OMS LO_2 boiloff		
21. Main LO_2 tank boiloff		
22. C-Band directional antenna		
23. Electrical wiring		
24. Hydraulic actuators		
25. APU		
26. LO_2 natural recirculating system		
27. Water chiller		

*Selected for prototype fab dwgs and test plans

**Selected for study under separate contract

Table 3-1 Shuttle Heat Pipe Applications Evaluation

Selected Concepts

Heat Pipe Augmented Cold Rail

The heat pipe augmented cold rail is made by inserting a heat pipe in the center of a standard two-passage fluid cold rail. The heat pipe, by distributing localized heat inputs over the length of the rail, allows it to accommodate the higher power densities of present generation power conditioning and control equipment. Without heat pipe augmentation, the previous generation electronics would have to be substituted for the newer and more compact equipment resulting in heavier avionics and fewer components mounted per rail. Since more cold rails would be required, not employing heat pipes in the cold rails causes increased weight and flow pressure losses. The heat pipe augmented cold rail is capable of transferring, simultaneously, an average heat load of .83 watts/inch/side and a concentrated load of 39 watts/inch/side (over 1.8 inches) to the fluid loop, while maintaining equipment flange root temperatures below 140°F.

Avionics Heat Pipe Circuit

This system consists of an equipment rack comprised of all-heat pipe cold rails, a heat pipe header to collect and carry the energy away from the rack, and a heat pipe-to-fluid heat exchanger to transfer the waste heat to the pumped fluid (water) loop system.

Heat pipe cold rails can more conveniently provide greater cooling capacity than all-fluid rails, both on a power density and a total load per rail basis, and since heat pipes operate near isothermal conditions they provide flexibility for equipment location within the circuit. In addition to the twin benefits of capacity and flexibility, the problems associated with flow balancing and pumping losses in the fluid rails are eliminated. The absence of fluid connections at the rails also minimizes fluid leaks and possible equipment contamination.

Heat Pipe/Phase Change Material Modular Sink

This system provides autonomous thermal control of heat generating packages located in remote portions of the vehicle, where fluid-loop cooling would require very long lines with their inherent installation and leak problems. As applied to the flight data/voice recorder electronics, the modular heat sink thermal control concept couples the electronics base plate (heat source), via heat pipes, to either a structural or phase change material heat sink, as required. During most phases, heat would normally be transferred to structure. During times of high structural temperatures the pipes would self-regulate, minimizing thermal feedback from structure, while utilizing the phase change sink for adequate equipment cooling. This system controls the baseplate temperature between -20°F and 130°F while the surrounding structure ranges between -40°F and 207°F.

The modular heat sink thermal control concept has broad applicability to remotely located heat sources utilizing any number of possible sinks -- e.g., structure, expendable fluids, phase change materials, isolated radiators.

Air-to-Heat-Pipe Heat Exchanger

The air cooling requirements of "off-the-shelf" available commercial and military electronics can be satisfied, without modification, by using a heat pipe-to-air heat exchanger in conjunction with an air circulating enclosure within which the equipment is mounted. The heat load picked up by the heat pipes is transferred to the main header of the heat transport system. Heat pipes are more attractive than a straight fluid-to-air heat exchanger because they do not require any fluid connections near the equipment, thereby decreasing the chance of fluid leakage and equipment contamination.

Heat Pipe Radiating Panel for Compartment Temperature Control

A heat pipe radiator system for compartment temperature control has power and weight advantages over an electrical system and control and reliability advantages over a conventional fluid radiator. The heat pipe radiator system described in this study has been designed for the orbiter's main landing gear compartment, although in principle and concept it can be used elsewhere.

It consists of a heat pipe radiator panel and a diode heat pipe header. Waste heat from a convenient fluid heat source (in this case the Freon-21 heat rejection loop) is extracted by a diode/heat exchanger coupling and directed to the feeder heat pipes of the radiator panel. The heat pipe radiator system, as described, is capable of maintaining the on-orbit main landing gear temperatures between 0°F and 117°F with a heat exchanger flow rate of 150 lb/hr, or only 30% of the maximum available rate.

SECTION 4
IDENTIFICATION/EVALUATION

4-1 IDENTIFICATION

No single configuration was used as "the" baseline Shuttle concept for the purpose of identifying and formulating heat pipe applications. Rather, the configurations of three Phase A/B Shuttle contractors were used (References 1-3). However, Grumman's concept served as the primary information source since it more closely reflected the then current NASA thinking and the supporting documentation was more readily available.

The approach was to analyze a typical shuttle vehicle, using design data from one contractor to supplement that of another, which served to incorporate the largest amount of available engineering information in the baseline configuration. However, there still was insufficient design data to provide detailed flight requirements for all of the shuttle's heat sources and sinks, e.g., temperature, heat load, operational timelines. As a result, many of the preliminary evaluations and tradeoffs were qualitative, relying heavily on sound engineering judgement. They were supported by analysis whenever possible.

Grumman's subsystem definition (see Table 4-1) was used to categorize the major functional areas on the shuttle.

Each of these subsystems were then reviewed in detail for feasible heat pipe applications by scrutinizing all of the heat sources and heat sinks which comprised them. Examples of typical shuttle heat sources are given in Table 4-2.

A list of shuttle items that have low operating temperatures and sufficient capacity to be designated as heat sinks is given in Table 4-3.

The factors considered in developing the applications were temperature and capacity requirements, physical location on the vehicle, mission environment, geometric or operational constraints, and effects of inertial forces on the movement of the working fluid.

Inertial forces during powered flight and entry can be as high as 3 g's and, as seen from Figure 4-1, the direction of these forces can vary through 180

TABLE 4-1

GAC SUBSYSTEM DEFINITIONS

A. Structural

1. Fuselage

   a. Nose Module
   b. Forward Mid Module (Crew Compartment, Payload Compartment)
   c. Aft Mid Module (ABPS Engine Support)
   d. Aft Fuselage (Thrust Structure and Mounts)
   e. Tanks (LO_2, OMS, ABPS)

2. Aero Surfaces

   a. Wing Elevon
   b. Elevon
   c. Fin
   d. Rudder

3. External LH_2 Tanks
4. Thermal Protection
5. Crew Station/Equipment and Passenger Accommodations
6. Flight Control Mechanical Equipment
7. Recovery System (Landing Gear)

B. Propulsion

1. Main Propulsion System
2. Air Breathing Propulsion System
3. Orbit Maneuvering System
4. Attitude Control Propulsion

C. Avionics

1. Guidance and Navigation
2. Flight Control
3. Data Management
4. Instrumentation
5. Telecommunications and Air Traffic Control
6. Displays and Controls

D. Power

1. Power Generation
2. Electrical Power Distribution
3. Hydraulic

E. Environmental Control Life Support

1. Atmospheric Revitalization
2. Heat Transport/Heat Rejection
3. Atmospheric Supply and Composition Control
4. Water Management
5. Waste Management

TABLE 4-2 TYPICAL HEAT SOURCES ON THE SPACE SHUTTLE

INTERNAL	
● AVIONICS – "BLACK BOXES" – AIRCRAFT AVIONICS – SPACECRAFT ELECTRONICS – RADAR ANTENNAE – HIGH POWER WIRING & CONNECTORS ● ELECTRICAL POWER EQUIPMENT – APU – FUEL CELLS – BATTERIES ● HYDRAULIC EQUIPMENT – PUMPS – HYDRAULIC LINES & CONTROL VALVING – ACTUATORS	● ENVIRONMENTAL CONTROL EQUIPMENT – RADIATORS – HEAT EXCHANGERS ● MAIN PROPULSION EQUIPMENT – GIMBAL RINGS – HEAT EXCHANGERS ● AIR BREATHING ENGINE EQUIPMENT – LUBRICANTS – PROPELLANT ● STRUCTURE – LANDING GEAR – ENGINE COMPARTMENTS – WHEEL WELLS – PIVOTS AND ATTACHMENTS
EXTERNAL	
● TPS – AEROTHERMODYNAMIC HEATING – POST FLIGHT SOAKBACK – PLUME IMPINGEMENT	● ENVIRONMENTAL – SOLAR RADIATION – ALBEDO – EARTH RADIATION – DIRECT AND REFLECTED RADIATION FROM OTHER SPACE VEHICLES

TABLE 4-3 TYPICAL SPACE SHUTTLE HEAT SINKS

INTERNAL	EXTERNAL
● STRUCTURE ● CRYOGENIC TANKAGE AND PIPING ● CRYOGENIC BOILOFF ● WATER BOILERS ● WATER SUBLIMATORS ● FREON BOILERS ● FLUID LOOP ELEMENTS ● COLD PLATES, COLD RAILS, HEAT EXCHANGERS ● AIR CYCLE EQUIPMENT ● PROPELLANTS	● SPACE ● DEPLOYABLE SPACE RADIATORS ● FIXED SPACE RADIATORS ● OTHER SPACE STATION MODULES OR VEHICLES ● GROUND SUPPORT EQUIPMENT ● AIR CONDITIONING, INERT GAS PURGE

degrees during the various mission phases. For a heat pipe mounted parallel to the fuselage reference line, these forces will drive the working fluid aft during ascent and generally forward during entry. Thus, if operation is required during other than orbital mission phases the pipe axis must be either normal to the gravity vector or a reflux condition must exist. The latter implies a gravity assisted return of the working fluid to the evaporator. Consideration was given to operation during five mission phases:

Phase	Description	Duration (Hr)
Launch	Prior to lift off	2
Boost	Ascent to orbit	2
On-orbit	270 n mi, i = 55°	164
Entry	De-orbit to sea level	2
Landing	Up to GSE hook-up	1/2

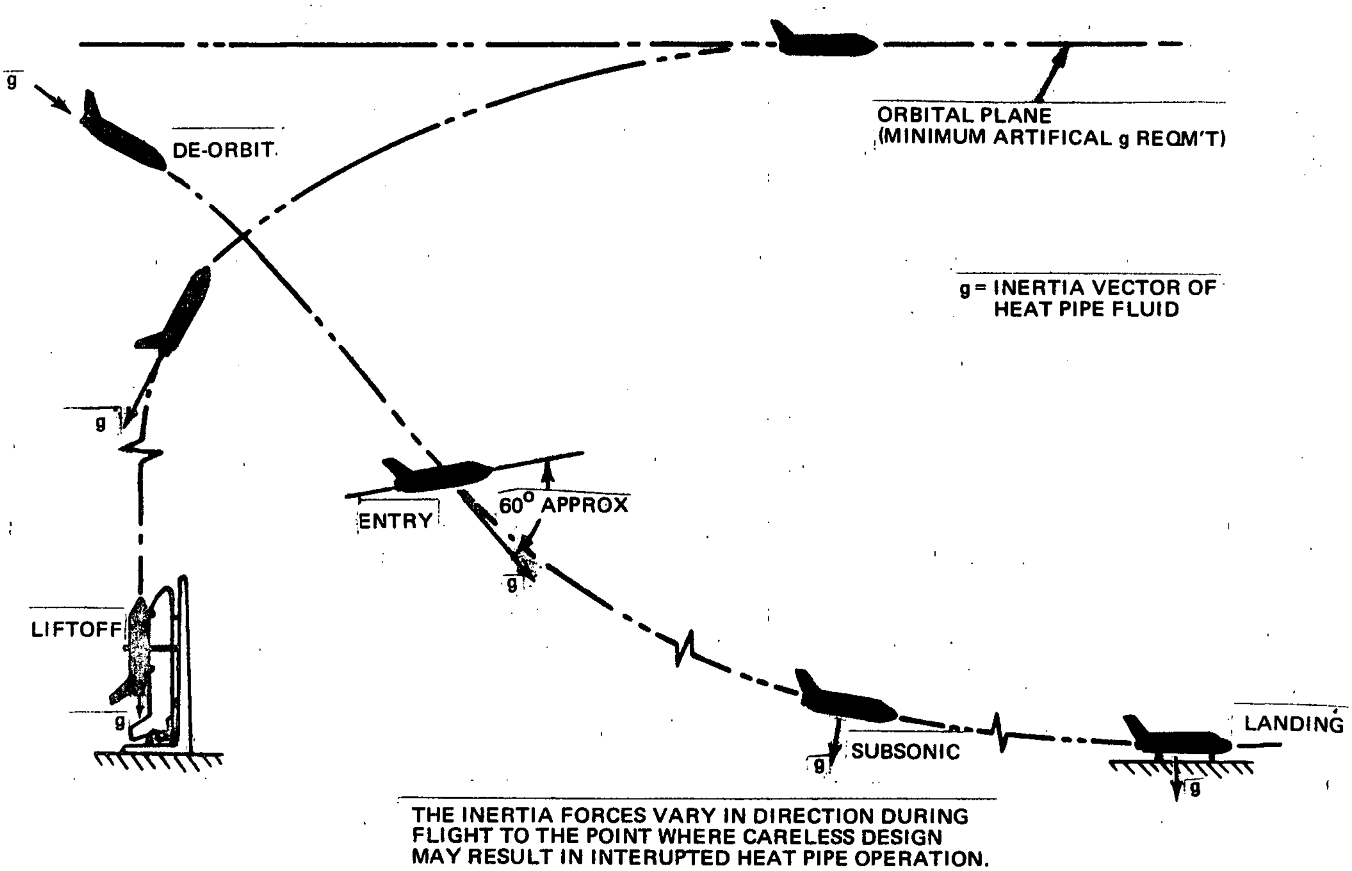


Figure 4-1: Inertia Force Variation During Shuttle Mission

Identification sheets for each proposed application were prepared which gave a description of the application, its requirements and its advantages and disadvantages. A representative sample, for the modular heat sink, is given in Figure 4-2.

The applications were given a preliminary evaluation and grouped into three general categories:

- Prime Contenders (Rating = 2): Those applications offering tangible benefits over the baseline thermal control system and a realistic chance of being implemented. They are minimum risk systems with potentially large payoffs
- Possible Contenders (Rating = 1): Those applications providing marginal improvements over the baseline systems. The potential benefits are uncertain and may not warrant the development effort
- Rejected (Rating = 0): Those applications offering no significant benefit over the baseline. Insufficient definition exists to warrant further consideration at this time

Table 4-4 lists the proposed applications and their ratings.

Additional work was done in areas related to applications previously identified, in response to changing shuttle definitions. These included:

- Coupling the wing leading edge to the upper wing surfaces to equalize temperatures
- Lowering the backface temperatures of superlight ablator and panels
- Using heat pipe in a lube oil/hydraulic fluid heat exchanger for a hypergolic fueled APU.

None of them, however, were evaluated as prime contenders.

4-2 EVALUATION

The most promising applications were chosen for detailed design and analysis based on how well they compared with their counterpart baseline thermal control systems. The comparisons were made on the basis of performance, weight and safety as gauged by six evaluation criteria: temperature gradient, capacity margin, power requirement, control requirement, weight and safety. Because of the lack of factual data, parameters such as cost, maintainability, reliability, durability, and development risk were only evaluated on a secondary basis.

Figure 4-2

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.4

APPLICATION: HP/Phase Change Remote Sinks (i.e., Modular Sinks)

SUBSYSTEM: Avionics LOCATION:

BASELINE T/C SYSTEM: ECS loop (Intermittently operated avionics)

MISSION PHASE	TEMP LEVEL (°F)	HEAT LOAD (BTU/HR)	TRANSPORT LENGTH (FT)
All	50-100	< 500	$\frac{1}{2}$

DESCRIPTION:

HP's imbedded in suitable phase change material and connected to a common mounting interface (e.g., cold plate or rail) to which the component to be cooled is mounted. Phase change material is connected in turn to a radiating surface or suitable structure.

ADVANTAGES

1. Eliminates special runs of ECS lines to remote equipment.
2. Self-sufficient system.
3. No electrical power req'd.
4. Flexible designs
5. Simple.
6. High Q's in short time can be designed as low Q over longer time.

DISADVANTAGES

1. Weight penalty must be traded off with total ECS savings.

COMMENTS:

Typical Equip. Applications: Control Electronics (Air Surface, engines), Rate Sensors, Communications equip. (L Band Tacan, C band altimeter, L band transponder, VHF ATC transceivers)

RATING: 2

TABLE 4-4

SHUTTLE HEAT PIPE APPLICATIONS CANDIDATES

Title	Subsystem
<u>Prime Contenders (Rating = 2)</u>	
1. TPS Leading Edge	Structure
2. Landing Gear	Structure
3. Avionics HP Circuit	Avionics
4. Modular Sinks	Avionics
5. Air Cooled Equipment	Avionics
6. Flight & Voice Recorders	Avionics
7. HP Radiator W/HP Header	ECS
8. HP Radiator with integral HP/Fluid Header	ECS
9. ECS Cold Rail	ECS
10. HP Radiator for Fuel Cell	Power
11. Air Breathing Engine Compartment	Propulsion
<u>Possible Contender (Rating = 1)</u>	
12. OMS LH_2 Boiloff	Structure
13. High Intensity Lights	Avionics
14. Battery	Avionics
15. Tracking Radar	Avionics
16. Fluid Evaporator	ECS
<u>Rejected (Rating = 0)</u>	
17. Fuselage TPS, Interference Heating	Structure
18. TPS Panel	Structure
19. Control Surface Pivots	Structure
20. OMS LO_2 Boiloff	Structure
21. Main LO_2 Tank Boiloff	Structure
22. C-Band Directional Antenna	Avionics
23. Electrical Wiring	Avionics
24. Hydraulic Actuators	Power
25. APU	Power
26. LO_2 Natural Recirculating System	Propulsion
27. Water Chiller	ECS

CRITERIA	RATING COMPARED TO BASELINE		
	WORSE THAN	BETTER THAN	COMMENTS
1. Temperature Gradient			
2. Capacity Margin			
3. Power Requirement			
4. Control Requirement			
5. Weight			
6. Safety			

Figure 4-3 - Evaluation Matrix

Figure 4-3 shows the evaluation matrix which was used for the better than/worse than comparisons. The definitions of the parameters are given below:

- Temperature Gradient - Adverse temperature differences which exist in a system due to thermal inefficiencies in the heat transport mechanism. These temperature losses can occur within a heat transfer element (e.g., heat pipe, fluid line) or across a required attachment interface (e.g., tube-saddle). The more thermally efficient design can transfer the required amount of heat from one point to another with the smaller resulting temperature drop. This impacts the overall design by requiring smaller capacity sinks, in terms of area and weight, to reject the specified heat load
- Capacity Margin - The usable heat transport capacity which is available in a system above the design requirement. It can be used to accommodate future increase in heat load or to afford a measure of redundancy in a heat transport system with dual transfer paths
- Power Requirement - The amount of electrical power required to operate the heat transport device; it includes power for control systems
- Control Requirement - Is an active control device (e.g., heater, valve) required for proper operation of the system? The preferred system is self-controlling needing no supplemental devices
- Weight - The total weight of the heat transport device including its basic components and any special attachments, fittings and control elements
- Safety - Freedom from chance of injury or loss to personnel and equipment

The evaluation of "safety" is quantified by using a scoring matrix similar to that used for the Safety criteria of Reference 4. As shown below, the factors comprising safety are assigned scores of 0, 1, or 2 - the highest score indicating the most desirable situation. The system with the highest cumulative score for the four factors is considered the safest.

FACTORS

Score	Inflammable Materials	Toxic Materials	Hi-Pressure Fluids	Potential Hazards
2	None	None	None	None
1	External*	External	External	External
0	Internal*	Internal	Internal	Internal

* With respect to the pressure shell

To help determine the relative worth of each application a simple quantitative rating system was established. A (-1) weighting factor was assigned to the "worse than" category and a (+1) weighting factor to the "better than" category. Criteria which were considered to be the same as the baseline, i.e., neither clearly worse than nor better than, had a zero weighting factor. The cumulative numerical rating for an application was determined by adding the weighted scores for the six criteria making up the complete evaluation sheet. The most desirable applications would be those with the highest positive cumulative score - the ones offering the most benefits and least disadvantages when compared to their corresponding baseline systems.

Table 4-5 contains a summary of the evaluations of the contending applications along with their net numerical ratings. The highest rated application scored +4 and there were seven of them. One scored +3; two +2, and one zero.

Since many of the applications were similar in that they used a heat pipe radiator, it was decided to consider six generic heat pipe application categories for the detailed design and analysis task. Thus, the landing gear wheel wells (1) and air breathing engine compartment (11) were considered part of the heat pipe radiator, compartment temperature control category. In the same fashion applications 7, 8, and 10 were considered heat pipe radiators for waste heat

TABLE 4-5

SHUTTLE HEAT PIPE APPLICATIONS EVALUATION - SUMMARY

Application	Compared to Baseline: Worse Than -1	Better Than +1	Same 0	Net Rating
1. TPS Leading Edge	1	2	3	+1
2. Landing Gear Compartments		4	2	+4
3a. HP Heat Transport System	3	3		0
3b. Avionics HP Circuit		4	2	+4
4. HP/Phase Change Remote Sink		2	4	+2
5. Air-cooled (ATR) Equipment		2	4	+2
6. Flight & Voice Recorder		3	3	+3
7. HP Radiator with HP Header		4	2	+4
8. HP Radiator with HP/Fluid Header		4	2	+4
9. HP Augmented Cold Rail		4	2	+4
10. HP Radiator for Fuel Cells		4	2	+4
11. HP Radiator for ABE Compartment		4	2	+4

rejection. The six heat pipe applications which resulted from this evaluation, and the ones selected for further development were:

- Heat pipe augmented cold rail
- Avionics heat pipe circuit
- Heat pipe/phase change material modular sink
- Air-to-heat-pipe heat exchanger
- Heat pipe radiating panel for compartment temperature control
- Heat pipe radiator for waste heat rejection

At this point in the program, NASA requested that work on the heat pipe radiator for waste heat rejection be suspended since such studies would be advanced under a separate effort. The remaining five applications then underwent detailed design and analysis studies, and drawings were prepared in sufficient detail to permit fabrication of all five. Based on the information made available in these studies, three applications were recommended for development and testing of prototype hardware. They were the heat pipe augmented cold rail, the heat pipe circuit, and the modular heat sink.

The following sections of this report summarize the detailed designs and analyses, featuring design drawings, specifications, and thermal performance predictions, for the five selected applications.

SECTION 5

GENERAL DESIGN CONCEPT

In formulating the initial concepts it was noted that many different heat pipes would be required to satisfy the various applications. However, a significant reduction is possible if a single or limited number of modular designs are developed. These standard modules could then be combined to satisfy the requirements of many applications. Two modular concepts were investigated: the self-contained modular design and the modular subassembly design. For either concept to be used in any shuttle location without restriction, it is necessary that they function satisfactorily with at least two working fluids: one suitable for use inside the pressure shell and one for use outside. This allows the same hardware to be used both inside the pressure shell, where low toxicity is important, or outside where thermal transport properties might be paramount. Thus, a standardized wick compatible and operating with either of two working fluids is central to a truly general design concept.

In the self-contained modular design, the heat pipe building blocks are single closed units capable of independent operation. Multiple modular units are used when the required performance exceeds that of a single unit's capability. These modules are placed in parallel for larger capacities and in series where long lengths are required. In the modular subassembly design, heat pipes are configured by combining standard lengths of major subassemblies. The basic subassemblies are the pipe envelope, wick, reservoir, low-k section, mitered joint and flexible joint. These pieces are joined together to create the desired heat pipe configuration with maintainable joints to allow for replacement of defective units.

After careful examination, it was determined that self-contained modular heat pipes would not satisfy enough applications to warrant further serious consideration. Satisfying the requirements of these varied applications with self-contained modules would mean many series/parallel circuits with many pipe-to-pipe conductive attachments resulting in installations which are inherently heavier and less efficient (large temperature drops) than customized designs. On the other hand, the modular subassembly approach presents a compromise between modular and custom designs. It offers some of the savings intrinsic in a modular design without weight and performance penalties. The subassemblies can be machined and fabricated in large lots beforehand, drawn from stock, cut to length, and assembled when required. The modular subassembly concept offers manufacturing, assembly, scheduling, and maintainability advantages over customized designs.

In addition to the outer shell of the heat pipe, modular wicks can be established for relatively efficient performance with two working fluids. Because it is nontoxic, the working fluid of choice for heat pipes inside the pressure shell is water. Outside this area, ammonia appears to be the best working fluid because of its high transport capacity. Neither fluid is suitable for use as a back-up for the other, as regions outside the pressure shell can attain temperatures below the freezing point of water, and ammonia, being extremely toxic, should not be used in a life-supporting area of a spacecraft. Further, the material usually used to construct heat pipes for each fluid may not be used for the other because of compatibility problems such as gas generation. The various restrictions that control the interchangeability of working fluid and wick design are given in Table 5-1 for ammonia, water and Freon-21. Freon-21 can be used as a back-up fluid for both water and ammonia because of its low freezing point and relative nontoxicity. It presents no compatibility problems with copper, aluminum, or stainless steel. Due to the materials problems, there can be no one multi-fluid wick used on the shuttle with all three fluids; two separate two-fluid wicks must be manufactured.

Before actually using a modular heat pipe design concept for the shuttle, the basic question of its practicality must be raised. The use of the modular heat pipe designs becomes competitive with customized systems only when an extensive commitment to heat pipe systems over baseline systems has been made. Certainly, it would make little sense to manufacture standard subassemblies beforehand if there are only one or two realistic heat pipe applications contemplated. For a limited number of heat pipe systems there is no alternative to using customized designs.

TABLE 5-1

HEAT PIPE WORKING FLUID/WICK DESIGN INTERCHANGEABILITY

Fluid	Inside Pressure Shell			Outside Pressure Shell		
	Water Wick	NH_3 Wick	F-21 Wick	Water Wick	NH_3 Wick	F-21 Wick
Water	X	No (1)	No (1)	No (2)	No (2)	No (2)
Ammonia	No (3)	No (3)	No (3)	X	X	X
Freon-21	No (4)	No (4)	X	No (4)	No (4)	X

STANDARD HEAT PIPE MATERIALS

Fluid	Pipe	Wick
Water	Monel, Copper	Monel, Copper
Ammonia	Aluminum Stainless Steel	Stainless Steel
Freon-21	Aluminum Stainless Steel	Stainless Steel

X - Working fluid/wick design gives acceptable performance

(1) - Materials incompatibility

(2) - Freezing problem

(3) - Toxicity problem

(4) - Doesn't self-prime

SECTION 6

DETAILED DESIGN AND ANALYSIS

The following subsections summarize the detailed designs and analyses of the five selected heat pipe applications:

- Heat Pipe Augmented Cold Rail
- Avionics Heat Pipe Circuit
- Heat Pipe/Phase Change Material Modular Sink
- Air-to-Heat Pipe Heat Exchanger
- Heat Pipe Radiator Panel for Compartment Temperature Control

A sixth application (the waste heat rejection radiator) was also selected, but NASA directed that work on it be suspended since further study would advance under a separate contract.

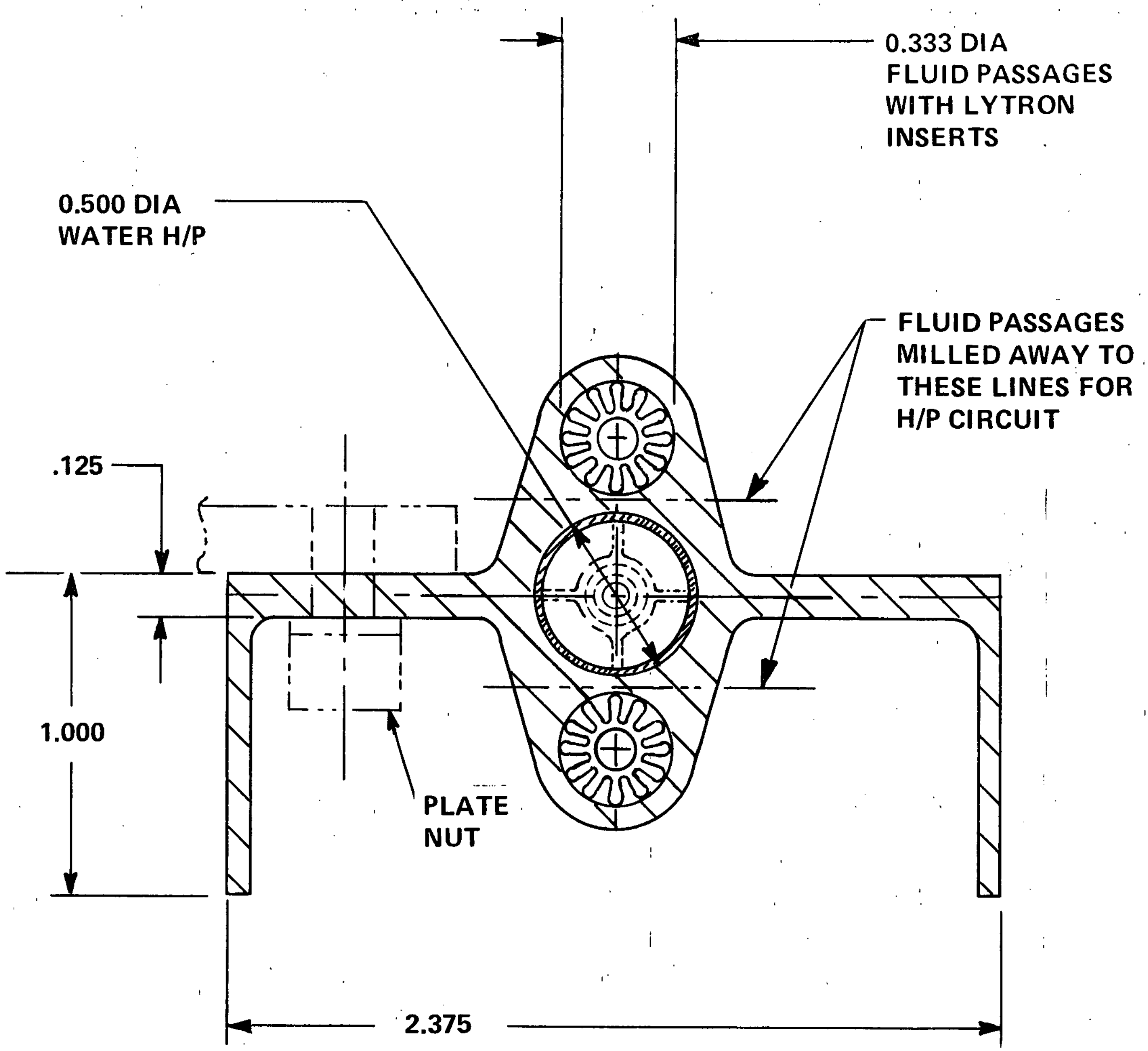
Details of the design and analysis efforts for each of the five applications are contained in Volume II of this report.

6-1 HEAT PIPE AUGMENTED COLD RAIL

The electronics proposed for the Space Shuttle depicts dissipation levels of some of the flange-mounted modules of 40 watts per linear inch per side. This value is approximately 16 times greater than the design values in the Apollo vehicle and exceeds the capability of simple fluid cold rails. If not thermally corrected, this would necessitate a less efficient redesign of electronics packages in terms of weight and volume. An increase in rail material thickness is simple, but the increase required at these loadings would cause an undesirable weight penalty. Using a heat pipe to provide longitudinal isothermalizing is simple and light. Figure 6-1.1 shows the proposed configuration cross-section. The heat pipe is an integral part of the extruded rail and serves to distribute a localized high heat load over the length of the rail, thereby increasing the coolant wetted area and reducing the temperature drop in the rail. The detailed design drawing for the HP augmented cold rail (SPL-104) is included at the end of this section.

The rail is made of an extrusion of 6101 aluminum. The tops of the rail flanges are finished to 64 micro-inch RMS to enhance thermal contact with the equipment. At those sites where high power density modules will not be mounted, the flange is machined down to 0.125 inch to minimize weight. Plate nut fittings are used at the mounting sites for box attachment. The fluid passages are internally finned using Lytron-type radial fins around an internal tube of 0.156 inch diameter. Gamah couplings are attached to short tubes inserted and welded into the entrances and exits of the fluid passages to provide a maintainable loop coupling.

The heat pipe has an envelope of 0.5 inch OD and is internally threaded to 100 threads/inch. The tubing is installed within the cold rail by reaming the extruded hole to allow for a push fit. The rail/tube interface is bonded to increase heat transfer. The structure and orientation of the wick is illustrated in Fig. 6-1.1. It is made of three wraps of 100/100 screen around a 0.062 inch diameter hollow core, and supported by four retainer legs oriented to facilitate vapor flow between the heat pipe wall section nearest the cold rail flange and the wall section nearest the fluid passage. The calculated



HEAT PIPE AUGMENTED COLD RAIL CROSS-SECTION

FIGURE 6-1.1

capacity of such a heat pipe, with the hollow core unprimed and 10-inch evaporator and condensor sections is 310 watts. If the core is primed (filled with working fluid), the capacity is 570 watts. Comparable watt-inch figures have been achieved in tests at Grumman using water heat pipes. The capacity of the overall rail is to be defined by two parameters: the peak localized input, limited to any two-inch long section of the rail's mounting flange, and the total heat absorption rate applicable to the entire rail length.

A typical lumped parameter, finite difference heat transfer technique was used to analyze the cold rail. Heat pipe operation and fluid flow are modeled by inserting a subroutine which is executed prior to each network iteration. The heat pipe vapor temperature required for steady-state equilibrium with the surrounding nodes is calculated and substituted for that used in the previous iteration. If the new heat pipe temperature indicates that, in relation to an adjacent temperature node, a region of the heat pipe acting as an evaporator is now acting as a condenser or vice versa, the appropriate conductance (corresponding to the film coefficient) is changed and a new steady-state temperature is calculated. Grumman has recorded film coefficients for water heat pipe evaporators in excess of 3000 $Btu/hr\text{-}ft^2\text{-}^oF$, so comparatively conservative values of 2000 and 2500 were used for heat pipe evaporators and condensers, respectively.

The 21.6-inch-long rail was broken into 12 stations each 1.8 inches in length. The analysis included a typical high power density module, i.e., the power switch module, 1.8 inches wide and generating 140 thermal watts, mounted on the cold rail. Both sides of the rest of the rail were loaded with uniform power densities. Flow rates were established using the Apollo limitation of 4 watts per pound of coolant per hour, divided evenly between the two 3/8 inch diameter passages.

The analytical studies resulted in the following conclusions:

- The heat pipe operates by distributing energy along the cold rail, thus lowering rail temperatures at high watt density equipment mounting sites

- The power module dissipating 40 watts per inch per side may be mounted anywhere on the proposed heat pipe augmented cold rail. In addition, other equipment with an average power dissipation of 0.83 watt per inch may be mounted at all the other locations of the rail
- The power module cannot be mounted on a conventional cold rail, even with thickened flanges, since it cannot remove enough heat to prevent the module box temperature from rising above 140°F
- All equipment mounting sites on the heat pipe augmented cold rail are equivalent. Unlike the conventional rail, there are no thermal advantages or disadvantages to mounting equipment at the fluid "upstream" or "downstream" end of the rail

Figure 6-1.2 bears out the first three conclusions. This figure shows the temperature distribution within the cold rail, with and without the heat pipe operational, with a 70 watt source mounted on one side of the rail at station 3 and 1.5 watt sources at all other mounting sites. As employed here, the heat pipe lowers the temperature of the attached module box by the same amount.

The curve showing temperature distribution with the heat pipe operational indicates that the box flange root temperature will not exceed 140°F with the imposed loading, and is therefore a thermally acceptable configuration, with a total load on the rail of 104.5 watts. The curve showing temperature distribution without the heat pipe indicates a box flange root temperature of the high power density module of 162°F, an unacceptable level.

In an effort to determine the high power density capability of the rail without the heat pipe, the cold rail model was then run with no thermal loads other than 70 watt source at station 3. The box flange root temperature dropped only 3°F to 159°F, still unacceptably high. The power module cannot be mounted on an augmented cold rail.

Figure 6-1.3 shows the temperature distribution in the augmented rail with the same thermal input used in Figure 6-1.2, but with the high density source moved to station 10 at the fluid downstream end of the cold rail. The flange root temperature of the module box is at 140°F, the same flange root temperature level obtained when the box was mounted upstream, substantiating the final conclusion above.

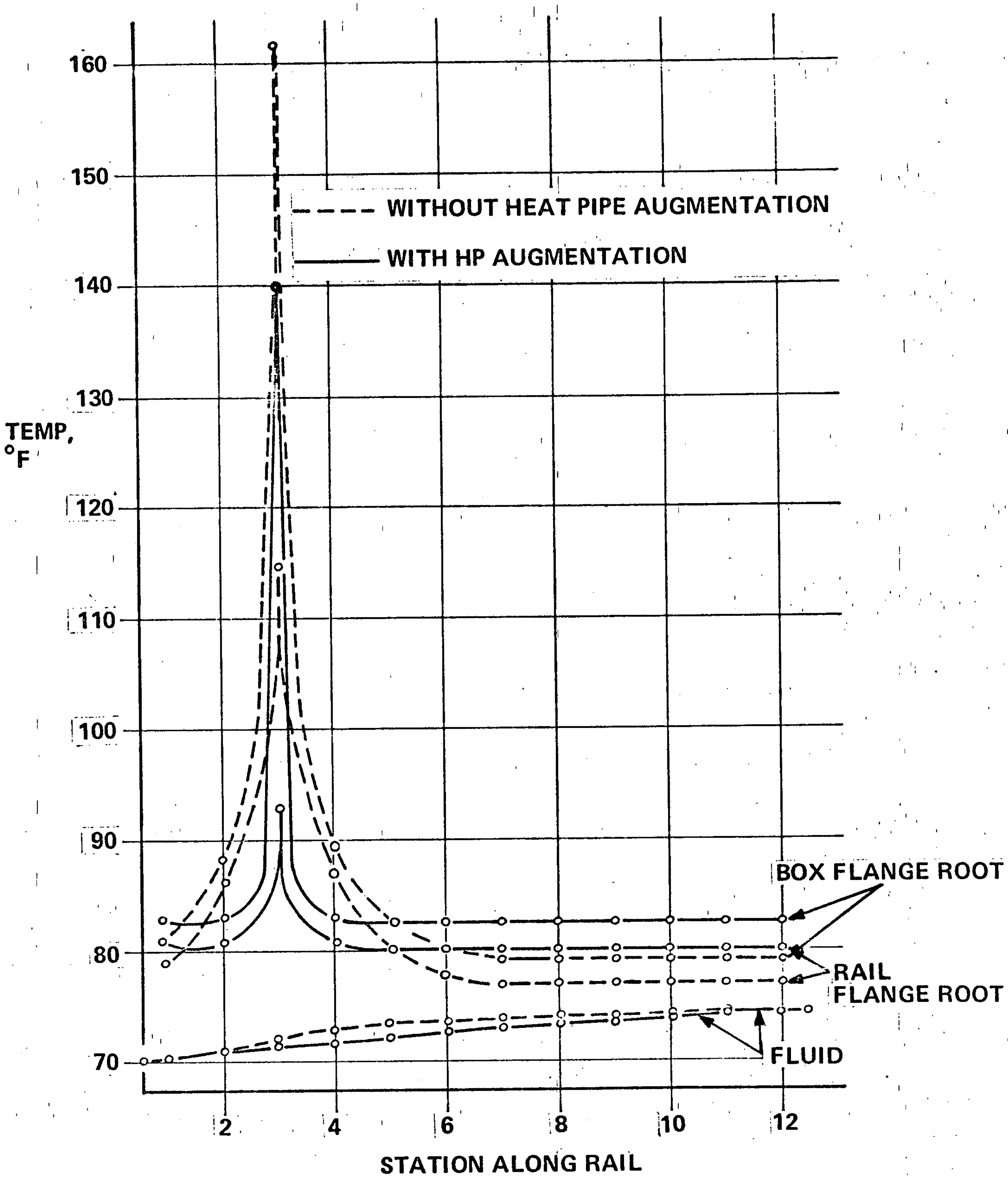


FIGURE 6-1.2 TEMPERATURE DISTRIBUTION
ALONG COLD RAIL

GRUMMAN

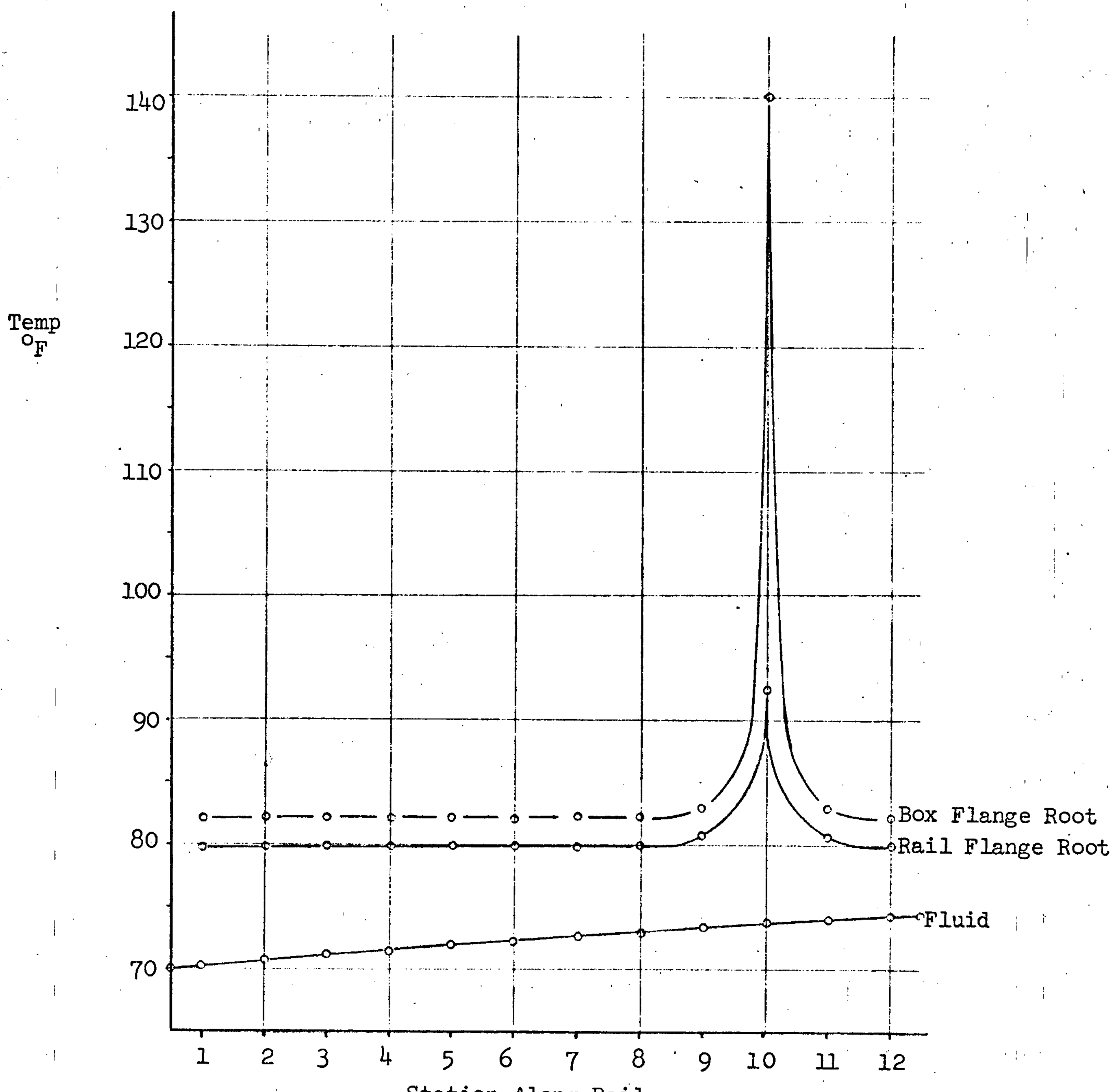
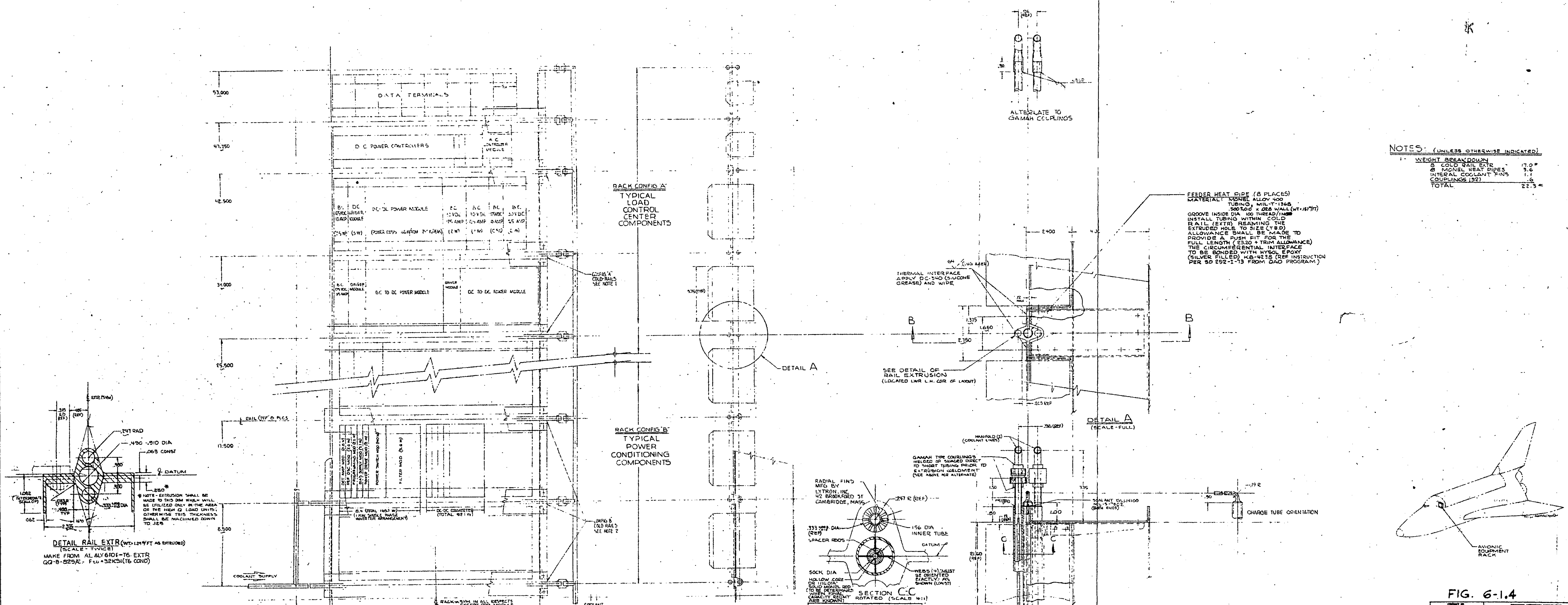


FIGURE 6-1.3 TEMPERATURE DISTRIBUTION ALONG
HP AUGMENTED COLD RAIL

FOLDOUT FRAME 1

FOLDOUT FRAME 2

FOLDOUT FRAME 3



MODULE EQUIPMENT RACK (SCALE - HALF)
VIEW LOOKING FWD OR AFT IN THE ORBITER VEHICLE

NOTES: (UNLESS OTHERWISE INDICATED)

1 - WEIGHT BREAKDOWN

8 COLD RAIL EXTR	17.0#
8 MONEL HEAT PIPES	3.6
INTERAL COOLANT FINS	1.1
COUPLINGS (32)	.6
TOTAL	22.3#

TABLE I (HEAT PIPE CONFIG)

ITEMS	DATA
WORKING FLUID	DISTILLED WATER
SPIRAL ARTERY	HOLLOW CORE DIA =.060 NO OF WRAPS = 3 GAP SPACE OR ROD DIA =.010
RETAINER	SOCK DIA = .202 NO OF WEBS = 4
TUBE	.500 O.D (NOM) x .028 WALL MAT'L - MONEL 400

MAKE FROM 100/100 MESH MONEL SCREENING

FIG. 6-1.4

CONTRACT NO. NAS 9-12034

J. F. Coscello 9/23/71

AUGMENTED COLD RAIL RACK CONFIG "A" & "B" (LAYOUT)

26512 SPL-104

SHEET 1 OF 1

AUGMENTED COLD RAIL SPL-104

6-8.1 E 6-8.2 6-8.3

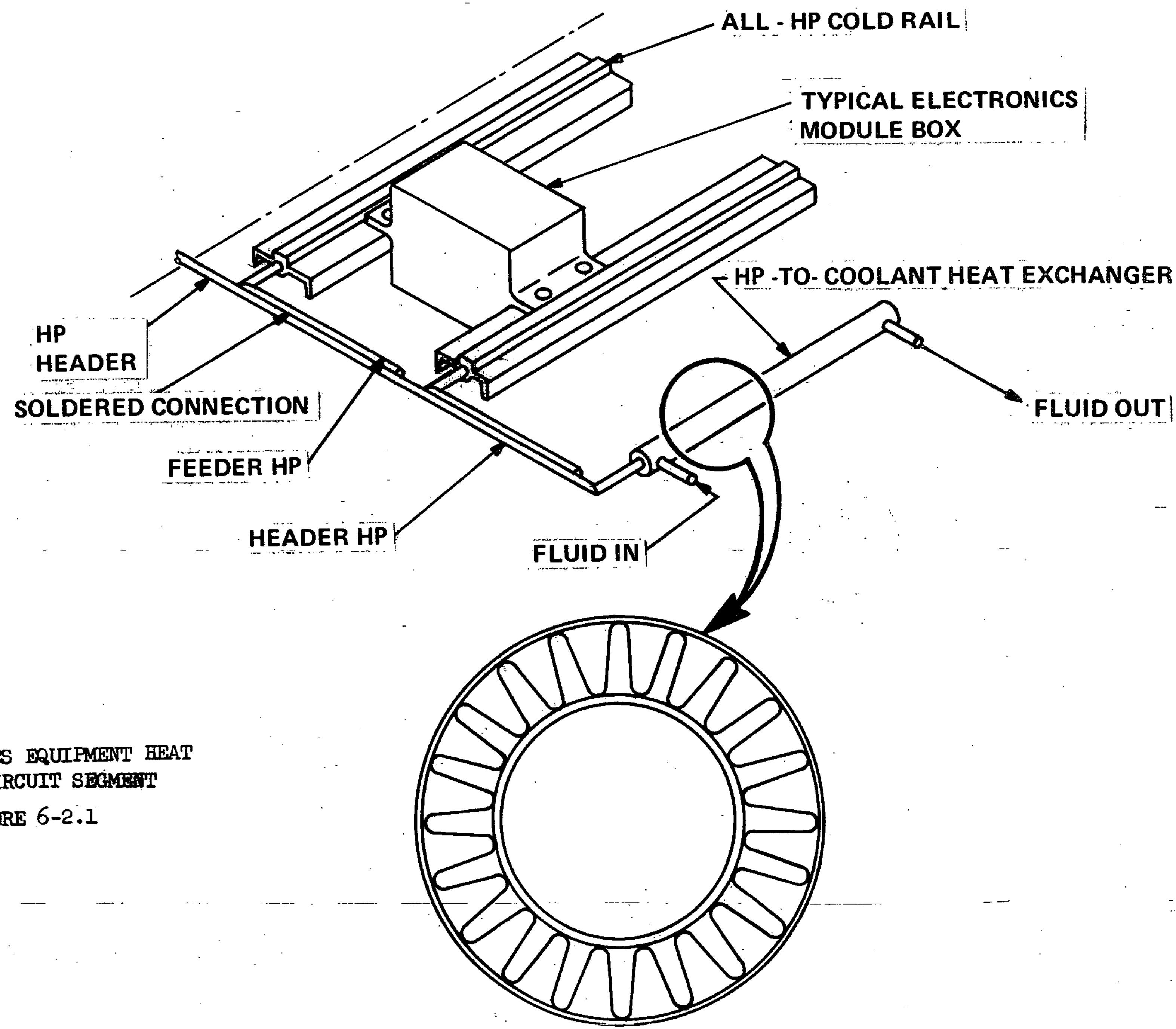
6-2 AVIONICS HEAT PIPE CIRCUIT

This concept for cooling rack-mounted avionics equipment is a logical extension of the heat pipe augmented cold rail. The system consists of a rack composed of seven all-heat pipe cold rails, a heat pipe header to collect and carry the energy away from the rack, and a heat pipe-to-fluid heat exchanger to transfer the waste heat to the pumped fluid (water) loop system. Figure 6-2.1 shows a typical segment of the circuit. All heat pipes in the system use water as their working fluid to avoid introducing toxic substances into the life support area in the event of a failure.

This equipment rack differs from one composed of conventional fluid or heat pipe augmented cold rails in that only fluid connections in the system are in the heat exchanger. Therefore, the probability of a fluid leak is less in this system, with only two fluid line connections versus 28 in an equivalent seven-rail conventional system. The detailed design drawing, SPL-102B, generated in support of this application is included at the end of this section.

The heat pipe cold rails, 44 inches long, are of the same extrusion used for the heat pipe augmented cold rail, with the fluid passages at the top and bottom milled away as shown in Figure 6-1.1. This concept is used to minimize costs for prototype development and would required additional design for flight hardware. The attachment of the feeder heat pipes (0.5 inch OD) to the header (0.875 inch OD) is accomplished by milling one side of both the feeder condenser and header evaporator sections into flat surfaces which are subsequently soldered and then clamped together. This interface is assumed to have the conductance of the heat pipe evaporator ($h = 2000$ btu/hr-ft^2-$^\circ$F) and condenser ($h = 2500$ btu/hr-ft^2-$^\circ$F) film coefficients in series.

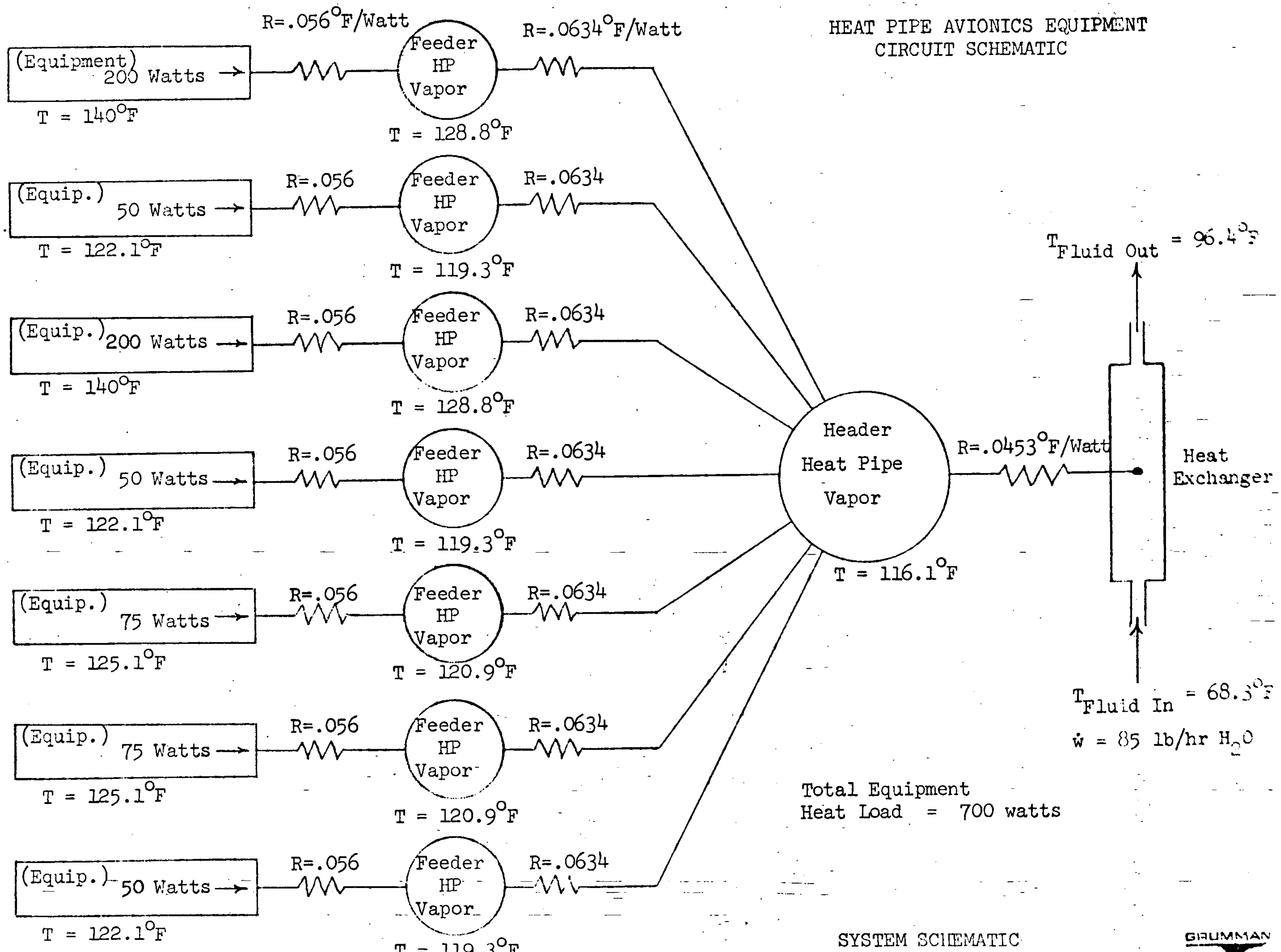
For design purposes, maximum loads of 200 and 700 watts were considered for the individual cold rail and header HP, respectively. An analysis of the system under various conditions has been performed and indicates satisfactory performance for all cases. Figure 6-2.2 presents the calculated circuit resistances and the resulting temperatures for the maximum load condition. This analysis was based on an imposed load of 200 watts distributed uniformly over the cold rail and a box flange root temperature of 140°F. Based on an allotment



AVIONICS EQUIPMENT HEAT
PIPE CIRCUIT SEGMENT

FIGURE 6-2.1

HEAT PIPE AVIONICS EQUIPMENT
CIRCUIT SCHEMATIC



SYSTEM SCHEMATIC
FIGURE 6-2.2

of 1 lb/hr of coolant for each 8.25 watts of power dissipated, a flow rate of 85 lb/hr was determined for the design load of 700 watts. Using this flow rate and a fluid inlet temperature of $68^{\circ}F$, an overall "UA Product" of 75 btu/hr$^{\circ}F$ was calculated for the heat exchanger.

The relatively simple fluid to heat pipe heat exchanger unit shown in Figure 6-2.1 was designed to satisfy this requirement. As shown, longitudinal fins are bonded to the header condenser which is then encased in a cylinderical water jacket containing fluid inlet and outlet ports. A pressure drop of less than one inch of water has been calculated for this design. Like the heat pipe augmented cold rail, this system is insensitive to the position of a module box on the rail.

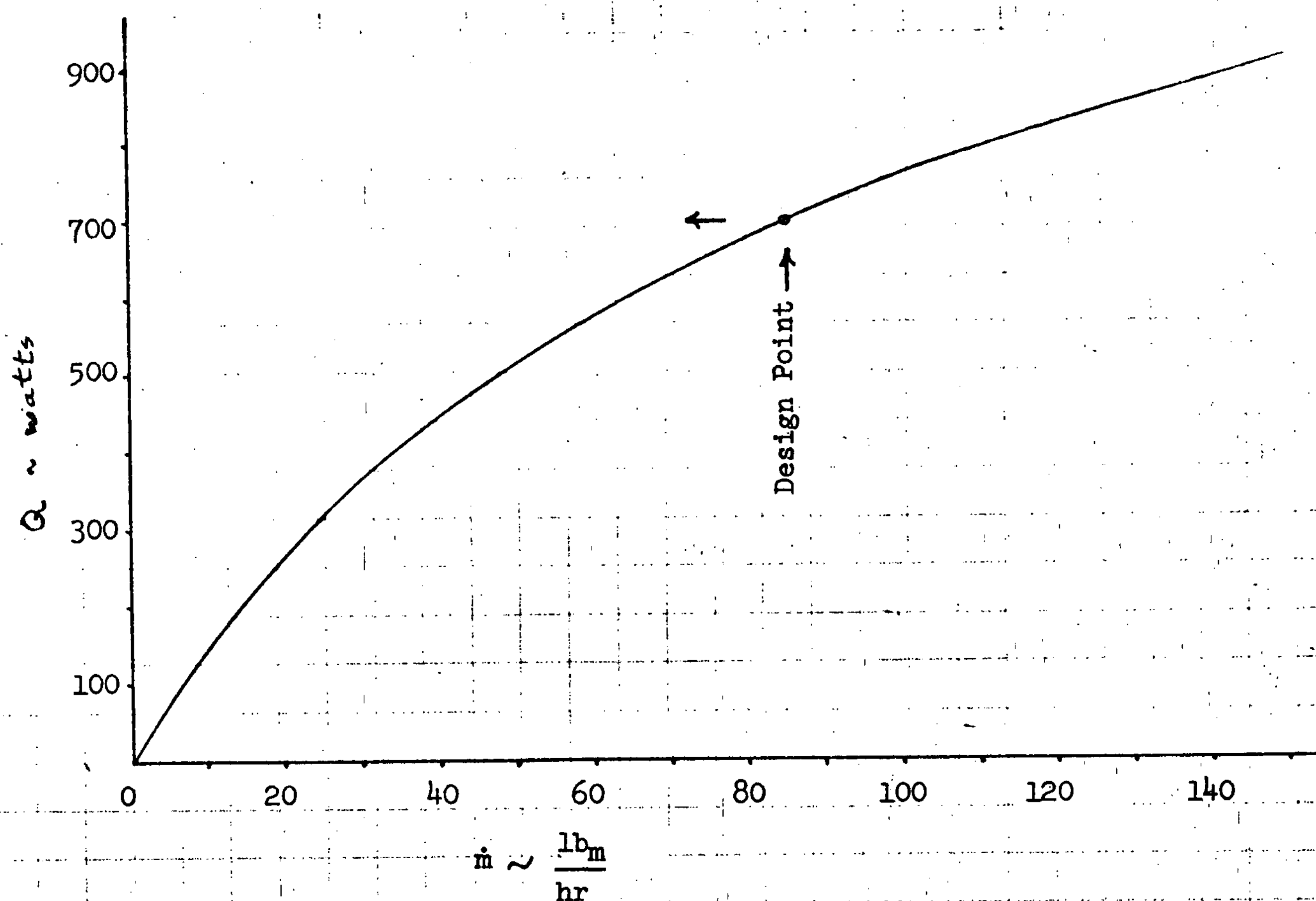
The total design capacity of this system can easily be increased because any desired change involves only the substitution of the appropriate new flow rate and new heat exchanger having the proper "UA product." If no hardware changes are feasible, an increase in the coolant flow rate alone increases the capacity of the system, as shown in Figure 6-2.3. The analysis conducted to produce this figure assumed the maximum load on any cold rail was 200 watts, which fixes the heat pipe header temperature at $116.1^{\circ}F$.

The high power density capabilities of a rail in this system were explored, within the design constraint of a 700 watt total load. This load fixes the header temperature at $116.1^{\circ}F$. The temperature drop from a $140^{\circ}F$ module box flange root to the feeder heat pipe vapor was calculated for a number of thermal loads of higher watt density than the baseline. (These heat loads were imposed over a two-inch section of flange.) From this temperature drop, the feeder heat pipe vapor temperature was calculated, and, from the temperature difference between feeder and header, the total amount of heat which can be removed from the feeder was determined.

The local high density loads obtained are shown plotted against the total load on that rail in Figure 6-2.4. If the total load on a rail is limited to 50 watts, the local load the rail can accept over two inches (with the assumed total load on the system) is increased to 8.4 watts. If a load of 9.4 watts is imposed locally, no other load may be placed on the rail. The limiting

FIGURE 6-2.3

SYSTEM CAPACITY AS A FUNCTION OF COOLANT FLOW RATE



$$Q \text{ into exchanger} = U_o A_o \frac{T_{Fluid\ Out} - T_{Fluid\ In}}{\ln \dfrac{T_{HP} - T_{Fluid\ In}}{T_{HP} - T_{Fluid\ Out}}}$$

$$Q \text{ into coolant} = \dot{m}\, Cp\, (T_{Fluid\ Out} - T_{Fluid\ In})$$

By combining the above expressions to eliminate $T_{Fluid\ Out}$, the equation of the curve above is derived:

$$Q = \dot{m}\, Cp \left(T_{HP} - T_{Fluid\ In} - \frac{T_{HP} - T_{Fluid\ In}}{\exp\left(\dfrac{U_o A_o}{\dot{m}\, Cp}\right)} \right)$$

where

$T_{HP} = 116.1^{o}F$

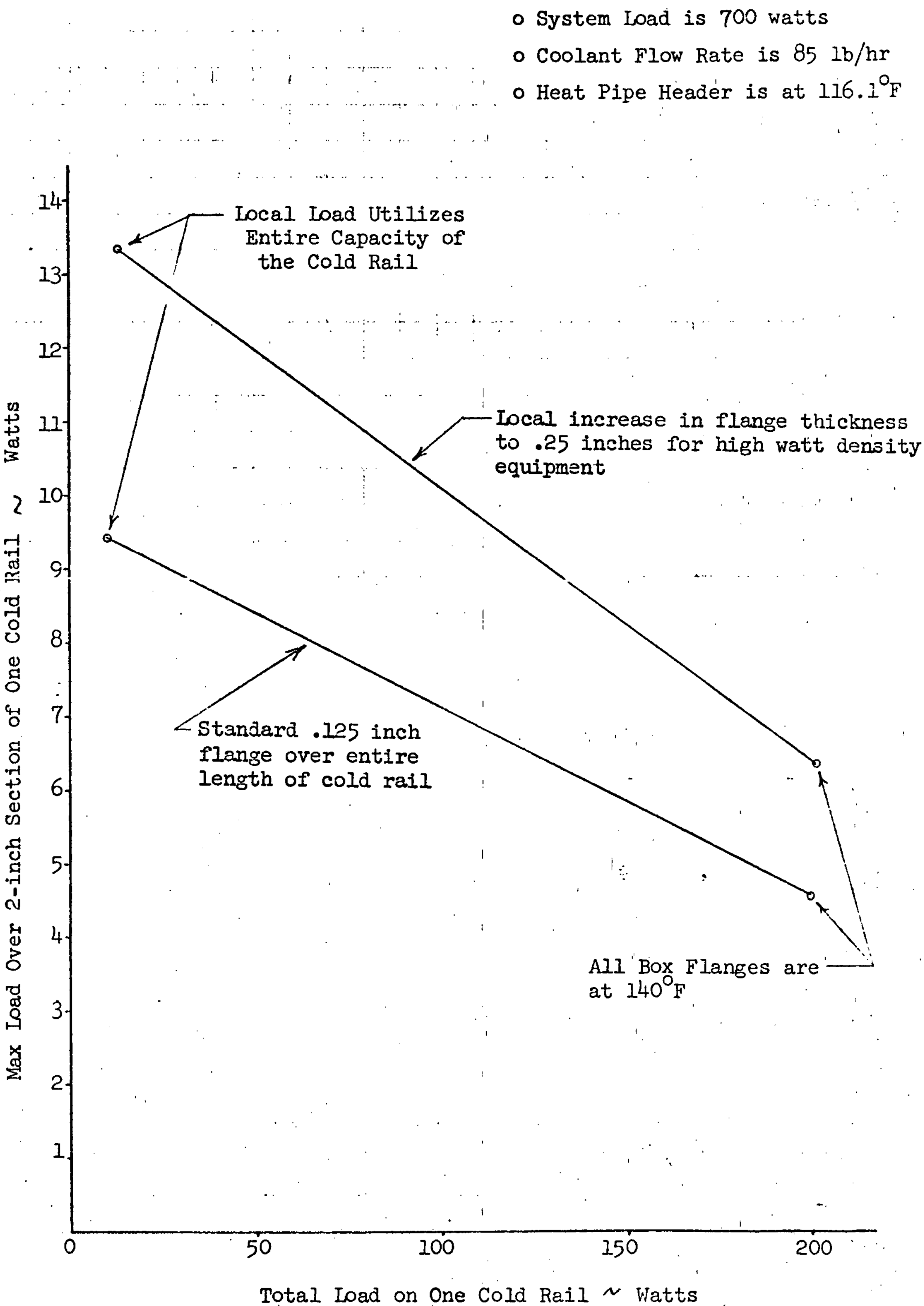
$T_{Fluid\ In} = 68.3°F$

$Cp = 1$, as coolant used is water

$U_o A_o$ is a function of flow rate and exchanger design

FIGURE 6-2.4

Local Max Load vs. Total Load on a Cold Rail



factor in each case is the feeder to header temperature drop which must be controlled to maintain the header at 116°F for the design temperature constraints to be met. When it becomes necessary to mount equipment with a higher watt density loading than the indicated limits, the equipment could be designed to accept higher flange root temperatures than 140°F, or the rail flange could be made thicker locally. A higher coolant flow rate or lower inlet temperature would also improve high density load capabilities, but such solutions to the problem involve new system design constraints. Calculations similar to those described above were also performed for the locally thick flange utilized in the augmented cold rail design, and the resulting curve plotted in Figure 6-2.4. All of these calculated values are higher than the local maximum watt density capability of an Apollo-type fluid cold rail, 2.25 watts/linear inch/side, or 4.5 watts over the same two-inch section.

A consideration of failure modes has shown that a heat pipe system offers advantages over a conventional system. The most catastrophic failure that could occur would be the loss of an individual pipe and a leak of its contained fluid within the vehicle. Each pipe contains a small amount of water, 50 grams, and it is anticipated that this quantity of fluid would not impair overall system performance. On the other hand, a leak in a fluid line could result in degraded insulation effectiveness and equipment damage before the line could be automatically bypassed.

Any analysis of this type of system is extremely dependent upon the values chosen for the heat pipe evaporator and condenser heat transfer coefficients. As heat pipe technology advances, the capabilities of a heat pipe circuit such as this should increase along with increasing film coefficients. Current Grumman test data support the values used for this study.

Heat pipe circuits such as the one analyzed have shown advantages over the baseline cooling systems on the shuttle orbiter. With only two fluid connections <u>per rack</u>, as opposed to four <u>per conventional rail</u>, leakage problems are minimized and flow balancing within a rack is eliminated. Unlike the conventional fluid cold rail, every mounting site is thermally the same as every other. A high density local load can be accepted on a rail if the penalty of lower rail capacity can be accepted. The entire capacity of the rack can be changed by replacing the heat pipe header/heat exchanger assembly with one having a larger exchanger.

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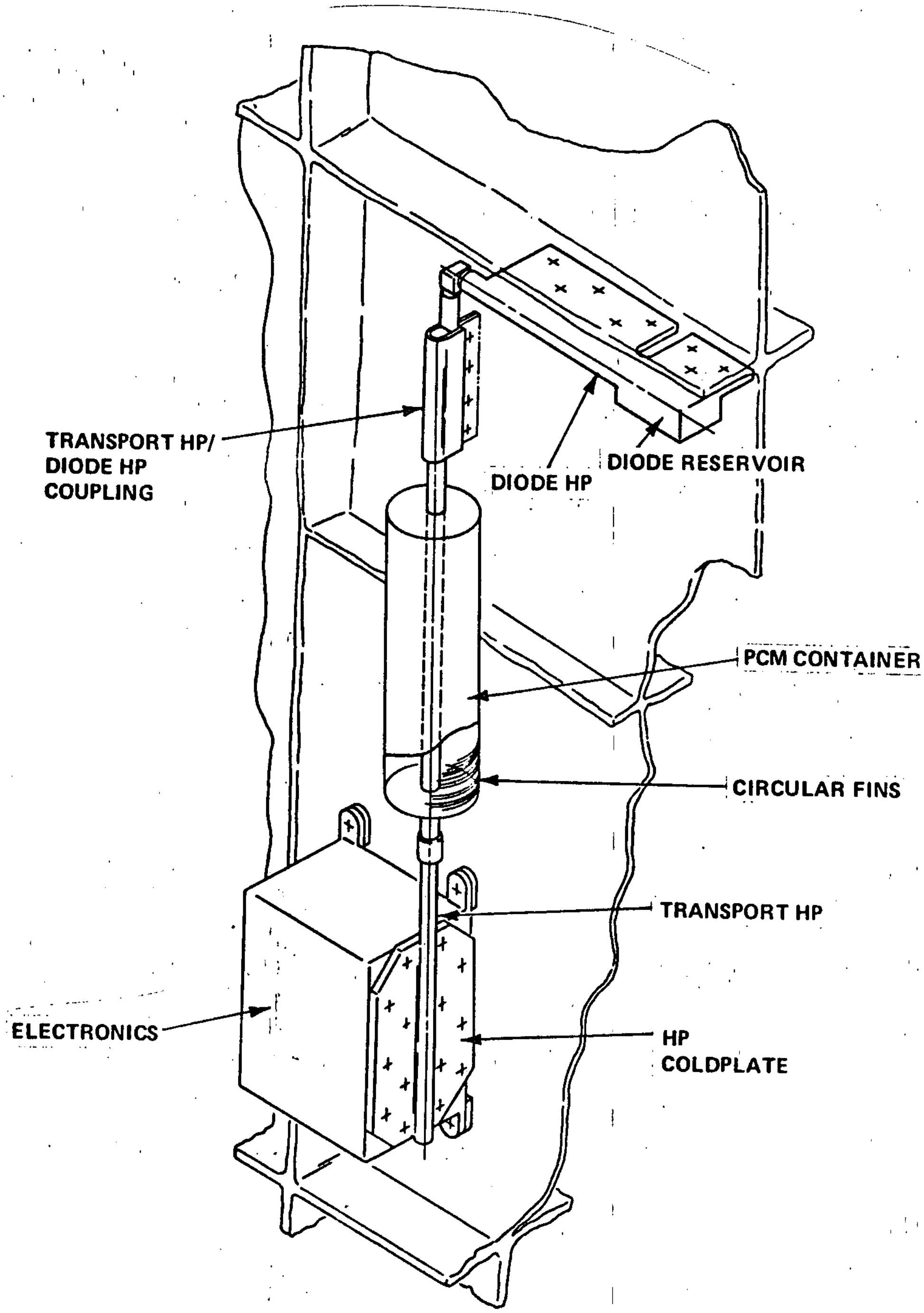
6-3 HEAT PIPE/PHASE CHANGE MATERIAL MODULAR SINK

A class of problem that has appeared on the shuttle involves heat generating packages located in remote portions of the vehicle where fluid loop cooling would require very long line lengths with their inherent installation and leak problems. One such package contains the electronics for the flight recorders and is located in the tail section. The modular heat sink thermal control concept couples the electronics base plate (heat source), via heat pipes, to either the structural or phase change material (PCM) heat sink, as required by the mission. As shown in Figure 6-3.1, it consists of a cold plate which interfaces with the electronics, a transport heat pipe whose evaporator is integral with the cold plate, a PCM container attached to the middle of the transport pipe, and a diode heat pipe which connects the transport pipe to the structural bulkhead. Drawing SPL-111, containing the system's design details, is included at the end of this section.

During boost and the entire on-orbit operation, the base plate is coupled directly to the structural sink (average bulkhead temperature $-40^{\circ}F$ to $+110^{\circ}F$) through the transport and diode heat pipe connection. During entry the structure becomes too hot to function as a sink and the diode reverses, decoupling the structure from the transport pipe. This creates an isolated system consisting of the base plate, transport HP and PCM container with the heat flow path now terminating at the PCM. After landing, when the structure has cooled, the diode once again completes the connection to the bulkhead allowing the liquefied PCM to unload its stored energy.

Although several working fluids can be used for the temperature range involved ($-40^{\circ}F$ to $+140^{\circ}F$), the selection was narrowed to ammonia and Freon-21. A design capacity of 100 watts was specified for each heat pipe. This accounted for the possible simultaneous transfer of 35 watts from the electronics, 35 watts from the PCM, and 30 watts from the environment. Half-inch ID heat pipes were selected since they provide reasonable circumferential heat transfer area. The designs of the arteries were optimized for the geometries involved and the analytical performance, in the form of capacity versus operating temperature, are presented in Figures 6-3.2 and 6-3.3 for the transport heat pipe and diode heat pipe, respectively. Based on these results ammonia was selected as the working fluid, with a simple artery wick for the diode heat pipe (Ref. 5) and a spiral artery/tunnel wick for the transport heat pipe (Ref. 6).



MODULAR HEAT SINK

FIGURE 6-3.1

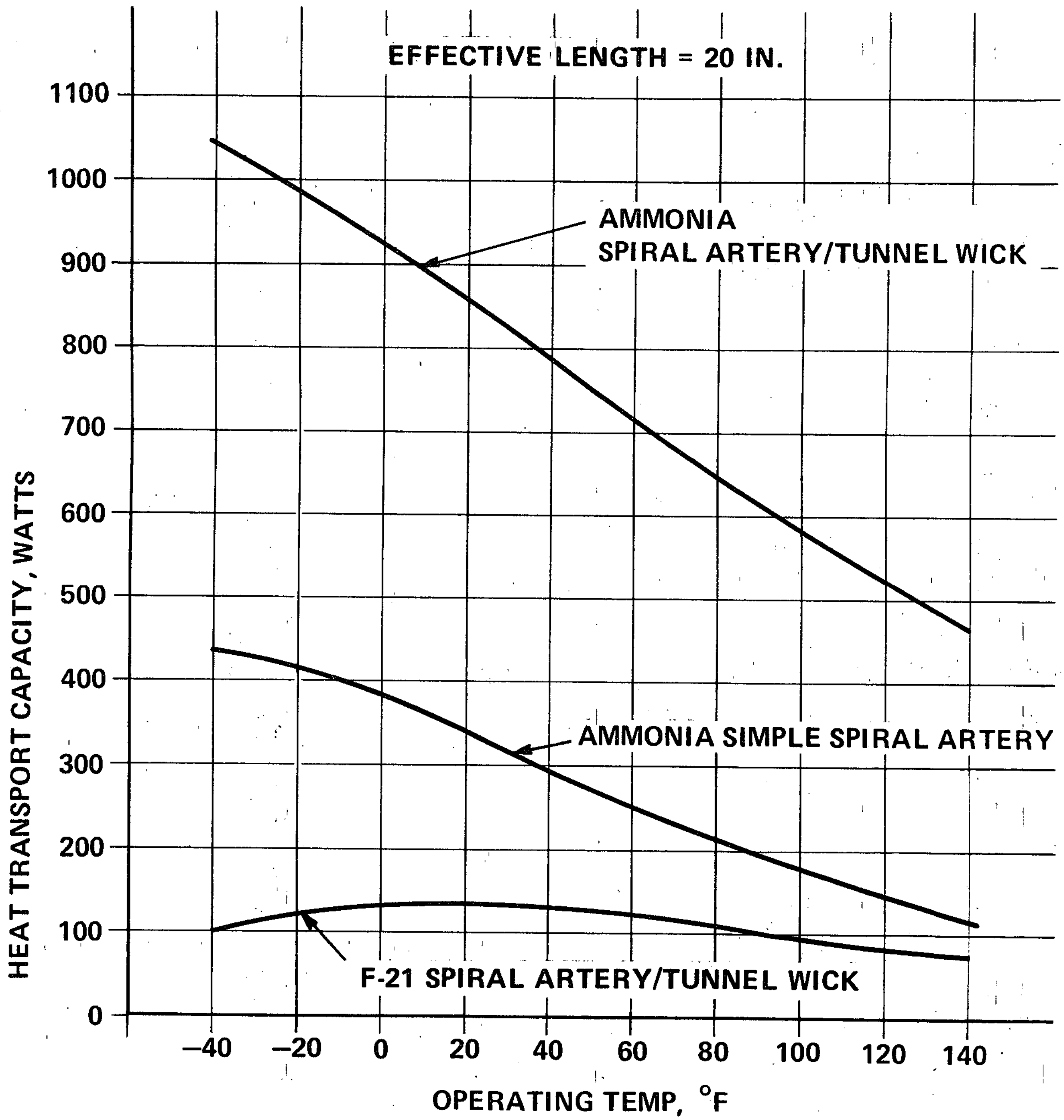


FIGURE 6-3.2 TRANSPORT HEAT PIPE CAPACITY

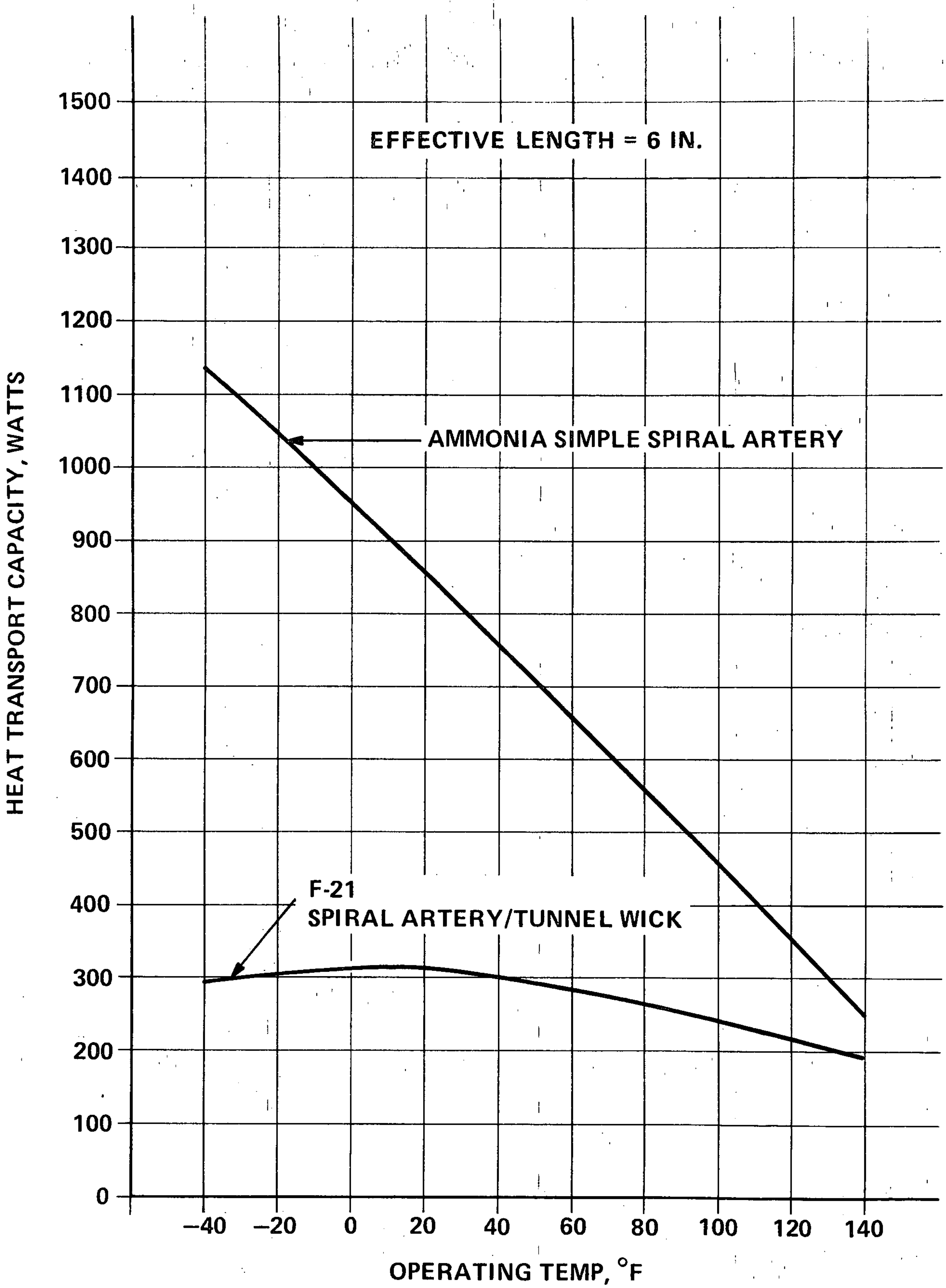


FIGURE 6-3.3 DIODE HEAT PIPE CAPACITY

The diode heat pipe operates on the liquid blockage principle (Ref. 7); the diode stops functioning as a normal heat pipe when excess fluid held in the reservoir at the condenser end is released and transported to the evaporator, thereby completely filling the evaporator vapor space. Since diode reversal is required during entry, the reservoir is designed to retain excess liquid until an inertia force greater than 1.25 g is experienced. For ammonia, this requires capillaries of 0.06 inch diameter for a 2.12 inch-long reservoir. The diode will also reverse in the more conventional manner when the temperature of the condenser exceeds that of the evaporator. The excess fluid vapor will travel to the now cooler evaporator, condense and fill the vapor space.

The electronic box is attached to the structure by steel bolts using insulating washers to minimize the conduction to structure. In addition, a fibrous insulation blanket completely encases the entire system. A single wrap of oxidized nickel foil (1/2 mil) is wrapped around the insulated system to create a thermal radiation barrier. The mechanically bonded interfaces are finished to 64 micro-inch RMS; before mating, silicon grease is applied to and then wiped from the contact surfaces. Both heat pipes are designed to withstand twice the maximum expected internal pressure of 800 psi. The maximum working pressure is 350 psi.

The cold plate is a flanged heat pipe evaporator which is fastened to the electronics base plate. It is a machined part which contains the evaporator section of the transport heat pipe as an integral part. The heat pipe wall incorporates fine internally machined circumferential grooves (150 grooves/inch). The charge tube is at one end; the other end attaches to the remainder of the transport heat pipe at a fillet welded interface. The cold plate flange is 0.070 inch thick with a contact surface area of 24.5 in.^{2}.

The transport heat pipe connects the cold plate flange to the PCM and to the diode heat pipe. Its condenser section mates with the evaporator of the diode heat pipe at a milled-flat interface mechanically joined by a clamp assembly. The transport heat pipe is charged with ammonia after the PCM container is attached and filled.

Hexacosane was the preferred choice for the phase change material because it has a high heat of fusion (110 Btu/lb_m) and a melting point (133°F) that exceeds the vapor temperature of the transport heat pipe during on-orbit operation. The latter requirement is necessary since the PCM is in contact with the transport heat pipe at all times but must melt only during entry. All of the PCM's which were investigated were non-toxic and showed good compatibility with aluminum, the container and heat pipe material.

Since phase change materials generally have very low thermal conductivities, they require metallic fillers to improve their heat conduction. Otherwise, very steep temperature gradients are required to transfer heat to the PCM, which would result in excessively high equipment temperatures during the melting process. The integration of a metallic filler into the PCM package provides low thermal resistance paths, thus reducing the temperature gradient necessary to dissipate the required heat flux. The actual design of the package is a tradeoff between the volume and weight of PCM required for the heat sink, and the effective conductance necessary to insure a reasonable temperature gradient.

Four basic PCM containers were analyzed to determine the most efficient configuration.

- Cylindrical container with circular (circumferential) fins
- Cylindrical container with straight (longitudinal) fins
- Rectangular container with straight fins mounted directly on the heat pipe cold plate
- Rectangular container with honeycomb mounted directly on the heat pipe cold plate

The circularly finned cylindrical container was chosen because it had the highest conductance per pound (PCM + fins) and it offered inherently greater design flexibility. Its effective conductance and weight, as a function of the number of fins per inch, are given in Figures 6-3.4 and 6-3.5, respectively.

At the PCM interface, near the midpoint of the transport heat pipe, the circular aluminum fins (6101-T6) are brazed to the 0.590 outside diameter of the pipe. The fins are 0.016 inch thick; each contains several notches and holes which act as flow passages to permit uniform distribution of PCM during filling.

FIG. 6-3.4 - EFFECTIVE CONDUCTANCE

o HP Wall to PCM

o Circular (Circumferential) Fins

o t = Fin Thickness, in.

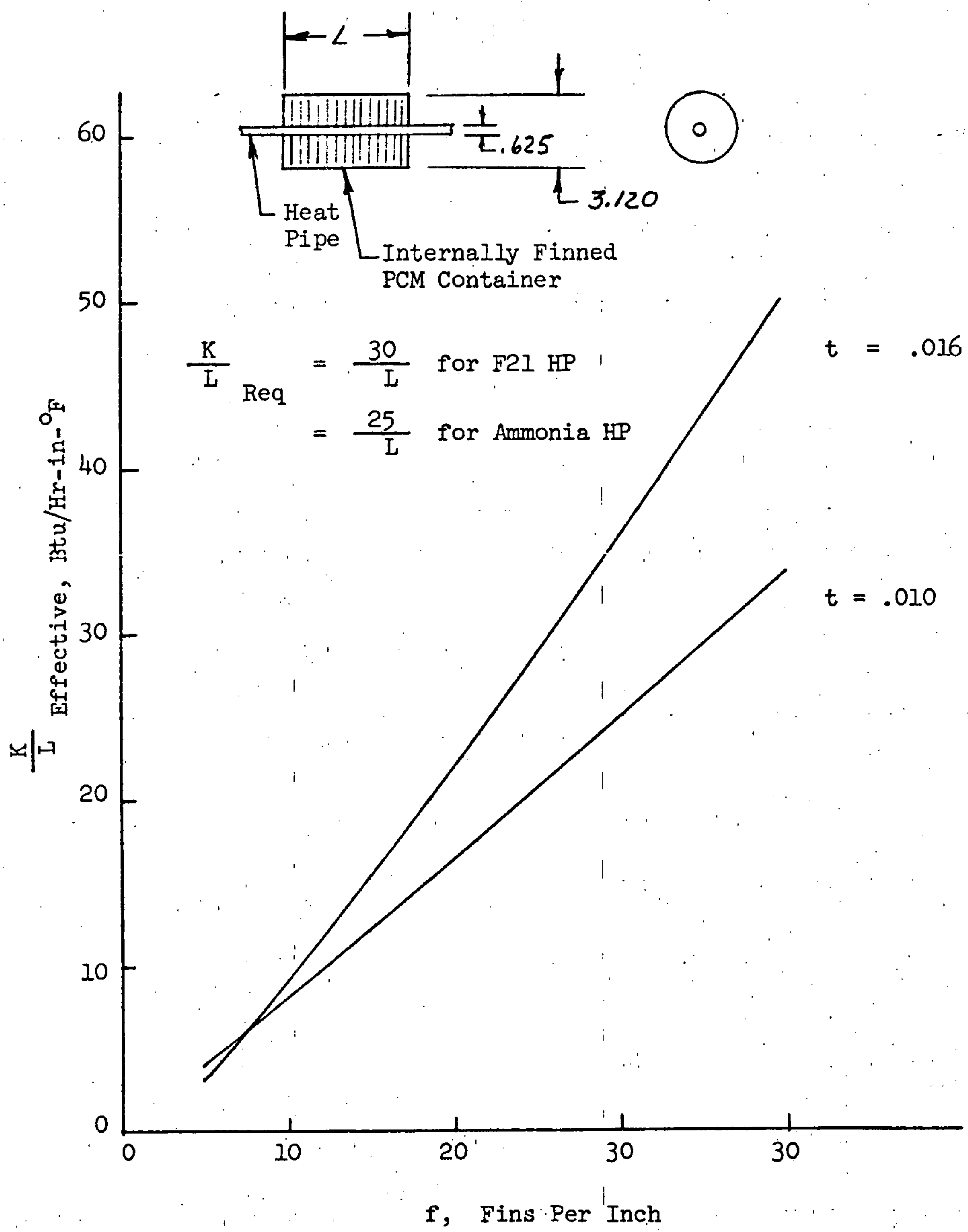
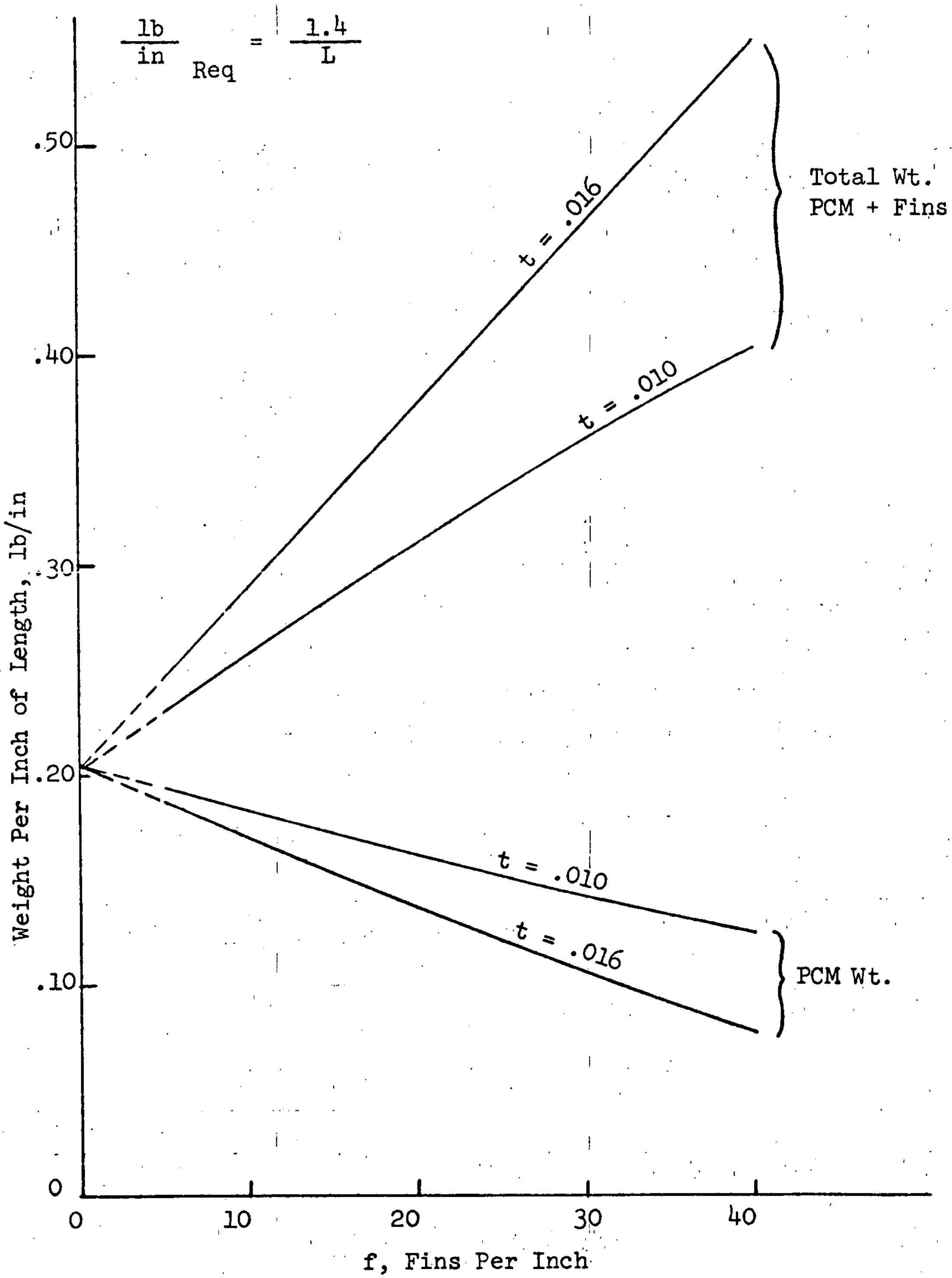


FIG. 6-3.5 - WEIGHTS - PCM AND FINS

- o Circular (Circumferential) Fins
- o t = Fin Thickness, in.
- o does not include heat pipe or container envelope



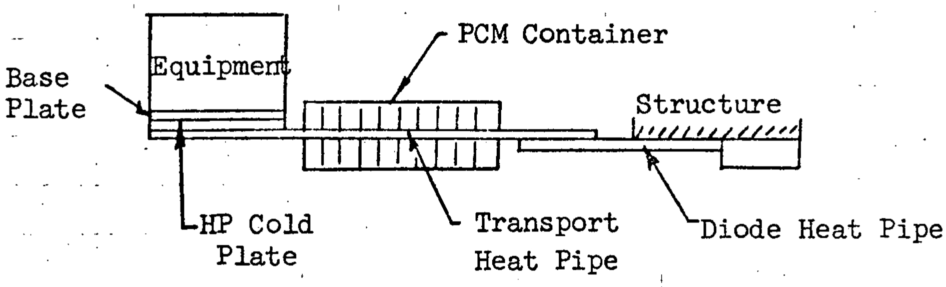
The PCM container, which houses the hexacosane and the conductive fins, completely surrounds a centrally positioned ten inch segment of the transport heat pipe. The container is a completely welded aluminum assembly whose major components are a 10-inch long, 3.125 ID tube (0.032 inch wall), with a fill port, and end baffles or discs which seal off the ends of the tube and support the central heat pipe. It contains 1.4 pounds of hexacosane, enough for 1.07 hours of operation with a 33% margin.

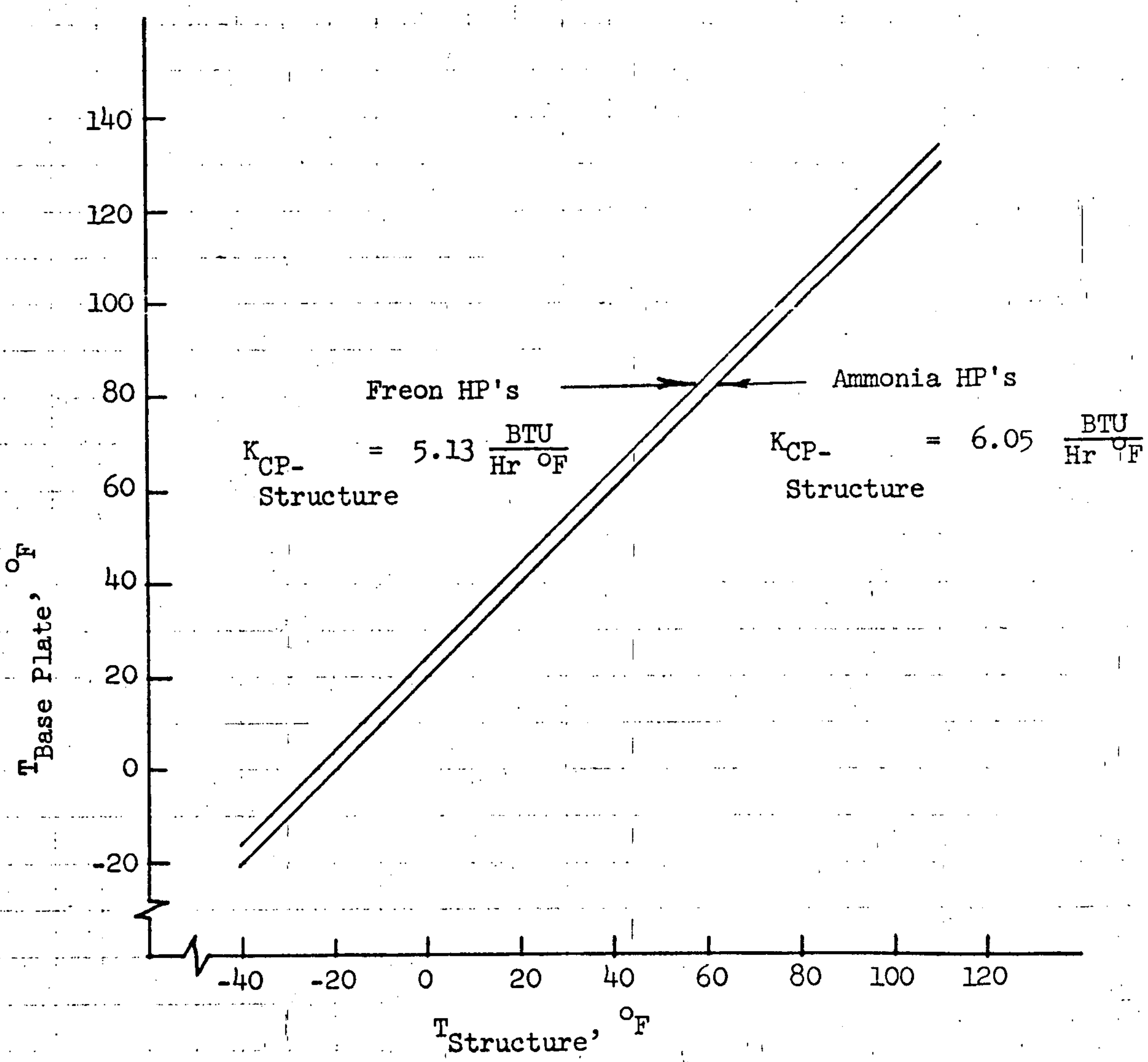
Figure 6-3.6 presents the electronics baseplate temperature as a function of the structural sink temperature. The base plate varies from $-20^{\circ}F$ to $+130^{\circ}F$ (within the design requirement) for an orbital sink variation of $-40^{\circ}F$ to $+110^{\circ}F$. The entire modular heat pipe system weighs 4.45 pounds versus a fluid cold plate system weight of 6.75 pounds, which includes the weight of the cold plate, fluid lines (40 feet) and coolant.

The modular heat sink thermal control concept has broad applicability to remotely located heat sources utilizing any number of possible sinks--e.g., structure, expendable fluids, phase change materials, isolated radiators. This design effort shows that it can be manufactured with current technology and methods.

FIG. 6-3.6 - MODULAR HEAT SINK PERFORMANCE (ON-ORBIT)

- o Heat Pipe Cold Plate Coupled to Structure
- o Q = 35 Watts





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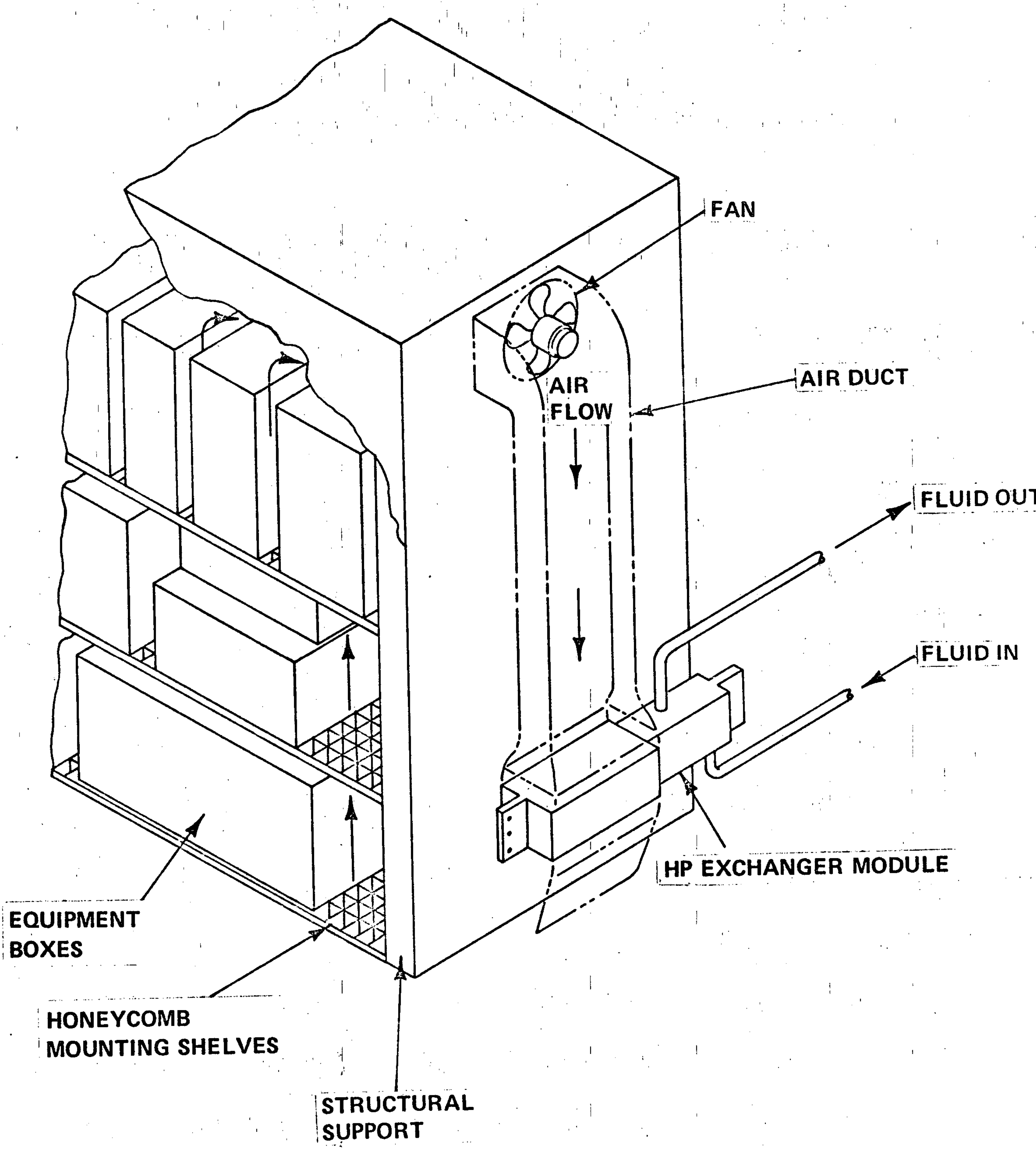
6-4 AIR-TO-HEAT PIPE HEAT EXCHANGER

Shuttle plans call for the utilization of qualified military and commercial avionics equipment as a cost effectiveness measure. Since these units are designed to be cooled by convection they will be housed in sealed racks (see Fig. 6-4.1) cooled by self-contained air circulation systems.

The heat pipe heat exchanger system transfers the heat from the air to the pumped loop (Fig. 6-4.2). Using heat pipes to separate the air and coolant sides of the exchanger eliminates the possibility of a leak discharging fluid into the air stream and damaging the electronic components. The HP unit is sized for an avionics load of 1800 watts plus a 10% allowance for the fan. Air temperatures in the rack are maintained below 131°F (MIL-E-5400, Class 1 equipment) down to pressures of 10 psia, with water temperatures to the exchanger of up to 70°F. Figure 6-4.2 is a schematic of the flow paths through the heat exchanger. Air enters the avionics rack and is heated at the rate of 1800 watts to the 131°F design value. The air moves past the fan and is heated to T_3 by the motor's heat load; it then enters the air side of the exchanger where it loses its heat to the evaporator sections of the heat pipes, exiting at temperature T_1. The heat absorbed at the evaporators is transmitted to the condenser and then to the fluid in the water side of the exchanger. Water enters the exchanger at 70°F, with a flow rate of 1 pound per hour per 8.25 watts of load, is heated by the pipes, and exits at temperature T_5.

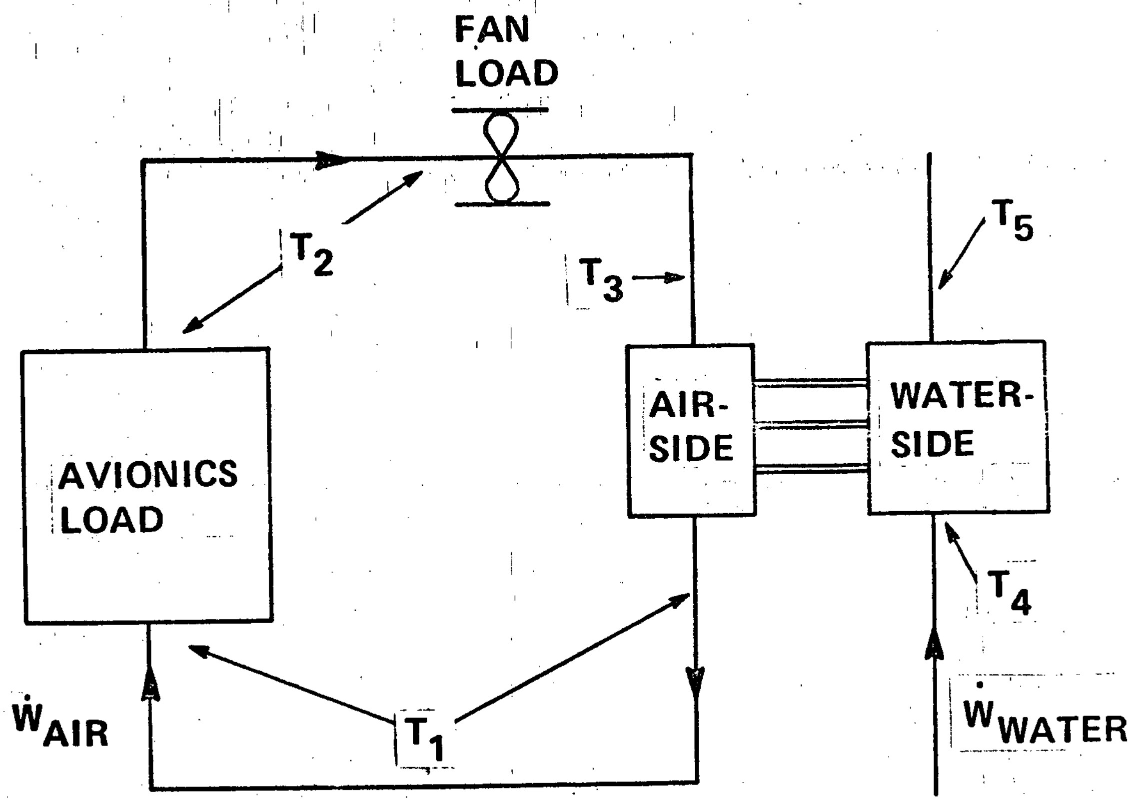
The fluid streams are counter-flow; thus the temperature rise in the water is equal to the temperature drop in the air stream. This means an equal heat load to each of the pipes since the temperature difference between the air and water streams along the flow path will be the same, facilitating the analysis and optimization of the system. These restrictions completely define the design envelope and permit calculation of the system temperatures, flow rates, and required UA_{Total}. The results are summarized in Fig. 6-4.2.

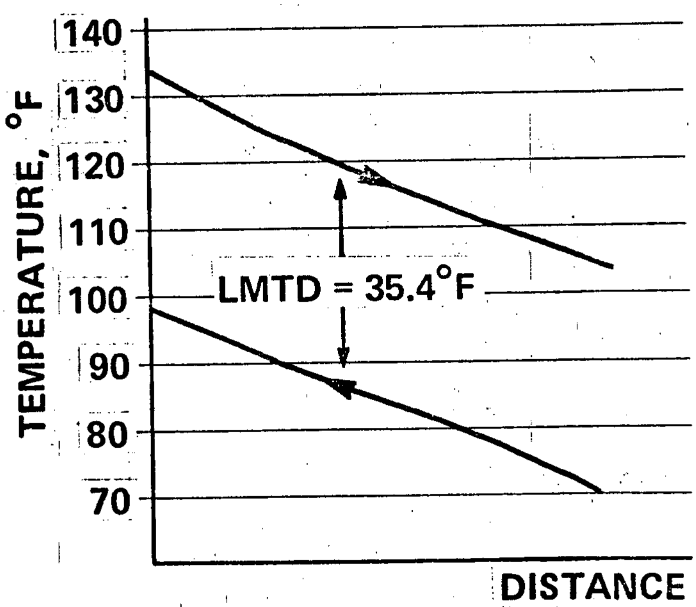
A maximum overall exchanger, and thus heat pipe length, is specified as 15 inches. The working fluid is water because of its relatively high thermal performance and low toxicity. Pipes of square cross-section facilitate assembly;



AVIONICS RACK SCHEMATIC

FIGURE 6-4.1





DESIGN DATA

Avionics Load = 1800 Watts

Fan Load Allowance = 180 Watts

$\dot{w}_{Water}$ (Allowance) = 240 Lb/Hr

$\dot{w}_{Air}$ (For Equal Fluid Capacity) = 1000 Lb/Hr

T_1 = 105.4°F

T_2 = 131.0°F

T_3 = 133.6°F

T_4 = 70°F

T_5 = 98.2°F

$UA_{TOTAL\ REQ'D} = \frac{Q}{LMTD} = 191.0$ BTU/Hr °F

HEAT EXCHANGER SCHEMATIC

FIGURE 6-4.2

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the planar outer surfaces of these pipes are readily bonded to the rectangular plate-fin type cores used for the fluid passages. Copper was chosen as the pipe material because of its proven compatibility with water and its high thermal conductivity (needed to increase the fin effectiveness of the two pipe sides not in direct contact with the exchanger's fluid cores). An optimum number of pipes and the relative evaporator and condenser section lengths were determined by relating the total heat pipe temperature gradient to the number of pipes and the evaporator section length. Additionally, the evaporator section unit heat flux was determined as a function of the same variables and was limited to a conservatively low 25 watts/in^2 to preclude the possibility of local dry-out.

The analyses showed seven pipes were optimum: fewer resulted in too large a temperature gradient while more are of decreasing incremental benefit. An evaporator length of 9.5 inches (5.25 inch condenser) was selected for the design. This results in a ΔT of 11°F and an evaporator flux of 19 watts/in^2. Eight pipes were included in the final design to provide redundancy in the event of the loss of any single pipe. With seven pipes, each will carry a load of 283 watts over an effective length of 7.6 inches. Analyses performed on 1/2-inch water pipes show transport capacities well in excess of this requirement.

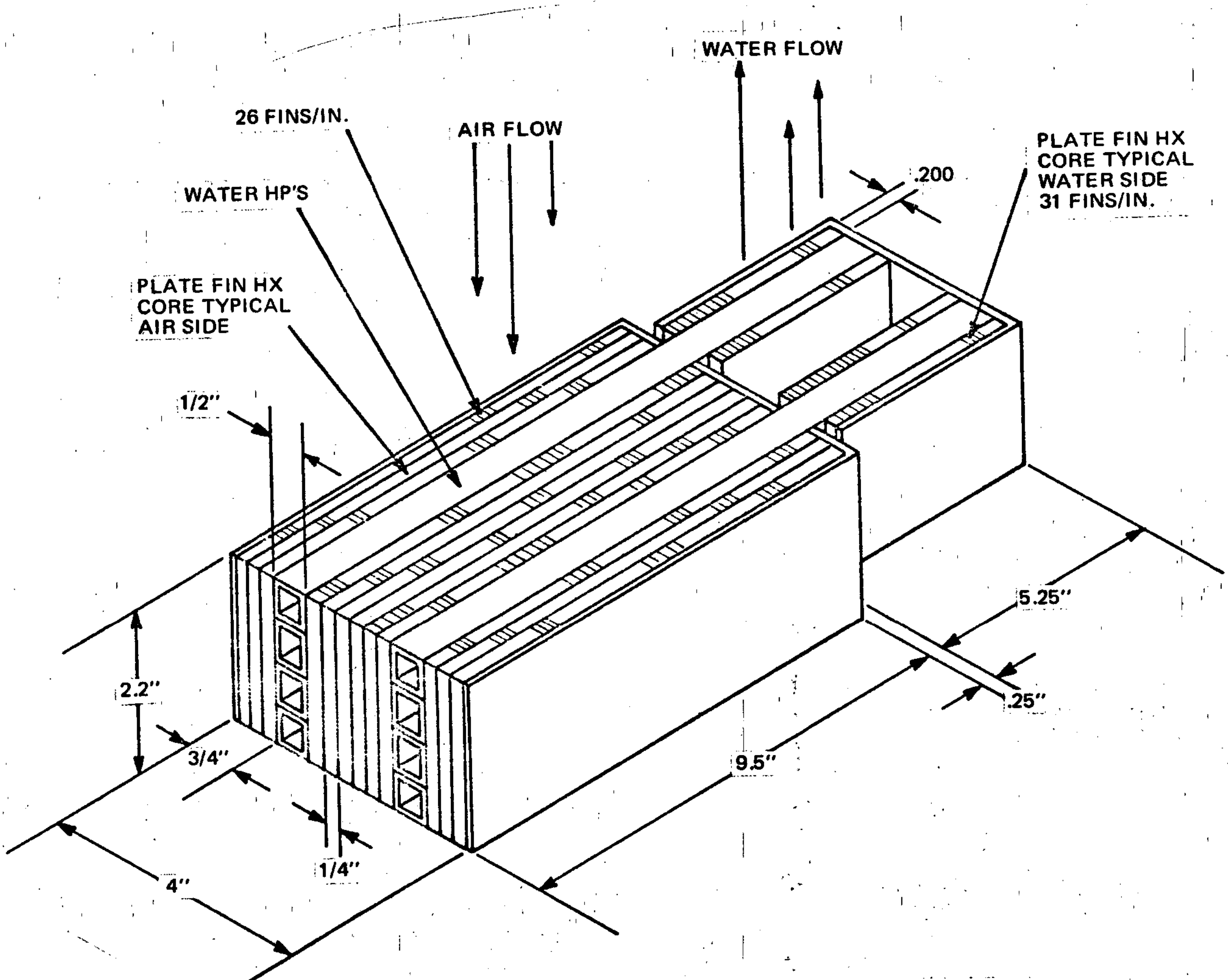
Knowing the ΔT and heat load, $\frac{1}{UA_{HP}}$ was calculated and subtracted from $\frac{1}{UA_{Total}}$ to yield $\frac{1}{UA_{Air}} + \frac{1}{UA_{Water}} = 0.00362^\circ F/Btu/hr$. The major portion of the allowable resistance was allocated to the air-side, and split as follows:

$$\frac{1}{UA_{Air}} = 0.00278^\circ F/Btu/hr$$

$$\frac{1}{UA_{Water}} = 0.00084^\circ F/Btu/hr$$

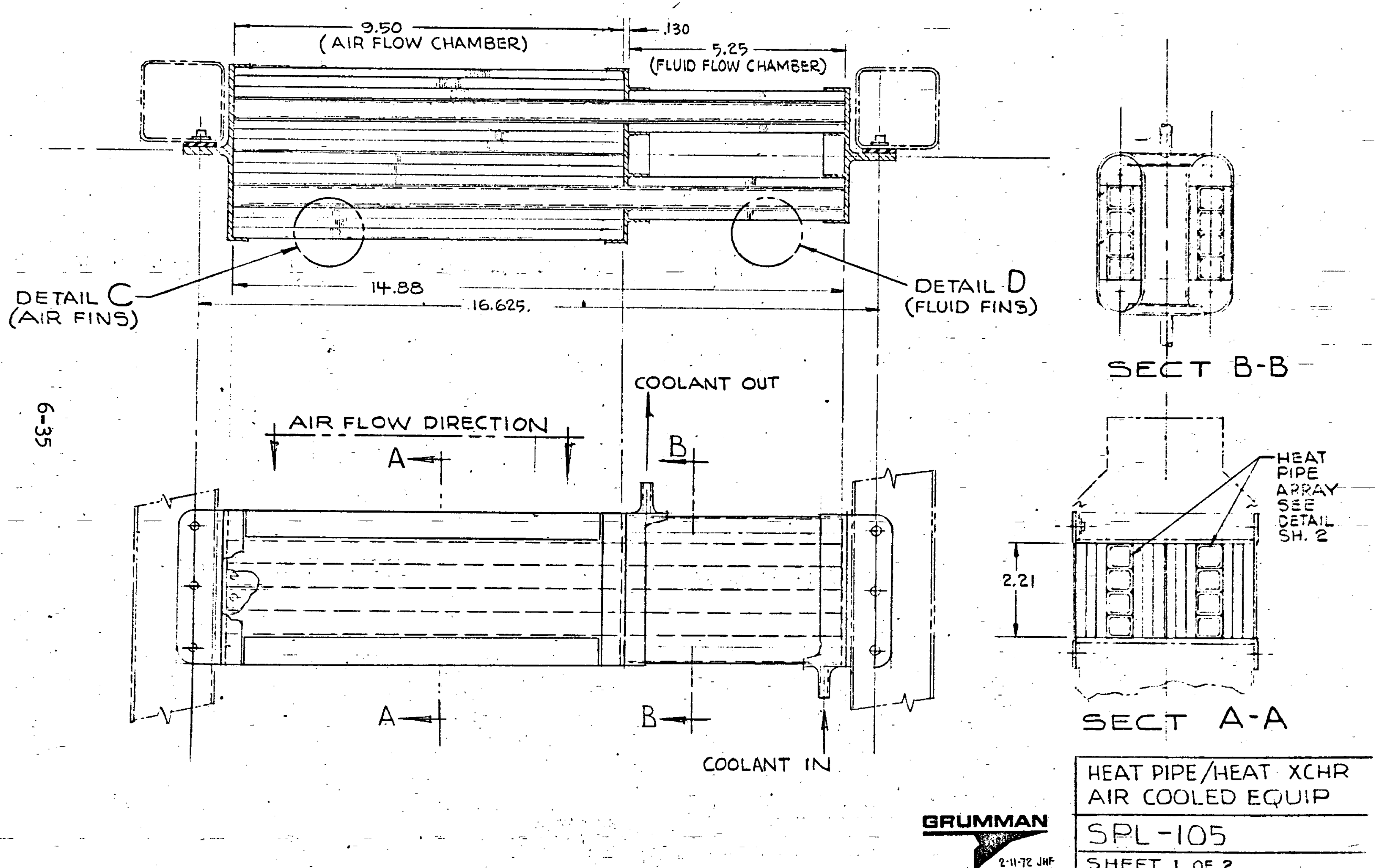
An allowable air core pressure drop was determined corresponding to a fan power allowance of 180 watts and a fan efficiency of 35%. This drop is 1.33 inches of water at 10 psia, 125°F. The design of the plate-fin type air and water cores was performed using a Grumman developed technique for sizing a laminar plate-fin heat exchanger. The method uses a mathematical correlation of heat transfer data for the flow of air in rectangular ducts (Ref. 8). By specifying allowable UA's, pressure drops, flow length and geometric core arrangement, the size and required number of core layers was determined as well as the numbers of fins per inch needed in the extended surface.

Drawing SPL-105, at the end of this section, gives some design details for the system; Figure 6-4.3 shows an isometric view of the final exchanger design. The pipes are arranged in two rows of four each. On the air-side, a three-layer core is arranged on either side of each row of pipes. On the water side, only a single layer core is required.



Avionics Rack HP Heat Exchanger

Figure 6-4.3



9.50
(AIR FLOW CHAMBER)
.130
5.25
(FLUID FLOW CHAMBER)
DETAIL C
(AIR FINS)
14.88
16.625
DETAIL D
(FLUID FINS)
SECT B-B
COOLANT OUT
AIR FLOW DIRECTION
A
B
HEAT PIPE ARRAY SEE DETAIL SH. 2
2.21
SECT A-A
COOLANT IN
GRUMMAN
2-11-72 JHF
HEAT PIPE/HEAT XCHR AIR COOLED EQUIP
SPL-105
SHEET 1 OF 2

FIG. 6-4.4

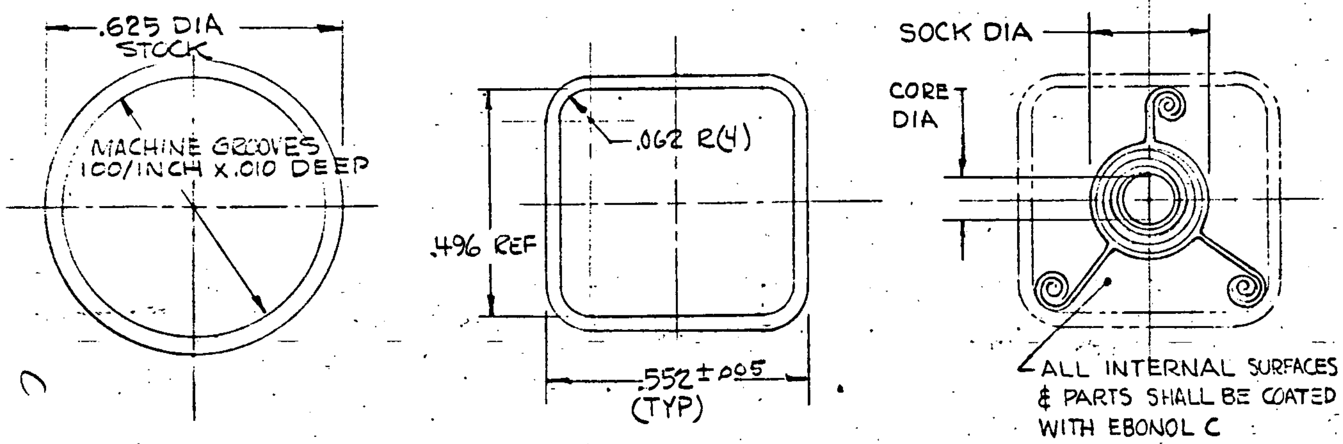
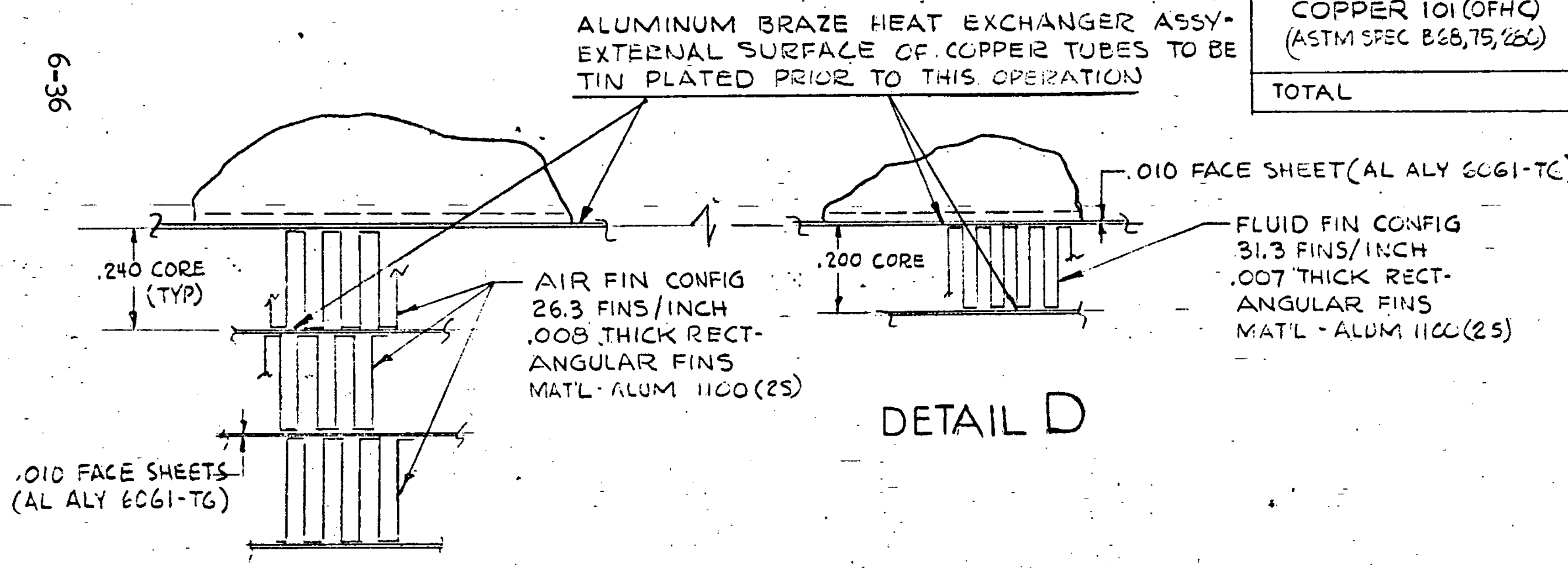


TABLE I

INTERNAL HEAT PIPE CONFIGURATION

ITEMS & DATA		WT #/FT
WORKING FLUID		
DISTILLED WATER		.024
SPIRAL ARTERY		.023
HOLLOW CORE DIA =	.110	
NO OF WRAPS =	3	
GAP SPACE (OR ROD DIA) =	.010	
NO OF RODS =	12	
RETAINER		.020 (EST)
SOCK DIA (O.D)	.250	
NO. OF WEBS	3	
TUBE		.173
COPPER 101 (OFHC)		
(ASTM SPEC B68,75,280)		
TOTAL		.24 #/FT



DETAIL C

DETAIL D

SCALE 4:1

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SHEET 2 OF 2

6-5 HEAT PIPE RADIATORS FOR COMPARTMENT TEMPERATURE CONTROL

The shuttle contains several structural compartments which require on-orbit thermal control. Typically, these are unpressurized and isolated by their remote locations or by insulation systems. There are compartments for two main landing gears and one nose gear; self-contained RCS modules, located at the extremities of the wing and tail assembly; and air breathing engine compartments. They all share the basic thermal requirement of maintaining their contents between prescribed temperature limits for all mission phases. In the baseline design insulation protects against a hot environment while the combination of heat addition and insulation controls the temperature in a cold environment.

Fluid radiators and electrical heaters are being considered as the heat sources in the baseline systems. The electrical heaters need thermostatic controls to protect against runaway circuits and they need power. In the case of the shuttle, the additional power is a concern because of the weight penalty it presents in terms of extra fuel cell capacity and reactant requirements, typically 0.435 pounds per watt. The fluid radiators utilize the sensible heat of the Freon-21 in the orbiter's heat rejection system; they use waste heat which would otherwise be rejected to space. The Freon-21 is taken from the fuel cell coolant outlet, the warmest point in the heat rejection loop, and circulated through aluminum panels which line the compartment walls and then returned to the heat rejection system. The radiators need supporting structure, bypass, and shut-off controls and protection from tube puncture when such a hazard exists. Controls are needed to prevent reverse heat transfer during entry.

A heat pipe radiator system for compartment temperature control has power and weight advantages over an electrical system and control and reliability advantages over a conventional fluid radiator. The heat pipe radiator system described herein has been designed for the orbiter's main landing gear compartment, although in principle and concept it can be used elsewhere. The compartment contains rubber tires and hydraulic actuators which cannot survive temperatures below -65°F. Also, the tires are limited to -50°F for landings, and -20°F is the desired lower limit for the hydraulic fluid. The upper temperature limits are 270°F for the tires and 400°F for the hydraulics.

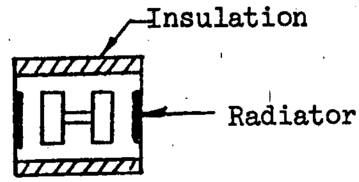
The thermal analyses for an insulated compartment without supplemental heat input indicate landing gear temperatures ranging from -85°F for a solar inertial orbit to -36°F for an earth oriented mode. As seen in Fig. 6-5.1, a compartment heat input of 10 Btu/hr/ft^2 of radiator (94 watts total) will maintain landing gear temperatures above 0°F and was selected as a reasonable lower limit design point for the heat pipe radiator system. Waste head from the Freon-21 heat rejection system is a convenient and reliable source of energy, with the fluid temperature ranging from 85 to 117°F, and a flow rate of 2000 lb/hr.

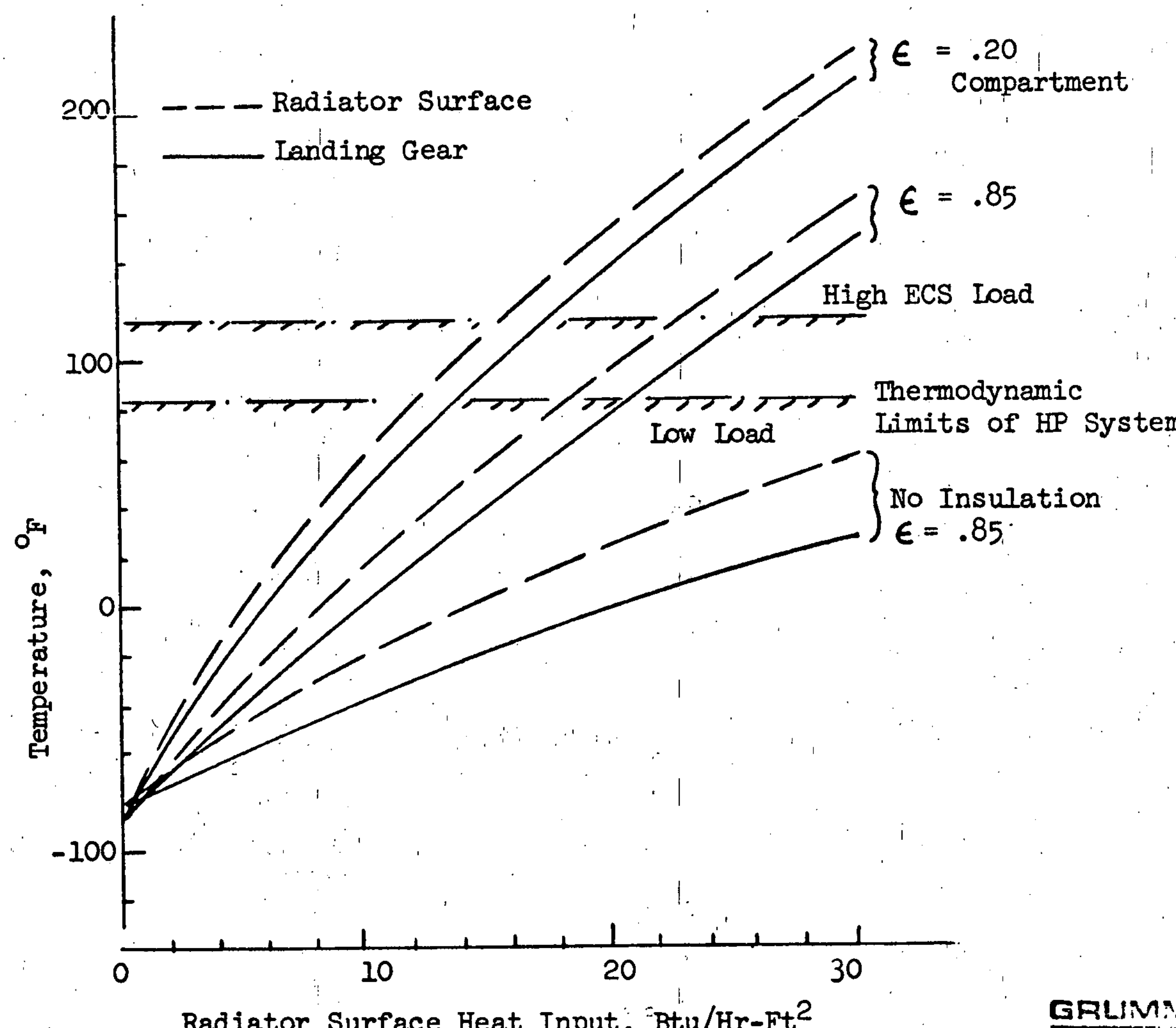
As shown in Fig. 6-5.2 the radiator system for each compartment consists of two heat pipe radiator panels, two diode heat pipes, and two heat pipe-to-fluid heat exchangers. The exchangers are placed in parallel with the fluid flow. In this case there are a total of four exchangers for both main landing gear compartments, resulting in a maximum available flow rate of 500 lb/hr for each exchanger. The heat pipe radiator panels are located on each side of the compartment facing a tire sidewall. In this configuration, approximately 80% of the energy emitted by the radiator is received by the landing gear with the remainder distributed to the rest of the compartment. Each panel is separately connected to the fluid loop by a diode heat pipe and heat exchanger. The diode permits heat transfer from the fluid to the compartment radiator panel only when the panel temperature is less than the fluid temperature. Energy is transferred from the fluid to the evaporator section of the diode heat pipe by means of the annular HP/fluid heat exchanger. The diode, functioning as a normal heat pipe, then transmits the heat to the attached evaporator sections of the radiator panel feeder heat pipes. The energy is ultimately transferred to the radiator surface from the condenser sections of the feeder heat pipes which are jointed to the panel. The detailed design drawing for a compartment radiator panel (SPL-103) is included at the end of this section.

Each radiator panel has 16 ft^2 of area and three 1/2 inch ID "L" shaped aluminum heat pipes, 6 inches by 52 inches. The long leg (condenser section) of the heat pipes are brazed to the 20 mil thick aluminum panel on 16.8 inch centers providing a fin effectiveness from 61% to 54% over the operating range of 40 to 115°F. The evaporator sections (short leg) interface with the condenser

FIG. 6-5.1 - LANDING GEAR TEMPERATURE vs. RADIATOR SURFACE HEAT INPUT

- o Solar Inertial Orbit
- o Insulation Thickness = 1 inch (TG-15000)





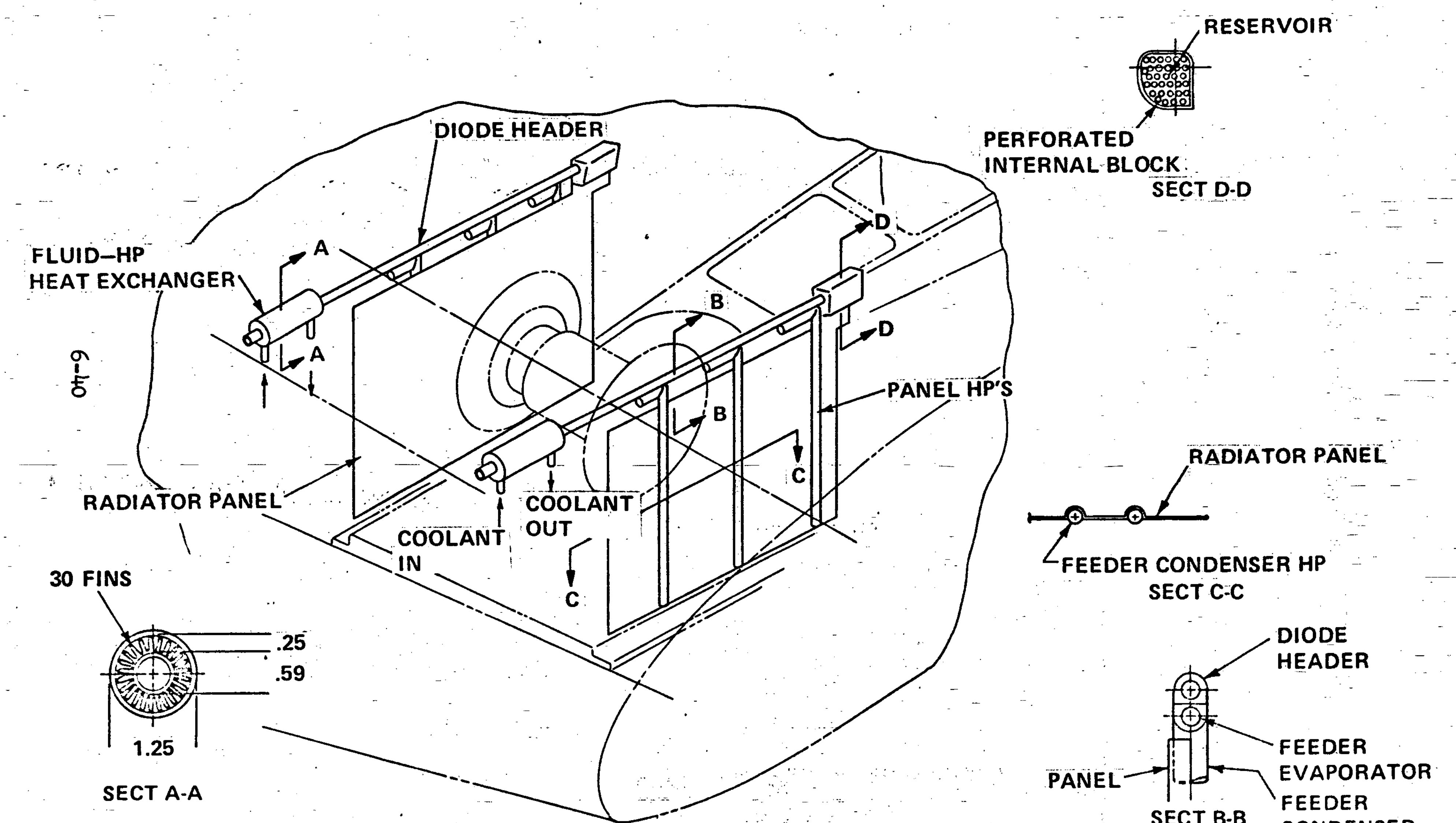


FIGURE 6-5.2 COMPARTMENT RADIATOR

of the diode heat pipe through a machined clamped joint. Each panel pipe is required to carry 16 watts; 47 watts per panel. The artery design was optimized for the specified dimensions and Freon-21 as the working fluid. This resulted in selecting a spiral artery/tunnel wick having a 95 watt capacity at 40°F, as seen in Fig. 6-5.3.

The diode heat pipe in this application operates on the liquid blockage principle. When the temperature of the condenser exceeds that of the evaporator, excess fluid, which is held in a reservoir at the condenser end, vaporizes. This excess fluid vapor condenses and fills the vapor space, thereby "choking" the heat pipe. The on-orbit upper temperature of the radiator is limited to the maximum fluid temperature that the diode can sense in the heat exchanger. The diode carries the entire 47 watt load of a radiator panel. It is a 1/2 inch ID aluminum pipe with a 5-inch evaporator, 52-inch condenser and 15-inch transport section. It uses ammonia as its working fluid and has an optimized spiral artery/tunnel wick which can provide a 370 watt capacity at the design point. Its performance versus temperature is given in Fig. 6-5.3. The reservoir contains 1.6 in^3 of ammonia, enough to flood the evaporator and half the transport section in the reverse mode.

Each heat pipe-to-fluid heat exchanger consists of a finned annular passage which completely surrounds the centrally positioned evaporator of the diode. There are 30 aluminum fins in the annulus; each 0.010 inch thick and 0.250 inch high brazed to the outer surface of the evaporator. The heat transfer length is 5 inches, consistent with the design heat flux density of 25 watts/in^2. The performance characteristics of the heat exchanger are given in Fig. 6-5.4.

The heat pipe radiator system, as described, is capable of maintaining the on-orbit main landing gear temperatures between 0°F and 117°F with a heat exchanger flow rate of 150 lb/hr, or only 30% of the maximum available rate. This can be seen by the performance estimates in Fig. 6-5.5, where at the stated conditions, the system will provide 14 Btu/hr/ft^2 and a minimum landing gear temperature of 32°F (from Fig. 6-5.1). The system weights 0.74 pounds

FIG. 6-5.3 - HEAT PIPE TRANSPORT CAPACITIES

- Diode Header
- Panel Feeder Pipe

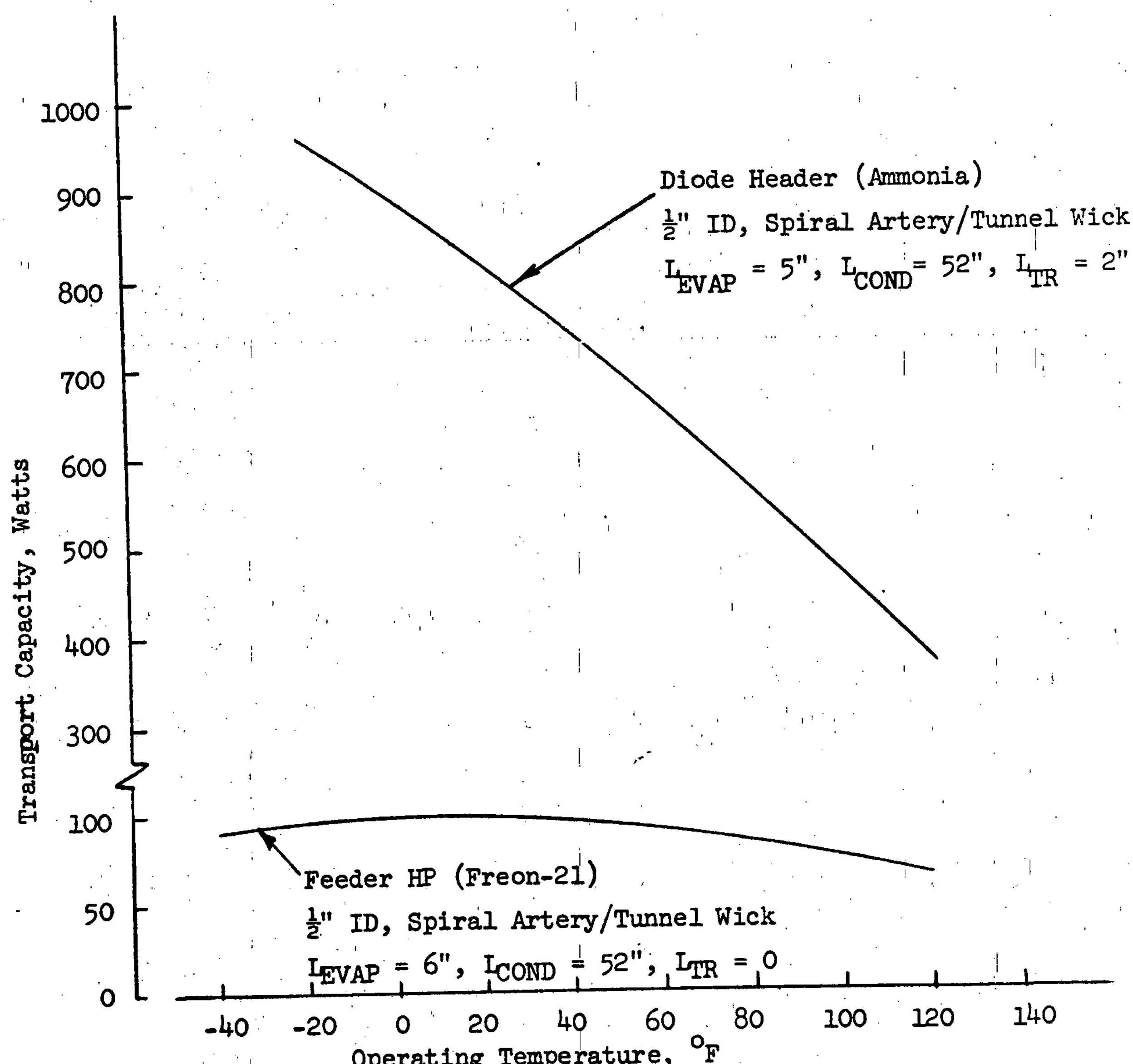
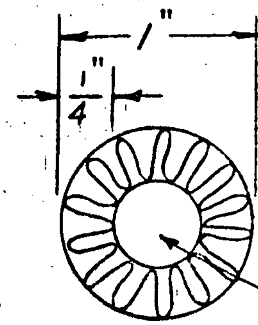


FIG. 6-5.4 - HEAT EXCHANGER PERFORMANCE CURVES

- Aluminum Fins (≈ 30) 11.1 Fins/In.
- Fin Thickness = .010"
- Freon-21



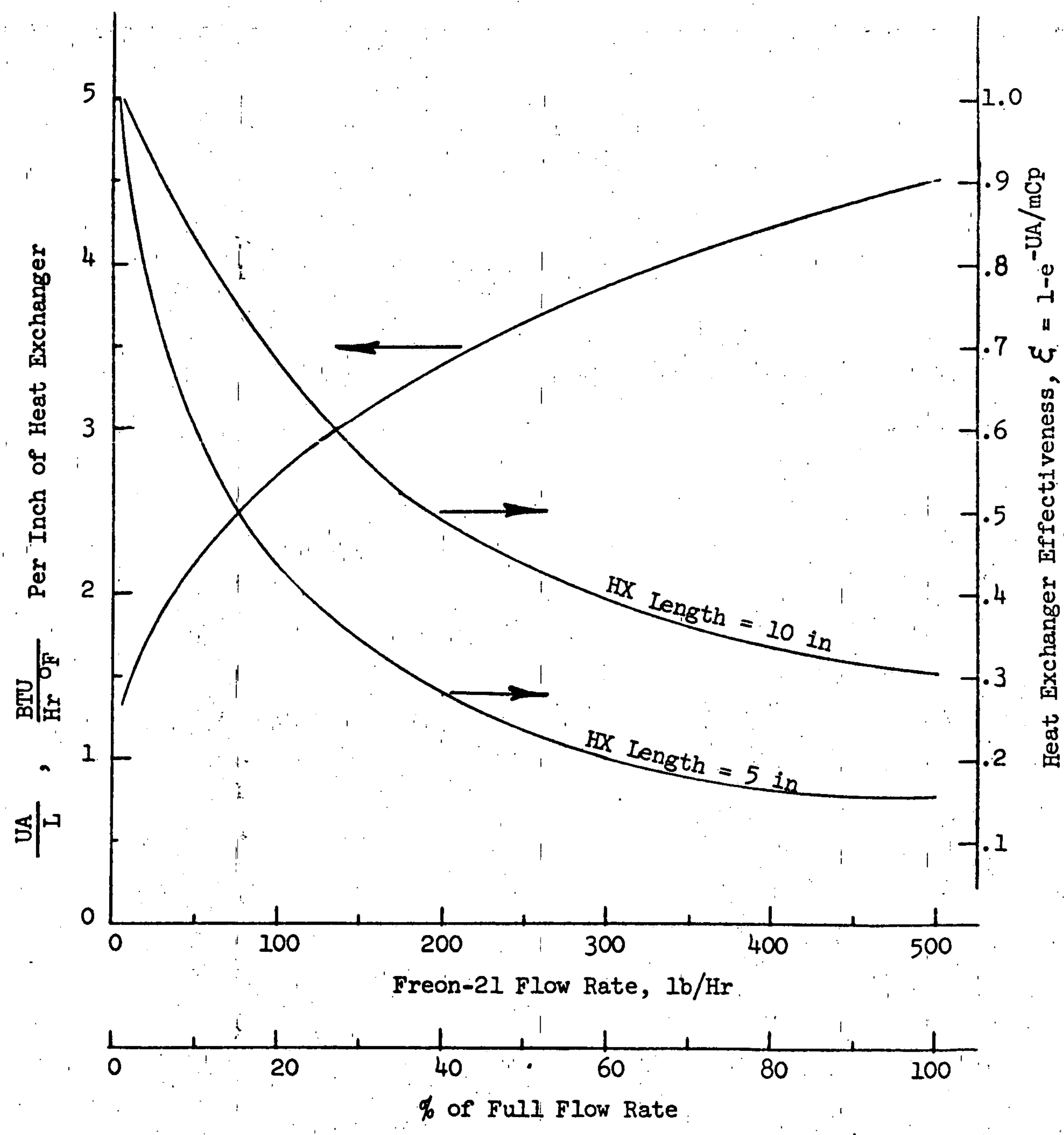
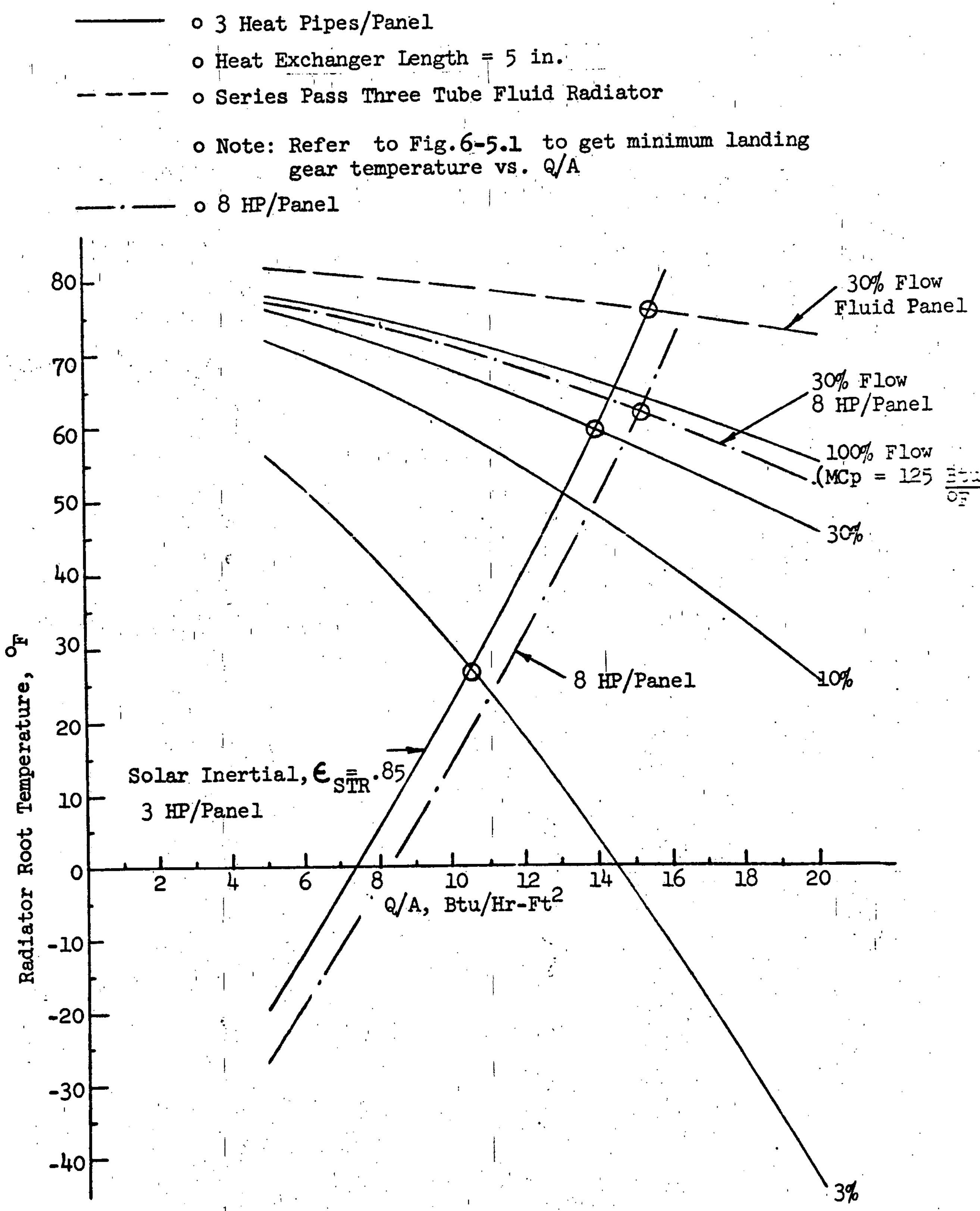
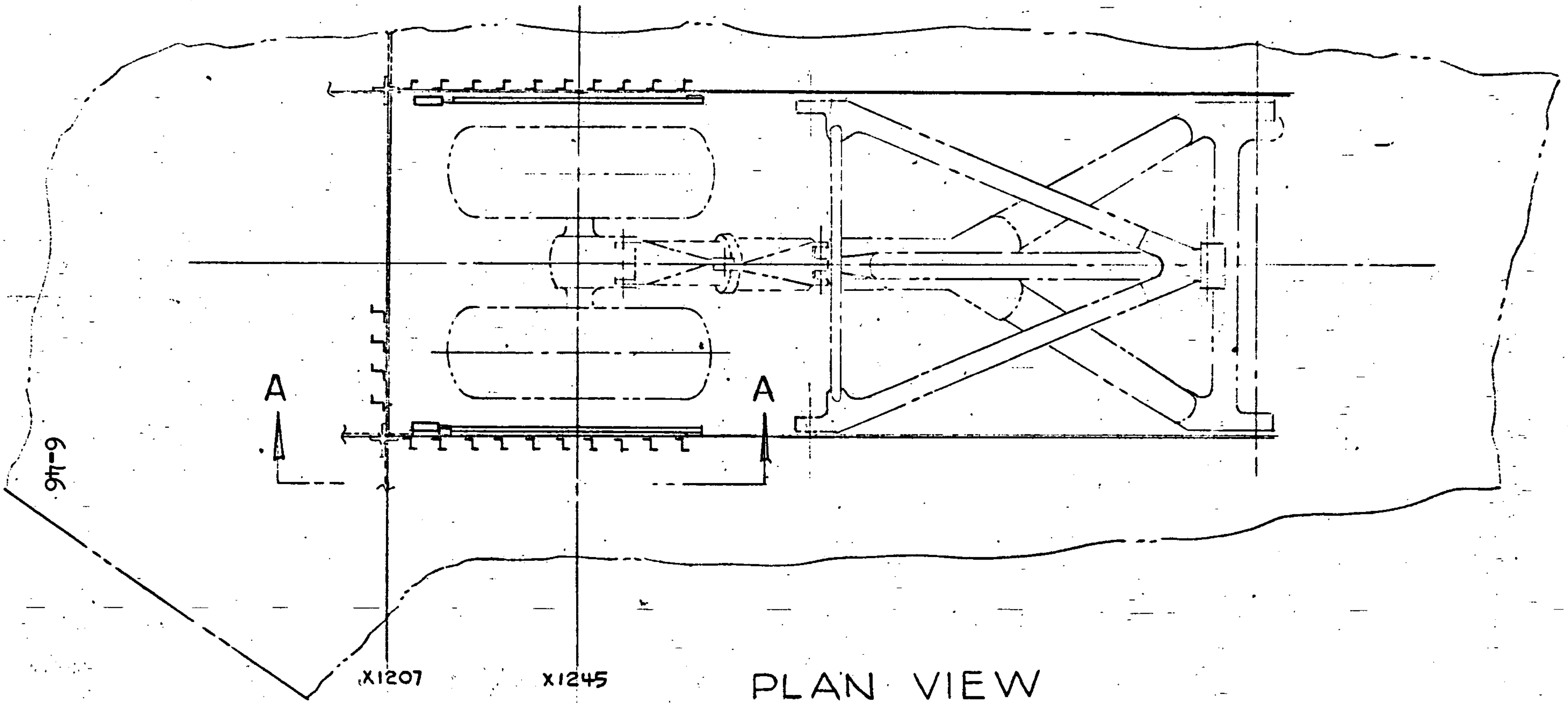


FIG. 6-5.5 - HP RADIATOR SYSTEM PERFORMANCE



per ft^2 of radiator surface versus 0.78 lb/ft^2 for a conventional fluid radiator, while the weight penalty for a system which uses electrical heaters would be 1.61 lb/ft^2. In addition to the power and weight advantages over baseline systems, the heat pipe system is simpler and more reliable due to the inherent self-controlling features of its diode header.



PLAN VIEW
MAIN L/G (L.H.) GENERAL ARRANGEMENT
(ORBITER VEHICLE)
SCALE - 1/20

FIG. 6-5.6

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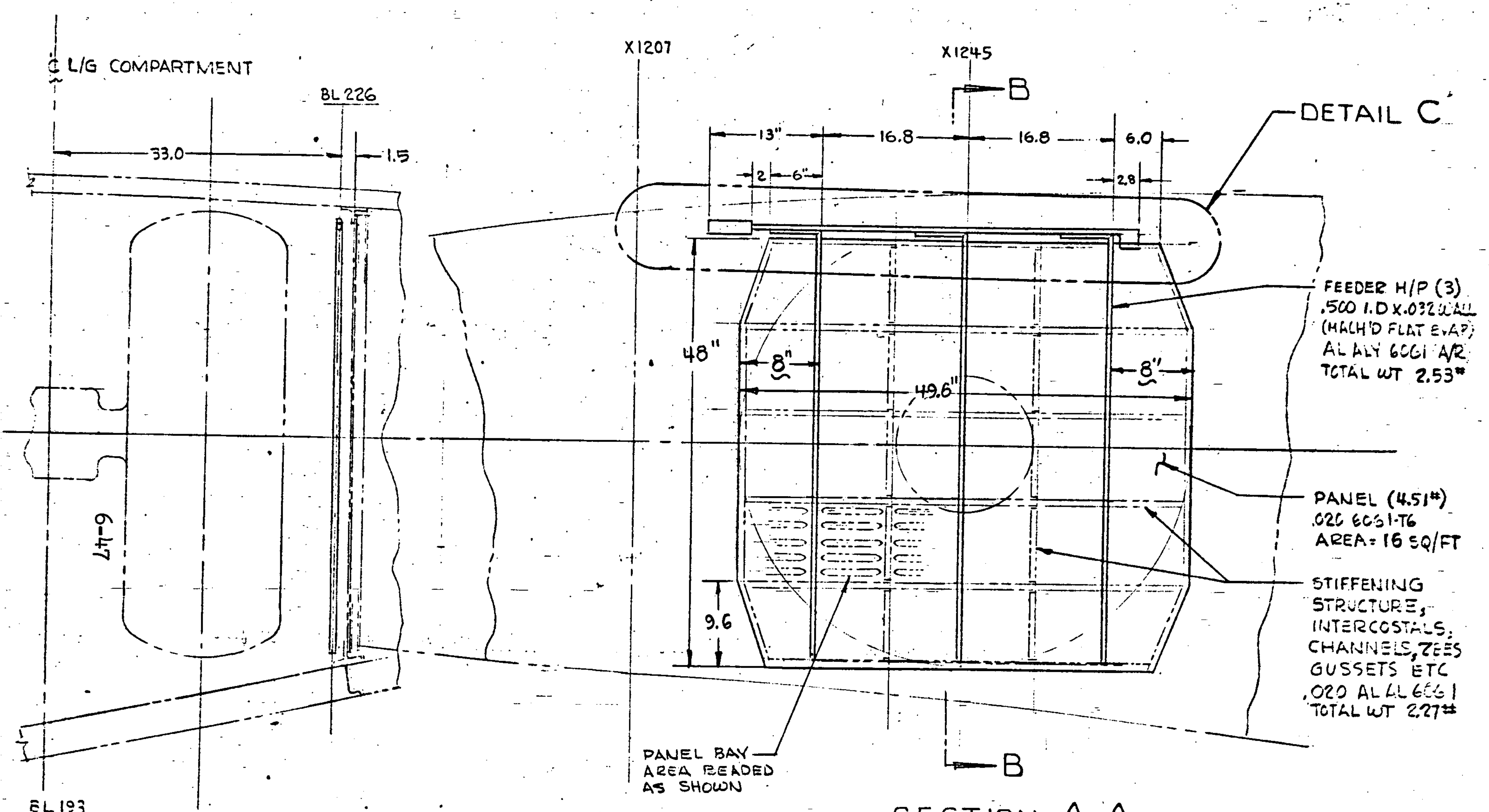
HEAT PIPE RADIATOR APPL MAIN L/G COMPARTMENT ORBITER	
SPL-103 SHEET 1 OF 4	



SECTION B-B

SECTION A-A
(SCALE 1/10)

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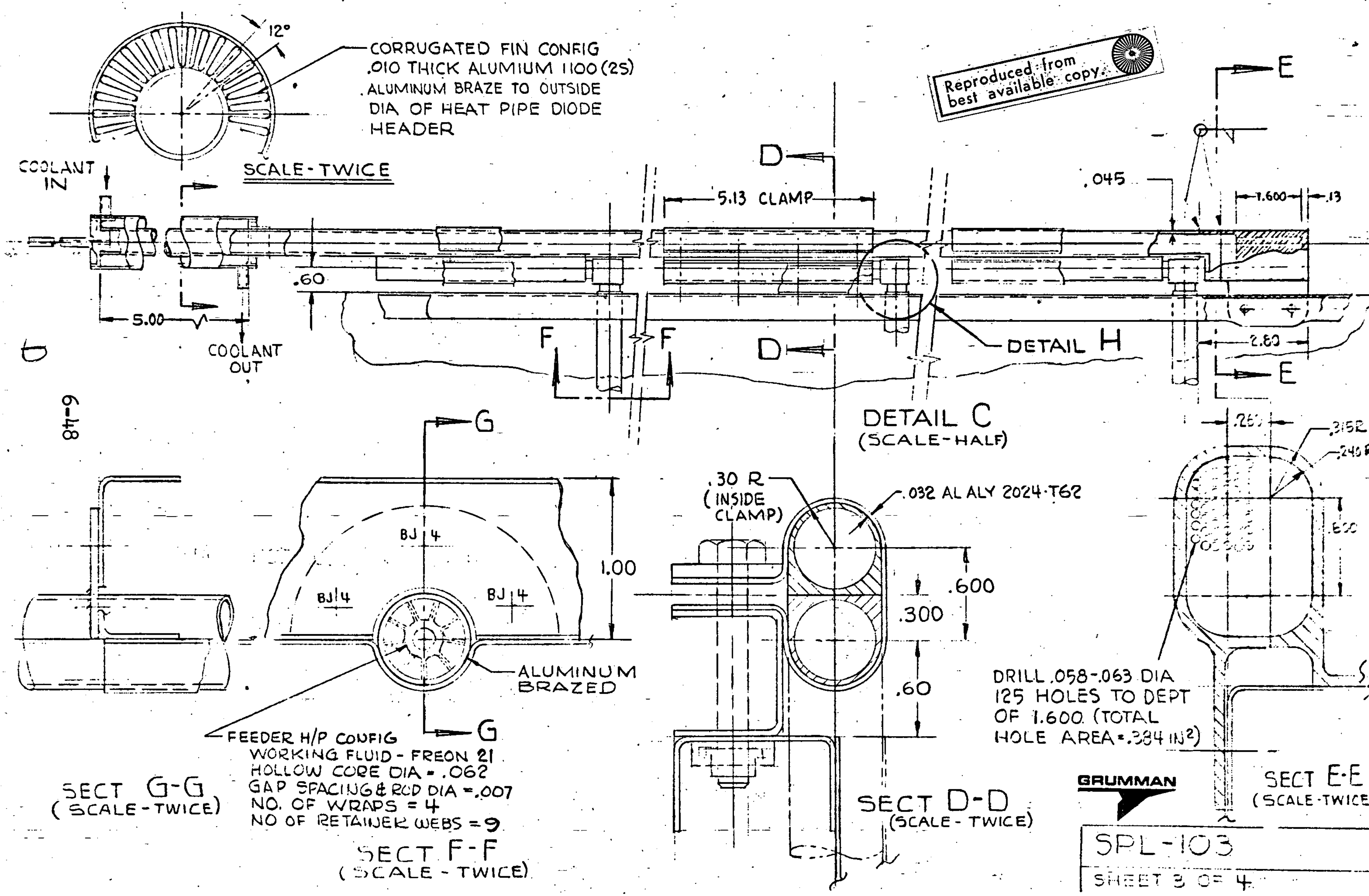
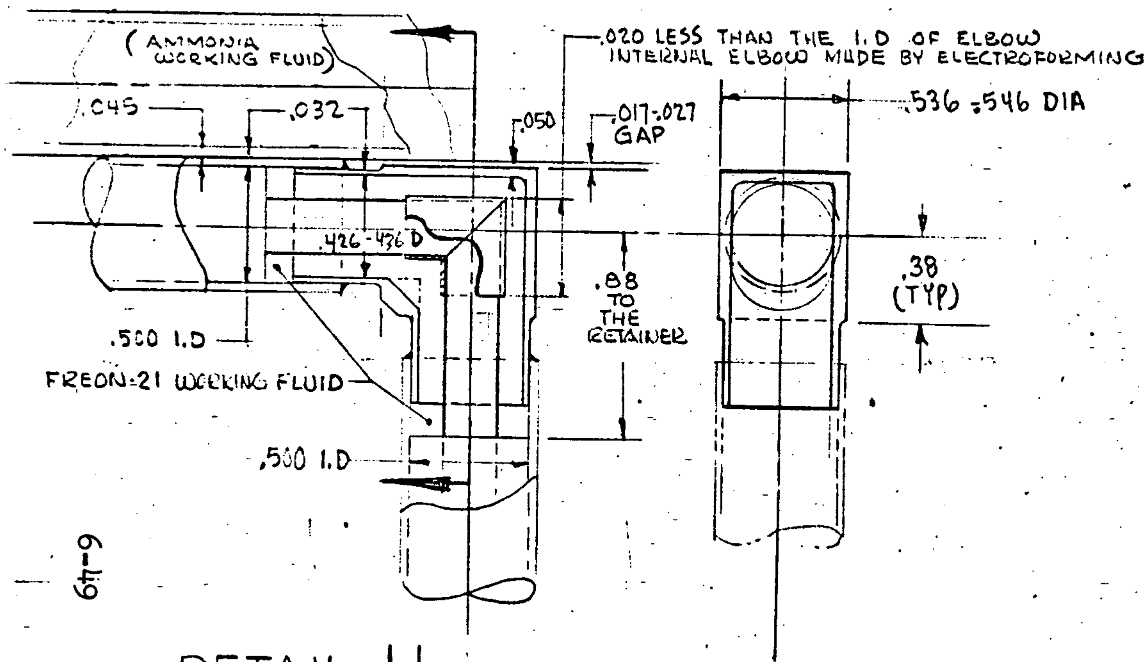
SHEET 2 OF 4


CORRUGATED FIN CONFIG
.010 THICK ALUMIUM 1100 (2S)
ALUMINUM BRAZE TO OUTSIDE
DIA OF HEAT PIPE DIODE
HEADER
12°
SCALE-TWICE
COOLANT IN
COOLANT OUT
5.00
.60
5.13 CLAMP
.045
1.600
.13
2.80
D
E
F
DETAIL H
DETAIL C
(SCALE-HALF)
G
BJ 4
1.00
ALUMINUM BRAZED
FEEDER H/P CONFIG
WORKING FLUID - FREON 21
HOLLOW CORE DIA = .062
GAP SPACING & ROD DIA = .007
NO. OF WRAPS = 4
NO OF RETAINER WEBS = 9
SECT G-G
(SCALE-TWICE)
SECT F-F
(SCALE - TWICE)
.30 R
(INSIDE CLAMP)
.032 AL ALY 2024-T62
.600
.300
.60
SECT D-D
(SCALE - TWICE)
.315R
.240R
.600
DRILL .058-.063 DIA
125 HOLES TO DEPT
OF 1.600 (TOTAL
HOLE AREA = .384 IN²)
SECT E-E
(SCALE-TWICE)
GRUMMAN
SPL-103
SHEET 3 OF 4



DETAIL H
(SCALE - TWICE)

ESTIMATED WEIGHT BREAKDOWN
FOR SINGLE PANEL CONFIGURATION

1- 16 FT² SURFACE FIN PANEL AL ALY 6061-T6 .020T x 48.0H x 49.6 W		4.51 #
2- FEEDER H/P (FREON-21) .843 # EACH x 3 REQ'D		2.53 #
3- PANEL STIFFENERS CHANNELS, ZEE, GUSSETS, ETC.		2.27 #
4- FEEDER H/P TO DIODE HEADER ATTACH'MT		.54 #
3 CLAMPS .032 AL ALY 2024-T62 =	.22	
(12) RADIUS BLOCK .125x.70x.75	.076	
(12) #10-32 BOLTS (1 3/8 LG) STL	.132	
(3) SPACER CHANNELS (5.13 LG) .032 AL ALY 2024-T62	.110	
5- DIODE HEADER (AMMONIA)		.70 #
6- HEAT EXCHANGER		.17 #
		10.72 #
10% ALLOWANCE, SUPPORT H'DWARE, BACKUP/STRUCT		1.07 #
TOTAL WT FOR 16 SQ/FT PANEL		11.79 #

OR $\frac{11.79}{16 \text{ SQ/FT AREA}}$ = .74# PER SQ/FT

TOTAL FOR ORBITER VEHICLE
FOR FOUR PANELS = 47.2 #

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SHEET 4 OF 4

SECTION 7

CONCLUSIONS

Heat pipe applications for the shuttle have been found which either supplement or replace conventional thermal control systems. They present viable alternatives offering possible performance, weight and reliability advantages. Existing technology has proven the performance capabilities of the heat pipe as an individual component and pointed out its unique control and reliability features. Heat pipes have been successfully operated as simple heat pipes, thermal diodes and variable conductance heat pipes.

The most practicable shuttle heat pipe applications, having the best chance of initial implementation, exist in the low temperature regime ($-40^{\circ}F$ to $150^{\circ}F$) and require moderate heat transport capacities (500 to 10,000 watt-inches). Artery-type heat pipes in finely grooved envelopes are best suited to meet these requirements because their high capacities provide large design margins, and their high evaporator and condenser film coefficients result in lower overall system temperature gradients - hence, more thermally efficient heat pipe systems. They also self-prime and function in a gravity field, which is mandatory since these systems must not only be ground tested in 1-'g' but might also be called upon to operate on the launch pad, during boost and entry, and possibly during earthbound ferry missions.

SECTION 8

RECOMMENDATIONS

The sustained, rapid advance made in heat pipe technology and hardware points to the imminent and widespread acceptance of heat pipe thermal control systems for both manned and unmanned spacecraft. Heat pipes have progressed beyond the laboratory curiosity stage and their unique performance capabilities cannot be denied - the time has come to put them to the proper test.

When properly integrated into the Shuttle, heat pipes could result in lighter, simpler, more reliable thermal control systems with greater operating efficiency. As a step toward realizing these potential benefits on the Shuttle, it is recommended that confidence in the capabilities of the selected heat pipe systems be firmly established by building and testing the prototype hardware.

SECTION 9

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SECTION 10

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